Basic Ship Theory

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Fifth edition

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Foreword to the fifth edition

Over the last quarter of the last century there were many changes in the maritime scene. Ships may now be much larger; their speeds are generally higher; the crews have become drastically reduced; there are many different types (including hovercraft, multi-hull designs and so on); much quicker and more accurate assessments of stability, strength, manoeuvring, motions and powering are possible using complex computer programs; on-board computer systems help the operators; ferries carry many more vehicles and passengers; and so the list goes on. However, the fundamental concepts of naval architecture, which the authors set out when *Basic Ship Theory* was first published, remain as valid as ever.

As with many other branches of engineering, quite rapid advances have been made in ship design, production and operation. Many advances relate to the effectiveness (in terms of money, manpower and time) with which older procedures or methods can be accomplished. This is largely due to the greater efficiency and lower cost of modern computers and proliferation of information available. Other advances are related to our fundamental understanding of naval architecture and the environment in which ships operate. These tend to be associated with the more advanced aspects of the subject: more complex programs for analysing structures, for example, which are not appropriate to a basic text book.

The naval architect is affected not only by changes in technology but also by changes in society itself. Fashions change as do the concerns of the public, often stimulated by the press. Some tragic losses in the last few years of the twentieth century brought increased public concern for the safety of ships and those sailing in them, both passengers and crew. It must be recognized, of course, that increased safety usually means more cost so that a conflict between money and safety is to be expected. In spite of steps taken as a result of thet experiences, there are, sadly, still many losses of ships, some quite large an some involving significant loss of life. It remains important, therefore, to strive to improve still further the safety of ships and protection of the environment. Steady, if somewhat slow, progress is being made by the national and international bodies concerned. Public concern for the environment impacts upon ship design and operation. Thus, tankers must be designed to reduce the risk of oil spillage and more dangerous cargoes must receive special attention to protect the public and nature. Respect for the environment including discharges into the sea is an important aspect of defining risk through accident or irresponsible usage.

A lot of information is now available on the Internet, including results of much research. Taking the Royal Institution of Naval Architects as an example of a learned society, its website makes available summaries of technical papers and enables members to join in the discussions of its technical groups. Other data is available in a compact form on CD-rom. Clearly anything that improves the amount and/or quality of information available to the naval architect is to be welcomed. However, it is considered that, for the present at any rate, there remains a need for basic text books. The two are complementary. A basic understanding of the subject is needed before information from the Internet can be used intelligently. **In** this edition we have maintained the objective of conveying principles and understanding to help student and practitioner in their work.

The authors have again been in a slight dilemma in deciding just how far to go in the subjects of each chapter. It is tempting to load the book with theories which have become more and more advanced. What has been done is to provide a glimpse into developments and advanced work with which students and practitioners must become familiar. Towards the end of each chapter a section giving an outline of how matters are developing has been included which will help to lead students, with the aid of the Internet, to all relevant references. Some web site addresses have also been given.

It must be appreciated that standards change continually, as do the titles of organizations. Every attempt has been made to include the latest at the time of writing but the reader should always check source documents to see whether they still apply in detail at the time they are to be used. What the reader can rely on is that the principles underlying such standards will still be relevant.

2001 K J R E C T

Acknowledgements

The authors have deliberately refrained from quoting a large number of references. However, we wish to acknowledge the contributions of many practitioners and research workers to our understanding of naval architecture, upon whose work we have drawn. Many will be well known to any student of engineering. Those early engineers in the field who set the fundamentals of the subject, such as Bernoulli, Reynolds, the Froudes, Taylor, Timoshenko, Southwell and Simpson, are mentioned in the text because their names are synonymous with sections of naval architecture.

Others have developed our understanding, with more precise and comprehensive methods and theories as technology has advanced and the ability to carry out complex computations improved. Some notable workers are not quoted as their work has been too advanced for a book of this nature.

We are indebted to a number of organizations which have allowed us to draw upon their publications, transactions, journals and conference proceedings. This has enabled us to illustrate and quantify some of the phenomena discussed. These include the learned societies, such as the Royal Institution of Naval Architects and the Society of Naval Architects and Marine Engineers; research establishments, such as the Defence Evaluation and Research Agency, the Taylor Model Basin, British Maritime Technology and MARIN; the classification societies; and Government departments such as the Ministry of Defence and the Department of the Environment, Transport and the Regions; publications such as those of the International Maritime Organisation and the International Towing Tank Conferences.

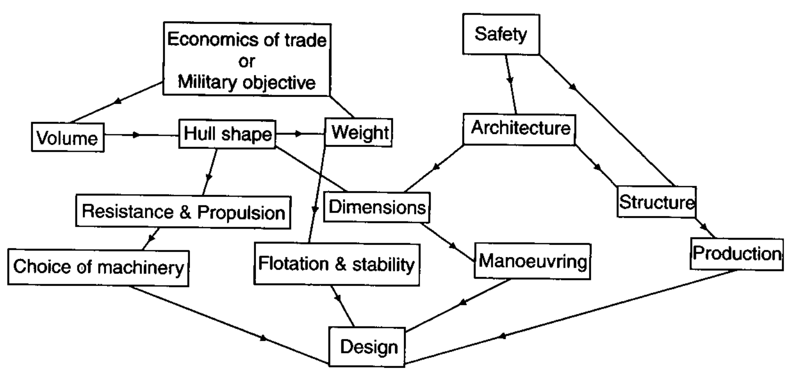
Introduction

In their young days the authors performed the calculations outlined in this work manually aided only by slide rule and, luxuriously, calculators. The arduous nature of such endeavours detracted from the creative aspects and affected the enjoyment of designing ships. Today, while it would be possible, such prolonged calculation is unthinkable because the chores have been removed to the care of the computer, which has greatly enriched the design process by giving time for reflection, trial and innovation, allowing the effects of changes to be examined rapidly.

It would be equally nonsensical to plunge into computer manipulation without knowledge of the basic theories, their strengths and limitations, which allow judgement to be quantified and interactions to be acknowledged. A simple change in dimensions of an embryo ship, for example, will affect flotation, stability, protection, powering, strength, manoeuvring and many sub-systems within, that affect a land architect to much less an extent. For this reason, the authors have decided to leave computer system design to those qualified to provide such important tools and to ensure that the student recognizes the fundamental theory on which they are based so that he or she may understand what consequences the designer's actions will have, as they feel their way towards the best solution to an owner's economic aims or military demands.

Manipulation of the elements of a ship is greatly strengthened by such a 'feel' and experience provided by personal involvement. Virtually every ship's characteristic and system affects every other ship so that some form of holistic approach is essential.

A crude representation of the process of creating a ship is outlined in the figure.



This is, of course, only a beginning. Moreover, the arrows should really be pointing in both directions; for example, the choice of machinery to serve speed and endurance reflects back on the volume required and the architecture of the ship which affects safety and structure. And so on. Quantification of the changes is effected by the choice of suitable computer programs. Downstream of this process lies design of systems to support each function but this, for the moment, is enough to distinguish between knowledge and application.

The authors have had to limit their work to presentation of the fundamentals of naval architecture and would expect readers to adopt whatever computer systems are available to them with a sound knowledge of their basis and frailties. The sequence of the chapters which follow has been chosen to build knowledge in a logical progression. The first thirteen chapters address elements of ship response to the environments likely to be met; Chapter 14 adds some of the major systems needed within the ship and Chapter 15 provides some discipline to the design process. The final chapter reflects upon some particular ship types showing how the application of the same general principles can lead to significantly different responses to an owner's needs. A few worked examples are included to demonstrate that there is real purpose in understanding theoretical naval architecture.

The opportunity, afforded by the publication of a fifth edition, has been taken to extend the use of SI units throughout. The relationships between them and the old Imperial units, however, have been retained in the Introduction to assist those who have to deal with older ships whose particulars remain in the old units.

Care has been taken to avoid duplicating, as far as is possible, work that students will cover in other parts of the course; indeed, it is necessary to assume that knowledge in all subjects advances with progress through the book. The authors have tried to stimulate and hold the interest of students by careful arrangement of subject matter. Chapter 1 and the opening paragraphs of each succeeding chapter have been presented in somewhat lyrical terms in the hope that they convey to students some of the enthusiasm which the authors themselves feel for this fascinating subject. Naval architects need never fear that they will, during their careers, have to face the same problems, day after day. They will experience as wide a variety of sciences as are touched upon by any profession.

Before embarking on the book proper, it is necessary to comment on the units employed.

UNITS

In May 1965, the UK Government, in common with other governments, announced that Industry should move to the use of the metric system. At the same time, a rationalized set of metric units has been adopted internationally, following endorsement by the International Organization for Standardization using the Systeme International d'Unites (SI).

The adoption of SI units has been patchy in many countries while some have yet to change from their traditional positions.

In the following notes, the SI system of units is presented briefly; a fuller treatment appears in British Standard 5555. This book is written using SI units.

The SI is a rationalized selection of units in the metric system. It is a coherent system, i.e. the product or quotient of any two unit quantities in the system is the unit of the resultant quantity. The basic units are as follows:

Quantity	Name of unit	Unit symbol
Length	metre	m
Mass	kilogramme	kg
Time	second	s
Electric current	ampere	A
Thermodynamic temperature	kelvin	K
Luminous intensity	candela	cd
Amount of substance	mole	mol
Plane angle	radian	rad
Solid angle	steradian	sr

Special names have been adopted for some of the derived SI units and these are listed below together with their unit symbols:

Physical quantity	SI unit	Unit symbol
Force	newton	N = kgm/s^2
Work, energy	joule	J=Nm
Power	watt	W = J/s
Electric'charge	coulomb	C=As
Electric potential	volt	V=W/A
Electric capacitance	farad	F = As/V
Electric resistance	ohm	0= VIA
Frequency	hertz	Hz = s-I
Illuminance	lux	Ix = Im/m2
Self inductance	henry	H = Vs/A
Luminous flux	lumen	1m= cdsr
Pressure, stress	pascal	Pa = N/m^2
	megapascal	MPa = N/mm^2
Electrical conductance	siemens	S = 1/0
Magnetic flux	weber	Wb=Vs
Magnetic flux density	tesla	T = Wb/m^2

The following two tables list other derived units and the equivalent values of some UK units, respectively:

Physical quantity	SI unit	Unit symbol
Area	square metre	m^2
Volume	cubic metre	m^3
Density	kilogramme per cubic metre	kg/m^3
Velocity	metre per second	m/s
Angular velocity	radian per second	radls
Acceleration	metre per second squared	m/s^2
Angular acceleration	radian per second squared	rad/s^2
Pressure, stress	newton per square metre	N/m^2
Surface tension	newton per metre	N/m
Dynamic viscosity	newton second per metre squared	$N\,s/m^2$
Kinematic viscosity	metre squared per second	m^2/s
Thermal conductivity	watt per metre kelvin	W/(mK)

Quantity	Imperial unit	Equivalent SI units
Length	1 yd	0.9144 m
	1 ft	0.3048 m
	1 in	0.0254 m
	1 mile	1609.344 m
	1 nautical mile (UK)	1853.18 m
	1 nautical mile (International)	1852 m
Area	1 in^2	$645.16 \times 10^{-6}\,m^2$
	1 ft^2	0.092903 m^2
	1 yd^2	0.836127 m^2
	1 $mile^2$	$2.58999 \times 10^6\,m^2$
Volume	1 in^3	$16.3871 \times 10^{-6}\,m^3$
	1 ft^3	0.0283168 m^3
	1 UK gal	0.004546092 m^3 = 4.546092 litres
Velocity	1 ft/s	0.3048 m/s
	1 mile/hr	0.44704 m/s; 1.60934 km/hr
	1 knot (UK)	0.51477 m/s; 1.85318 km/hr
	1 knot (International)	0.51444 m/s; 1.852 km/hr
Standard acceleration, g	32.174 ft/s^2	9.80665 m/s^2
Mass	1 lb	0.45359237 kg
	1 ton	1016.05 kg = 1.01605 tonnes
Mass density	1 lb/in^3	$27.6799 \times 10^3\,kg/m^3$
	1 lb/ft^3	16.0185 kg/m^3
Force	1 pdl	0.138255 N
	1 lbf	4.44822 N
Pressure	1 lbf/in^2	6894.76 N/m^2 0.0689476 bars
Stress	1 $tonf/in^2$	$15.4443 \times 10^6\,N/m^2$ 15.443 MPa or N/mm^2
Energy	1 ft pdl	0.0421401 J
	1 ft lbf	1.35582 J
	1 cal	4.1868 J
	1 Btu	1055.06 J
Power	1 hp	745.700 W
Temperature	1 Rankine unit	5/9 Kelvin unit
	1 Fahrenheit unit	5/9 Celsius unit

Note that, while multiples of the denominators are preferred, the engineering industry has generally adopted N/mm^2 for stress instead of MN/m^2 which has, of course, the same numerical value and are the same as MPa.

Prefixes to denote multiples and sub-multiples to be affixed to the names of units are:

Factor by which the unit is multiplied	Prefix	Symbol
$1\,000\,000\,000\,000 = 10^{12}$	tera	T
$1\,000\,000\,000 = 10^{9}$	giga	G
$1\,000\,000 = 10^{6}$	mega	M
$1\,000 = 10^{3}$	kilo	k
$100 = 10^{2}$	hecto	h
$10 = 10^{1}$	deca	da
$0.1 = 10^{-1}$	deci	d
$0.01 = 10^{-2}$	centi	c
$0.001 = 10^{-3}$	milli	m
$0.000\,001 = 10^{-6}$	micro	μ
$0.000\,000\,001 = 10^{-9}$	nano	n
$0.000\,000\,000\,001 = 10^{-12}$	pico	p
$0.000\,000\,000\,000\,001 = 10^{-15}$	femto	f
$0.000\,000\,000\,000\,000\,001 = 10^{-18}$	atto	a

We list, finally, some preferred metric values (values preferred for density of fresh and salt water are based on a temperature of 15 °C (59 °F)).

Item	Accepted Imperial figure	Direct metric equivalent	Preferred SI value
Gravity, g	$32.17\ \mathrm{ft/s^2}$	$9.80665\ \mathrm{m/s^2}$	$9.807\ \mathrm{m/s^2}$
Mass density salt water	$64\ \mathrm{lb/ft^3}$ $35\ \mathrm{ft^3/ton}$	$1.0252\ \mathrm{tonne/m^3}$ $0.9754\ \mathrm{m^3/tonne}$	$1.025\ \mathrm{tonne/m^3}$ $0.975\ \mathrm{m^3/tonne}$
Mass density fresh water	$62.2\ \mathrm{lb/ft^3}$ $36\ \mathrm{ft^3/ton}$	$0.9964\ \mathrm{tonne/m^3}$ $1.0033\ \mathrm{m^3/tonne}$	$1.0\ \mathrm{tonne/m^3}$ $1.0\ \mathrm{m^3/tonne}$
Young's modulus E (steel)	$13{,}500\ \mathrm{tonf/in^2}$	$2.0855 \times 10^7\ \mathrm{N/cm^2}$	$209\ \mathrm{GN/m^2}$ or GPa
Atmospheric pressure	$14.7\ \mathrm{lbf/in^2}$	$101{,}353\ \mathrm{N/m^2}$ $10.1353\ \mathrm{N/cm^2}$	$10^5\ \mathrm{N/m^2}$ or Pa or 1.0 bar
TPI (salt water)	$\frac{A_w}{420}$ tonf/in	$1.025\,A_w$(tonnef/m)	$1.025\,A_w$tonnef/m
	$A_w(\mathrm{ft^2})$	$A_w(\mathrm{m^2})$	
NPC		$100.52\,A_{w(N/cm)}$	
NPM	$A_w(\mathrm{m^2})$	$10{,}052\,A_w(\mathrm{N/m})$	$10^4\,A_w(\mathrm{N/m})$
MCT 1″ (salt water) (Units of tonf and feet)	$\frac{\Delta \overline{GM}_L}{12L} \frac{\text{tonf ft}}{\text{in}}$		
One metre trim moment, (Δ in MN or $\frac{\text{tonnef m}}{\text{m}}$, Δ in tonnef)		$\frac{\Delta \overline{GM}_L}{L}\left(\frac{\text{MN m}}{\text{m}}\right)$	$\frac{\Delta \overline{GM}_L}{L}\left(\frac{\text{MN m}}{\text{m}}\right)$
Force displacement Δ	1 tonf	1.01605 tonnef 9964.02N	1.016 tonnef 9964 N
Mass displacement Σ	1 ton	1.01605 tonne	1.016 tonne
Weight density: Salt water Fresh water			 $0.01\ \mathrm{MN/m^3}$ $0.0098\ \mathrm{MN/m^3}$
Specific volume: Salt water Fresh water			 $99.5\ \mathrm{m^3/MN}$ $102.0\ \mathrm{m^3/MN}$

Of particular significance to the naval architect are the units used for displacement, density and stress. The force displacement .6., under the SI scheme must be expressed in terms of newtons. In practice the meganewton (MN) is a more convenient unit and I MN is approximately equivalent to 100 tonf (100.44 more exactly). The authors have additionally introduced the tonnef (and, correspondingly, the tonne for mass measurement) as explained more fully in Chapter 3.

EXAMPLES

A number of worked examples has been included in the text of most chapters to illustrate the application of the principles enunciated therein. Some are relatively short but others involve lengthy computations. They have been deliberately chosen to help educate the student in the subject of naval architecture, and the authors have not been unduly influenced by the thought that examination questions often involve about 30 minutes' work.

In the problems set at the end of each chapter, the aim has been adequately to cover the subject matter, avoiding, as far as possible, examples involving mere arithmetic substitution in standard formulae.

REFERENCES AND THE INTERNET

References for each chapter are given in a Bibliography at the end of the book with a list of works for general reading. Because a lot of useful information is to be found these days on the Internet, some relevant web sites are quoted at the end of the Bibliography.

Symbols and nomenclature

GENERAL

a	linear acceleration
A	area in general
B	breadth in general
D,d	diameter in general
E	energy in general
F	force in general
g	acceleration due to gravity
h	depth or pressure head in general
hw, (w	height of wave, crest to trough
H	total head, Bernoulli
L	length in general
Lw,'\	wave-length
m	mass
n	rate of revolution
p	pressure intensity
Pv	vapour pressure of water
Poo	ambient pressure at infinity
P	power in general
q	stagnation pressure
Q	rate of flow
Y, R	radius in general
s	length along path
t	time in general
to	temperature in general
T	period of time for a complete cycle
u	reciprocal weight density, specific volume,
u, v, w	velocity components in direction of x-, y-, z-axes
U, V	linear velocity
w	weight density
W	weight in general
x,y, z	body axes and Cartesian co-ordinates Right-hand system fixed in the body, z-axis vertically down, x-axis forward. Origin at c.g.
xo, Yo, Zo	fixed axes Right-hand orthogonal system nominally fixed in space, zo-axis vertically down, xo-axis in the general direction of the initial motion.
a	angular acceleration
'Y	specific gravity
r	circulation
8	thickness of boundary layer in general
()	angle of pitch
J.l	coefficient of dynamic viscosity
,I	coefficient of kinematic viscosity
p	mass density
¢>	angle of roll, heel or list
X	angle of yaw
w	angular velocity or circular frequency
V'	volume in general

GEOMETRY OF SHIP

AM	midship section area
Aw	waterplane area
Ax	maximum transverse section area
B	beam or moulded breadth
-BM-	metacentre above centre of buoyancy
Cn	block coefficient
CM	midship section coefficient
Cp	longitudinal prismatic coefficient
Cyp	vertical prismatic coefficient
Cwp	coefficient of fineness of waterplane
D	depth of ship
F	freeboard
-OM-	transverse metacentric height
-GM-L	longitudinal metacentric height
h	longitudinal moment of inertia of waterplane about C
Ip	polar moment of inertia
h	transverse moment of inertia
L	length of ship--generally between perps
LoA	length overall
Lpp	length between perps
LWL	length of waterline in general
S	wetted surface
T	draught
d	displacement force
,\	scale ratio-ship/model dimension
V'	displacement volume
r;	displacement mass

PROPELLER GEOMETRY

AD	developed blade area
AE	expanded area
Ao	disc area
Ap	projected blade area
b	span of aerofoil or hydrofoil
c	chord length
d	boss or hub diameter
D	diameter of propeller
1M	camber
P	propeller pitch in general
R	propeller radius
t	thickness of aerofoil
Z	number of blades of propeller
a	angle of attack
¢>	pitch angle of screw propeller

RESIST ANCE AND PROPULSION

a	resistance augment fraction
CD	drag caeff.
CL	lift caeff.
CT	specific total resistance caetf.
Cw	specific wave-making resistance caetf.
D	drag force
Fn	Froude number
I	idle resistance
J	advance number of propeller
KQ	torque coeff.
KT	thrust caetf.
L	lift force

P_D	delivered power at propeller
PE	effective power
PI	indicated power
Ps	shaft power
PT	thrust power
Q	torque
R	resistance in general
Rn	Reynolds number
R_F	frictional resistance
RR	residuary resistance
RT	total resistance
Rw	wave-making resistance
SA	apparent slip ratio
t	thrust deduction fraction
T	thrust
U	velocity of a fluid
Uoo	velocity of an undisturbed flow
V	speed of ship
VA	speed of advance of propeller
w	Taylor wake fraction in general
wF	Proude wake fraction
Wn	Weber number
fJ	appendage scale effect factor
fJ	advance angle of a propeller blade section
8	Taylor's advance coeff.
'f/	efficiency in general
'f/B	propeller efficiency behind ship
'f/D	quasi propulsive coefficient
'f/H	hull eff.
'f/o	propeller eff. in open water
'f/R	relative rotative efficiency
a	cavitation number

SEAKEEPING

c	wave velocity
f	frequency
fE	frequency of encounter
Ixx, Iyy, /zz	real moments of inertia
Ixy, Ix.z, Iyz	real products of inertia
k	radius of gyration
mn	spectrum moment where n is an integer
ML	horizontal wave bending moment
MT	torsional wave bending moment
Mv	vertical wave bending moment
S	relative vertical motion of bow with respect to wave surface
S«w), So(w), etc.	one-dimensional spectral density
S«w,f.1), So(w,f.1), etc.	two-dimensional spectral density
T	wave period
TE	period of encounter
Tz	natural period in smooth water for heaving
To	natural period in smooth water for pitching
T¢	natural period in smooth water for rolling
Yo«w)	response amplitude operator-pitch
Y¢«w)	response amplitude operator-roll
Yx«w)	response amplitude operator-yaw
fJ	leeway or drift angle
8R	rudder angle
c	phase angle between any two harmonic motions
(	instantaneous wave elevation

(A	wave amplitude
(w	wave height, crest to trough
()	pitch angle
()A	pitch amplitude
r;,	wave number
WE	frequency of encounter
A	tuning factor

MANOEUVRABILITY

Ae	area under cut-up
AR	area of rudder
b	span of hydrofoil
c	chord of hydrofoil
K,M,N	moment components on body relative to body axes
O	origin of body axes
p, q, r	components of angular velocity relative to bOdy axes
X, Y,Z	force components on body
a	angle of attack
fJ	drift angle
8R	rudder angle
χ	heading angle
We	steady rate of turn

STRENGTH

a	length of plate
b	breadth of plate
C	modulus of rigidity
c	linear strain
E	modulus of elasticity, Young's modulus
a	direct stress
ay	yield stress
g	acceleration due to gravity
I	planar second moment of area
J	polar second moment of area
j	stress concentration factor
k	radius of gyration
K	bulk modulus
l	length of member
L	length
M	bending moment
Mp	plastic moment
MAR	bending moment at A in member AB
m	mass
P	direct load, externally applied
PE	Euler collapse load
p	distributed direct load (area distribution), pressure
pi	distributed direct load (line distribution)
τ	shear stress
r	radius
S	internal shear force
S	distance along a curve
T	applied torque
t	thickness, time
U	strain energy
W	weight, external load
y	lever in bending
8	deflection, permanent set, elemental (when associated with element of breadth, e.g. 8b)
p	mass density
v	Poisson's ratio

NOTES

(a) A distance between two points is represented by a bar over the letters defining the two points, e.g. $\overline{GM}$ is the distance between G and M.

(b) When a quantity is to be expressed in non-dimensional form it is denoted by the use of the prime $'$. Unless otherwise specified, the non-dimensionalizing factor is a function of ρ, L and V, e.g. $m' = m/\frac{1}{2}\rho L^3$, $x' = x/\frac{1}{2}\rho L^2V^2$, $L' = L/\frac{1}{2}\rho L^3V^2$.

(c) A lower case subscript is used to denote the denominator of a partial derivative, e.g. $Y_u = \partial Y/\partial u$.

(d) For derivatives with respect to time the dot notation is used, e.g. $\dot{X} = dX/dt$.

1 Art or science?

Many thousands of years ago when people became intelligent and adventurous, those tribes who lived near the sea ventured on to it. They built rafts or hollowed out tree trunks and soon experienced the thrill of moving across the water, propelled by tide or wind or device. They experienced, too, the first sea disasters; their boats sank or broke, capsized or rotted and lives were lost. It was natural that those builders of boats which were adjudged more successful than others, received the acclaim of their fellows and were soon regarded as craftsmen. The intelligent craftsman observed perhaps, that capsizing was less frequent when using two trunks joined together or when an outrigger was fixed, or that it could be manoeuvred better with a rudder in a suitable position. The tools were trial and error and the stimulus was pride. He was the first naval architect.

The craftsmen's expertise developed as it was passed down the generations: the Greeks built their triremes and the Romans their galleys; the Vikings produced their beautiful craft to carry soldiers through heavy seas and on to the beaches. Several hundred years later, the craftsmen were designing and building great square rigged ships for trade and war and relying still on knowledge passed down through the generations and guarded by extreme secrecy. Still, they learned by trial and error because they had as yet no other tools and the disasters at sea persisted.

The need for a scientific approach must have been felt many hundreds of years before it was possible and it was not possible until relatively recently, despite the corner stone laid by Archimedes two thousand years ago. Until the middle of the eighteenth century the design and building of ships was wholly a craft and it was not, until the second half of the nineteenth century that science affected ships appreciably.

Isaac Newton and other great mathematicians of the seventeenth century laid the foundations for so many sciences and naval architecture was no exception. Without any doubt, however, the father of naval architecture was Pierre ~ Bouguer who published in 1746, *Traite du Navire*. **In** his book, Bouguer laid the foundations of many aspects of naval architecture which were developed later in the eighteenth century by Bernoulli, Euler and Santacilla. Lagrange and many others made contributions but the other outstanding figure of that century was the Swede, Frederick Chapman who pioneered work on ship resistance which led up to the great work of William Froude a hundred years later. A scientific approach to naval architecture was encouraged more on the continent than in Britain where it remained until the 1850s, a craft surrounded by pride and secrecy. On 19 May 1666, Samuel Pepys wrote of a Mr Deane:

> And then he fell to explain to me his manner of casting the draught of water which a ship will draw before-hand; which is a secret the King and

all admire in him, and he is the first that hath come to any certainty beforehand of foretelling the draught of water of a ship before she be launched.

The second half of the nineteenth century, however, produced Scott Russell, Rankine and Froude and the development of the science, and dissemination of knowledge in Britain was rapid.

NAVAL ARCHITECTURE TODAY

It would be quite wrong to say that the art and craft built up over many thousands of years has been wholly replaced by a science. The need for a scientific approach was felt, first, because the .art had proved inadequate to halt the disasters at sea or to guarantee the merchant that he or she was getting the best value for their money. Science has contributed much to alleviate these shortcomings but it continues to require the injection of experience of successful practice. Science produces the correct basis for comparison of ships but the exact value of the criteria which determine their performances must, as in other branches of engineering, continue to be dictated by previous successful practice, i.e. like most engineering, this is largely a comparative science. Where the scientific tool is less precise than one could wish, it must be heavily overlaid with craft; where a precise tool is developed, the craft must be discarded. Because complex problems encourage dogma, this has not always been easy.

The question, 'Art or Science?' is therefore loaded since it presupposes a choice. Naval architecture is art and science.

Basically, naval architecture is concerned with ship safety, ship performance and ship geometry, although these are not exclusive divisions.

With ship safety, the naval architect is concerned that the ship does not capsize in a seaway, or when damaged or even when maltreated. It is necessary to ensure that the ship is sufficiently strong so that it does not break up or fracture locally to let the water in. The crew must be assured that they have a good chance of survival if the ship does let water in through accident or enemy action.

The performance of the ship is dictated by the needs of trade or war. The required amount of cargo must be carried to the places which the owner specifies in the right condition and in the most economical manner; the warship must carry the maximum hitting power of the right sort and an efficient crew to the remote parts of the world. Size, tonnage, deadweight, endurance, speed, life, resistance, methods of propulsion, manoeuvrability and many other features must be matched to provide the right primary performance at the right cost. Over 90 per cent of the world's trade is still carried by sea.

Ship geometry concerns the correct interrelation of compartments which the architect of a house considers on a smaller scale. In an aircraft carrier, the naval architect has 2000 rooms to relate, one with another, and must provide up to fifty different piping and ducting systems to all parts of the ship. It is necessary to provide comfort for the crew and facilities to enable each member to perform his or her correct function. The ship must load and unload in harbour with the utmost speed and perhaps replenish at sea. The architecture of the ship must be such that it can be economically built, and aesthetically pleasing. The naval

architect is being held increasingly responsible for ensuring that the environmental impact of the product is minimal both in normal operation and following any foreseeable accident. There is a duty to the public at large for the safety of marine transport. In common with other professionals the naval architect is expected to abide by a stringent code of conduct.

It must be clear that naval architecture involves complex compromises of many of these features. The art is, perhaps, the blending in the right proportions. There can be few other pursuits which draw on such a variety of sciences to blend them into an acceptable whole. There can be few pursuits as fascinating.

SHIPS

Ships are designed to meet the requirements of owners or of war and their features are dictated by these requirements. The purpose of a merchant ship has been described as conveying passengers or cargo from one port to another in the most efficient manner. This was interpreted by the owners of *Cutty Sark* as the conveyance of relatively small quantities of tea in the shortest possible time, because this was what the tea market demanded at that time. The market might well have required twice the quantity of tea per voyage in a voyage of twice the length of time, when a fundamentally different design of ship would have resulted. The economics of any particular market have a profound effect on merchant ship design. Thus, the change in the oil market following the second world war resulted in the disappearance of the 12,000 tonf deadweight tankers and the appearance of the 400,000 tonf deadweight supertankers. The economics of the trading of the ship itself have an effect on its design; the desire, for example, for small tonnage (and therefore small harbour dues) with large cargo-carrying capacity brought about the three island and shelter deck ships where cargo could be stowed in spaces not counted towards the tonnage on which insurance rates and harbour dues were based. Such trends have not always been compatible with safety and requirements of safety now also vitally influence ship design. Specialized demands of trade have produced the great passenger liners and bulk carriers, the natural-gas carriers, the trawlers and many other interesting ships. Indeed, the trend is towards more and more specialization in merchant ship design (see Chapter 16).

Specialization applies equally to warships. Basically, the warship is designe3 to meet a country's defence policy. Because the design and building of warships takes several years, it is an advantage if a particular defence policy persists for at least ten years and the task of long term defence planning is an onerous and responsible one. The Defence Staff interprets the general Government policy into the needs for meeting particular threats in particular parts of the world and the scientists and technologists produce weapons for defensive and offensive use. The naval architect then designs ships to carry the weapons and the men to use them to the correct part of the world. Thus, nations like Britain and the USA with commitments the other side of the world, would be expected to expend more of the available space in their ships on facilities for getting the weapons and crew in a satisfactory state to a remote, perhaps hot, area than a nation which expects to make short harrying excursions from its home ports. It is important, therefore, to regard the ship as a complete weapon system and weapon and ship designers must work in the closest possible contact.

Nowhere, probably, was this more important than in the aircraft carrier. The type of aircraft carried so vitally affects an aircraft carrier that the ship is virtually designed around it; only by exceeding all the minimum demands of an aircraft and producing monster carriers, can any appreciable degree of flexibility be introduced. The guided missile destroyer results directly from the Defence Staff's assessment of likely enemy aircraft and guided weapons and their concept of how and where they would be used; submarine design is profoundly affected by diving depth and weapon systems which are determined by offensive and defensive considerations. The invention of the torpedo led to the motor torpedo boat which in turn led to the torpedo boat destroyer; the submarine, as an alternative carrier of the torpedo, led to the design of the anti-submarine frigate; the missile carrying nuclear submarine led to the hunter killer nuclear submarine. Thus, the particular demand of war, as is natural, produces a particular warship.

Particular demands of the sea have resulted in many other interesting and important ships: the self-righting lifeboats, surface effect vessels, container ships, cargo drones, hydrofoil craft and a host of others. All are governed by the basic rules and tools of naval architecture which this book seeks to explore. Precision in the use of these tools must continue to be inspired by knowledge of sea disasters; Liberty ships of the second world war, the loss of the *Royal George*, the loss of HMS *Captain*, and the loss of the *Vasa:*

> In 1628, the *Vasa* set out on a maiden voyage which lasted little more than two hours. She sank in good weather through capsizing while still in view of the people of Stockholm.

That disasters remain an influence upon design and operation has been tragically illustrated by the losses of the *Herald of Free Enterprise* and *Estonia* in the 1990s, while ferry losses continue at an alarming rate, often in nations which cannot afford the level of safety that they would like.

Authorities

CLASSIFICATION SOCIETIES

The authorities with the most profound influence on shipbuilding, merchant ship design and ship safety are the classification societies. Among the most dominant are Lloyd's Register of Shipping, det Norske Veritas, the American Bureau of Shipping, Bureau Veritas, Registro Italiano, Germanische Lloyd and Nippon Kaiji Kyokai. These meet to discuss standards under the auspices of the International Association of Classification Societies (lACS).

It is odd that the two most influential bodies in the shipbuilding and shipping industries should both derive their names from the same owner of a coffee shop, Edward Lloyd, at the end of the seventeenth century. Yet the two organizations

are entirely independent with quite separate histories. Lloyd's Insurance Corporation is concerned with mercantile and other insurance. Lloyd's Register of Shipping is concerned with the maintenance of proper technical standards in ship construction and the classification of ships, i.e. the record of all relevant technical details and the assurance that these meet the required standards. Vessels so registered with Lloyd's Register are said to be classed with the Society and may be awarded the classification + 100 AI. The cross denotes that the ship has been built under the supervision of surveyors from Lloyd's Register while 100 A shows that the vessel is built in accordance with the recommended standards. The final 1 indicates that the safety equipment, anchors and cables are as required. Other provisos to the classification are often added.

The maintenance of these standards is an important function of Lloyd's Register who require surveys of a specified thoroughness at stated intervals by the Society's surveyors. Failure to conform may result in removal of the ship from class and a consequent reduction in its value. The total impartiality of the Society is its great strength. It is also empowered to allot load line (see Chapter 5) certificates to ships, to ensure that they are adhered to and, as agents for certain foreign governments, to assess tonnage measurement and to ensure compliance with safety regulations. Over 1000 surveyors, scattered all over the world, carry out the required surveys, reporting to headquarters in London or other national centres where the classification of the ships are considered.

The standards to which the ships must be built and maintained are laid down in· the first of the two major publications of Lloyd's Register, *Rules and Regulations for the Classification of Ships.* This is issued annually and kept up to date to meet new demands. The other major publication is the *Register Book* in several volumes, which lists every known ship, whether classed with the Society or not, together with all of its important technical particulars. Separate books appear for the building and classification of yachts and there are many other publications to assist surveyors.

A number of classification societies, including LR, DNV and ABS, offer a service for classifying naval craft. Typically, such rules cover the ship and its systems including those that support the fighting capability of the craft. They do not cover the military sensors, weapons or command and control systems, fill' that the classification society concentrates on the ship as a fit for purpose weapon platform. The navy concerned acts as buyer and owner and can continue to specify any special military requirements. The technical requirements that make a ship fit for naval service, and which would be defined by the navy concerned, and make the ship different from a typical merchant ship, are:

1. different strength requirements to give a design able to accept damage;
2. weapon and sensor supports taking account of possible deformation of structure;
3. the ability to withstand enemy action, including appropriate strength, stability, shock and redundancy features;
4. allowance for the effects of impacting weapons.

The main elements of the class rules are common for naval and civilian craft. This ensures compliance with international regulations such as those of SOLAS and MARPOL. The warship is issued with the same range of technical and operational certificates as would be the case for a merchant ship.

One advantage is that the navy, through its chosen shipbuilder, has access to the world wide organization of the classification society in relation to material and equipment acceptance.

GOVERNMENT BODIES

The statutory authority in the United Kingdom for declaring the standards of safety for merchant ships, related to damage, collision, subdivision, life saving equipment, loading, stability, fire protection, navigation, carriage of dangerous goods, load line standards and many allied subjects, is the Department of the Environment, Transport and the Regions (DETR). This department is also the authority on tonnage measurement standards. It is responsible for seeing that safety standards, many of which are governed by international agreements, are maintained. Executive authority for marine safety was invested in 1994 in the Marine Safety Agency (MSA) created from the former Surveyor General's organization. Then in 1998, an executive agency of DETR, the Maritime and Coastguard Agency (MCA), was formed by merging the MSA and the Coastguard Agency (CA). The MCA provides three functions, survey and inspection of vessels, co-ordination of search and rescue, and marine pollution control response. DETR is responsible for enquiring into sea disasters through the Marine Accident Investigation Branch. Responsibility for the safety of offshore structures was transferred in 1994 from the Department of Eneray to the Health and Safety Executive following the *Piper Alpha* disaster.

Ship surveyors in the Marine Division and similar national authoritiea in other countries, like the US Coastguard, carry, thus, an enormous responlibility.

INTERNATIONAL BODIES

The International Maritime Organization (IMO), represents over 150 of the maritime nations of the world. The organization sponsors international action with a view to improving and standardizing questions relatinl to IlUp .flty and measurement. It sponsors the International Conventions on Safety ofLif.at Sea which agree to the application of new standards of safety. The IUD' orpnization sponsors, also, international conferences on the load line and Itlndudizing action on tonnage measurement and many other maritime probleml.

LEARNED SOCIETIES

The Institution of Naval Architects was formed in 1860 when In_l'IIt In the subject in Britain quickened and it has contributed much to the ㄊ of naval architecture. It became the Royal Institution of Naval A••••••• iD 1960. Abroad, among the many societies worthy of mention are the Alloctation Technique Maritime et Aeronautique in France, the Society of Nav.a Arc:Ihitects and Marine Engineers in the USA and the Society of Naval Archl", of Japan.

2 Some tools

No occupation can be properly developed without tools, whether it be gardening or naval architecture or astro-navigation. Many tools needed for the study of naval architecture are already to hand, provided by mathematics, applied mechanics and physics and it will be necessary to assume as the book progresses that knowledge in all allied subjects has also progressed. Knowledge, for example, of elementary differential and integral calculus is assumed to be developing concurrently with this chapter. Moreover, the tools need to be sharp; definitions must be precise, while the devices adopted from mathematics must be pointed in such a way as to bear directly on ship shapes and problems. As a means of examining this science, these are the tools.

It is convenient too, to adopt a terminology or particular language and a shorthand for many of the devices to be used. This chapter lays a firm foundation from which to build up the subject. Finally, there are short notes on statistics and approximate formulae.

Basic geometric concepts

The main parts of a typical ship together with the terms applied to the principal parts are illustrated in Fig. 2.1. Because, at first, they are of little interest or influence, superstructures and deckhouses are ignored and the hull of the ship is considered as a hollow body curved in all directions, surmounted by a watertight deck. Most ships have only one plane of symmetry, called the *middle line plane* which becomes the principal plane of reference. The shape of the ship cut by this plane is known as the sheer plan or profile. The *design waterplane* is a plane perpendicular to the middle line plane, chosen as a plane of reference at or near the horizontal: it mayor may not be parallel to the keel. Planes perpendicular to both the middle line plane and the design waterplane are

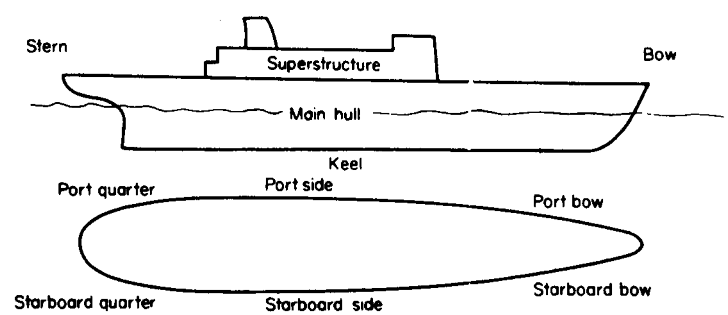


Fig. 2.1

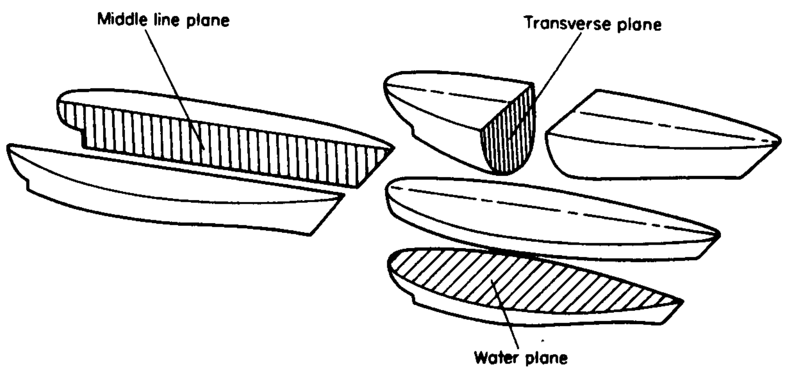


Fig. 2.2

called *transverse planes* and a transverse section of the ship does, normally, exhibit symmetry about the middle line. Planes at right angles to the middle line plane, and parallel to the design waterplane are called *waterplanes,* whether they are in the water or not, and they are usually symmetrical about the middle line. Waterplanes are not necessarily parallel to the keel. Thus, the curved shape of a ship is best conveyed to our minds by its sections cut by orthogonal planes. Figure 2.2 illustrates these planes.

Transverse sections laid one on top of the other form a *body plan* which, by convention, when the sections are symmetrical, shows only half sections, the forward half sections on the right-hand side of the middle line and the after half sections on the left. Halfwaterplanes placed one on top of the other form a *half breadth plan.* Waterplanes looked at edge on in the sheer or body plan are called *waterlines.* The sheer, the body plan and the half breadth collectively are called the *lines plan* or *sheer drawing* and the three constituents are clearly related (see Fig. 2.3).

It is convenient if the waterplanes and the transverse planes are equally spaced and datum points are needed to start from. That waterplane to which

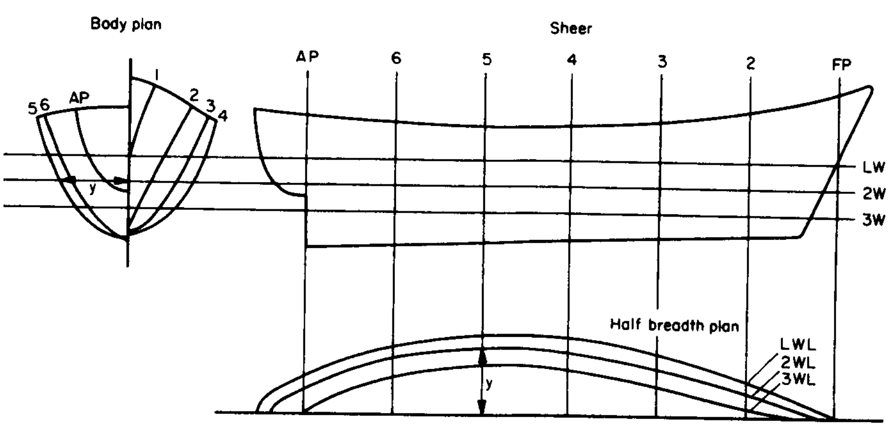


Fig. 2.3 Lines plan

the ship is being designed is called the *load waterplane* (LWP) or *design waterplane* and additional waterplanes for examining the ship's shape are drawn above it and below it, equally spaced, usually leaving an uneven slice near the keel which is best examined separately.

A reference point at the fore end of the ship is provided by the intersection of the load waterline and the stem contour and the line perpendicular to the LWP through this point is called the *fore perpendicular* (FP). It does not matter where the perpendiculars are, provided that they are precise and fixed for the ship's life, that they embrace most of the underwater portion and that there are no serious discontinuities between them. The *after perpendicular* (AP) is frequently taken through the axis of the rudder stock or the intersection of the LWL and transom profile. If the point is sharp enough, it is sometimes better taken at the after cut up or at a place in the vicinity where there is a discontinuity in the ship's shape. The distance between these two convenient reference lines is called the *length between perpendiculars* (LBP or L_{pp}). Two other lengths which will be referred to and which need no further explanation are the *length overall* and the *length on the waterline.*

The distance between perpendiculars is divided into a convenient number of equal spaces, often twenty, to give, including the FP and the AP, twenty-one evenly spaced ordinates. These ordinates are, of course, the edges of transverse planes looked at in the sheer or half breadth and have the shapes half shown in the body plan. Ordinates can also define any set of evenly spaced reference lines drawn on an irregular shape. The distance from the middle line plane along an ordinate in the half breadth is called an *offset* and this distance appears again in the body plan where it is viewed from a different direction. All such distances for all waterplanes and all ordinates form a *table of offsets* which defines the shape of the hull and from which a lines plan can be drawn. A simple table of offsets is used in Fig. 3.30 to calculate the geometric particulars of the form.

A reference plane is needed about mid-length of the ship and, not unnaturally, the transverse plane midway between the perpendiculars is chosen. It is called *amidships* or *midships* and the section of the ship by this plane is the midship section. It may not be the largest section and it should have no significance other than its position halfway between the perpendiculars. ItJr position is usually defined by the symbol :~

The shape, lines, offsets and dimensions of primary interest to the theory of naval architecture are those which are wetted by the sea and are called *displacement* lines, ordinates, offsets, etc. Unless otherwise stated, this book refers normally to displacement dimensions. Those which are of interest to the shipbuilder are the lines of the frames which differ from the displacement lines by the thickness of hull plating or more, according to how the ship is built. These are called *moulded* dimensions. Definitions of displacement dimensions are similar to those which follow but will differ by plating thicknesses.

The *moulded draught* is the perpendicular distance in a transverse plane from the top of the flat keel to the design waterline. If unspecified, it refers to amidships. The draught amidships is the mean draught unless the mean draught is referred directly to draught mark readings.

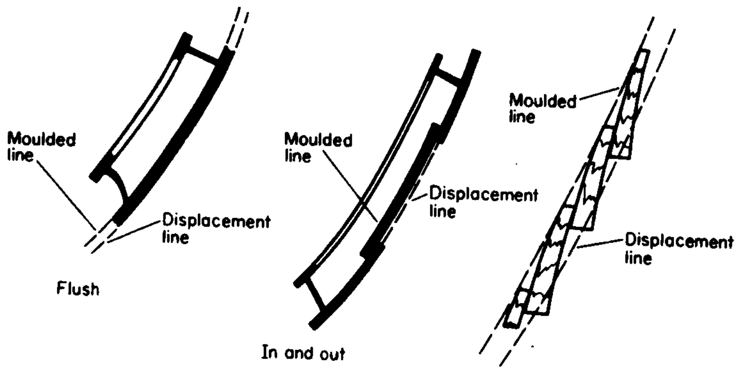


Fig. 2.4 Moulded and displacement lines

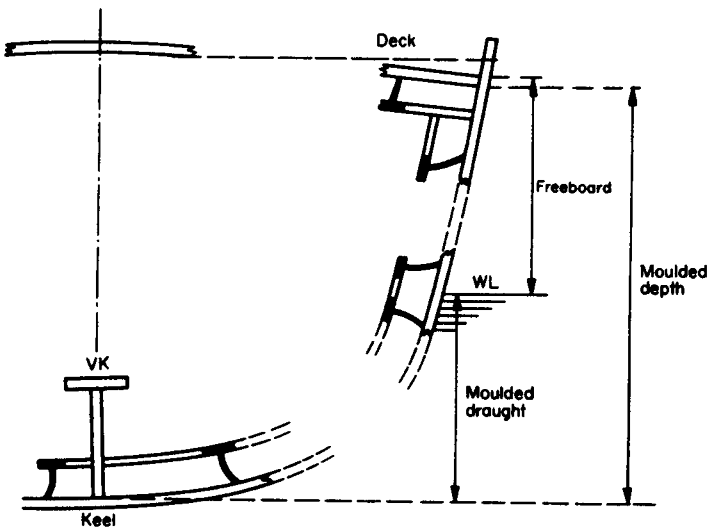


Fig. 2.5

The *moulded depth* is the perpendicular distance in a transverse plane from the top of the flat keel to the underside of deck plating at the ship's side. If unspecified, it refers to this dimension amidships.

Freeboard is the difference between the depth at side and the draught. It is the perpendicular distance in a transverse plane from the waterline to the upperside of the deck plating at side.

The *moulded breadth extreme* is the maximum horizontal breadth of any frame section. The terms breadth and beam are synonymous.

Certain other geometric concepts of varying precision will be found useful in defining the shape of the hull. *Rise of floor* is the distance above the keel that a tangent to the bottom at or near the keel cuts the line of maximum beam amidships (see Fig. 2.6).

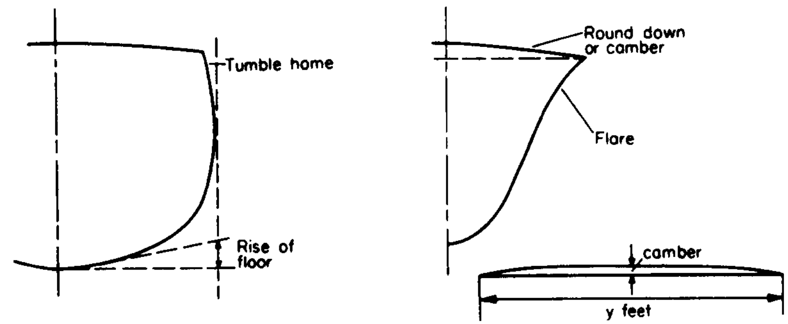


Fig. 2.6

Tumble home is the tendency of a section to fall in towards the middle line plane from the vertical as it approaches the deck edge. The opposite tendency is called *flare* (see Fig. 2.6).

Deck camber or *round down* is the curve applied to a deck transversely. It is normally concave downwards, a parabolic or circular curve, and measured as x centimetres in y metres.

Sheer is the tendency of a deck to rise above the horizontal in profile.

Rake is the departure from the vertical of any conspicuous line in profile such as a funnel, mast, stem contour, superstructure, etc. (Fig. 2.7).

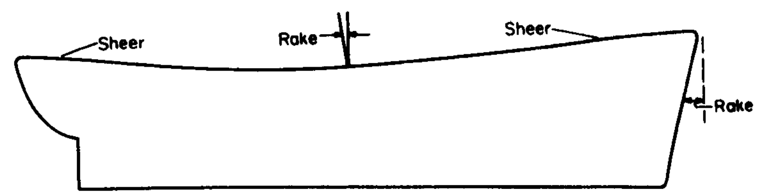


Fig. 2.7

There are special words applied to the angular movements of the whole ship from equilibrium conditions. Angular bodily movement from the vertical in a transverse plane is called *heel*. Angular bodily movement in the middle lY4\ plane is called *trim*. Angular disturbance from the mean course of a ship in the horizontal plane is called *yaw* or *drift*. Note that these are all angles and not rates, which are considered in later chapters.

There are two curves which can be derived from the offsets which define the shape of the hull by areas instead of distances which will later prove of great value. By erecting a height proportional to the area of each ordinate up to the LWP at each ordinate station on a horizontal axis, a curve is obtained known as the *curve of areas*. Figure 2.8 shows such a curve with number 4 ordinate, taken as an example. The height of the curve of areas at number 4 ordinate represents the area of number 4 ordinate section; the height at number 5 is proportional to the area of number 5 section and so on. A second type of area curve can be obtained by examining each ordinate section. Figure 2.8 again

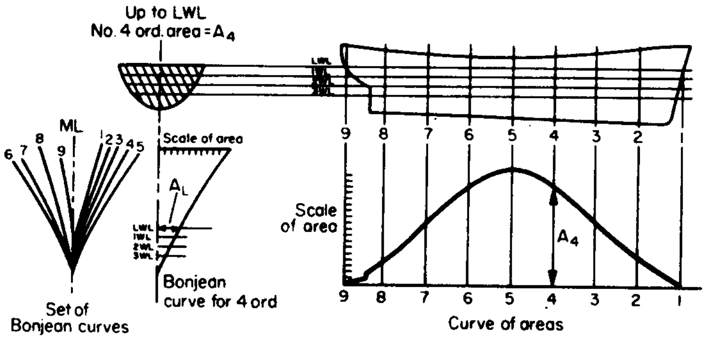


Fig. 2.8

takes 4 ordinate section as an example. Plotting outwards from a vertical axis, distances corresponding to the areas of a section up to each waterline, a curve known as a *Bonjean curve* is obtained. Thus, the distance outwards at the LWL is proportional to the area of the section up to the LWL, the distance outwards at 1WL is proportional to the area of section up to 1WL and so on. Clearly, a Bonjean curve can be drawn for each section and a set produced.

The *volume of displacement*, ∇, is the total volume of fluid displaced by the ship. It is best conceived by imagining the fluid to be wax and the ship removed from it; it is then the volume of the impression left by the hull. For convenience of calculation, it is the addition of the volumes of the main body and appendages such as the slices at the keel, abaft the AP, rudder, bilge keels, propellers, etc., with subtractions for condensor inlets and other holes.

Finally, in the definition of hull geometry there are certain coefficients which will later prove of value as guides to the fatness or slimness of the hull.

The *coefficient of fineness of waterplane*, C_{WP}, is the ratio of the area of the waterplane to the area of its circumscribing rectangle. It varies from about 0.70 for ships with unusually fine ends to about 0.90 for ships with much parallel middle body.

$$C_{WP} = \frac{A_W}{L_{WL}B}$$

The *midship section coefficient*, C_M, is the ratio of the midship section area to the area of a rectangle whose sides are equal to the draught and the breadth extreme amidships. Its value usually exceeds 0.85 for ships other than yachts.

$$C_M = \frac{A_M}{BT}$$

The *block coefficient*, C_B, is the ratio of the volume of displacement to the volume of a rectangular block whose sides are equal to the breadth extreme, the mean draught and the length between perpendiculars.

$$C_B = \frac{\nabla}{BTL_{PP}}$$

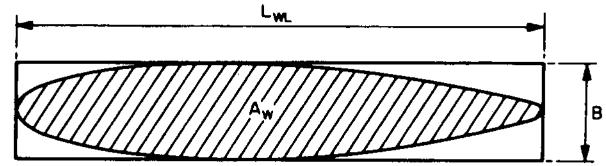


Fig. 2.9 Waterplane coefficient

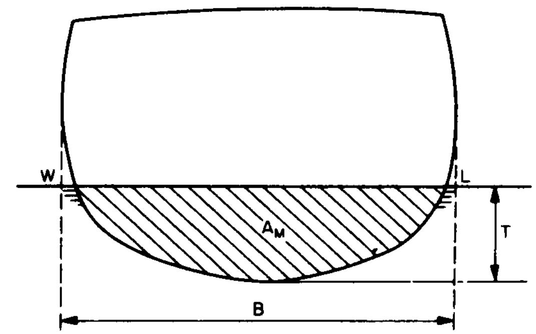


Fig. 2.10 Midship coefficient

Mean values of block coefficient might be 0.88 for a large oil tanker, 0.60 for an aircraft carrier and 0.50 for a yacht form.

The *longitudinal prismatic coefficient*, C_P, or simply *prismatic coefficient* is the ratio of the volume of displacement to the volume of a prism having a length equal to the length between perpendiculars and a cross-sectional area equal to the midship sectional area. Expected values generally exceed 0.55.

$$C_P = \frac{\nabla}{A_M L_{PP}}$$

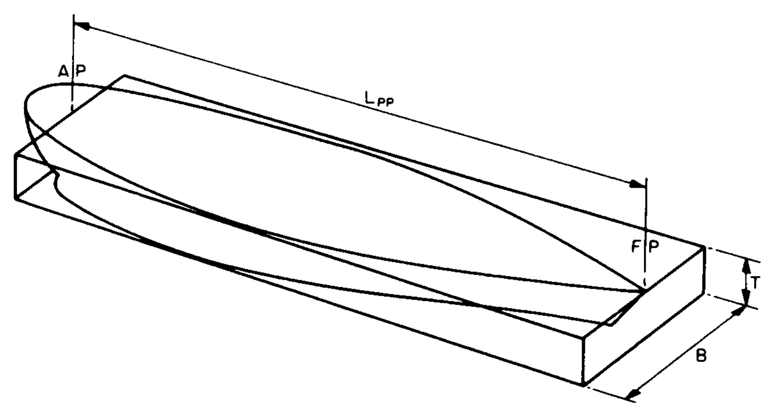


Fig. 2.11 Block coefficient

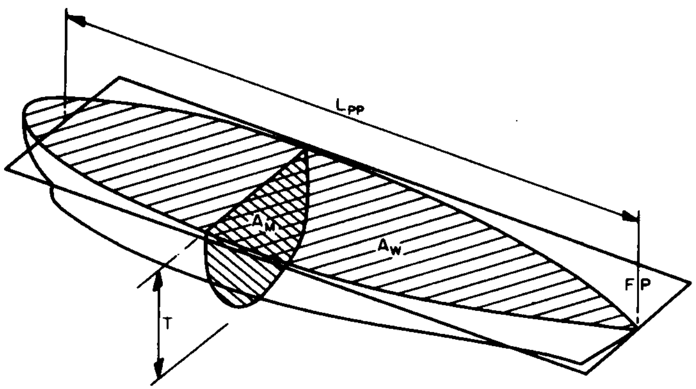


Fig. 2.12 Longitudinal prismatic coefficient

The *vertical prismatic coefficient*, C_{VP} is the ratio of the volume of displacement to the volume of a prism having a length equal to the draught and a cross-sectional area equal to the waterplane area.

$$C_{VP} = \frac{\nabla}{A_W T}$$

Before leaving these coefficients for the time being, it should be observed that the definitions above have used displacement and not moulded dimensions because it is generally in the very early stages of design that these are of interest. Practice in this respect varies a good deal. Where the difference is significant, as for example in the structural design of tankers by Lloyd's Rules, care should be taken to check the definition in use. It should also be noted that the values of the various coefficients depend on the positions adopted for the perpendiculars.

Properties of irregular shapes

Now that the geometry of the ship has been defined, it is necessary to anticipate what properties of these shapes are going to be useful and find out how to calculate them.

PLANE SHAPES

Waterplanes, transverse sections, flat decks, bulkheads, the curve of areas and expansions of curved surfaces are some of the plane shapes whose properties are of interest. The area of a surface in the plane of Oxy defined in Cartesian co-ordinates, is

$$A = \int y\,\mathrm{d}x$$

in which all strips of length y and width 8x are summed over the total extent of x. Because y is rarely, with ship shapes, a precise mathematical function of x the integration must be carried out by an approximate method which will presently be deduced.

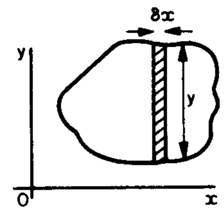


Fig. 2.13

There are first moments of area about each axis. (For the figures shown in Fig. 2.14, x_1 and y_1 are lengths and x and y are co-ordinates.)

$$M_{yy} = \int xy_1\,\mathrm{d}x \quad \text{and} \quad M_{xx} = \int x_1 y\,\mathrm{d}y$$

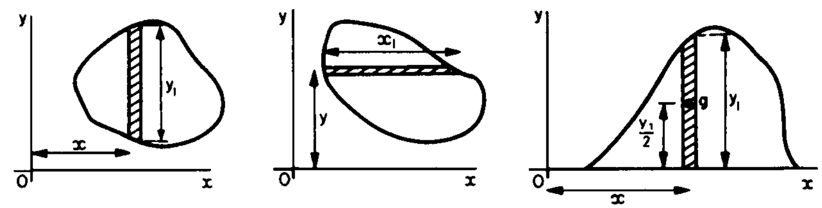


Fig. 2.14

Dividing each expression by the area gives the co-ordinates of the centre of area, $(\bar{x}, \bar{y})$:

$$\bar{x} = \frac{1}{A}\int xy_1\,\mathrm{d}x \quad \text{and} \quad \bar{y} = \frac{1}{A}\int x_1 y\,\mathrm{d}y$$

For the particular case of a figure bounded on one edge by the x-axis

$$M_y^* = \int \frac{1}{2}y^2\,\mathrm{d}x \quad \text{and} \quad \bar{y} = \frac{1}{2A}\int y_1^2\,\mathrm{d}x$$

For a plane figure placed symmetrically about the x-axis such as a waterplane, $M_{xx} = \int x_1 y\,\mathrm{d}y = 0$ and the distance of the centre of area, called in the particular case of a waterplane, the *centre of flotation* (CF), from the y-axis is given by

$$\bar{x} = \frac{M_{yy}}{A} = \frac{\int xy_1\,\mathrm{d}x}{\int y_1\,\mathrm{d}x}$$

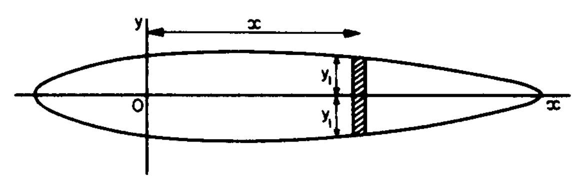


Fig. 2.15

*Note that $M_y \equiv M_{xx}$.

It is convenient to examine such a symmetrical figure in relation to the second moment of area, since it is normally possible to simplify work by choosing one symmetrical axis for ship shapes. The second moments of area or moments of inertia about the two axes for the waterplane shown in Fig. 2.15 are given by

$$I_T = \frac{1}{3}\int y_1^3\,dx \quad \text{about O}x \text{ for each half}$$

$$I_{yy} = \int x^2 y_1\,dx \quad \text{about O}y \text{ for each half}$$

The parallel axis theorem shows that the second moment of area of a plane figure about any axis, Q, of a set of parallel axes is least when that axis passes through the centre of area and that the second moment of area about any other axis, R, parallel to Q and at a distance h from it is given by (Fig. 2.16)

$$I_R = I_Q + Ah^2$$

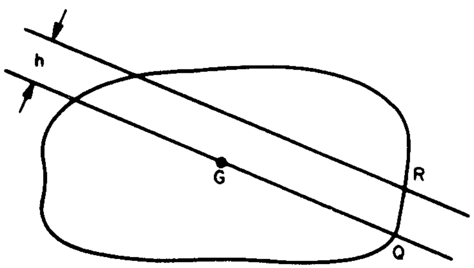


Fig. 2.16

From this, it follows that the least longitudinal second moment of area of a waterplane is that about an axis through the centre of flotation and given by (Fig. 2.17)

$$I_L = I_{yy} - A\bar{x}^2$$

i.e.

$$I_L = \int x^2 y_1\,dx - A\bar{x}^2$$

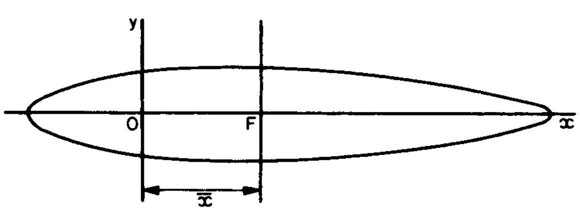


Fig. 2.17

THREE-DIMENSIONAL SHAPES

It has already been shown how to represent the three-dimensional shape of the ship by a plane shape, the curve of areas, by representing each section area by a length (Fig. 2.8). This is one convenient way to represent the three-dimensional shape of the main underwater form (less appendages). The volume of displacement is given by

$$\nabla = \int_{x_1}^{x_2} A\,dx$$

i.e. it is the sum of all such slices of cross-sectional area A over the total extent of x (Fig. 2.18).

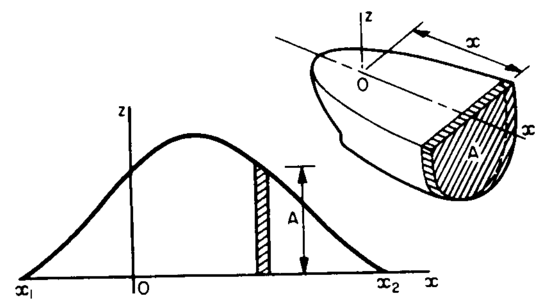


Fig. 2.18

The shape of the ship can equally be represented by a curve of waterplane areas on a vertical axis (Fig. 2.19), the breadth of the curve at any height, z, above the keel representing the area of the waterplane at that draught. The volume of displacement is again the sum of all such slices of cross-sectional area A_w, over the total extent of z from zero to draught T,

$$\nabla = \int_0^T A_w\,dz$$

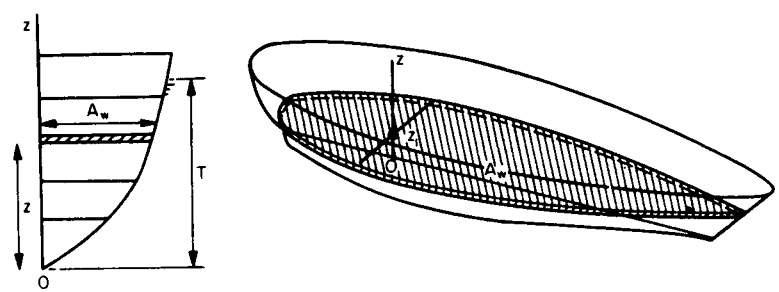


Fig. 2.19

The first moments of volume in the longitudinal direction about Oz and in the vertical direction about the keel are given by

$$M_L = \int Ax\,dx \quad \text{and} \quad M_V = \int_0^T A_W z\,dz$$

Dividing by the volume in each case gives the co-ordinates of the centre of volume. The centre of volume of fluid displaced by a ship is known as the *centre of buoyancy*; its projections in the plan and in section are known as the longitudinal centre of buoyancy (LCB) and the vertical centre of buoyancy (VCB)

$$\text{LCB from O}y = \frac{1}{\nabla}\int Ax\,dx$$

$$\text{VCB above keel} = \frac{1}{\nabla}\int A_W z\,dz$$

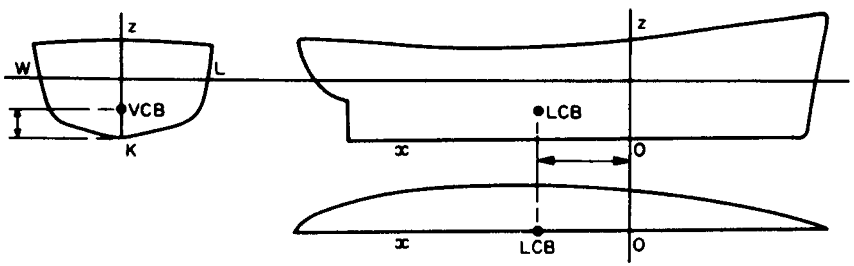


Fig. 2.20 Centre of buoyancy projections

Should the ship not be symmetrical below the waterline, the centre of buoyancy will not lie in the middle line plane. Its projection in plan may then be referred to as the transverse centre of buoyancy (TCB). Had z been taken as the distance below the waterline, the second expression would, of course, represent the position of the VCB below the waterline. Defining it formally, the *centre of buoyancy* of a floating body is the centre of volume of the displaced fluid in which the body is floating. The first moment of volume about the centre of volume is zero.

The *weight* of a body is the total of the weights of all of its constituent parts. First moments of the weights about particular axes divided by the total weight, define the co-ordinates of the centre of weight or centre of gravity (CG) relative to those axes. Projections of the centre of gravity of a ship in plan and in section are known as the longitudinal centre of gravity (LCG) and vertical centre of gravity (VCG) and transverse centre of gravity (TCG).

$$\text{LCG from O}y = \frac{1}{W}\int x\,dW$$

$$\text{VCG above keel} = \frac{1}{W}\int z\,dW$$

$$\text{TCG from middle line plane} = \frac{1}{W}\int y\,dW$$

Defining it formally, the *centre of gravity* of a body is that point through which, for statical considerations, the whole weight of the body may be assumed to act. The first moment of weight about the centre of gravity is zero.

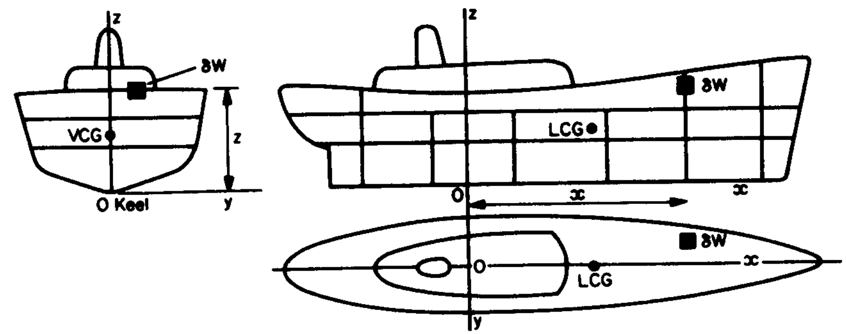


Fig. 2.21 Centre of gravity projections

METACENTRE

Consider any body floating upright and freely at waterline WL, whose centre of buoyancy is at B. Let the body now be rotated through a small angle in the plane of the paper without altering the volume of displacement (it is more convenient to draw if the body is assumed fixed and the waterline rotated to W_1L_1). The centre of buoyancy for this new immersed shape is at B_1. Lines through B and B_1

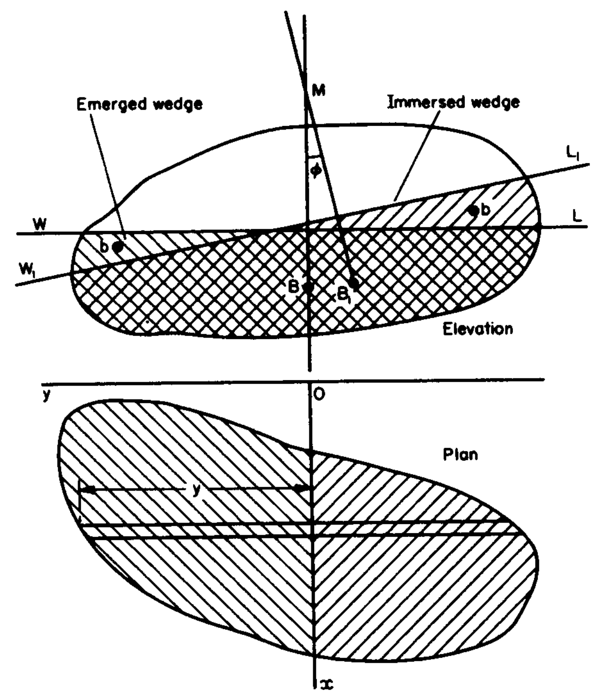


Fig. 2.22

normal to their respective waterlines intersect at M which is known as the *metacentre* since it appears as if the body rotates about it for small angles of rotation. The *metacentre* is the point of intersection of the normal to a slightly inclined waterplane of a body, rotated without change of displacement, through the centre of buoyancy pertaining to that waterplane and the vertical plane through the centre of buoyancy pertaining to the upright condition. The term metacentre is reserved for small inclinations from an upright condition. The point of intersection of normals through the centres of buoyancy pertaining to successive waterplanes of a body rotated infinitesimally at any angle of inclination without change of displacement, is called the *pro-metacentre.*

If the body is rotated without change of displacement, the volume of the immersed wedge must be equal to the volume of the emerged wedge. Furthermore, the transfer of this volume from the emerged to the immersed side must be responsible for the movement of the centre of buoyancy of the whole body from B to B_1; from this we conclude:

(a) that the volumes of the two wedges must be equal
(b) that the first moments of the two wedges about their line of intersection must, for equilibrium, be equal and
(c) that the transfer of first moment of the wedges must equal the change in first moment of the whole body.

Writing down these observations in mathematical symbols,

$$\text{Volume of immersed wedge} = \int y \times \tfrac{1}{2} y\phi \,\mathrm{d}x$$
$$= \text{Volume of emerged wedge}$$
$$\text{1st moment of immersed wedge} = \int \left(\tfrac{1}{2} y^2\phi\right) \times \tfrac{2}{3} y \,\mathrm{d}x$$
$$= \text{1st moment of emerged wedge}$$
$$\text{Transfer of 1st moment of wedges} = 2 \times \int \tfrac{1}{3} y^3 \phi \,\mathrm{d}x = \tfrac{2}{3}\phi \int y^3 \,\mathrm{d}x$$

$$\text{Transfer of 1st moment of whole body} = \nabla \times \overline{\mathrm{BB}'} = \nabla \cdot \overline{\mathrm{BM}} \cdot \phi$$
$$\therefore\ \nabla \cdot \overline{\mathrm{BM}} \cdot \phi = \tfrac{2}{3}\phi \int y^3 \,\mathrm{d}x$$

But we have already seen that $I = \frac{2}{3}\int y^3 \,\mathrm{d}x$ about the axis of inclination for both half waterplanes

$$\therefore\ \overline{\mathrm{BM}} = \frac{I}{\nabla}$$

This is an important geometric property of a floating body. If the floating body is a ship there are two $\overline{\mathrm{BM}}$s of particular interest, the transverse $\overline{\mathrm{BM}}$ for rotation about a fore-and-aft axis and the longitudinal $\overline{\mathrm{BM}}$ for rotation about a transverse axis, the two axes passing through the centre of flotation of the waterplane.

HOLLOW SHAPES

The hull of a ship is, of course, a hollow body enclosed by plating. It will be necessary to find the weight and positions of the centre of gravity of such shapes. A pseudo-expansion of the shape is first obtained by a method described fully in a textbook on laying off. Briefly, the girths of section are plotted at each ordinate and increased in height by a factor to allow for the difference between projected and slant distances in plan. A mean value of this factor is found for each ordinate.

It is now necessary to apply to each ordinate a mean plating thickness which must be found by examining the plating thicknesses (or weights per unit area, sometimes called poundages) along the girth at each ordinate (Fig. 2.23). The variation is usually not great in girth and an arithmetic mean t' will be given by dividing the sum of each plate width x plate thickness by the girth. If the weight density of the material is w, the weight of the bottom plating is thus given by $W = w\int g't'\,dx$ and the position of the LCG is given by

$$\bar{x} = \frac{w}{W}\int g't'x \,\mathrm{d}x$$

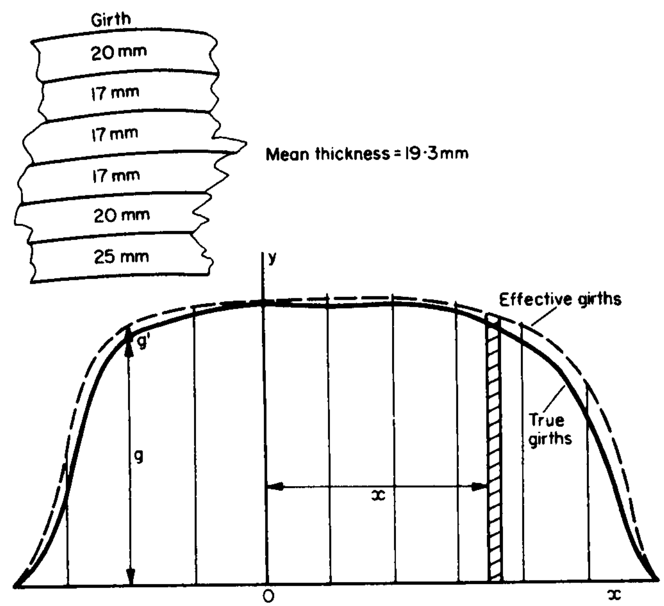


Fig. 2.23 Bottom plating

To find the position of the VCG, it is necessary to return to the sections and to find the position by drawing. Each section is divided by trial and error, into four lengths of equal weight. The mid-points of two adjacent sections are joined and the mid-points of these lines are joined. The mid-point of the resulting line

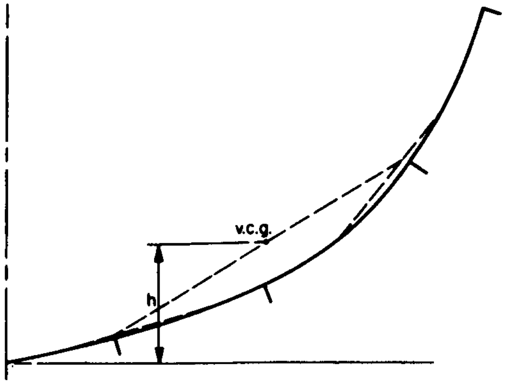


Fig. 2.24 VCG of bottom plating

is the required position of the c.g. and its height h above the keel measured. For the whole body, then, the position of the VCG above the keel is given by

$$\bar{h} = \frac{w}{W}\int g' t' h \, \mathrm{d}x$$

Various factors can be applied to the weight density figure to account for different methods of construction. An allowance for the additional weight of laps, if the plating is raised and sunken or clinker, can be made; an addition can be made for rivet heads. It is unwise to apply any general rule and these factors must be calculated for each case.

SYMBOLS AND CONVENTIONS

It would be simpler if everyone used the same symbols for the same things. Various international bodies attempt to promote this and the symbo,ls used in this book, listed at the beginning, follow the general agreements. The symbols and units associated with hydrodynamics are those agreed by the International Towing Tank Conference.

Approximate integration

A number of different properties of particular interest to the naval architect have been expressed as simple integrals because this is a convenient form of shorthand. It is not necessary to be familiar with the integral calculus, however, beyond understanding that the elongated S sign, $\int$, means the sum of all such typical parts that follow the sign over the extent of whatever follows d. Some textbooks at this stage would use the symbol ~ which is simply the Greek letter S. It is now necessary to adopt various rules for calculating these integrals. The obvious way to calculate $A = \int y\,\mathrm{d}x$ is to plot the curve on squared paper and then count up all the small squares. This could be extended to calculate the first moment, $M = \int xy\,dx$, in which the number of squares in a column, y, is multiplied by the number of squares from the origin, x, and this added for all

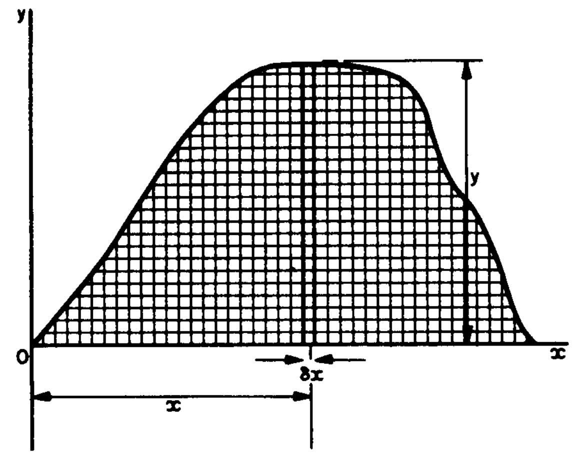


Fig. 2.25 The squared paper approach

of the columns that go to make up the shape. Clearly, this soon becomes laborious and other means of determining the value of an integral must be found. The integral calculus will be used to deduce some of the rules but those who are not yet sufficiently familiar with that subject-and indeed, by those who are-they should be regarded merely as tools for calculating the various expressions shown above in mathematical shorthand.

TRAPEZOIDAL RULE

A trapezoid is a plane four-sided figure having two sides parallel. **If** the lengths of these sides are y_1 and y_2 and they are h apart, the area of the trapezoid is given by

$$A = \tfrac{1}{2}h(y_1 + y_2)$$

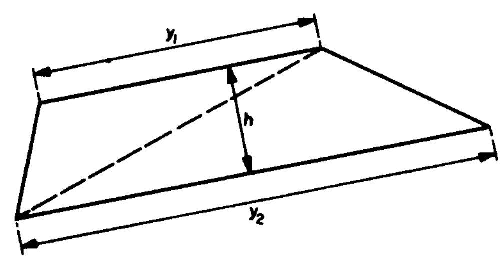


Fig. 2.26 A trapezoid

A curvilinear figure can be divided into a number of approximate trapezoids by covering it with n equally spaced ordinates, h apart, the breadths at the ordinates in order being $y_1, y_2, y_3, \cdots, y_n$.

Commencing with the left-hand trapezoid, the areas of each trapezoid are given by

$$\tfrac{1}{2}h(y_1 + y_2)$$
$$\tfrac{1}{2}h(y_2 + y_3)$$
$$\tfrac{1}{2}h(y_3 + y_4) \ldots$$

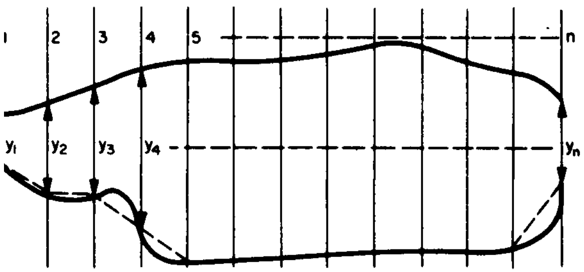


Fig. 2.27 Curvilinear figure represented by trapezoids

By addition, the total area A of the figure is given by

$$A = \tfrac{1}{2}h(y_1 + 2y_2 + 2y_3 + \cdots + y_n)$$
$$= h(\tfrac{1}{2}y_1 + y_2 + y_3 + \cdots + \tfrac{1}{2}y_n)$$

This is termed the *Trapezoidal Rule*. Clearly, the more numerous the ordinates, the more accurate will be the answer. Thus, to evaluate the expression $A = \int y\,dx$ the shape is divided into evenly spaced sections h apart, the ordinates measured and substituted in the rule given above. If the ordinates represent cross-sectional areas of a solid, then the integration gives the volume of that solid, $\nabla = \int A\,dx$.

Expressions can be deduced for moments, but these are not as convenient to use as those that follow.

SIMPSON'S RULES

Generally known as Simpson's rules, these rules for approximate integration were, in fact, deduced by other mathematicians many years previously. They are a special case of the Newton–Cotes' rules. Let us deduce a rule for integrating a curve y over the extent of x. It will be convenient to choose the origin to be in the middle of the base $2h$ long, having ordinates y_1, y_2 and y_3. The choice of origin in no way affects the results as the student should verify for himself.

Assume that the curve can be represented by an equation of the third order,

$$y = a_0 + a_1x + a_2x^2 + a_3x^3$$

The area under the curve is given by

$$A = \int_{-h}^{h} y\,dx = \int_{-h}^{h}(a_0 + a_1x + a_2x^2 + a_3x^3)\,dx$$
$$= \left[a_0x + a_1\frac{x^2}{2} + a_2\frac{x^3}{3} + a_3\frac{x^4}{4}\right]_{-h}^{+h} = 2a_0h + \tfrac{2}{3}a_2h^3 \qquad (1)$$

Assume now that the area can be given by the expression

$$A = Ly_1 + My_2 + Ny_3 \qquad (2)$$

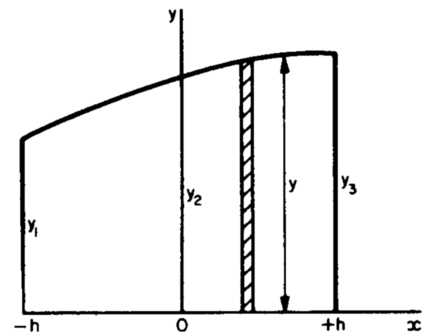


Fig. 2.28

Now

$$y_1 = a_0 - a_1h + a_2h^2 - a_3h^3$$
$$y_2 = a_0$$
$$y_3 = a_0 + a_1h + a_2h^2 + a_3h^3$$

Substituting in Equation (2)

$$A = (L + M + N)a_0 - (L - N)a_1h + (L + N)a_2h^2 - (L - N)a_3h^3 \qquad (3)$$

Equating the coefficients of a in equations (1) and (3)

$$\left.\begin{aligned} L + M + N &= 2h \\ L - N &= 0 \\ L + N &= \tfrac{2}{3}h \end{aligned}\right\} \therefore\ L = N = \tfrac{1}{3}h \quad \text{and} \quad M = \tfrac{4}{3}h$$

The area *can* be represented by Equation (2), therefore, provided that the coefficients are those deduced and

$$A = \tfrac{1}{3}hy_1 + \tfrac{4}{3}hy_2 + \tfrac{1}{3}hy_3 = \tfrac{1}{3}h(y_1 + 4y_2 + y_3)$$

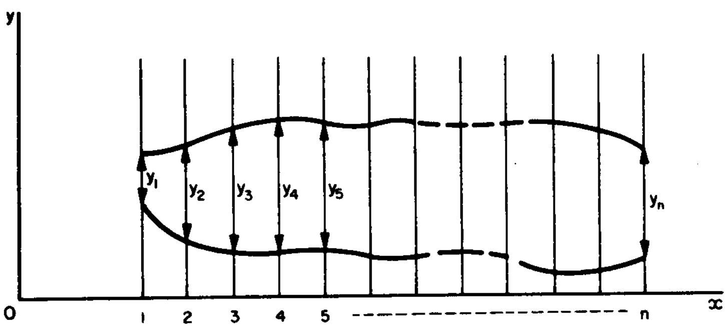


Fig. 2.29

This is known as *Simpson's First Rule* or *3 Ordinate Rule*. A curved figure can be divided by any uneven number of equally spaced ordinates h apart. The area within ordinates numbers 1 and 3 is

$$A_1 = \tfrac{1}{3}h(y_1 + 4y_2 + y_3)$$

within 3 and 5 ordinates

$$A_2 = \tfrac{1}{3}h(y_3 + 4y_4 + y_5)$$

within 5 and 7 ordinates

$$A_3 = \tfrac{1}{3}h(y_5 + 4y_6 + y_7)$$

and so on.
The total area is therefore given by

$$A = \tfrac{1}{3}h(y_1 + 4y_2 + 2y_3 + 4y_4 + 2y_5 + 4y_6 + 2y_7 + \cdots + y_n)$$

i.e.

$$A = \tfrac{2}{3}h(\tfrac{1}{2}y_1 + 2y_2 + y_3 + 2y_4 + y_5 + 2y_6 + y_7 + \cdots + \tfrac{1}{2}y_n)$$

This is the generalized form of the first rule applied to areas. The common multiplier is $\frac{1}{3} \times$ the common interval h and the individual multipliers are 1, 4, 2, 4, 2, 4, ..., 2, 4, 1.

The rule is one for evaluating an integral. While it has been deduced using area as an example, it is equally applicable to any integration using Cartesian or polar co-ordinates. To evaluate the integral $M_x = \int xy\,dx$, for example, instead of multiplying the value of y at each ordinate by the appropriate Simpson multiplier, the value of xy is so treated. Similarly, to evaluate $I_x = \int x^2y\,dx$, the value of x^2y at each ordinate is multiplied by the appropriate Simpson multiplier.

All of these operations are best performed in a methodical fashion as shown in the following example and more fully in worked examples later in the chapter. Students should develop a facility in the use of Simpson's rules by practice.

EXAMPLE 1. Calculate the value of the integral $\int P\,dv$ between the values of v equal to 7 and 15. The values of P at equal intervals of v are as follows:

v	7	9	11	13	15
P	9	27	36	39	37

Solution: The common interval, i.e. the distance between successive values of v, is 2. Setting out Simpson's rule in tabular form,

v	P	Simpson's multipliers	Functions of $\int P\,dv$
7	9	1	9
9	27	4	108
11	36	2	72
13	39	4	156
15	37	1	37
			382

$$\int P\,dv = \tfrac{1}{3} \times 2 \times 382 = \underline{254.7}$$

The units are, of course, those appropriate to $P \times v$.

To apply the trapezoidal rule to a curvilinear shape, we had to assume that the relationship between successive ordinates was linear. In applying Simpson's first rule, we have assumed the relationship to be bounded by an expression of the third order. Much greater accuracy is therefore to be expected and for most functions in naval architecture the accuracy so obtained is quite sufficient. Where there is known to be a rapid change of form, it is wise to put in an intermediate ordinate and the rule can be adapted to do this. Suppose that the rapid change is known to be between ordinates 3 and 4 (Fig. 2.30).

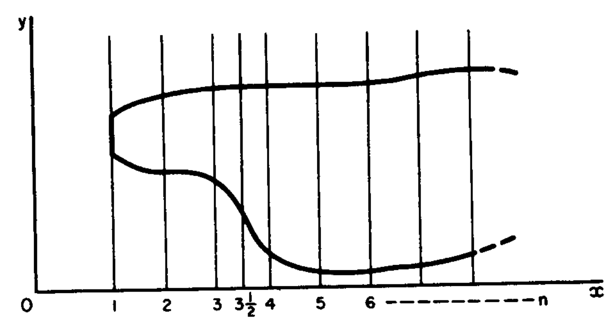


Fig. 2.30

$$\text{Area between 1 and 3 ords.} = \tfrac{1}{3}h(y_1 + 4y_2 + y_3)$$

$$\text{Area between 3 and 4ords.} = \tfrac{1}{3}\frac{h}{2}(y_3 + 4y_{3\frac{1}{2}} + y_4) = \tfrac{1}{3}h\left(\tfrac{1}{2}y_3 + 2y_{3\frac{1}{2}} + \tfrac{1}{2}y_4\right)$$

$$\text{Area between 4 and 6 ords.} = \tfrac{1}{3}h(y_4 + 4y_5 + y_6)$$

$$\text{Total area} = \tfrac{1}{3}h\left(y_1 + 4y_2 + 1\tfrac{1}{2}y_3 + 2y_{3\frac{1}{2}} + 1\tfrac{1}{2}y_4 + 4y_5 + 2y_6 + \cdots + y_n\right)$$

Note that, unless a second half ordinate is inserted, n must now be even.

Rules can be deduced, in a similar manner, for figures bounded by unevenly spaced ordinates or different numbers of evenly spaced ordinates. For four evenly spaced ordinates the rule becomes

$$A = \tfrac{3}{8}h(y_1 + 3y_2 + 3y_3 + y_4)$$

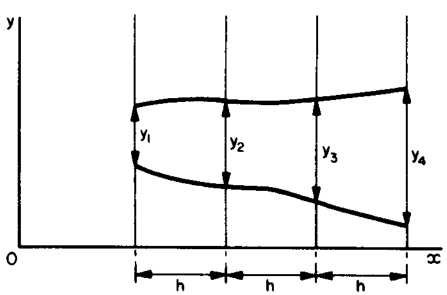


Fig. 2.31

This is known as *Simpson's Second Rule*. Extended for a large number of ordinates, it becomes

$$A = \tfrac{3}{8}h(y_1 + 3y_2 + 3y_3 + 2y_4 + 3y_5 + 3y_6 + 2y_7 + \cdots + y_n)$$

Thus, the common multiplier in this case is $\frac{3}{8}$ times the common interval and the individual multipliers, 1, 3, 3, 2, 3, 3, 2, 3, 3, 2, ..., 3, 3, 1. It is suitable for 4, 7, 10, 13, 16, etc., ordinates. It can be proved in a manner exactly similar to that employed for the first rule, assuming a third order curve, and it can be used like the first rule to integrate any continuous function.

Another particular Simpson's rule which will be useful is that which gives the area between two ordinates when three are known. The area between ordinates 1 and 2 is given by

$$A_1 = \tfrac{1}{12}h(5y_1 + 8y_2 - y_3)$$

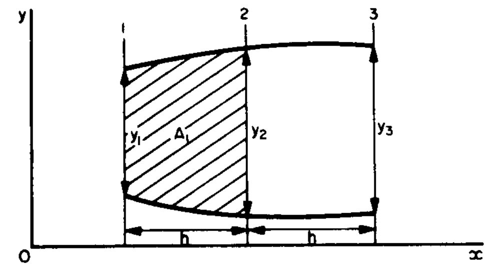


Fig. 2.32

This rule cannot be used for moments. The first moment of area of that portion between ordinates 1 and 2 about number 1 ordinate is

$$M_x = \tfrac{1}{24}h^2(3y_1 + 10y_2 - y_3)$$

These two rules are known loosely as the 5,8 *minus one Rule* and the 3,10 *minus one Rule*. They are somewhat less accurate than the first two rules. Incidentally, applying the 5,8 minus one rule backwards, the unshaded area of Fig. 2.32 is

$$A_2 = \tfrac{1}{12}h(-y_1 + 8y_2 + 5y_3)$$

and adding this to the expression for A_1, the total area is

$$A = \tfrac{1}{3}h(y_1 + 4y_2 + y_3)$$

If the common multiplier has been forgotten, the student can quickly deduce it by applying the particular rule to a rectangle.

Rules can be combined one with another just as the unit for each rule is combined in series to deal with many ordinates. It is important that any discontinuity in a curve falls at the end of a unit, e.g. on the 2 multiplier for the first and second rules; if this is so, the rules can be used on curves with discontinuities. In general, because of differences in the common interval necessary each side of a discontinuity, it will be convenient to deal with the two parts separately.

The first rule deals with functions bounded by 3, 5, 7, 9, 11, 13, etc., ordinates and the second rule with 4, 7, 10, 13, etc., ordinates. Let us deduce a rule for six ordinates

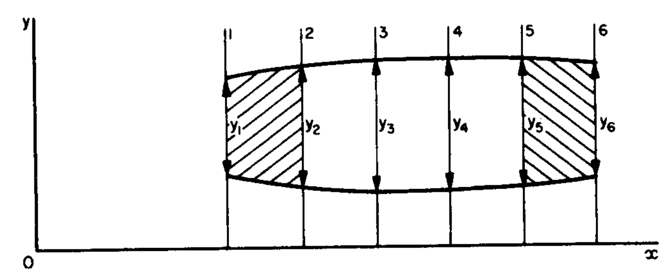


Fig. 2.33

$$\text{Area between 1 and 2 ords.} = \tfrac{1}{12}h(5y_1 + 8y_2 - y_3)$$

$$= h\left(\tfrac{5}{12}y_1 + \tfrac{8}{12}y_2 - \tfrac{1}{12}y_3\right)$$

$$\text{Area between 2 and 5 ords.} = \tfrac{3}{8}h(y_2 + 3y_3 + 3y_4 + y_5)$$

$$= h\left(\tfrac{3}{8}y_2 + \tfrac{9}{8}y_3 + \tfrac{9}{8}y_4 + \tfrac{3}{8}y_5\right)$$

$$\text{Area between 5 and 6 ords.} = \tfrac{1}{12}h(-y_4 + 8y_5 + 5y_6)$$

$$= h\left(-\tfrac{1}{12}y_4 + \tfrac{8}{12}y_5 + \tfrac{5}{12}y_6\right)$$

Total area

$$A = h\left(\tfrac{5}{12}y_1 + \tfrac{25}{24}y_2 + \tfrac{25}{24}y_3 + \tfrac{25}{24}y_4 + \tfrac{25}{24}y_5 + \tfrac{5}{12}y_6\right)$$

$$= \tfrac{25}{24}h(0.4y_1 + y_2 + y_3 + y_4 + y_5 + 0.4y_6)$$

Table 2.1
Newton–Cotes' rules

Number of ordinates	Multipliers for ordinates numbers 1	2	3	4	5
2	$\frac{1}{2}$	$\frac{1}{2}$			
3	$\frac{1}{6}$	$\frac{4}{6}$	$\frac{1}{6}$		
4	$\frac{1}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{8}$	
5	$\frac{7}{90}$	$\frac{32}{90}$	$\frac{12}{90}$	$\frac{32}{90}$	$\frac{7}{90}$
6	$\frac{19}{288}$	$\frac{75}{288}$	$\frac{50}{288}$	$\frac{50}{288}$	$\frac{75}{288}\cdots$
7	$\frac{41}{840}$	$\frac{216}{840}$	$\frac{27}{840}$	$\frac{272}{840}$	$\frac{27}{840}\cdots$
8	$\frac{751}{17280}$	$\frac{3577}{17280}$	$\frac{1323}{17280}$	$\frac{2989}{17280}$	$\frac{2989}{17280}\cdots$
9	$\frac{989}{28350}$	$\frac{5888}{28350}$	$-\frac{928}{28350}$	$\frac{10496}{28350}$	$-\frac{4540}{28350}\cdots$

Area $= L \times \Sigma$ (Multiplier $\times$ ordinate)

Ordinates are equally spaced with end ordinates coinciding with ends of curve. Multipliers are always symmetrical and are indicated by dots.

These few Simpson's rules, applied in a repetitive manner, have been found satisfactory for hand computation for many years. The digital computer makes somewhat different demands and the more generalized Newton-Cotes' rules, summarized in Table 2.1, may be found more suitable for some purposes.

TCHEBYCHEFF'S RULES

Returning to Equation (2) under Simpson's rules, the rule required was forced to take the form of the sum of equally spaced ordinates, each multiplied by a coefficient. The rule could have been forced to take many forms, most of them inconvenient.

One form which does yield a convenient rule results from assuming that the area can be represented by the sum of ordinates placed a special distance x (which may be zero) from the origin, all multiplied by the same coefficient, i.e. instead of assuming the form as before, assume that the area can be represented by

$$A = p(y_1 + y_2 + y_3), \quad y_2 \text{ being at the origin}$$

Now

$$y_1 = a_0 - a_1x + a_2x^2 - a_3x^3$$

$$y_2 = a_0$$

$$y_3 = a_0 + a_1x + a_2x^2 + a_3x^3$$

adding:

$$A = p(3a_0 + 2a_2x^2)$$

equating coefficients of a above with those of Equation (1) on p. 25

$$3p = 2h \quad \therefore\ p = \tfrac{2}{3}h$$

and

$$2px^2 = \tfrac{2}{3}h^3 = ph^2$$

$$\therefore\ x = \frac{1}{\sqrt{2}}h = 0.7071\,h$$

The total shaded area in Fig. 2.34 can therefore be calculated by erecting ordinates equal to 0.7071 of the half length from the mid-point, measuring their heights and that of the mid-ordinate, adding the three heights together and multiplying the total by two-thirds of the half length.

$$A = \tfrac{2}{3}h(y_1 + y_2 + y_3)$$

where y_1 and y_3 are 0.7071 of the half length h, from the mid-point.

This is Tchebycheff's rule for three ordinates. Similar rules can be deduced for 2, 3, 4, 5, 6, 7 and 9 ordinates when spacings become as shown in Table 2.2.

The eight and ten ordinate rule spacings have been deduced by applying the four and five ordinate rules each side of the mid-point of the half length.

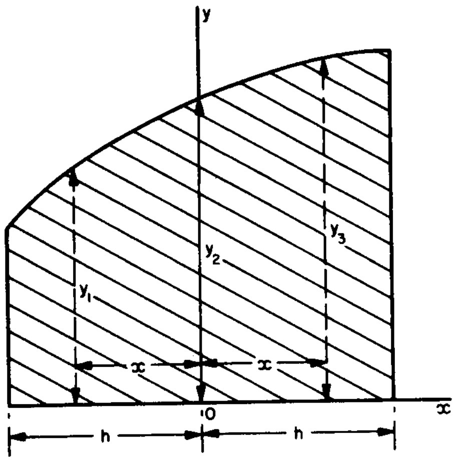


Fig. 2.34

Table 2.2
Tchebycheff's rule spacings

Number of ordinates, n	Spacing each side of mid-ordinate as a factor of the half length h					Degree of curve
2	0.57735					3
3	0	0.70711				3
4	0.18759	0.79465				5
5	0	0.37454	0.83250			5
6	0.26664	0.42252	0.86625			7
7	0	0.32391	0.52966	0.88386		7
8	0.10268	0.40620	0.59380	0.89733		5
9	0	0.16791	0.52876	0.60102	0.91159	9
10	0.08375	0.31273	0.50000	0.68727	0.91625	5

The common multiplier for all rules is the whole length $2h$ divided by n, the number of ordinates used, $2h/n$.

Tchebycheff's rules are used not infrequently, particularly the ten ordinate rule, for calculating displacement from a 'Tchebycheff body plan', i.e. a body plan drawn with ordinate positions to correspond to the Tchebycheff spacings. Areas of the sections are calculated by Simpson's rules or by other convenient means, merely added together and multiplied by $2h/n$ to give volume of displacement. Lines are, in fact, often faired on a Tchebycheff body plan to avoid the more prolonged calculation by Simpson's rules with each iteration. Since fairing is basically to a curve or areas, this assumes the use of Tchebycheff ordinates to define the body plan.

GAUSS RULES

It has been seen that the Simpson rules and Newton-Cotes' rules employ equally spaced ordinates with unequal multipliers and the Tchebycheff rules use constant

multipliers with unequal ordinate spacing. A third set of rules, the Gauss rules, uses unequal spacing of ordinates and unequal multipliers as shown in Table 2.3.

Table 2.3
Gauss' rules

Number of ordinates	Spacing each side of mid-ordinate as a factor of the half length. Multiplier			
2	Spacing	0.57735		
	Multiplier	0.50000		
3	Position	0	0.77460	
	Multiplier	0.44444	0.27778	
4	Position	0.33998	0.86114	
	Multiplier	0.32607	0.17393	
5	Position	0	0.53847	0.90618
	Multiplier	0.28445	0.23931	0.11846
	Integral = Sum of products × whole base			

The Gauss rules have the merit of being more accurate than either the Simpson or Tchebycheff rules, but their application involves more tedious calculation when manual methods are used. By using Gauss rules, the naval architect can either obtain greater accuracy by using the same number of ordinates or obtain the same accuracy with fewer ordinates and in less time.

It can be shown that:

(*a*) a Simpson rule with an even number of ordinates is only marginally more accurate than the next lower odd ordinate rule; odd ordinate Simpson rules are therefore to be preferred,
(*b*) a Tchebycheff rule with an even number of ordinates gives the same accuracy as the next highest odd ordinate rule. Even ordinate Tchebycheff rules are therefore to be preferred,
(*c*) a Tchebycheff rule with an even number of ordinates gives an accuracy rather better than the next highest odd ordinate Simpson rule, i.e. the two-ordinate Tchebycheff rule is more accurate than the three-ordinate Simpson rule,
(*d*) The five-ordinate Gauss rule gives an accuracy comparable with that achieved with nine-ordinate Simpson or Tchebycheff rules.

EXAMPLE 2. Integrate $y = \tan x$ from $x = 0$ to $x = \pi/3$ by the five ordinate rules of (*a*) Simpson, (*b*) Newton–Cotes, (*c*) Tchebycheff, (*d*) Gauss.

Solution: The precise solution is

$$\int_0^{\pi/3} \tan x = [-\log\cos x]_0^{\pi/3}$$
$$= 0.69315$$
$$= 0.22064\pi$$

(*a*) Simpson

x	y	SM	$f(A)$
0	0	$\frac{1}{2}$	0
$\pi/12$	0.26795	2	0.53590
$\pi/6$	0.57735	1	0.57735
$\pi/4$	1.00000	2	2.00000
$\pi/3$	1.73205	$\frac{1}{2}$	0.86603
			3.97928

$$\text{Area} = \frac{2}{3} \times \frac{\pi}{12} \times 3.97928 = 0.22107\pi$$

(*b*) Newton–Cotes

x	y	M	$f(A)$
0	0	7	0
$\pi/12$	0.26795	32	8.57440
$\pi/6$	0.57735	12	6.92820
$\pi/4$	1.00000	32	32.00000
$\pi/3$	1.73205	7	12.12435
			59.62695

$$\text{Area} = \frac{1}{90} \times \frac{\pi}{3} \times 59.62695 = 0.22084\pi$$

(*c*) Tchebycheff

x	y
$\pi/6(1 - 0.83250) = 5.025°$	0.08793
$\pi/6(1 - 0.37454) = 18.764°$	0.33972
$\pi/6 = 30°$	0.57735
$\pi/6(1 + 0.37454) = 41.236°$	0.87655
$\pi/6(1 + 0.83250) = 54.975°$	1.42674
	3.30829

$$\text{Area} = \frac{\pi}{15} \times 3.30829 = 0.22055\pi$$

(*d*) Gauss

x	y	M	$f(A)$
$\pi/6(1 - 0.90618) = 2.815°$	0.04918	0.11846	0.00583
$\pi/6(1 - 0.53847) = 13.846°$	0.24647	0.23931	0.05898
$\pi/6 = 30°$	0.57735	0.28445	0.16423
$\pi/6(1 + 0.53847) = 46.154°$	1.04114	0.23931	0.24916
$\pi/6(1 + 0.90618) = 57.185°$	1.55090	0.11846	0.18372
			0.66192

$$\text{Area} = \frac{\pi}{3} \times 0.66192 = 0.22064\pi$$

Computers

DIGITAL COMPUTERS

A digital computer is an electronic device capable of holding large amounts of data and of carrying out arithmetical computations and groups of logical processes at high speed. As such it is as much a tool for the naval architect to use, as the slide rule, or the rule for calculating areas and volumes. It is, of course, much more powerful but that makes it even more important to understand. It is not necessary to know in detail how it works but its basic characteristics, strengths and weaknesses, should be understood.

The system includes input units which accept information in a suitably coded form (CD-rom or disk readers, keyboards, optical readers or light pens); storage or memory units for holding instructions; a calculation unit by which data is manipulated; a control unit which calls up data and programs from storage in the correct sequence for use by the calculation unit and then passes the results to output units; output units for presenting results (visual display units, printers, or plotters); and a power unit.

The immediate output may be a magnetic tape or disk which can be decoded later on separate print-out devices. Sometimes the output is used directly to control a machine tool or automatic draughting equipment. Input and output units may be remote from the computer itself, providing a number of outstations with access to a large central computer, or providing a network with the ability to interact with other users.

As with any other form of communication, that between the designer and the computer must be conducted in a language understood by both. There are many such languages suitable for scientific, engineering and commercial work. The computer itself uses a compiler to translate the input language into the more complex machine language it needs for efficient working.

Input systems are usually interactive, enabling a designer to engage in a dialogue with the computer or, more accurately, with the software in the computer. Software is of two main types; that which controls the general activities within the computer (e.g. how data is stored and accessed) and that which directs how a particular problem is to be tackled. The computer may prompt the operator by asking for more data or for a decision between possibl~ options it presents to him. The software can include decision aids, i.e. it can recommend a particular option as the best in the circumstances-and give its reasons if requested. If the operator makes, or appears to make, a mistake the machine can challenge the input.

Displays can be in colour and present data in graphical form. Colour can be useful in differentiating between different elements of the total display. Red can be used to highlight hazardous situations because humans associate red with danger. However, for some applications monochrome is superior. Shades of one colour can more readily indicate the relative magnitude of a single parameter, e.g. shades of blue indicating water depth. Graphical displays are often more meaningful to humans than long tabulations of figures. Thus a plot of points which should lie on smooth curves will quickly highlight a rogue reading.

This facility is used as an input check in large finite element calculations. The computer can cause the display to rotate so that a complex shape, a ship's hull for instance, can be viewed from a number of directions. The designer can view a space or equipment from any chosen position. In this way checks can be made, as the design progresses, on the acceptability of various sight lines. Can operators see all the displays in, say, the Operations Room they need to in order to carry out their tasks? Equally, maintainers can check that there is adequate space around an equipment for opening it up and working on it.

Taking this one stage further, the computer can generate what is termed virtual reality (VR). The user of VR is effectively immersed in, and interacts with, a computer generated environment. Typically a helmet, or headset, is worn which has a stereoscopic screen for each eye. Users are fitted with sensors which pick up their movements and the computer translates these into changing pictures on the screens. Thus an owner could be taken on a 'walk through' of a planned vessel, or those responsible for layouts can be given a tour of the space. Colours, surface textures and lighting can all be represented. Such methods are capable of replacing the traditional mock-ups and the 3-D and 2-D line outs used during construction. All this can be done before any steel is cut. To enhance the sense of realism gloves, or suits, with force feed back devices can be worn to provide a sense of touch. Objects can be 'picked up' and 'manipulated' in the virtual environment.

It does not follow that because a computer can be used to provide a service it should be so used. It can be expensive both in money and time. Thus in the example above, the computer may be cheaper than a full scale mock-up but would a small scale model suffice? A computer is likely to be most cost effective as part of a comprehensive system. That is, as part of a *computer aided design and manufacture system (CAD/CAM).* In such systems the designer uses a terminal to access data and a complete suite of design programs. Several systems have been developed for ship design, some concentrating on the initial design phase and others on the detailed design process and its interaction with production. Thus once the computer holds a definition of the hull shape that shape can be called up for subsequent manipulation and for all calculations, layouts, etc. for which it is needed. This clearly reduces the chance of errors. Again, once the structure has been designed the computer can be programmed to generate a materials ordering list. Then given suitable inputs it can keep track of the material through the stores and workshops. The complexity of a ship, and the many inter-relationships between its component elements, are such that it is an ideal candidate for computerization. The challenge lies in establishing all the interactions.

In the case of the design and build of a Landing Platform Dock (LPD) for the UK MOD a system was used involving 250 work stations for creating 2-D and 3-D geometry and data, 125 work stations for viewing the data and 140 PCs for accessing and creating the *product definition model data.* The system enabled 'virtual prototyping' and early customer approval of subjective areas of the ship. Among other uses of the computer which are of interest to the naval architect, are:

Simulation modelling. Provided that the factors governing a real life situation are understood, it may be possible to represent it by a set of mathematical relationships. In other words it can be modelled and the model used to study the effects of changing some of the factors much quicker, cheaper and safer than could be achieved with full scale experimentation. Consider tankers arriving at a terminal. Factors influencing the smooth operation are numbers of ships, arrival intervals, ship size, discharge rate and storage capacities. A simulation model could be produced to study the problem, reduce the queuing time and to see the effects of additional berths and different discharge rates and ship size. The economics of such a procedure is also conducive to this type of modelling.

Expert systems and decision aiding. Humans can reason and learn from previous experience. *Artificial Intelligence* (AI) is concerned with developing computer-based systems endowed with such higher intellectual processes. It has been possible to program them to carry out fairly complex tasks such as playing chess. The computer uses its high speed to consider all possible options and their consequences and to develop a winning strategy. Such programs are called *expert systems.* These systems can make use of human inputs to help decide on the significance of certain situations and what action is advisable. These *knowledge-based expert systems* have been used as *decision aids.* An early application of such techniques was in medicine. Medical officers could not be provided for all ballistic missile submarines, but they did carry a qualified sick berth attendant (SBA). The SBA would examine a sick crew member, taking temperature and other readings to feed into a computer program containing contributions from distinguished doctors. The computer then analyzed the data it received, decided what might be wrong with the patient and asked for additional facts to narrow down the possibilities. The end result was a print out of the most likely complaints and their probability. This enabled the command to decide whether, or not, to abort the mission.

In the same way, the naval architect can develop decision aids for problems where a number of options are available and experience is useful. Such aids can enlist the help of leading designers, making their expertise available to even inexperienced staff.

SIMULATORS

Anticipating some of the work of later chapters, the behaviour of a ship in response to applied forces and moments can be represented by a mathematical equation. The applied forces may arise from the deliberate action of those on board in moving a control surface such as a rudder or from some external agency such as the seaway in which the ship is operating. In its simplest form, the equation may be a linear differential equation involving one degree of freedom but, to achieve greater accuracy, may include non-linear and cross-coupling terms.

The same form of equation can be represented by a suitably contrived electrical circuit. In the case of a ship turning under the action of the rudder, if the componefts of the electrical circuit are correctly chosen, by varying an input signal in (nformity with the rudder movements, the voltage across two points of the circuit can be measured to represent the ship's heading. By extending the circuitry, more variables can be studied such as the angle of heel, drift angle and advance. This is the fundamental principle of the *analogue computer.*

The correct values of the electrical components can be computed by theoretical means, or measured in model experiments or full scale trials. Having set up the circuit correctly it will represent faithfully the response of the ship. That is, it will 'simulate' the ship's behaviour. It can be made to do this in real time. The realism can be heightened by mounting the set-up on a enclosed platform which turns and tilts in response to the output signal~ Furthermore, the input can be derived from a steering wheel turned by an operator who then gains the impression of actually being on a ship. The complete system is called a *simulator* and such devices are used to train personnel in the operation of ships and aircraft. They are particularly valuable for training people to deal with emergency situations which can arise in service but which are potentially too dangerous to reproduce deliberately in the vehicle itself. Simulators for pilotage in crowded and restricted waters are an example. The degree of realism can be varied to suit the need, the most comprehensive involving virtual reality techniques. By varying the electrical constants of the circuit the responses of different ships can be represented.

If desired, components of a real shipboard system can be incorporated into the electrical system. If the behaviour of a particular hydraulic control system is difficult to represent (it may be non-linear in a complex manner) the system itself can be built into the simulator. The operator can be presented with a variety of displays to see which is easiest to understand and act upon. Research of this type was done in the early days of one-man control systems for submannes.

In many applications the high speed digital computer has replaced, or is used in conjunction with, the analogue computer giving greater flexibility. Computer graphics allow the external environment to be represented pictorially in a realistic manner. Thus in a pilotage simulator the changing external view, as the ship progresses through a harbour, can be projected on to screens reproducing what a navigator would see from the bridge if negotiating that particular harbour. Other vessels can be represented, entering or leaving port. Changed visibility under night time or foggy conditions can be included. Any audio cues, such as fog sirens or bells on buoys, can be injected for added realism.

Another useful simulator is one representing the motions of a ship in varying sea conditions. If the outputs are used to drive a cabin in a realistic way, including the vertical motions, the effects of motion on a human being can be established. Some subjects can be cured of seasickness by a course of treatment in a simulator. By fitting out the cabin with various work stations, say a sonar operator's position, the human's ability to perform under motion conditions can be studied. Apart from physical symptoms, such as nausea or loss of balance, the mental processes of the operator may be degraded. It may prove more difficult to concentrate or to distinguish contacts on the screen. Optimum orientation of displays to the axes of motion can be developed.

Approximate formulae and rules

Approximate formulae and rules grew up with the craftsman approach to naval architecture and were encouraged by the secrecy that surrounded it. Many were bad and most have now been discarded. There remains a need, however, for coarse approximations during the early, iterative processes of ship design. Usually, the need is met by referring to a similar, previous design and correcting the required figure according to the dimensions of new and old, e.g. supposing an estimate for the transverse -B-M= *h*/,V' is required, the transverse second moment of area *h* is proportional to L x B^3 and V' is proportional to *LxBxT*

$$\therefore\ \overline{\mathrm{BM}} \propto \frac{LB^3}{LBT} = \frac{B^2}{T}$$

$$\therefore\ \overline{\mathrm{BM}} = \frac{B^2}{KT}$$

where K is a constant for geometrically similar ships.

$$\text{or}\quad \frac{\overline{\mathrm{BM}}_1}{\overline{\mathrm{BM}}_2} = \frac{B_1^2}{B_2^2} \times \frac{T_2}{T_1}$$

This presumes that the forms are similar. This formula is one that may suffice for a check; K varies between 10 and 15 for ship shapes. More important, it shows how ship dimensions affect this geometric property, viz. that the beam contributes as its square, the draught inversely and the length not at all. The effects of changes to dimensions can, therefore, be assessed.

An approximate formula for the longitudinal $\overline{\mathrm{BM}}$ is

$$\overline{\mathrm{BM}}_{\mathrm{L}} = \frac{3A_{\mathrm{w}}^2 L}{40B\nabla}$$

NORMAND'S FORMULA

This is known also as Morrish's rule. It gives the distance of the centre of buoyancy of a ship-like form below the waterline.

$$\text{Distance of VCB below waterline} = \frac{1}{3}\left(\frac{T}{2} + \frac{\nabla}{A_{\mathrm{w}}}\right)$$

The formula is remarkably accurate for conventional ship forms and is used frequently in the early design stages.

WEIGHT CONVENTIONS

Plating thickness in ships is usually specified in millimetres. Sections used in ship construction are also specified in millimetres, web × table × thickness with a mass per metre in kg/m.

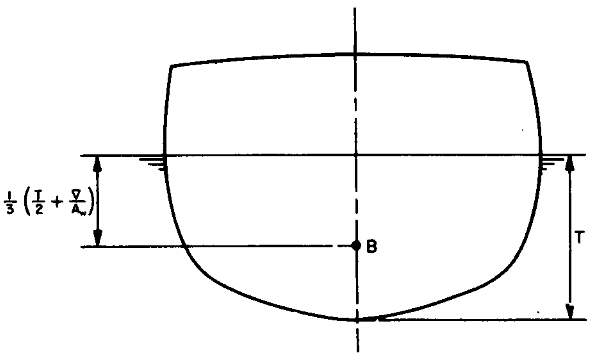


Fig. 2.35

Statistics

A feature of present day naval architecture, as in other engineering disciplines, is the increasing use made of statistics by the practising naval architect. This is not because the subject itself has changed but rather that the necessary mathematical methods have been developed to the stage where they can be applied to the subject.

It will be concluded, for example in Chapter 6, that the hull girder stress level accepted from the standard calculation should reflect the naval architect's opinion as to the probability of exceeding the standard assumed loading during the life of the ship. Again, in the study of ship motions the extreme amplitudes of motion used in calculations must be associated with the probability of their occurrence and probabilities of exceeding lesser amplitudes are also of considerable importance.

It is not appropriate in a book of this nature to develop in detail the statistical approach to the various aspects of naval architecture. Students should refer to a textbook on statistics for detailed study. However, use is made in several chapters of certain general concepts of which the following are important.

PROBABILITY

Consider an aggregate of n experimental results (e.g. amplitudes of pitch from a ship motion trial) of which m have the result Rand $(n - m)$ do not have this result. Then, the probability of obtaining the result R is $p = min$. The probability that R will not occur is I - p. If an event is impossible its probability is zero. If an event is a certainty its probability is unity.

PROBABILITY CURVE

When a large amount of information is available, it can be presented graphically by a curve. The information is plotted in such a way that the area under the curve is unity and the probability of the experimental result lying between say Rand $R + DR$ is represented by the area under the curve between these values of the abscissa. There are a number of features about this probability curve which may best be defined and understood by an example.

Consider the following example of experimental data.

EXAMPLE 3. Successive amplitudes of pitch to the nearest half degree recorded during a trial are:

$4, 2, 3\frac{1}{2}, 2\frac{1}{2}, 3, 2, 3\frac{1}{2}, 1\frac{1}{2}, 3, 1, 3\frac{1}{2}, \frac{1}{2}, 2, 1, 1\frac{1}{2}, 1, 2, 1\frac{1}{2}, 1\frac{1}{2}, 4, 2\frac{1}{2}, 3\frac{1}{2}, 3, 2\frac{1}{2}, 2, 2\frac{1}{2}, 2\frac{1}{2}, 3, 2, 1\frac{1}{2}.$

Solution: As a string of figures these values have little significance to the casual reader. Using the concepts given above however, the occurrence of specific pitch amplitudes is given in columns (1) and (2) below:

(1) Pitch amplitude	(2) Number of occurrences	(3) (1) × (2)	(4) (1) − μ	(5) $(4)^2$	(6) (5) × (2)
$\frac{1}{2}$	1	$\frac{1}{2}$	−1.82	3.312	3.31
1	3	3	−1.32	1.741	5.23
$1\frac{1}{2}$	5	$7\frac{1}{2}$	−0.82	0.672	3.36
2	6	12	−0.32	0.102	0.61
$2\frac{1}{2}$	5	$12\frac{1}{2}$	+0.18	0.032	0.16
3	4	12	0.68	0.462	1.85
$3\frac{1}{2}$	4	14	1.18	1.392	5.57
4	2	8	1.68	2.822	5.64
	30	$69\frac{1}{2}$			25.73

Selecting the information from this table,

$$\text{ARITHMETIC MEAN} = \mu = \frac{69\frac{1}{2}}{30} = 2.32 \text{ deg}$$

MEDIAN is the mid measurement of the 30 which occurs midway between 2 and $2\frac{1}{2} = 2.25$ deg. The median bisects the area

MODE = 2 deg. The mode is the most common value

RANGE $= 4 - \frac{1}{2} = 3\frac{1}{2}$ deg. The range is the difference of extreme values of variate

$$\text{STANDARD DEVIATION} = \sigma = \sqrt{\left(\frac{25.73}{30}\right)} = 0.926 \text{ deg}$$

$$\text{VARIANCE} = \sigma^2 = 0.858 \text{ (deg)}^2$$

These values provide a much clearer physical picture of the motion being measured than does the original test data. The significance of these various features of the probability curve will be clear to the student with some knowledge of statistics (see Fig. 2.36).

Worked examples

EXAMPLE 4. Calculate the area, position of the centre of flotation and the second moments of area about the two principal axes of the waterplane defined by the following ordinates, numbered from forward. It is 220 m long.

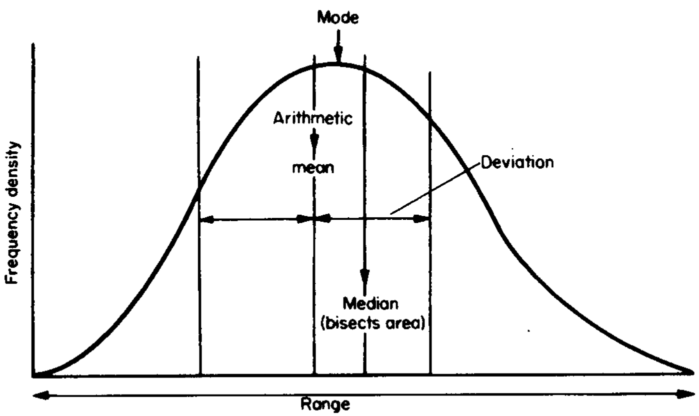


Fig. 2.36

Ordinate number	1	2	3	4	5	6	7	8	9	10	11
$\frac{1}{2}$Ord. (m)	0.2	2.4	4.6	6.7	8.1	9.0	9.4	9.2	8.6	6.3	0.0

Solution:

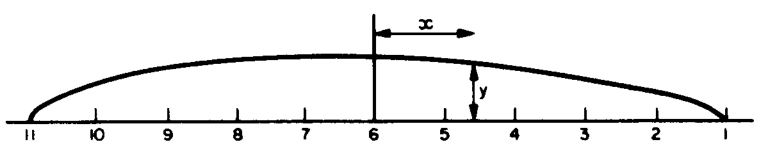


Fig. 2.37

Ord. no.	$\frac{1}{2}$ Ord. y	S.M.	Func. of y	Lever x	Func. of xy	Lever x	Func. of x^2y	$\frac{1}{2}$Ord.^{3} y^3	Func. of y^3
1	0.2	$\frac{1}{2}$	0.1	5	0.5	5	2.5	0.0	0.0
2	2.4	2	4.8	4	19.2	4	76.8	13.8	27.6
3	4.6	1	4.6	3	13.8	3	41.4	97.3	97.3
4	6.7	2	13.4	2	26.8	2	53.6	300.8	601.6
5	8.1	1	8.1	1	8.1	1	8.1	531.4	531.4
6	9.0	2	18.0	0	68.4	0	0.0	729.0	1458.0
7	9.4	1	9.4	−1	−9.4	−1	9.4	830.6	830.6
8	9.2	2	18.4	−2	−36.8	−2	73.6	778.7	1557.4
9	8.6	1	8.6	−3	−25.8	−3	77.4	636.1	636.1
10	6.3	2	12.6	−4	−50.4	−4	201.6	250.0	500.0
11	0.0	$\frac{1}{2}$	0.0	−5	0.0	−5	0.0	0.0	0.0
			98.0		−122.4		544.4		6240.0
					68.4				
					−54.0				

$$\text{Area} = \int y\,dx \text{ for each half}$$

There are eleven ordinates and therefore ten spaces, $\frac{220}{10} = 22$ m apart. The total of the $f(y)$ column must be multiplied by $\frac{2}{3}$ times 22, the common interval, to complete the integration and by 2, for both sides of the waterplane.

$$\text{Total area} = 2 \times \tfrac{2}{3} \times 22 \times 98.0 = 2{,}874.7\,\text{m}^2$$

$$\text{1st moment} = \int xy\,dx \text{ for each half}$$

Instead of multiplying each ordinate by its actual distance from Oy, we have made the levers the number of ordinate distances to simplify the arithmetic so that the total must be multiplied by a lever factor 22. We have also chosen Oy as number 6, the mid-ordinate as being somewhere near the centre of area, to ease the arithmetic; it may well transpire that number 7 ordinate would have been closer. The moments each side are in opposition and must be subtracted to find the out-of-balance moment. First moment about number 6 ordinate, $M_x = 2 \times \frac{2}{3} \times 22 \times 22 \times 54.0$.

Now the distance of the centre of area from O$y = M_x/A$; three of the multipliers are common to both and cancel out, leaving

$$\text{CF abaft 6 ord. } \bar{x} = 22 \times \frac{f(xy)}{f(y)} = \frac{22 \times 54}{98} = 12.1\,\text{m}$$

$$\text{2nd Moment about O}y = \int x^2y\,dx \text{ for each half}$$

We have twice multiplied by the number of ordinate spacings instead of the actual distances so that the lever factor this time is 22 × 22. The second moments all act together (x is squared and therefore always positive) and must be added

$$\therefore\ I \text{ about 6 ord.} = 2 \times \tfrac{2}{3} \times 22 \times 22 \times 22 \times 544.4 = 7{,}729{,}000\,\text{m}^4$$

Now this is not the least I and is not of much interest; the least is always that about an axis through the centre of area and is found from the parallel axis theorem

$$I_{\text{L}} = I - A\bar{x}^2 = 7{,}729{,}000 - 2874.7 \times (12.1)^2 = 7{,}309{,}000\,\text{m}^4$$

$$\text{2nd Moment about O}x = \tfrac{1}{3}\int y^3\,dx \text{ for each half}$$

To integrate $y^3, f(y)^3$ must be multiplied by $\frac{2}{3}$ times the common interval.

$$I_{\text{T}} = 2 \times \tfrac{1}{3} \times \tfrac{2}{3} \times 22 \times 6240.0 = 61{,}000\,\text{m}^4$$

This time the axis already passes through the centre of area.

EXAMPLE 5. A ship has a main body defined by the waterplane areas given below. The waterlines are 0.5 m apart. In addition, there is an appendage having a volume of displacement of $10\,m^3$ with a centre of volume 0.1 m below No. 4 WL.

What are the volume of displacement and the position of the VCB?

Waterline	1	2	3	4
Area (m^2)	123	110	87	48

Solution:

WL	Area A	S.M.	$f(A)$	Lever, y	$f(Ay)$
1	123	1	123	0	0
2	110	3	330	1	330
3	87	3	261	2	522
4	48	1	48	3	144
			762		996

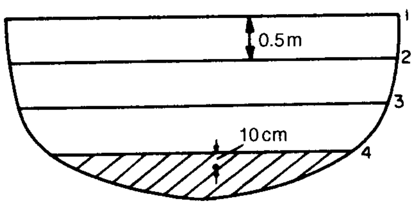


Fig. 2.38

Volume of displacement for main body $= \frac{3}{8} \times 0.5 \times 762 = 143\,m^3$

$$\text{VCB below 1 WL for main body} = \frac{996}{762} \times 0.5 = 0.654\,m$$

Adding volumes and moments for main body and appendage:

Main body volume $= 143\,m^3$

Appendage volume $= 10\,m^3$

Total volume $= 153\,m^3$

Main body moment below 1 WL $= 143 \times 0.654 = 93.52\,m^4$

Appendage moment below 1 WL $= 10(1.5 + 0.1) = \underline{16.00\,m^4}$

$109.52\,m^4$

$$\text{Whole body VCB below 1 WL} = \frac{109.52}{153} = 0.716\,m$$

The value of a sketch, however simple, cannot be over-emphasized in working through examples.

All of these examples are worked out by slide rule. The number of digits worked to in any number should be pruned to be compatible with this level of accuracy.

EXAMPLE 6. Calculate by the trapezoidal rule the area of a transverse half section of a tanker bounded by the following waterline offsets. Waterlines are 3 m apart:

WL	1	2	3	4	5	6
Offset (m)	24.4	24.2	23.7	22.3	19.1	3.0

Compare this with the areas given by Simpson's rules,

(*a*) with six ordinates
(*b*) with five and a half ordinate equal to 16.0 m

Calculate for (*b*) also the vertical position of the centre of area.

Solution:

WL	Offset, y	Trap. mult.	$f_1(y)$	Simp. mult.	$f_2(y)$
1	24.4	$\frac{1}{2}$	12.2	0.4	9.8
2	24.2	1	24.2	1	24.2
3	23.7	1	23.7	1	23.7
4	22.3	1	22.3	1	22.3
5	19.1	1	19.1	1	19.1
6	3.0	$\frac{1}{2}$	1.5	0.4	1.2
			103.0		100.3

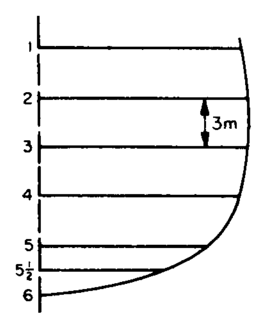


Fig. 2.39

Area by trapezoidal rule $= 3 \times 103.0 = 309.0\,m^2$

Area by Simpson's rule $= \frac{25}{24} \times 3 \times 100.3 = 313.4\,m^2$

WL	Offset, y	S.M.	$f(y)$	Lever, x	$f(xy)$
1	24.4	1	24.4	0	0
2	24.2	4	96.8	1	96.8
3	23.7	2	47.4	2	94.8
4	22.3	4	89.2	3	267.6
5	19.1	$1\frac{1}{2}$	28.65	4	114.6
$5\frac{1}{2}$	16.0	2	32.0	$4\frac{1}{2}$	144.0
6	3.0	$\frac{1}{2}$	1.5	5	7.5
			319.95		725.3

$$\text{Area} = \tfrac{1}{3} \times 3 \times 319.95 = 319.95\,\text{m}^2$$

$$\text{Centre of area below 1 WL} = \frac{725.3}{319.95} \times 3.0 = 6.80\,\text{m}$$

EXAMPLE 7. An appendage to the curve of areas abaft the after perpendicular, ordinate 21, is fair with the main body curve and is defined as follows:

Ordinate	20	21	22
Area (m²)	114.6	110.0	15.0

The ordinates are equally spaced 18 m apart. Calculate the volume of the appendage and the longitudinal position of its centre of buoyancy.

Solution:

Ord. no.	Area, A	S.M.	$f(A)$	S.M.	$f_M(A)$
20	114.6	−1	−114.6	−1	−114.6
21	110.0	8	880.0	10	1100.0
22	15.0	5	75.0	3	45.0
			840.4		1030.4

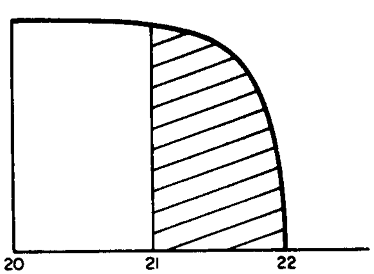


Fig. 2.40

$$\text{Volume of appendage} = \tfrac{1}{12} \times 18 \times 840.4 = 1260.6\,\text{m}^3$$

$$\text{c.b. from 22 ordinate} = \tfrac{1}{24} \times 18^2 \times 1030.4 \times \tfrac{12}{18 \times 840.8}$$

$$= \frac{18 \times 1030.4}{2 \times 840.8} = 11.03\,\text{m}$$

$$\text{c.b. abaft the AP} = 18.0 - 11.03 = 6.97\,\text{m}$$

EXAMPLE 8. It is necessary to calculate the volume of a wedge of fluid immersed by the rotation of a vessel through 20 degrees. The areas and distances of the centres of areas from the axis of rotation of the immersed half water planes have been calculated at 5 degree intervals as follows:

Angle of inclination (deg)	0	5	10	15	20
Area (m²)	650	710	920	1030	1810
Centre of area from axis (m)	3.1	3.2	3.8	4.8	6.0

Calculate the volume of the immersed wedge.

Solution: The Theorem of Pappus Guldinus states that the volume of a solid of revolution is given by the area of the plane of revolution multiplied by the distance moved by its centre of area. If r is the distance of the centre of area of a typical plane from the axis, for a rotation $\delta\theta$, it moves a distance $r\,\delta\theta$ and the volume traced out by the area A is

$$\delta V = Ar\,\delta\theta \quad (\theta \text{ in radians})$$

The total volume is therefore given by

$$V = \int Ar\,\mathrm{d}\theta$$

This time, therefore, an approximate integration has to be performed on Ar over the range of θ.

Ord.	A	r	Ar	S.M.	$f(Ar)$
0	650	3.1	2015	1	2015
5	710	3.2	2272	4	9088
10	920	3.8	3496	2	6992
15	1030	4.8	4944	4	19776
20	1810	6.0	10860	1	10860
					48731

$$\text{Volume} = \tfrac{1}{3} \times 5 \times \tfrac{\pi}{180} \times 48731 = 1420\,\text{m}^3$$

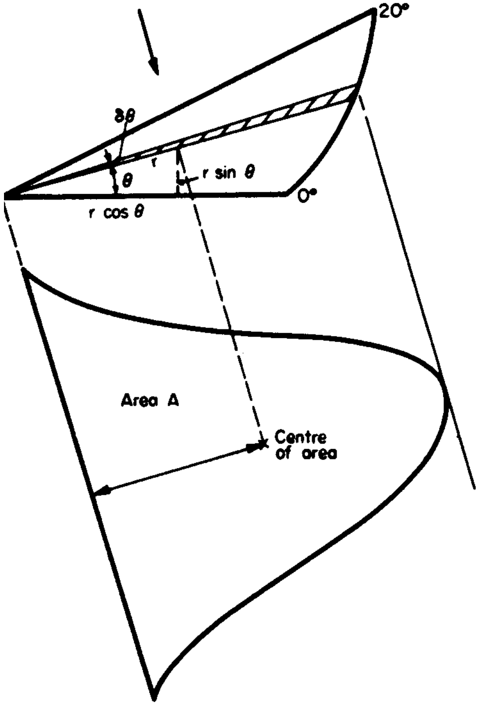


Fig. 2.41

Problems

1. A ship, 200 m between perpendiculars, has a beam of 22 m and a draught of 7 m. If the prismatic coefficient is 0.75 the area of the waterplane 3500 m^2 and mass displacement in salt water is 23,000 tonnes, estimate

   (a) block coefficient
   (b) waterplane coefficient
   (c) midship section coefficient and
   (d) the distance of the centre of buoyancy above the keel.

2. The length, beam and mean draught of a ship are respectively 115, 15.65 and 7.15 m. Its midship section coefficient is 0.921 and its block coefficient is 0.665. Find

   (a) displacement in tonnef and newtons in salt water
   (b) area of immersed midship section
   (c) prismatic coefficient of displacement.

3. Two similar right circular cones are joined at their bases. Each cone has a height equal to the diameter of its base. The composite body floats so that both apexes are in the water surface. Calculate

   (a) the midship section coefficient
   (b) the prismatic coefficient
   (c) the waterplane coefficient.

4. A curve has the following ordinates, spaced 1.68 m apart: 10.86, 13.53, 14.58, 15.05, 15.24, 15.28, 15.22 m. Calculate the area by Simpson's first rule and compare it with the area given by the trapezoidal rule. What is the ratio of the two solutions?
5. The half ordinates of the load waterplane of a vessel are 1.2, 4.6, 8.4, 11.0, 12.0, 11.7, 10.3, 7.5 and 3.0 m respectively and the overall length is 120 m. What is its area?
6. A curvilinear figure has the following ordinates at equidistant intervals: 12.4,27.6,43.8, 52.8,44.7,29.4 and 14.7. Calculate the percentage difference from the area found by Simpson's first rule when finding the area by *(a)* the trapezoidal rule, *(b)* Simpson's second rule.
7. The effective girths of the outer bottom plating of a ship, 27.5 m between perpendiculars, are given below, together with the mean thickness of plating at each ordinate. Calculate the volume of the plating. If the plating is of steel of mass density 7700 kg/m^3, calculate the weight in meganewtons.

Ord. No.	AP	10	9	8	7	6	5	4	3	2	FP
Girth (m)	14.4	22.8	29.4	34.2	37.0	37.4	36.8	28.6	24.2	22.6	23.2
Thickness (mm)	10.2	10.4	10.6	11.4	13.6	13.6	12.8	10.4	10.1	10.1	14.2

8. The half ordinates of a vessel, 144 m between perpendiculars, are given below.

Ord. No.	AP	8	7	6	5	4	3	2	FP
!Ord. (m)	17.0	20.8	22.4	22.6	21.6	18.6	12.8	5.6	0.0

   In addition, there is an appendage, 21.6 m long, abaft the AP, whose half ordinates are:

Ord. No.	12	11	10	AP
!Ord. (m)	0.0	9.6	14.0	17.0

   Find the area and position of the centre of area of the complete waterplane.

9. The loads per metre due to flooding, at equally spaced positions on f!f transverse bulkhead are given below. The bulkhead is 9.5 m deep. Calculate the total load on the bulkhead and the position of the centre of pressure

Ord. No.	1	2	3	4	5	6	7	8	9	10
Load (tonnef/m)	0	15	30	44	54	63	69	73	75	74

10. A tank is 8 m deep throughout its length and 20 m long and its top is flat and horizontal. The sections forward, in the middle and at the after end are all triangular, apex down and the widths of the triangles at the tank top are respectively 15, 12 and 8 m.

    Draw the calibration curve for the tank in tonnes of fuel against depth and state the capacity when the depth of oil is 5.50 m. SG of oil fuel = 0.90. Only five points on the curve need be obtained.

11. Areas of waterplanes, 2.5 m apart, of a tanker are given below.
Calculate the volume of displacement and the position of the VCB. Compare the latter with the figure obtained from Normand's (or Morrish's) rule.

Waterplane	1	2	3	4	5	5!	6
Area(mi	4010	4000	3800	3100	1700	700	200

12. The waterline of a ship is 70 m long. Its half ordinates, which are equally spaced, are given below. Calculate the least second moment of area about each of the two principal axes in the waterplane.

Ord. No.	1	2	3	4	5	6	7	7!	8
! Ord. (m)	0.0	3.1	6.0	8.4	10.0	10.1	8.6	6.4	0.0

13. The half ordinates of the waterplane of a ship, 440 m between perpendiculars, are given below. There is, in addition, an appendage abaft the AP with a half area of 90 m² whose centre of area is 8 m from the AP; the moment of inertia of the appendage about its own centre of area is negligible.
Calculate the least longitudinal moment of inertia of the waterplane.

Section	AP	10	9	8	7	6	5	4	3	2	FP
! Ord. (m)	6.2	16.2	22.5	26.0	27.5	27.4	23.7	19.2	14.5	8.0	0.0

14. The half breadths of the 16 m waterline of a ship which displaces 18,930 tonnef in salt water are given below. In addition, there is an appendage abaft the AP, 30 m long, approximately rectangular with a half breadth of 35.0 m. The length BP is 660 m.
Calculate the transverse -BM- and the approximate value of -K-M-.

Station	FP	2	3	4	5	6	7	8	9	10	AP
! Breadth (m)	0.0	21.0	32.0	37.0	40.6	43.0	43.8	43.6	43.0	40.0	37.0

15. The shape of a flat, between bulkheads, is defined by the ordinates, spaced 4 m apart, given below. If the plating weighs 70 N $1m2$, calculate the weight of the plating and the distance of the c.g. from No.1 ordinate.

Ord. No.	1	2	3	4	5	6
Breadth (m)	53.0	50.0	45.0	38.0	30.0	14.0

16. Each of the two hulls of a catamaran has the following dimensions.

Ord. No.	1	2	3	4	5	5!	6
! Ord. (cm)	0.0	4.0	6.2	7.2	6.4	4.9	0.0

The length and volume of displacement of each hull are respectively 18 m and 5.3 m³. The hull centre lines are 6 m apart. Calculate the transverse -BM- of the boat.

17. Compare the areas given by Simpson's rules and the trapezoidal rule for the portions of the curve defined below:

(a) between ordinates 1 and 4
(b) between ordinates 1 and 2

Ord. No.	1	2	3	4
Ord. (m)	39.0	19.0	12.6	10.0

The distance between ordinates is 10m.

18. Apply Normand's (or Morrish's) rule to a right circular cylinder floating with a diameter in the waterplane. Express the error from the true position of the VCB as a percentage of the draught.
19. Deduce a trapezoidal rule for calculating longitudinal moments of area.
20. Deduce the five ordinate rules of *(a)* Newton-Cotes, *(b)* Tchebycheff.
21. Compare with the correct solution to five decimal places J; sin x dx by the three ordinate rules of *(a)* Simpson, *(b)* Tchebycheff, *(c)* Gauss.
22. A quadrant of 16 m radius is divided by means of ordinates parallel to one radius and at the following distances: 4, 8, 10, 12, 13, 14 and 15m. The lengths of these ordinates are respectively: 15.49, 13.86, 12.49, 10.58, 9.33, 7.75 and 5.57m.
Find:

(a) the area to two decimal places by trigonometry
(b) the area using only ordinates 4m apart by Simpson's rule
(c) the area using also the half ordinates
(d) the area using all the ordinates given
(e) the area using all the ordinates except 12.49.

23. Calculate, using five figure tables, the area of a semicircle of 10 m radius by the four ordinate rules of *(a)* Simpson, *(b)* Tchebycheff, *(c)* Gauss and compare them with the correct solution.
24. Show that -K-B- is approximately T/6(5 - 2Cvp)·
25. From strains recorded in a ship during a passage, the following table was deduced for the occurrence of stress maxima due to ship motion. Calculate for this data *(a)* the mean value, *(b)* the standard deviation.

Max. stress *(MN/m²)*	10	20	30	40	50	60
Occurrences	852	1335	772	331	140	42

26. Construct a probability curve from the following data of maximum roll angle from the vertical which occurred in a ship crossing the Atlantic. Wha~ are *(a)* the mean value, *(b)* the variance, *(c)* the probability of exceeding ~ roll of 11 degrees

Max. roll angle, deg.	1	3	5	7	9	11	13	15	17
Occurrences	13,400	20,550	16,600	9720	4420	1690	510	106	8

3 Flotation and trim

A ship, like any other three-dimensional body, has six degrees of freedom. That is to say, any movement can be resolved into movements related to three orthogonal axes, three translations and three rotations. With a knowledge of each of these six movements, any combination movement of the ship can be assessed. The three principal axes have already been defined in Chapter 2. This chapter will be confined to an examination of two movements

(*a*) behaviour along a vertical axis, O*z* in the plane O*xz*
(*b*) rotation in the plane O*xz* about a transverse horizontal axis, O*y*.

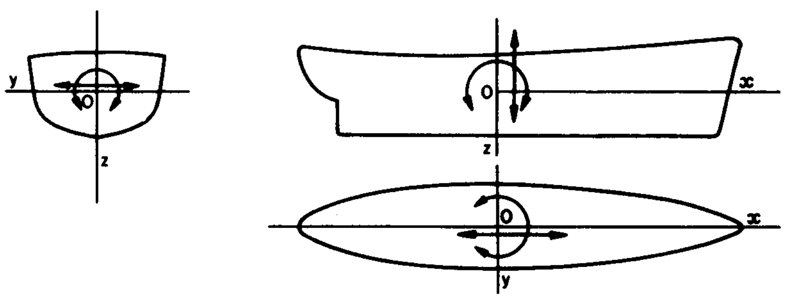


Fig. 3.1

Attention will be confined to static behaviour, i.e. conditions applying when the ship is still. Generally, it is the change from one static condition to another that will be of interest and so it is convenient to imagine any movement occurring very slowly. Dynamic behaviour, involving time, motion and momentum will be dealt with in later chapters.

Flotation

PROPERTIES OF FLUIDS

The *mass density* of a fluid ρ, is the mass of the fluid per unit volume. The *weight density* w, of a fluid is the weight of the fluid per unit volume. In SI units, $w = \rho g$ so that, if ρ is in kg/m^3, w is in $newtons/m^3$. Since they vary with pressure and temperature, the values must be related to a standard condition of pressure and temperature. The former is normally taken to be one atmosphere, $10^5\,Pa = 1\,Bar$ and the latter sometimes 15 °C and for water sometimes 4 °C when its density is a maximum.

The ratio of the density of a solid or a liquid to the density of pure water is the *specific gravity*, γ. Since it is the basic reference for all such materials, the weight properties of pure distilled water are reproduced in Fig. 9.1.

The inverse of the weight density is called the *reciprocal weight density* u, or *specific volume*. The value for salt water is $0.975\,m^3/tonnef$ or $99.5\,m^3/MN$. Corrections are applied for variations of reciprocal weight density from this value. Table 3.1 gives values of mass density for common fluids and for steel, air and mahogany.

Table 3.1
Properties of some common materials

Material	Mass density, ρ (kg/m^3)	Reciprocal mass density (m^3/Mg)	Specific gravity, γ
Fresh water (standard)	1000	1.00	1.00
Fresh water (British preferred value)	996	1.00	1.00
Salt water	1025	0.975	1.03
Furnace fuel oil	947	1.05	0.95
Diesel oil	841	1.19	0.84
Petrol	697	1.44	0.70
Steel	7689	0.13	7.70
Mahogany	849	1.18	0.85
Air	1.293	774.775	—

The reciprocal weight density is found merely by inverting the weight density and adjusting the units.

In SI units, the reciprocal weight density u must be expressed in m^3/MN which involves g. Hence, weight density reciprocal

$$u = \frac{1}{\rho}\frac{m^3}{kg}\;\frac{s^2}{9.807\,m}\;\frac{kg\,m}{s^2\,newton}\;\frac{10^6\,N}{MN}$$
$$= \frac{10^6}{9.807\rho}\,m^3/MN \quad \text{with } \rho \text{ in } kg/m^3$$

for steel, for example,

$$u = \frac{10^6}{9.807 \times 7689} = 13.26\,m^3/MN$$

The student is advised always to write in the units in any calculation to ensure that they cancel to the required dimensions.

ARCHIMEDES' PRINCIPLE

This states that when a solid is immersed in a liquid, it experiences an upthrust equal to the weight of the fluid displaced.

Thus, the tension in a piece of string by which a body is suspended, is reduced when the body is immersed in fluid by an amount equal to the volume of the body times the weight density of the fluid; a diver finds an article heavier to lift

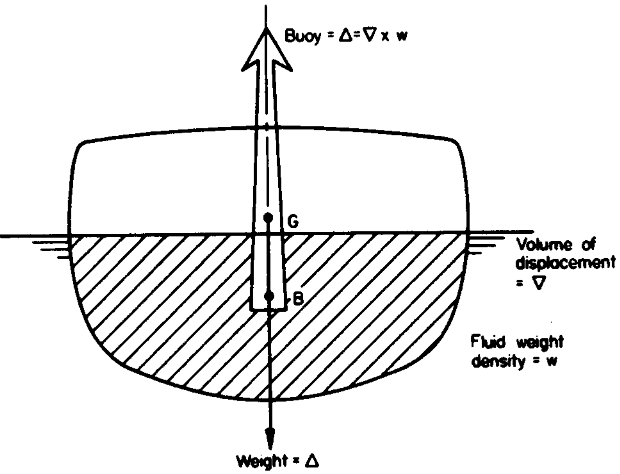


Fig. 3.2

out of water than under it, by an amount equal to its volume times the weight density of water. This upthrust is called the buoyancy of the object. **If,** by chance, the body has the same weight density as the fluid, the upthrust when it was totally immersed would be equal to its weight; the string would just go limp and the diver would find the object to be apparently weightless. **If** the body were to have a smaller weight density than the fluid, only sufficient of the body to cause an upthrust equal to its weight could be immersed without force; if the body is pushed further down the buoyancy exceeds the weight and it bobs up, like a beach ball released from below its natural position in the sea.

This leads to a corollary of Archimedes' principle known as the *Law of Flotation.* When a body is floating freely in a fluid, the weight of the body equals the buoyancy, which is the weight of the fluid displaced.

The *buoyancy* of a body immersed in a fluid is the vertical upthrust it experiences due to displacement of the fluid (Fig. 3.2). The body, in fact, experiences all of the hydrostatic pressures which obtained before it displaced the fluid.

The buoyancy is the resultant of all of the forces due to hydrostatic pressure on elements of the underwater portion (Fig. 3.3). Now, the hydrostatic pressure at a point in a fluid is equal to the depth of the point times the weight density of the fluid, i.e. it is the weight of a column of the fluid having unit cross-section and length equal to the depth of immersion, T

$$p = Tw$$

Let us examine the pressure distribution around a rectangular block $a \times b \times c$ floating squarely in a fluid at a draught T. The pressures on the vertical faces of the block all cancel out and contribute nothing to the vertical resultant; the hydrostatic pressure at the bottom face is Tw and so the total vertical upthrust is this pressure multiplied by the area:

$$\text{upthrust} = (Tw)ab$$

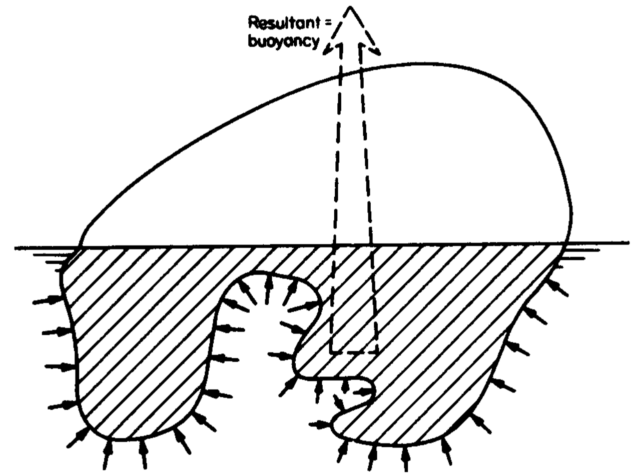


Fig. 3.3

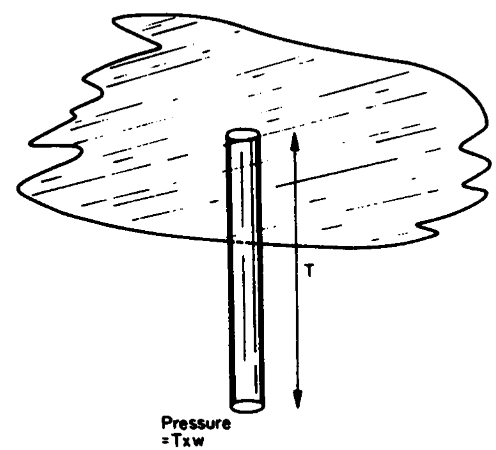


Fig. 3.4 Hydrostatic pressure at a point

But this is the displaced volume, abT, times the weight density of the fluid, w, which is in accordance with the law of flotation.

In the general case, imagine a body floating freely in a fluid as shown in Fig. 3.3. The body is supported by the summation of all the pressure forces acting on small elements of the surface area of the body. Now imagine the body removed and replaced by a thin film of material with the same surface shape, and further imagine that the interior volume created by this film is filled with the same fluid as that in which the body is floating. If the film has negligible weight, a state of equilibrium will be produced when the level of
the outside level. Thus, the forces acting on the film from
support those acting on the film from the inside, i.e. the
displaced by the body.

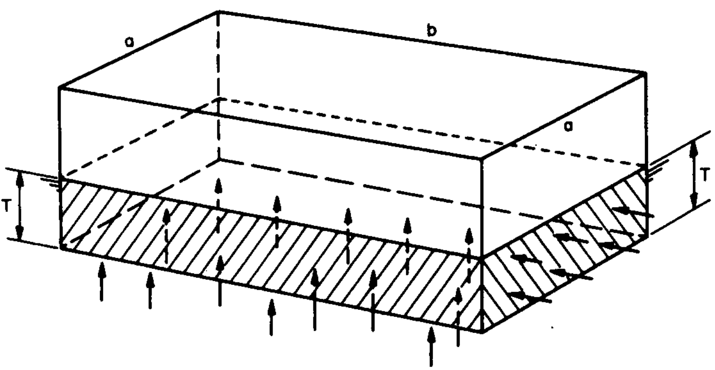


Fig. 3.5

Buoyancy is plainly a force. It is the upthrust caused by displacement of fluid. For very many years naval architects have used the terms *buoyancy* and *displacement* interchangeably and the latter has become an alternative word for the upthrust of the fluid upon a floating vessel. Both weight and buoyancy vary in the same way with gravity so that anywhere in the world, the ship will float at exactly the same waterline in water of the same density. For static considerations we are interested wholly in forces and weights and so the concept of a *force displacement* Δ identical to buoyancy is satisfactory and convenient and is retained in this book.

The same is not quite true for the study of dynamic behaviour of a vessel which depends upon mass rather than weight. It is necessary to introduce the concept of *mass displacement* Σ. However it is force which causes change and normally the single word displacement refers to a force and is defined by the symbol Δ. Thus:

$$\Delta = g\Sigma = \rho g\nabla = w\nabla$$

Retention of the metric tonneforce (tonnef) seems likely for many years yet in rule-of-thumb practice, although students should find no difficulty in dealing with displacement in terms of newtons. The tonnef is the force due to gravity acting on a mass of one tonne.

It is convenient thus to work in terms of the mass of a ship $\Sigma = n$ tonnes which, under gravity leads to a force displacement, $\Delta = \Sigma g = n \times g = n$ tonnef.

With an understanding of the relationship of buoyancy and weight, the wonder at why steel objects can float diminishes. It is natural to expect a laden cargo ship to wallow deeply and a light ballasted ship to tower high, and we now know that the difference in buoyancy is exactly equal to the difference in loading. An important and interesting example is the submarine; on the surface, it has a buoyancy equal to its weight like any other floating body. When submerged, sufficient water must have been admitted to make weight and buoyancy roughly equal, any small out-of-balance force being counteracted by the control surfaces. A surface effect machine, or cushion craft obeys Archimedes' principle when it hovers over the sea; the indentation of the water beneath the machine has a volume which, multiplied by the weight density of the sea, equals the weight of the craft (ignoring air momentum effects).

The watertight volume of a ship above the water line is called the *reserve of buoyancy*. It is clearly one measure of the ship's ability to withstand the effects of flooding following damage and is usually expressed as a percentage of the load displacement.

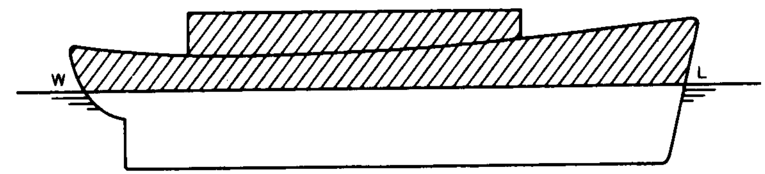


Fig. 3.6 Reserve of buoyancy

VERTICAL MOVEMENT

Figure 3.2 shows the forces acting upon a floating body which are

(*a*) the weight, vertically downwards, which may be taken for static considerations as acting as if it were all concentrated at the centre of gravity, as for any rigid body;

(*b*) The buoyancy, vertically upwards, which may be assumed concentrated at the centre of buoyancy, which is the centre of volume of the underwater shape.

It must be made clear that when the ship is still, the weight and buoyancy forces must act in the same straight line BG, otherwise a couple would act upon the ship, causing it to change its attitude. What happens when a small weight is placed on the vertical line through BG? (Strictly speaking, over the centre of flotation as will become clear later.) Clearly the vessel sinks—not completely but by an amount so that the additional buoyancy equals the additional weight and vertical equilibrium is again restored. The ship undergoes a parallel sinkage having a buoyancy W and the centre of buoyancy B moves towards the addition by an amount $\overline{BB'}$. Taking moments about B

$$W\overline{Bb} = (\Delta + W)\overline{BB'}$$

$$\therefore \quad \overline{BB'} = \frac{W\overline{Bb}}{\Delta + W} \qquad (1)$$

Thus, the new buoyancy $(\Delta + W)$ acts now through the new centre of buoyancy B′, whose position has just been found. It has been assumed that there is no trim; when there is, Fig. 3.7 shows the projections on tó a transverse plane and the same result holds.

In the same way, the ship has a new centre of gravity. Taking moments about G (see Fig. 3.8)

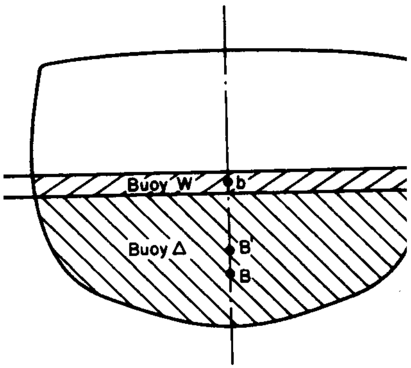


Fig. 3.7

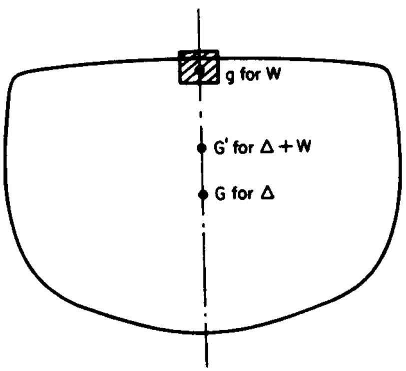


Fig. 3.8

$$W\overline{\text{Gg}} = (\Delta + W)\overline{\text{GG}'}$$

$$\therefore \quad \overline{\text{GG}'} = \frac{W\overline{\text{Gg}}}{\Delta + W}$$

Thus, the new weight (+ W) acts now through the new centre of gravity G'. whose position has just been found. Note the similarity of the expressions.

Equation (1) involves a knowledge of the position of b, the centre of buoyancy of the added layer of buoyancy. If the layer is thick, it must be determined from the ship geometry-in fact, it is probably as quick to determine the position of B directly from the shape of the whole ship. Frequently, however, additions are small in comparison with the total displacement and b can be taken halfway up a slice assumed to have parallel sides. It is convenient to find the approximate thickness of this slice by using a device called the *tonne! per centimetre immersion* (TPC) or tonf (or tonnef) parallel immersion (TPI). This latter has the merit of retaining a long established set of initials. When TPI is used in this book it has this meaning.

In old metric units, the weight required to effect a parallel sinkage of one metre is

$$A_W \text{m}^2 \rho \frac{\text{kg}}{\text{m}^3} \frac{\text{Mg}}{1000\,\text{kg}} \frac{\text{g}}{\text{g}}$$

and TPC $= \rho A_W 10^{-5}$ tonnef per cm with A_W in m^2 and ρ in kg/m^3. For salt water, $\rho = 1025\,\text{kg/m}^3$, whence TPC $= 0.01025\, A_W$.

In SI units, meganewtons per metre immersion

$$= A_W \text{m}^2 \rho \frac{\text{kg}}{\text{m}^3} \frac{9.807\,\text{m}}{\text{s}^2} \frac{\text{newton}\,\text{s}^2}{\text{kg}\,\text{m}} \frac{\text{MN}}{10^6\,\text{N}}$$

$$= 9.807 \rho A_W 10^{-6}$$

for which salt water becomes $0.01005\, A_W$ MN/m.

Of course, arguments are completely reversed if weights are removed and there is a parallel rise. What happens if the weight of the ship remains the same and the density of the water in which it is floating is changed? Let us examine, at first, a body of displacement Δ floating in water of weight density w_1 which passes into water of lower weight density w_2. It will sink deeper because the water is less buoyant. The weight and buoyancy have not changed because nothing has been added to the body or taken away.

$$\therefore \quad \Delta = \nabla_1 w_1 = \nabla_2 w_2$$

$$\frac{\nabla_1}{\nabla_2} = \frac{w_2}{w_1}$$

$$\text{The volume of the layer} = \nabla_2 - \nabla_1$$

$$= \nabla_2 \left(1 - \frac{w_2}{w_1}\right)$$

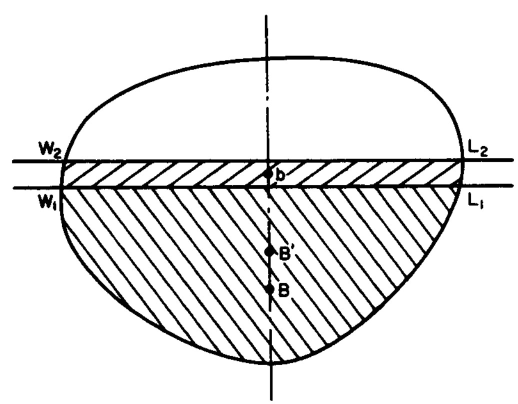


Fig. 3.9

Now, this layer is made up of water of weight density w_2

$$\therefore \text{ weight of layer} = \nabla_2 w_2 \left(1 - \frac{w_2}{w_1}\right)$$
$$= \Delta\left(1 - \frac{w_2}{w_1}\right)$$

The approximate thickness of this layer is given by this expression divided by the TPC or TPI in w_2 water; if the TPI is known in w_1 water,

$$\text{TPI}_2 = \text{TPI}_1 \frac{w_2}{w_1} = \text{TPI}_1 \frac{u_1}{u_2}$$

The approximate thickness of the layer is also, of course, given by its volume above, divided by the waterplane area

$$A_W = \text{TPI}_1 u_1 = \frac{\text{TPI}_1}{w_1}$$

Taking first moments of the volume about B the rise of the centre of buoyancy,

$$\overline{\text{BB}^1} = \frac{(\nabla_2 - \nabla_1)\overline{\text{Bb}}}{\nabla_2}$$
$$= \left(1 - \frac{w_2}{w_1}\right)\overline{\text{Bb}}$$

It has been assumed that the increases in draught are small and that TPI is sensibly constant over the change. There are occasions when a greater accuracy is needed. What is the relationship between the change in displacement and TPI? It will be remembered from Chapter 2 (Fig. 2.19) that the change in volume of displacement.

$$\text{change in } \nabla = \int_{\text{WL}_1}^{\text{WL}_2} A_W \, \mathrm{d}T$$

$$\therefore \text{ change in } \Delta = \int_{\text{WL}_1}^{\text{WL}_2} \frac{A_W}{u} \, \mathrm{d}T$$

Writing this another way,

$$\frac{\mathrm{d}\Delta}{\mathrm{d}T} = \text{TPI} = \text{TPC}$$

With an adjustment for units, the slope of the displacement curve with respect to draught, gives the TPC; displacement is the integral curve of TPC, i.e. displacement is represented by the area under the curve of TPC plotted against draught.

Displacement of a ship is calculated with some accuracy during its early design to a series of equally spaced waterlines. Although the curve of

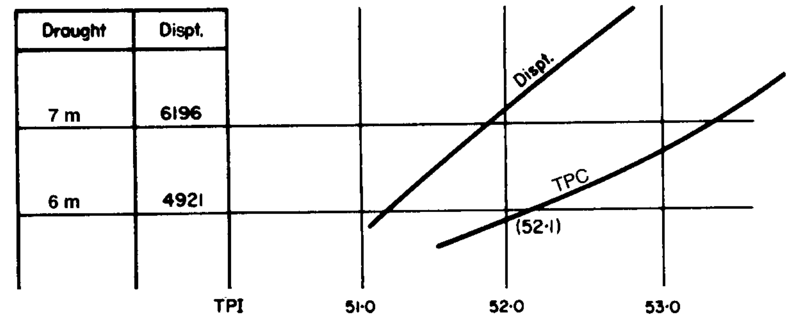


Fig. 3.10

displacement is plotted, it cannot be read to the required accuracy at intermediate waterlines. To find the displacement at an intermediate waterline, therefore, to the nearest tabulated displacement is added (or subtracted) the integral of the TPC curve between the tabulated waterline and the required waterline. For example, suppose Fig. 3.10 shows an extract of the curves of displacement and TPC at the 6m and 7m waterlines. The displacement at say 6.7m cannot accurately be read from the displacement curve: TPC can be read with sufficient accuracy at say, 6.0m, 6.35m and 6.7m, integrated by Simpson's first rule and the displacement of the slice added. Worked example 4, illustrates this.

Trim

Trim is the difference in draughts forward and aft. An excess draught aft is called trim *by the stern,* while an excess forward is called trim *by the bow.* It is important to know the places at which the draughts are measured and trim, unless it is obvious, is usually referred to *between perpendiculars* or *between*

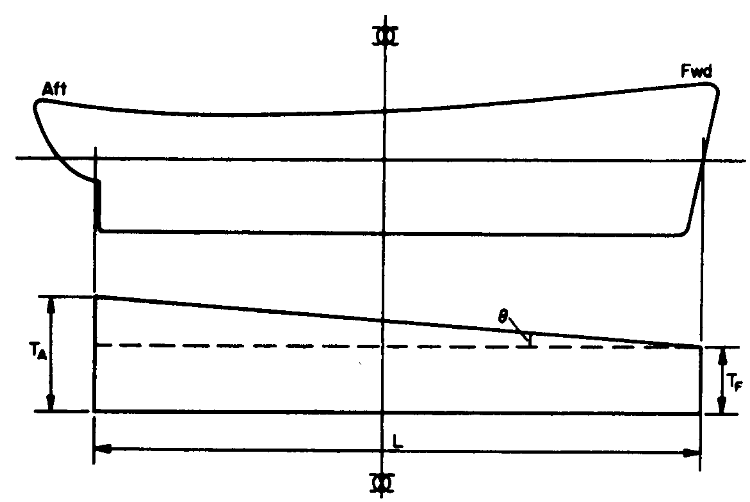


Fig. 3.11 Trim

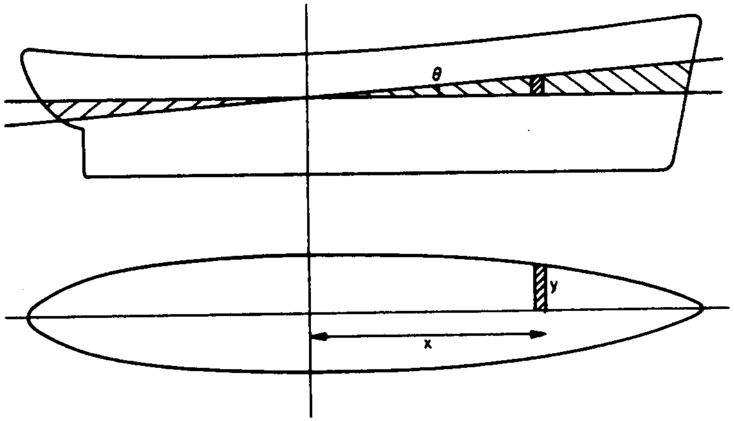


Fig. 3.12

marks. Used as a verb, trim refers to the act of angular rotation about the Oy-axis, from one angular position to another. There is also another important use of the word trim which must be excluded for the time being; in relation to submarines, trim is also the relationship between weight and buoyancy. So far as surface ships are concerned,

$$\text{trim} = T_A - T_F$$

$$\text{and the angle of trim } \theta = \frac{T_A - T_F}{L}$$

where L is the horizontal distance between the points at which T_A and T_F are measured.

If a ship is trimmed without change of displacement, it must rotate about the centre of flotation. This can be shown by writing down the condition that there shall be no change in displacement, i.e. that the volumes of emerged and immersed wedges are equal (Fig. 3.12)

$$2\int y_F(x_F\theta)\,dx = 2\int y_A(x_A\theta)\,dx$$

i.e. $\int xy\,dx$ forward $= \int xy\,dx$ aft, which is the condition for the centre of area of the waterplane.

A ship trims, therefore, about the centre of flotation of the waterplane. If a small weight is added to the ship, to avoid trim it must be added over the centre of flotation; the centre of buoyancy of the additional layer will also be at the centre of flotation and there is no out-of-balance moment (Fig. 3.13).

CHANGES OF DRAUGHT WITH TRIM

Because a ship trims about the centre of flotation, the draught at that position does not alter with trim. It does alter everywhere else, including amidships where the mean draught between perpendiculars occurs. Now, the displacement of a ship is recorded for different mean draughts but at a specified design trim.

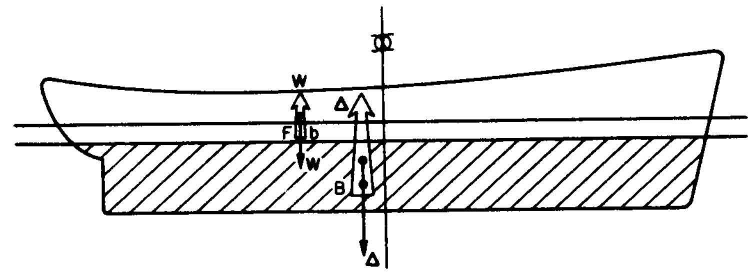


Fig. 3.13 Addition of weight at the centre of flotation

Should the ship be floating at a different trim, it is necessary to imagine the ship being trimmed about the centre of flotation (which does not alter the displacement) until it is brought to the designed trim; the revised mean draught is calculated and so the displacement pertinent to that draught can be found. Suppose that a ship is floating t out of designed trim at waterline WL and needs to be brought to the designed trim W_1L_1 (Fig. 3.14). The change of trim $t = W_1W + LL_1 = NL_1$. By the principle of similar triangles, the correction to the mean draught,

$$d = \frac{a}{L}t$$

where a is the distance of the CF from amidships. From this revised mean draught, the displacement can be found from the ship's hydrostatic data. Had the displacement been found first from the hydrostatic data for the ship floating

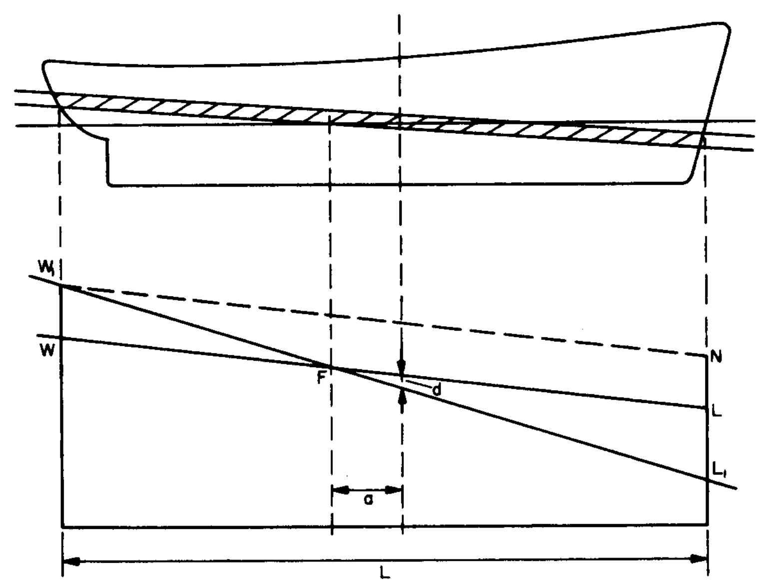


Fig. 3.14

at the actual mean draught, it would have been necessary to correct the displacement for the amount out of designed trim ; this is clearly the displacement of a layer of thickness at/L which is

$$\text{trim correction} = \frac{at}{L} \cdot \text{TPI}$$

Whether this is positive or negative depends upon the position of the CF and whether the excess trim is by bow or stern. Instead of attempting to remember rules to govern all cases, the student is advised to make a sketch similar to Fig. 3.14 on every occasion. For the case drawn, when the CF is abaft amidships and there is an excess trim by the bow, the trim correction is negative, i.e. the displacement given by the hydrostatic data for the actual mean draught to waterline WL is too great.

The principle of similar triangles is especially useful in calculating draughts at various positions along the ship, e.g. at the draught marks which are rarely at the perpendiculars. To calculate the effect of changes of trim on draughts at the marks, a simple sketch should be made to illustrate what the change at the marks must be; in Fig. 3.15, a change of trim t between perpendiculars causes a change of draught at the forward marks of bt/L and at the after marks of ct/L. Remember, all distances are measured from the CF!

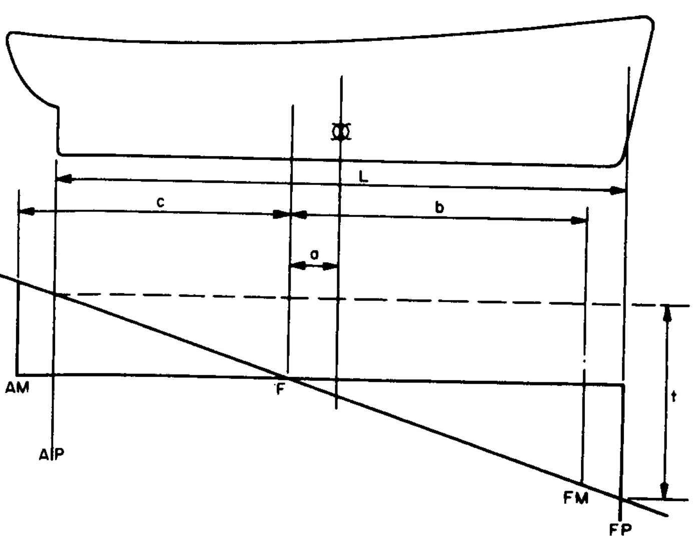


Fig. 3.15

MOMENT CAUSING TRIM

The trim rotation θ is caused by a moment M. What is the relationship between M and θ? In Chapter 2, we examined the rotation of any floating body, and discovered that it appeared to rotate about a point called the metacentre which was at a height I/∇ above the centre of buoyancy. Examine

now the particular case of rotation in the longitudinal vertical plane. Let us apply a moment M at the CF causing a rotation θ, immersing a wedge of water aft and causing a wedge to emerge forward. The change in underwater shape due to the movement of the wedges, causes B to move aft to, say, B_1. The buoyancy Δ acts vertically upwards, at right angles to the new waterline; the weight continues to act vertically downwards through the centre of gravity G which has not moved. The application of a moment M therefore has caused a couple $\Delta\overline{GZ}$, $\overline{GZ}$ being the perpendicular distance between the lines of buoyancy and weight, i.e.

$$M = \Delta\overline{GZ}$$

For small angles, $\sin\theta$ is approximately equal to θ

$$\therefore\ M = \Delta\overline{GM}_L\theta$$

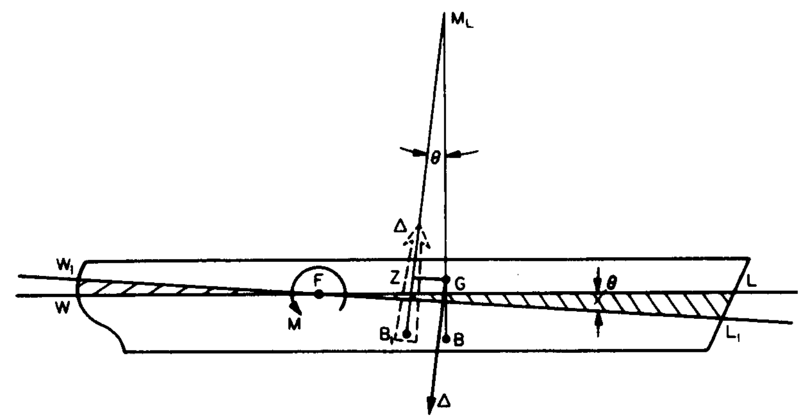


Fig. 3.16 Moment applied at the CF

If the angle of trim is such as to cause one metre of trim, $\theta = 1/L$ and the moment to cause one metre of trim is

$$\frac{\Delta\overline{GM}_L}{L} = \text{MCT}$$

This is an important new tool for calculating changes of trim and it is tabulated for each waterline in the hydrostatic data.

Thus, the change of trim in metres, caused by the application of a moment M is given by the moment divided by the moment needed for a metre change of trim. Just as the length over which the trim is measured requires specifying, so does the moment to change trim (MCT) require the length to be stated and, frequently, this is the length between perpendiculars.

In practice, $\overline{GM}_L$ and $\overline{BM}_L$ are both large numbers and close together. ($\overline{BG}$ may be one per cent of $\overline{BM}_L$). An approximation 1 is given by

$$\text{MCT} = \frac{\Delta \overline{\text{BM}}_{\text{L}}}{\text{L}}$$

In metric units, the moment to change trim one metre or the *one metre trim moment* is $\Delta \overline{\text{GM}}_{\text{L}}/L$ tonnef/m with Δ in tonnef or MN m with Δ in MN. For salt water, approximately,

$$\text{One metre trim moment} = \frac{\Delta \overline{\text{BM}}_{\text{L}}}{L} = 1.025\frac{I}{L}\,\text{tonnef/m}$$
$$= 0.01005\frac{I}{L}\,\text{MN m}$$

MCT can be used for the general case, typical values being:

	tonnef/m	MN m
18 m fishing vessel	30	0.3
90 m submarine, surfaced	1800	18
110 m frigate	5500	54
150 m dry cargo vessel	20000	210
240 m aircraft carrier	70000	650
300 m supertanker	200000	1900

ADDITION OF WEIGHT

It is a principle of applied mechanics that, for statical considerations, the effects of a force P acting on a body are exactly reproduced by a parallel force P at a distance h from the original line plus a moment Ph. In other words, the body shown in Fig. 3.17 behaves in precisely the same way under the action of the force and moment shown dotted as it does under the action of the force shown full. This is a useful principle to apply to the addition of a weight to a ship which causes it to sink by a shape awkward to assess. If the weight be assumed

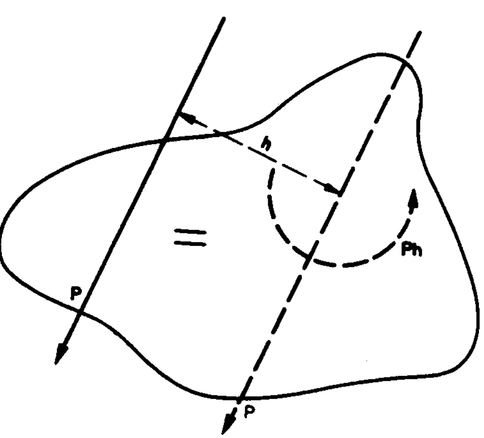


Fig. 3.17

replaced by a force W and a moment Wh at the centre of flotation, the separate effects of each can be readily calculated and added:

(*a*) the force W at the centre of flotation causes a parallel sinkage W/TPI,
(*b*) the moment Wh causes a change of trim Wh/MCT.

The changes of draught at any position of the ship's length can be calculated for each movement and added, algebraically, taking care to note the sign of each change. See worked example 5.

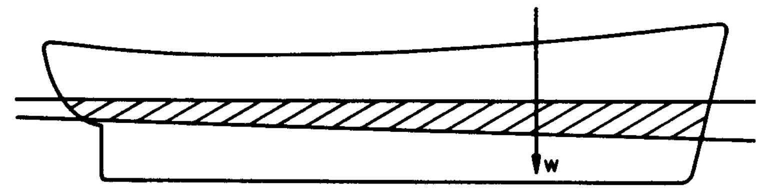


Fig. 3.18

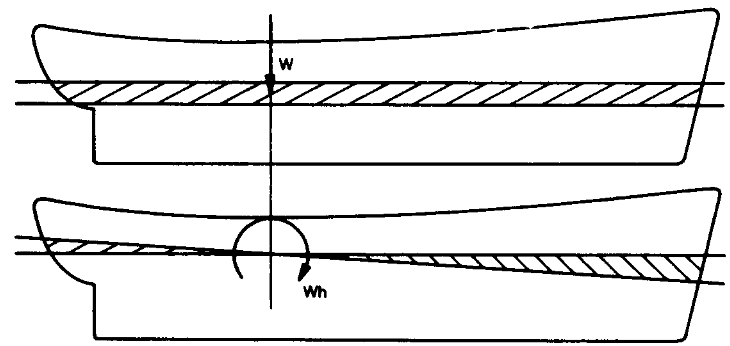


Fig. 3.19

One particular case of added weight (a negative addition, in fact) is worthy of special mention. When a ship grounds or docks, there is an upwards force at the point of contact. This is assumed replaced by a vertical force at the CF plus a moment which cause, respectively, a parallel rise and a trim. What causes the force is of little interest in this context. A fall of water level, equal to the change of draught at the point of contact is calculated from the combined effects of a parallel rise and a trim. Any attempt to treat the problem in one step is likely to meet with failure and the approach of Fig. 3.20 is essential. Worked example 6 illustrates the method.

While we are now in a position to assess the effects of a small added weight, let us return for a moment to the exact behaviour of the ship so that we have a real

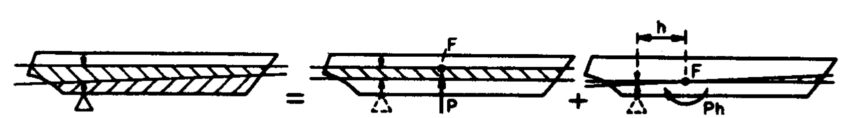


Fig. 3.20

understanding of what is happening. Referring to Fig. 3.21, before the addition of weight W, B, G and M were in a vertical line. The addition of the weight alters the position of all three but they must, of course, finally be in a straight line, a line perpendicular to the new waterline. What are the movements of B, G and M? G must move along a straight line joining G and g and the distance it moves could be calculated by an expression similar to Equation (2) on p. 58. B moves upwards along the straight line Bb_1 due to the parallel sinkage to B_1 and then moves to B_2 such that B_1B_2 is parallel to bb. $\overline{BB_1'}$ can be calculated from Equation (1) on p. 57. M moves up by virtue of B moving up to B_1 and, assuming I constant, down because ∇ has increased; $\overline{BM} = I/\nabla = wI/\Delta$ and $\overline{B_1M_1} = wI/(\Delta + W)$ so that

$$\overline{BM} - \overline{B_1M_1} = wI\left(\frac{1}{\Delta} - \frac{1}{\Delta + W}\right)$$

$$= \frac{wIW}{\Delta(\Delta + W)}$$

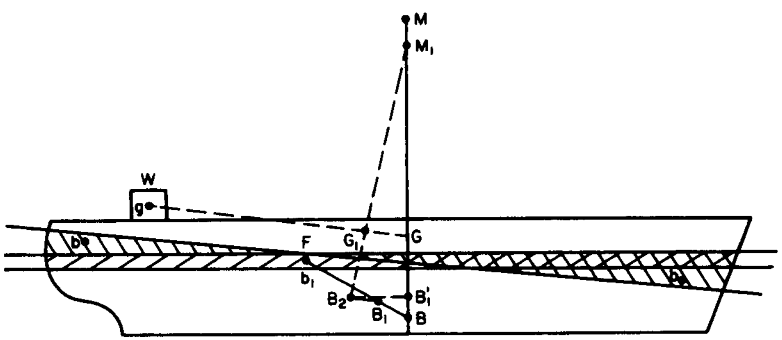


Fig. 3.21

We have assumed that I has not appreciably changed and that the vertical component of $\overline{B_1B_2}$ is negligible. Thus, from the three movements calculated, a drawing could be made and the angle of trim measured. Clearly, the use of MCT is quicker.

LARGE WEIGHT ADDITIONS

While the use of MCT and TP1 or TPC is permissible for moderate additions of weight, the assumptions implicit in their use break down if the addition becomes large or the resulting trim is large. No longer is it possible to assume that I does not change, that TPI is constant or the longitudinal position of the CF unchanged. An approach similar to Fig. 3.21 is essential, viz.:

(*a*) calculate new position of G as already described or by considering the two components,

$$\overline{GG'} = \frac{Wc}{\Delta + W} \quad \text{and} \quad \overline{G'G_1} = \frac{Wd}{\Delta + W}$$

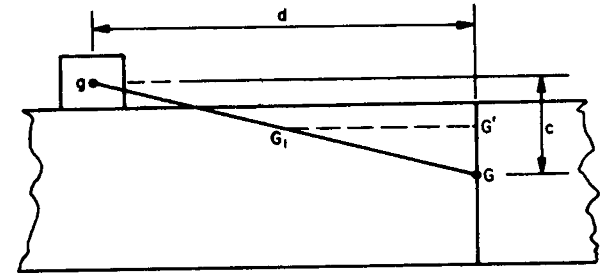


Fig. 3.22 Movement of G

(*b*) calculate new position of B in two steps

(i) the change in B due to the addition of a parallel slab; the displacement of the slab must, of course, be equal to the additional weight and its centre of buoyancy b_1 may be assumed on a line joining the centres of flotation of the two waterplanes at a height determined by the areas of the two waterplanes. Then

$$\overline{BB_1'} = \frac{We}{\Delta + W} \quad \text{and} \quad \overline{B_1'B_1} = \frac{Wf}{\Delta + W}$$

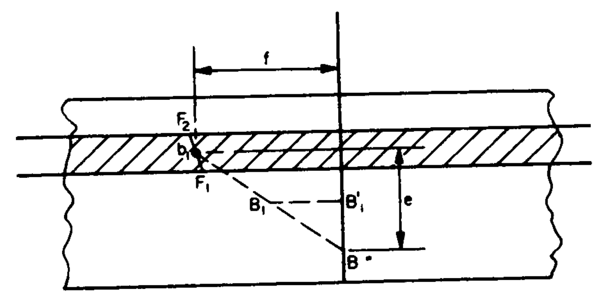


Fig. 3.23 Movement of B

(ii) the change in position of B_1 due to the trimming of the waterplanes (Fig. 3.24). The geometry of the wedges must be examined and the positions of b found by integration or rule. At this stage however, we encounter a difficulty—we do not know what θ is. The calculation is

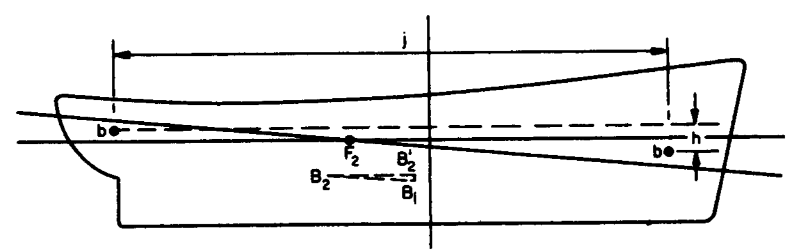


Fig. 3.24 Trim effects on B_1

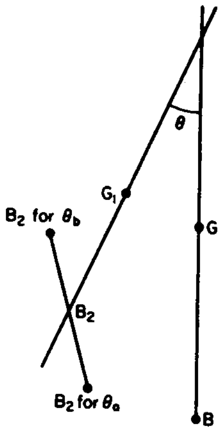


Fig. 3.25

therefore made for two or more trial values of θ, giving positions of B_2 applicable to each. As before

$$\overline{B_1B_2'} = \frac{Wh}{\Delta + W} \quad \text{and} \quad \overline{B_2'B_2} = \frac{Wj}{\Delta + W}$$

(*c*) examine the trigonometry ofB and G, knowing that they must be in vertical lines (Fig. 3.25). Having plotted the trial values of B_2, the choice of correct position of B_2 to correspond to the correct ()is best carried out graphically by trial and error. Having made such a judgment, the position is re- checked.

This is a laborious process. A preferable approach is to regard the ship as an entirely new one with a revised position of G and a trim found from the Bonjean curves to give B vertically beneath G and the correct displacement. This, in fact, is how the design trim of the ship is initially determined.

DETERMINATION OF DESIGN TRIM

As described later, the early stages of ship design involve a trial and error process of matching the longitudinal positions of Band G. For a given displacement and position of G, it will at some stage be necessary to find the position of the LCB and the trim resulting. Similar requirements occur in bilging and launching calculations, which are discussed in later chapters.

To find the draught and trim corresponding to a specified displacement and LCG position, it is necessary first to estimate the approximate likely waterline. Accuracy in the estimate is not important. The trial waterline is placed on a profile on which the Bonjean curve has been drawn for each ordinate so that the areas up to the waterline can be read off (Fig. 3.26). These areas and their longitudinal moments are then integrated in the usual fashion to find the displacement and LCB position. The waterplane offsets are then found, by projecting the waterline on to an offset body plan and the properties of the

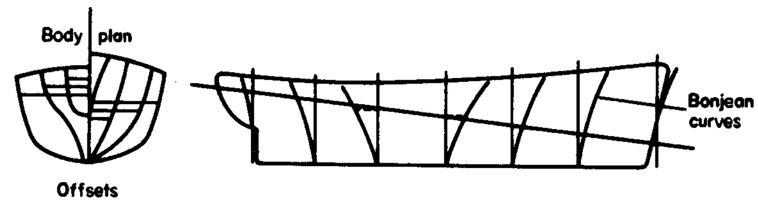


Fig. 3.26

waterplane, in particular LCF, TPI and MCT (using the approximation suggested earlier) are calculated. The ship may then be subjected to the calculations for parallel sinkage to alter the trial waterline to a new one giving the required displacement; it may then be trimmed to give the required position of LCB. If the parallel sinkage or trim are large to achieve this, a second trial waterline will be necessary and a similar adjustment made.

CHANGE OF WATER DENSITY

A ship moving from sea water to river will be subjected to a parallel sinkage and a trim. Let us examine why.

It is necessary, to study the problem clearly, again to separate parallel sinkage and trim. Because the river water has a smaller weight density and is therefore less buoyant, the ship will sink deeper. We have already found on pp. 59 and 60 that

$$\begin{aligned}\text{the volume of layer} &= \nabla_2 - \nabla_1 \\ &= \nabla_2\left(1 - \frac{w_2}{w_1}\right)\end{aligned}$$

and

$$\begin{aligned}\text{weight of layer} &= \nabla_2 w_2\left(1 - \frac{w_2}{w_1}\right) \\ &= \Delta\left(1 - \frac{w_2}{w_1}\right)\end{aligned}$$

w being weight density, not the reciprocal.

Now the volume that previously provided sufficient buoyancy had its centre of buoyancy at B, in line with G. The addition of a layer, which has its centre of buoyancy at F, the centre of flotation, causes an imbalance. As drawn (Fig. 3.27), a trim by the bow is caused and the wedges of added and lost buoyancy introduced by the trim restore equilibrium (Fig. 3.28).

If e is the horizontal distance between LCF and LCB, the moment causing trim is

$$\Delta e\left(1 - \frac{w_2}{w_1}\right)$$

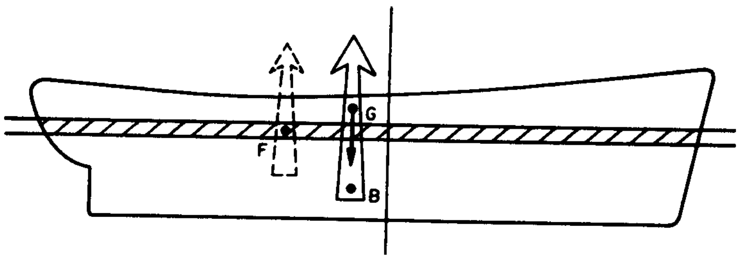


Fig. 3.27

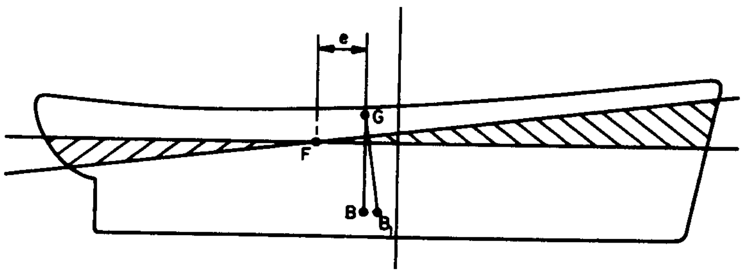


Fig. 3.28

The trim is then

$$\frac{\Delta e}{MCT} = \Delta e\left(1 - \frac{w_2}{w_1}\right) \times \frac{L}{\Delta GM_L} = \text{approx } \Delta e\left(1 - \frac{w_2}{w_1}\right) \times \frac{L}{Iw_2}$$

It must be remembered that the trim occurs in w_2 water and MCT must be related to that; if the value of MCT is given for w water, approximately, since MCT is proportional to wI/L,

$$\text{MCT}_2 = \text{MCT} \times \frac{w_2}{w}$$

It is not difficult to produce formulae for changes in draught for particular changes of condition and to invent rules which give the direction of trim. The student is advised to commit none to memory. Each problem should be tackled with a small diagram and an understanding of what actually happens. Summarizing, the process is

(*a*) using Archimedes' principle, calculate the volume and thence the weight of the layer,
(*b*) calculate the parallel sinkage using TPI pertinent to the new water,
(*c*) calculate the trim due to the movement of the layer buoyancy from B to F, using MCT pertinent to the new water. Worked example 7 is an illustration.

Hydrostatic data

Throughout its life, a ship changes its weight and disposition of cargo, its draught and trim, and freeboard; the density of the water in which it floats varies. Its stability, discussed later, also changes. If its condition in any stated set of circumstances is to be estimated, its condition in a precise state must be known so that the effect of the changes from that state can be calculated. This precise state is known as the design condition. For ocean going ships, a specific volume of 99.5 m^3/MN or 0.975 m^3/tonnef is employed for salt water. For this water the tools necessary to calculate changes from the design or load waterline are calculated for a complete range of waterlines. Also calculated are the geometrical properties of the underwater form for the range of waterlines.

Collectively, this information is known as the hydrostatic data. It is presented either in tabular form when intermediate values are interpolated or as a set of curves, which are called the *hydrostatic curves.*

HYDROSTATIC CURVES

The following properties are plotted against draught to form the hydrostatic curves

(i) Centre of bouyancy above keel, -K-B
(ii) Transverse metacentre above keel, -K-M-
(iii) CB aft of amidships
(iv) CF aft of amidships
(v) Displacement
(vi) Tonnef per centimetre immersion
(vii) Change of displacement for one metre change of trim
(viii) Moment to change trim one metre.

Occasionally, certain other properties are also plotted, such as longitudinal-K-M-, area of wetted surface and some of the geometric coefficients.

The first four items are all items of ship geometry and are affected neither by the density of the water nor by the ship's weight. Items (v), (vi) and (vii) are all related to the weight density of the water. Item (viii) is also related to the weight density of the water and alone is affected by the vertical position of the ship's centre of gravity, albeit only slightly. Accepting the approximation to MCT and pp. 65 and 66 all of these last four properties are corrected in the same way when their values are required in water of different densities; if the property is known in water of reciprocal weight density u_1 and it is required in water of reciprocal density u_2,

$$\text{property iii } u_2 \text{ water} = \text{property iii } u_1 \text{ water} \times \frac{u_1}{u_2}$$

u_1 will often be equal to 0.975 m^3/tonnef.

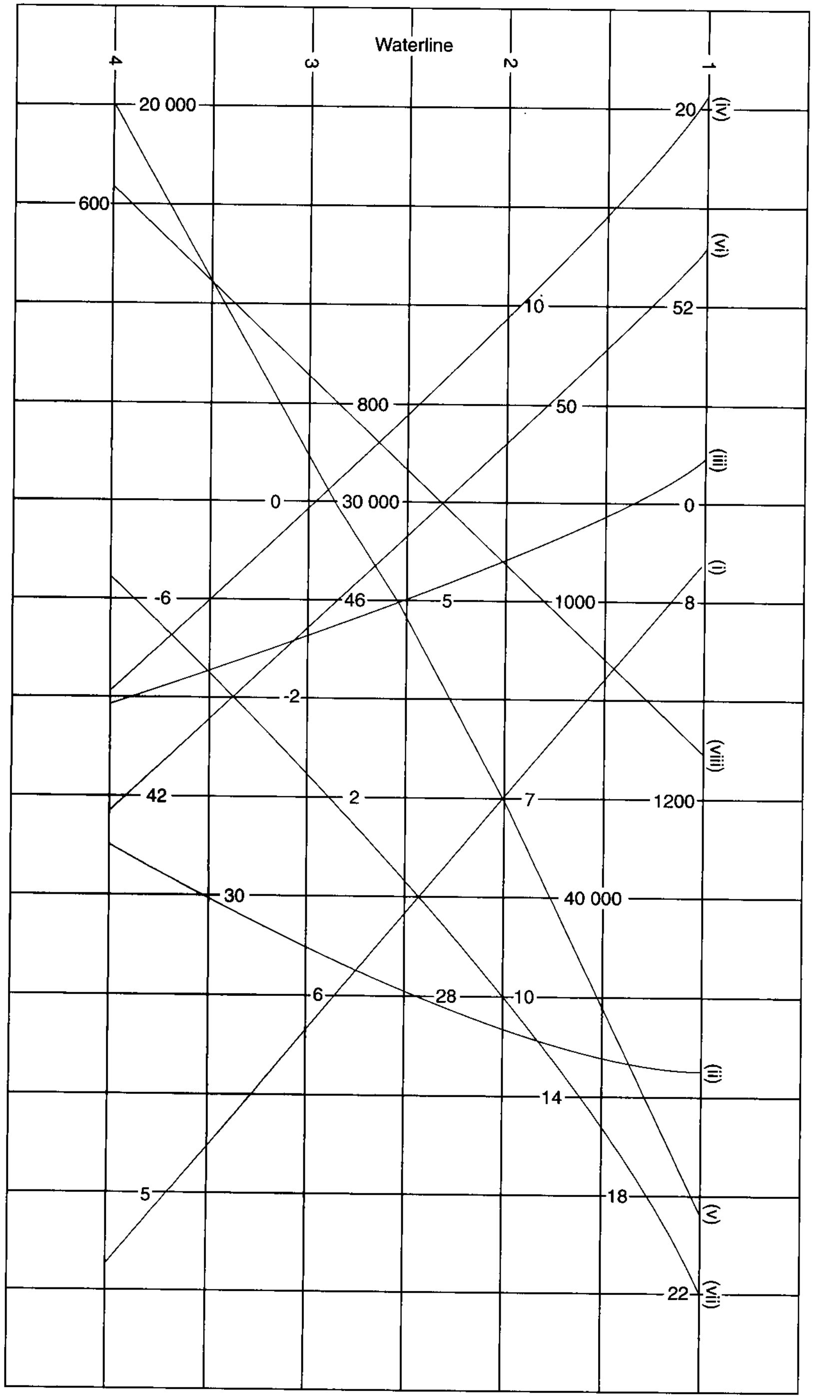


Fig. 3.29 Typical hydrostatic curves

(i) $\overline{KB}$, metres
(ii) $\overline{KM}$, metres
(iii) CB abaft amidships, metres
(iv) CF abaft amidships, metres
(v) Displacement, tonnef
(vi) Tonnef per cm immersion
(vii) Change of displacement for 1m change of trim
(viii) Moment to change trim one metre, tonnef/m

For example see Fig 3.29.

WL	Mean draught (m)	Disp.	TPC	CF abaft amidships	CB abaft amidships	$\overline{KB}$	$\overline{KM}$
1	14	47900	53.19	20.25	2.01	8.17	26.51
2	12	37600	49.43	9.19	−3.18	6.95	27.38
3	10	28500	45.58	−0.39	−7.17	5.76	28.79
4	8	20000	41.69	−9.94	−10.31	4.57	31.01

CALCULATION OF HYDROSTATIC DATA

The traditional manner of calculating the hydrostatic data is so methodical that it lends itself very readily to programming for a computer. Each of the properties listed above is calculated for each of the waterlines of the ship by the methods already described in this and the previous chapters. The offsets are

Ordinate number		WATERLINES										Vertical sections		Lever	FNS for moms
		5		4		3		2		1		FNS of area	Mults of area		
	S.M.	1		4		2		4		1					
1	1	1	1	1	1	1	1	1	1	1	1				
		1		4		2		4		1		12	12	3	36
2	4	5	20	6	24	7	28	8	32	9	36				
		5		24		14		32		9		84	336	2	672
3	2	7	14	8	16	9	18	10	20	11	22				
		7		32		18		40		11		108	216	1	216
4	4	9	36	10	40	11	44	12	48	13	52				
		9		40		22		48		13		132	528	0	924
5	2	6	12	7	14	8	16	9	18	10	20				
		6		28		16		36		10		96	192	1	192
6	4	4	16	5	20	6	24	7	28	8	32				
		4		20		12		28		8		72	288	2	576
7	1	1	1	1	1	1	1	1	1	1	1				
		1		4		2		4		1		12	12	3	36
FNS of area		100		116		132		148		164			1584		804
Mults. area		100		464		264		592		164		1584			120
Levers		4		3		2		1		0					
FNS for mts.		400		1392		528		592		0		2912			

Fig. 3.30 A simplified displacement sheet

set down in tabular form as shown in Fig. 3.30 which has been compiled for illustration purposes for a simple ship shape-in practice 21 ordinates and several more waterlines are usual. Offsets are set down in tabular fashion to correspond to the orthogonal sections explained by Fig. 2.3. The table is known as the *displacement table* and the complete sheet of calculations as a *displacement sheet.*

The waterline offsets are set down in vertical columns and each is operated upon by the vertical column of Simpson's multipliers. The sums of these columns are functions (i.e. values proportional to) of waterplane areas which, operated upon by the horizontal multipliers and summed give a function of the displacement. Levers applied to these functions result in a function of VCB. This whole process applied, first, to the ordinates by the horizontal multipliers and, secondly, to the functions of ordinate areas by the vertical multipliers and levers results in a function of displacement and a function of LCB. Clearly, the two functions of displacement should be the same and this comprises the traditional corner check.

There will frequently be an appendage below the lowest convenient waterline which must be added to the main body calculations. Areas of this appendage at each ordinate are measured (e.g. by planimeter) and positions of centres of areas either spotted or, if the appendage is considerable, measured. The effects of the appendage are then included, as described in Chapter 2.

With the offsets for each waterplane so conveniently presented by the displacement table, it is a simple matter to calculate by the methods described in Chapter 2, area, position of CF and the second moments of area for each waterplane. Oisplacements and CB positions are calculated by adding or subtracting slices from the nearest waterplane calculations.

If Tchebycheff spacings of ordinates are adopted, the Simpson multipliers are, of course, dispensed with in one direction. A form of displacement table in which this is done is not unusual.

THE METACENTRIC DIAGRAM

As has already been discussed, the positions of B and M are dependent only on the geometry of the ship and the waterplane at which it is floating. They can, therefore, be determined without any knowledge of the actual condition of ship loading which causes it to float at that waterline.

The metacentric diagram is a convenient way of defining variations in relative heights of B and M for a series of waterlines parallel to the design or load waterplane. Such a diagram is shown in Fig. 3.31 and it is constructed as follows. The vertical scale is used to represent draught and a line is drawn at 45 degrees to this scale. For a given draught T_1 a horizontal line is drawn intersecting the 45 degree line in O_1 and a vertical line is drawn through O_1. On this vertical line, a distance O_1M_1 is set out to represent the height of the metacentre above the waterplane and O_1B_1 to represent the depth of the centre of buoyancy below the waterplane. This process is repeated a sufficient number of times to define adequately the loci of the metacentre and centre of buoyancy. These two loci are termed the M and B curves. As drawn, M_1 lies above the waterplane but this is not necessarily always the case and the M curve may cross

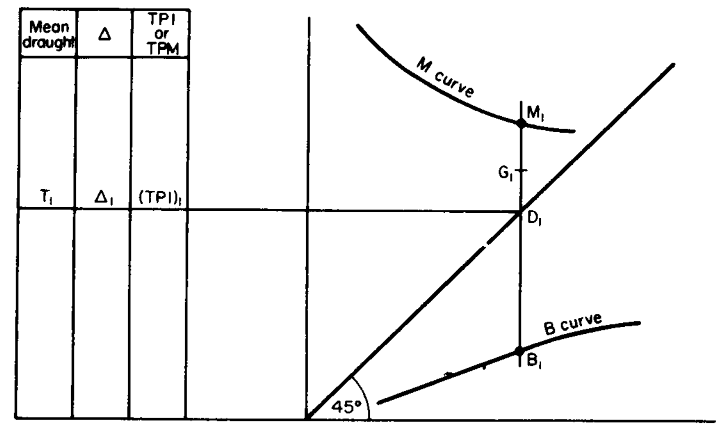


Fig. 3.31 *Metacentric diagram*

the 45 degree line. A table is constructed to the left of the diagram, in which are listed the displacements and TPI values for each of a number of draughts corresponding to typical ship conditions. These conditions must, strictly, be reduced to a common trim so that the loci of M and B will be continuous. In practice, however, not much error is involved in using the diagram for conditions other than the standard condition provided that the difference in trim is not excessive and provided that the draught is that at the centre of flotation.

In addition to the M and B curves, it is usual to show the positions of G for the conditions chosen. These positions can be obtained by methods discussed in Chapter 4.

Since KB is approximately proportional to draught over the normal operating range, the B curve is usually nearly straight for conventional ship shapes. The M curve, on the other hand, usually falls steeply with increasing draught at shallow draught then levels out and may even begin to rise at very deep draught.

Worked examples

EXAMPLE 1. Construct a metacentric diagram for a body with constant triangular cross-section floating with its apex down and at level draught. Dimensions of cross-section are shown in the figure.

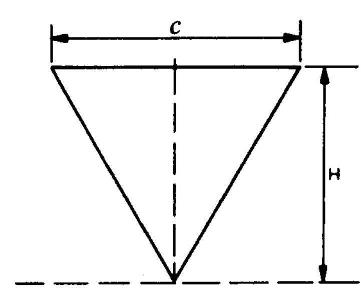


Fig. 3.32

Solution:

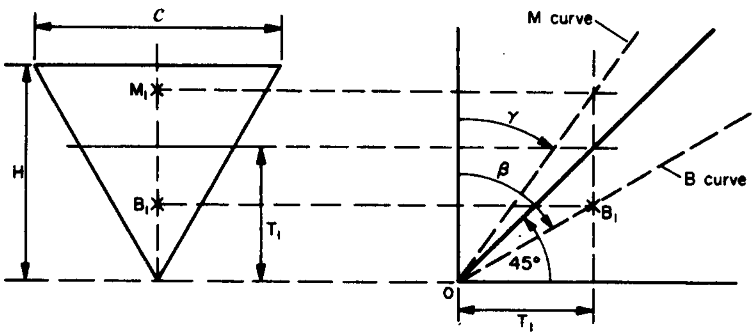


Fig. 3.33

For a given draught T_1, $\overline{KB} = \frac{2}{3}T_1$. Hence, the B curve is a straight line passing through 0, at an angle β to the vertical such that

$$\tan\beta = \frac{T_1}{\frac{2}{3}T_1}$$

i.e.

$$\tan\beta = 1.5$$

Breadth of waterplane at draught $T_1 = c\dfrac{T_1}{H}$

$$\therefore\ \overline{B_1M_1} = \frac{I}{\nabla} = \frac{\frac{1}{12}(cT_1/H)^3L}{\frac{1}{2}(cT_1/H)T_1L}$$

$$= \frac{1}{6}\frac{c^2T_1}{H^2}$$

$$\therefore\ \overline{KM_1} = \overline{KB_1} + \overline{B_1M_1} = \frac{2}{3}T_1 + \frac{1}{6}\left(\frac{c^2}{H^2}\right)T_1$$

$$= \frac{T_1}{6}\left(4 + \frac{c^2}{H^2}\right)$$

Hence, in this case, since c and H are constants, the M curve is also a straight line through 0 but at an angle γ to the vertical such that

$$\tan\gamma = \frac{T_1}{T_1(4 + c^2/H^2)/6} = \frac{6H^2}{4H^2 + c^2}$$

EXAMPLE 2. Construct a metacentric diagram for a vessel of constant circular cross-section floating at uniform draught.

Solution:

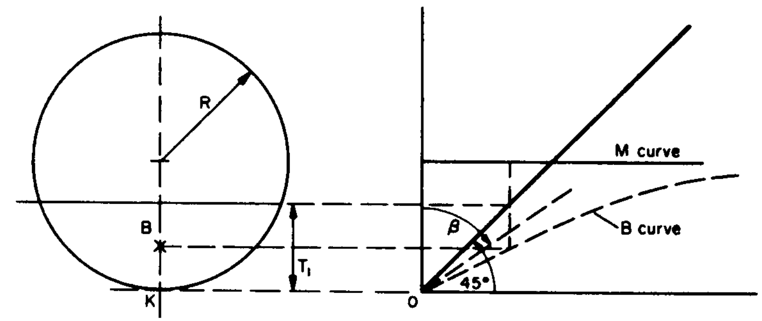


Fig. 3.34

For any inclination at any draught the buoyancy force must, from geometric considerations, pass through the centre of the circular cross-section. Hence, the M curve is a horizontal straight line.

The expression for $\overline{KB}$ is more complicated, but general considerations show that the curve must become tangential to the M curve when $T_1 = 2R$ and that its general form will be as indicated. If it is assumed that for small draughts the circle can be approximated to by a parabola, it can be shown that the tangent to the B curve at 0 is given by

$$\beta = \tan^{-1}\frac{5}{3}$$

EXAMPLE 3. When the tide falls, a moored barge grounds in soft homogeneous mud of specific gravity 1.35. With the keel just touching the mud, the barge was drawing a level 1.65 m of salt water of reciprocal weight density 0.975 m^3/tonnef. The tide falls a further 18 cm.

Assuming that the barge is box shaped 25 m long and 5 m in breadth, calculate the draught after the tide has fallen. What weight must be removed to free the barge from the mud?

Solution: When floating free

$$\Delta = \frac{1.65 \times 25 \times 5}{0.975}\ \mathrm{m}^3\ \frac{\text{tonnef}}{\mathrm{m}^3} = 211.5\ \text{tonnef}.$$

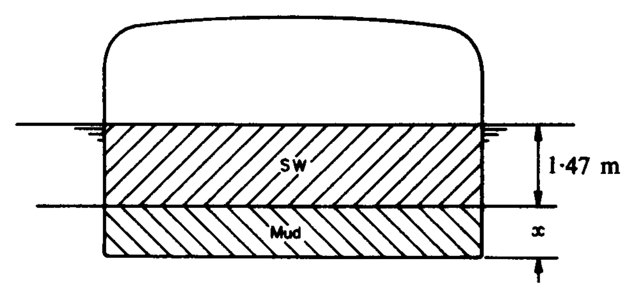


Fig. 3.35

When in mud,

$$\text{buoyancy supplied by salt water} = \frac{1.47}{1.65} \times 211.5 = 188.4\,\text{tonnef}$$

$$\text{buoyancy needed from mud} = 211.5 - 188.4 = 23.1\,\text{tonnef}$$

$$\text{buoyancy provided by mud} = x \times 25 \times 5 \times 1.35 = 23.1$$

$$\therefore\ x = \frac{23.1}{25 \times 5 \times 1.35} = 0.14\,\text{m}$$

$$\therefore\ \text{draught when tide has fallen} = 1.61\,\text{m}$$

$$\text{weight to be removed to clear mud} = 23.1\,\text{tonnef.}$$

(The draught x could also have been obtained by taking a proportion of the 18 cm fall in tide in the ratio of the S.Gs of mud and salt water,

$$(1.35 \times 0.975)x = 0.18.$$

EXAMPLE 4. Hydrostatic data for a ship show the following

Draught	6 m	7 m	8 m
Displacement (tonnef)	8421	10,900	13,700
TPC (tonnef/cm)	25.3	26.0	26.8

Calculate the displacement at a draught of 7.3 m.

Solution: The TPC curve gives values of TPC at 7.15 m and 7.3 m. of 26.10 and 26.22 by plotting

WL	TPC	SM	f(V)
7.3 m	26.22	1	26.22
7.15 m	26.10	4	104.40
7.00 m	26.00	1	26.00
			156.62

$$\text{displacement of addition} = \frac{0.15}{3} \times 156.62 \times 100\,\text{tonnef}$$
$$= 783.1\,\text{tonnef}$$

$$\therefore\ \Delta \text{ to 7.3 m. waterline} = 11{,}683\,\text{tonnef.}$$

(The best obtainable from interpolation of the displacement curve would have been a straight line relationship giving $\Delta = 10{,}900 + \frac{0.3}{10}(13{,}700 - 10{,}900) = 11{,}740$ tonnef. Alternatively we would have obtained a sufficiently accurate answer in this case by assuming the TPC to be constant at its 7.15 m. value, linearly interpolated between the two top waterplanes, i.e.

$$26.00 + \frac{0.15}{1.00} \times 0.80 = 26.12$$

then

$$\Delta = 10{,}900 + 0.3 \times 26.12 \times 100 = 11{,}684\,\text{tonnef.}$$

EXAMPLE 5. During a voyage, a cargo ship uses up 320 tonnef of consumable stores and fuel from the fore peak, 85 m forward of midships.

Before the voyage, the forward draught marks 7 m aft of the forward perpendicular, recorded 5.46 m and the after marks, 2 m aft of the after perpendicular, recorded 5.85 m. At this mean draught, the hydrostatic data show the ship to have tonnef per centimetre = 44, one metre trim moment BP = 33,200 tonnef/m BP, CF aft of midships = 3 m. Length BP = 195 m. Calculate the draught mark readings at the end of the voyage, assuming that there is no change in water density.

Solution (see Fig. 3.36):

$$\text{Parallel rise} = \frac{320}{4400} = 0.073\,\text{m}$$

$$\text{Moment trimming} = 320(85 + 3)\text{tonnef/m.}$$

$$\text{Trim BP (195 m)} = \frac{320 \times 88}{33{,}200} = 0.848\,\text{m.}$$

$$\text{Draught change at FM due to trim} = \frac{93.5}{195} \times 0.848 = 0.406\,\text{m.}$$

$$\text{Draught change at AM due to trim} = \frac{96.5}{195} \times 0.848 = 0.420\,\text{m.}$$

$$\text{New draught at FM} = 5.46 - 0.07 - 0.41 = 4.98\,\text{m.}$$

$$\text{New draught at AM} = 5.85 - 0.07 + 0.42 = 6.20\,\text{m.}$$

A clear sketch is essential for a reliable solution to problems of this sort.

EXAMPLE 6. A fishing vessel grounds on a rock at a point 13.4 m from the FP when the depth of water is 0.3 m above low tide. Before grounding, the vessel was drawing 1.60 m at the FP and 2.48 m at the AP. At this mean draught, TPI in MN per metre immersion = 3.5, one metre trim moment = 13 MN m and CF is 1 m forward of midships. The ship is 49 m BP. Calculate the force at the keel and the new draughts at low tide.

Solution: Let the force at the rock be P MN and replace it by a force P MN and a moment 10.1 P MN m at the CF.

$$\text{Parallel rise due to } P = \frac{P}{3.5}\,\text{m}$$

$$\text{Change of trim due to } 10.1\,P = \frac{10.1P}{13}\,\text{m BP}$$

$$\text{Change of draught at rock due to trim} = \frac{10.1}{49} \times \frac{10.1P}{13}\,\text{m}$$

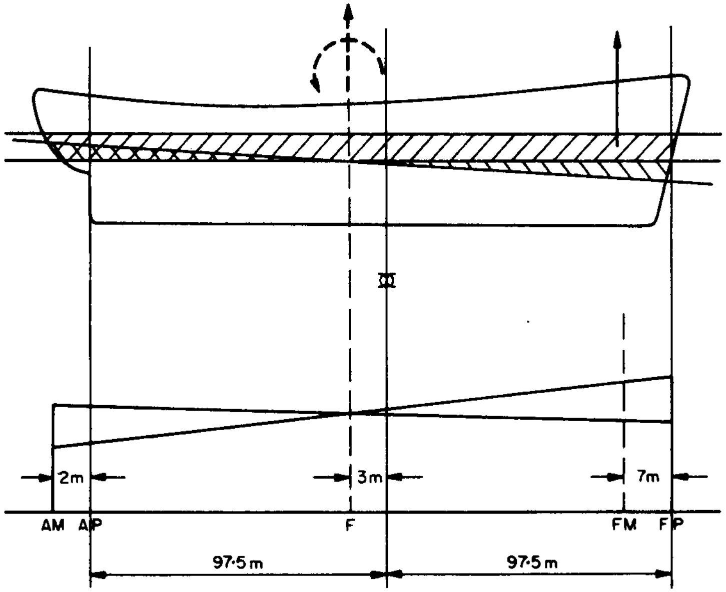


Fig. 3.36

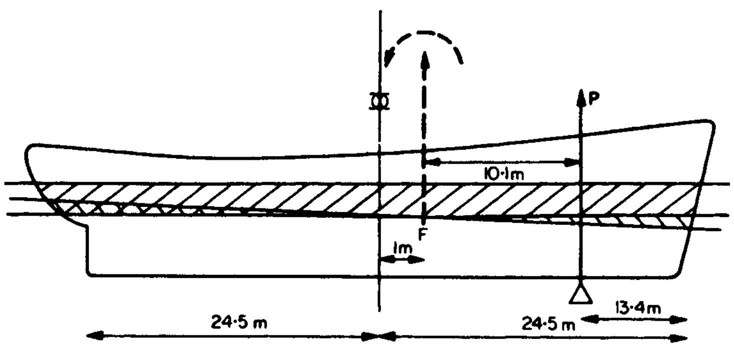


Fig. 3.37

$$\text{Total change of draught at rock} = \frac{P}{3.5} + \frac{10.1}{49} \times \frac{10.1P}{13} \text{ m}$$

$$= \text{fall of tide} = 0.3 \text{ m}$$

$$\therefore\ P(0.286 + 0.160) = 0.3$$

$$\therefore\ P = 0.673 \text{MN}$$

Then parallel rise due to $P = 0.192$ m.

Change of trim due to 10.1 $P = 0.523$ m.

$$\text{Change of draught at FP due to trim} = \frac{23.5}{49} \times 0.523 = 0.251 \text{ m.}$$

$$\text{Change of draught at AP due to trim} = \frac{25.5}{49} \times 0.523 = 0.272 \text{ m.}$$

New draught at $FP = 1.600 - 0.192 - 0.251 = 1.157$ m.

New draught at AP $= 2.480 - 0.192 + 0.272 = 2.560$ m.

(Because the final trim is large, it is unlikely that the answer can be relied upon to within 10 cm.)

EXAMPLE 7. The hydrostatic curves (prepared for water of reciprocal weight density 0.975 m^3/tonnef) for a guided missile destroyer show it to have the following particulars at a mean draught of 5.42 m at which it is floating with a 0.36 m stern trim in water of reciprocal weight density 1 m^3/tonnef:

Length BP	170 m
Designed trim	0.75 m by stern
Displacement	8070 tonnef
CB abaft amidships	0.73 m
CF abaft amidships	6.22 m
TPC	27.8 tonnef/cm
MCT	21,000 tonnef/m per metre

Calculate (*a*) the true displacement, (*b*) the new draught and trim when the ship puts to sea in water of reciprocal weight density 0.981 m^3/tonnef.

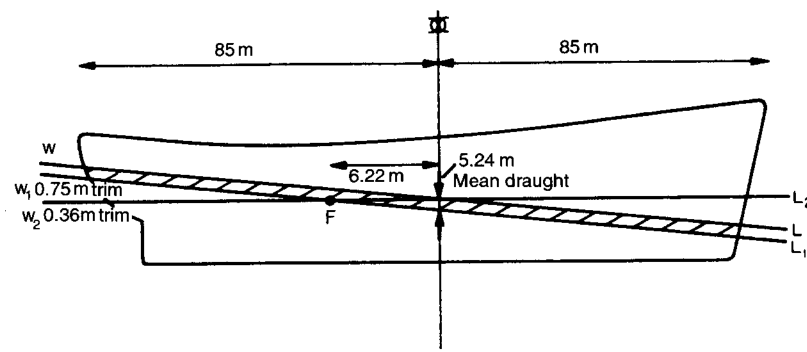


Fig. 3.38

Solution: As shown in Fig. 3.38, because the hydrostatic curves assume a 0.75 m trim, they overestimate the displacement by the shaded layer of thickness

$$\frac{6.22}{170} \times (0.75 - 0.36) = 0.014 \text{ m}$$

This layer is composed of 1 m^3/tonnef water, so that the relevant TPC is

$$27.8 \times \frac{0.975}{1} = 27.11$$

and the displacement of the layer

$$100 \times 0.014 \times 27.8 \times \frac{0.975}{1} = 37.9\,\text{tonnef}$$

$$\text{Now the displacement to WL} = 8070 \times \frac{0.975}{1}$$

$$= 8070 - 8070\left(\frac{1-0.975}{1}\right)^*$$

$$= 7868\,\text{tonnef}$$

$\therefore$ displacement to W_1L_1 (and W_2L_2) $= 7868 - 37.9 = 7830$ tonnef.

In moving from 1.0 to 0.981 water, which is more buoyant, there is a parallel rise:

$$\text{Volume of layer} = 7830(1 - 0.981) = 149\,\text{m}^3$$

This is a layer of 1.00 water, weight

$$\frac{149}{1.0} = 149\,\text{tonnef}$$

$$\text{TPC in 1.00 water} = 27.8\left(\frac{0.975}{1.000}\right)$$

$$\therefore \text{layer thickness} = \frac{149}{27.8 \times 0.975} = 5.5\,\text{cm}$$

$$\text{Moment causing trim} = 149(6.22 - 0.73) = 818\,\text{tonnef/m}$$

$$\text{Trim occurs in 0.981 water for which MCT by 1 m} = 21{,}000 \times \frac{0.975}{0.981}$$

$$\therefore \text{trim by stern} = \frac{818}{21{,}000} \times \frac{0.975}{0.981} \times 100 = 3.87\,\text{cm}.$$

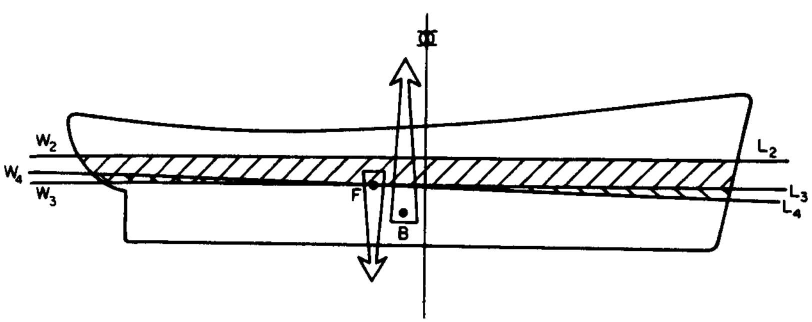


Fig. 3.39

* Note the important artifice used to obtain the displacement to WL by calculating the difference by pocket calculator.

Change in mean draught due to 3.87 cm trim by stern

$$= -\frac{6.22}{170} \times 3.87 = -0.14\,\text{cm}$$

total reduction in mean draught = 5.64 cm

$\therefore$ in water of RWD 0.981 m^3/tonnef

mean draught = 5.18 m

stern trim = 0.4 m.

Problems

1. (*a*) Calculate the reciprocal weight density of petrol in m^3/MN, given that its specific gravity is 0.70.
   (*b*) A pressure of 3.75 MPa is experienced by a submarine. If 1 lb = 0.4536 kg and 1 in. = 2.54 cm, convert this pressure to lbf/in^2. Put $g = 9.81\,\text{m/s}^2$.
2. A floating body has a constant triangular section, vertex downwards and has a constant draught of 12 m in fresh water, the breadth at the waterline being 24 m. The keel just touches a quantity of mud of specific gravity 2. The water level now falls 6 m. How far will the body sink into the mud?
3. A wallsided ship goes aground on a mudbank so that its draught marks record a mean draught of 2.13 m. Before going aground, the draught of the ship in salt water was 2.69 m. The area of the waterplane is 388.3 m^2. What volume of oil of specific gravity 0.92 should be pumped out to free the ship?
   Supposing, before pumping can begin, the tide falls a further metre and the ship sinks a further 50 cm. into the mud, displacing a total of 53.77 m^3 of mud. What would be the reciprocal weight density of the mud?
4. A homogeneous wooden pontoon of square section, having a specific gravity of 0.5 is made to float in fresh water with its sides vertical by covering one of its faces completely with a strip of metal having a specific gravity of 8.0.
   Show that G coincides with M if the thickness of the strip is approximately one ninetieth ($\frac{1}{90}$) of its width.
5. A model of a ship is built to float in water of 9823 N/m^3 weight density. It has the following geometric properties:

   Length BP, 1.83 m; Area of waterplane, 4461 cm^2
   Draught, 14 cm; Trans. MI of waterplane, $5.182 \times 10^{-3}\,\text{m}^4$
   $\overline{\text{KB}}$, 7.62 cm; Volume of displacement, $68 \times 10^{-3}\,\text{m}^3$

   The centre of gravity as built is 18.29 cm above the keel. Where must a weight of 111 N be placed to bring G below M, presuming the model to be wallsided at the waterline?
6. The TPC (tonnef per centimetre) of a wallsided ship floating at a draught of 2.75 m is 9.40. Its displacement at this draught is 3335 tonnef and the VCB is

1.13m below the waterline. The centre of gravity of the ship is 3.42m above the keel. The transverse second moment of area of the waterplane is 7820m^4.

Calculate the shifts of Band G and the new position of M when a load of 81.2 tonnef is placed 3.05m off the centre line on a deck 8.54m above the keel and by constructing the movements on squared paper, estimate, to the nearest degree, the heel of the ship.

7. The half ordinates of the waterplane of a barge, 50m long, are 0.0, 3.0,4.5, 5.0, 4.2 and 1.0m. Its volume of displacement is 735m^3 and the CG is 2.81m above the keel. The draught is 3.0m.

A block of SG 7.0, having a volume of 10m^3 has to be raised from the deck in a cage. Using an approximate' formula for the position of B, estimate the height to which the weight can be raised before G is higher than M.

8. The values of MN/m immersion of a ship at the waterlines, which are 1.52m apart, are given below. In addition, there is an appendage having a displacement of 14.75MN with its centre of buoyancy 1 metre below 7 WL.

Waterline	1	2	3	4	5	6	7
MN/cm	0.76	0.72	0.66	0.60	0.53	0.42	0.18

Calculate the displacement of the ship and the vertical position of the centre of buoyancy of the ship.

9. The values of MN/m of the waterlines of a vessel, which are 2.13m apart, are:

WL	6	5$\frac{1}{2}$	5	4	3	2	1
MN/cm	0.08	0.17	0.25	0.34	0.42	0.49	0.52

If there is a 112 tonnef appendage whose c.b. is 0.91m below 6WL, find the displacement and the position of the CB of the whole vessel.

10. A yacht of mass displacement 9.5 tonnes to 1 WL in fresh water has a CB 0.25m below 1 WL. If the distance between WLs is 0.15m, plot a curve of VCB against draught. The areas of A, 1, 2 and 3 WLs are respectively 28.4, 28.0,26.8 and 25.2m^2. Quote VCB at 2! WL.

11. A guided missile destroyer is 155m long BP and displaces 6228 tonnef in salt water when the draught mark readings are 4.53m. forward and 4.66m aft. The distances of the draught marks from amidships are 70.1m forward and 83.8m aft. The second moment of area of the waterplane is 0.71 x 10^{6}m^4 about the centre of flotation which is 1.52m abaft amidships. The waterplane area is 1626m^2.

Calculate the new draughts when 142 tonnef of missiles are embarked at a mean distance of 57.9m abaft amidships. Illustrate on a bold diagram, the forces and movements involved and state any assumptions made.

12. The longitudinal moment of inertia about the CF of the waterplane of a depot ship floating in fresh water is 1.64 x 10^{6}m^4. The length BP is 208m and the draught marks are 9.75m abaft the FP and 2.44m abaft the AP. The area of the waterplane is 4459 m^2. Draught marks read 5.26m. forward and 6.77m aft.

When 451 tonnef of oil with its c.g. 61m abaft amidships are pumped out, the reading of the after marks is observed to change to 5.14m.

Deduce the position of the CF of the waterplane and the reading of the forward marks.

13. For a mean draught of 4.0m with a 0.5m trim by the stern, the ship's book shows a destroyer to have the following particulars:

displacement, mass	= 2300 tonne
TPC	= 9.10
MCT BP	= 3010 tonnef/m
CF abaft amidships	= 6.85m
CB abaft amidships	= 2.35m

The ship is 102m BP and is floating where the water has an SG = 1.02 at draughts of 3.50m forward and 4.50m aft.

Find, to the nearest half centimetre the draught at the propellers, which are 13m forward of the AP, when the ship has moved up river to water of SG = 1.01.

14. A ship, displacement 7620 tonnef, floats at draughts of 4.10m forward and 4.70m aft. The forward marks are 2.1m aft of the FP and the after marks are 19.8m forward of the AP. The ship is 170m between perpendiculars, has a tonnef per centimetre of 22.4 and the CF is 5.8m aft of amidships. Determine the new draughts at the marks if a weight of 100 tonnefis placed 18.3m forward of amidships.

One metre trim moment BP = 20,700 tonnef m.

15. The following are the particulars of an aircraft carrier: Displacement = 44,700 tonnef, mean draught BP = 11.70m trim by stern between marks = 2.23m, TPC = 33.7, MCT 1cm = 3.06MN/m. CF abaft amidships = 12.2m, LBP=219.5m, forward marks abaft FP=IOm, after marks before AP = 15.8m

Calculate the new draughts at the marks when 538 tonnef of aircraft ar~ embarked with their c.g. 79.25m before amidships.

16. A box-shaped barge of length 20m beam 5m and mean draught 1.45m floats with a trim of 30 cm by the stern in water of specific gravity 1.03. If it moves into water of SG = 1.00, what will be its new draughts forward and aft?

17. The buoy mooring cable on a badly moored vessel is vertical and just taut when it has an even draught of 3m. The vessel is 50m long and has a TPM = 250 and CF 4m abaft amidships. MCT is 1300 tonnef m. The mooring cable passes over the bow bullring 5m before the FP.

Calculate the new draughts at the perpendiculars when the tide has risen 0.8m.

18. A cargo ship, displacing 5588 tonnef, floats at draughts of 4.57 m forward and 5.03 m aft, measured at marks 70.1 m forward of and 51.8 m abaft amidships.

The vessel grounds on a rock 22.9 m forward of amidships. What is the force on the bottom when the tide has fallen 30 cm and what are then the draughts at the marks? TPC = *0.204MNjcm;* CF is 4.57m abaft amidships; MCT BP is 1560MN/m per m; LBP is 146m.

19. The draught marks of a ship, 61 m long BP, are 6.1 m aft of the FP and 12.2 m before the AP. The MN per m immersion is 3.570 and the one metre trim moment BP is 7.66 MN/m. The CF is 2.45 m abaft amidships and the draught mark readings forward and aft are respectively 2.67 and 3.0 m.

The vessel goes aground on a ridge of rock 9.15 m from the FP. If the tide is falling at an average rate of 81 cm/hr, how long will it be before the freeboard at the AP is reduced by 1.22 m?

What will then be the draught mark readings?

20. A ship 110m long, has a design trim of 0.61 m by the stern, an after cut-up 22.9 m from the after perpendicular and a $\overline{KG}$ of 4.88 m. When the after cut-up just touches level blocks, the trim out of normal is measured as 0.91 m by the stern and the water level in the dock is 4.05 m above the blocks.

Calculate (i) the initial displacement and LCG; (ii) the force on the after cut-up and the water level above the blocks when the ship just sues all along.

Hydrostatic data for design trim:

Mean draught (m)	Displacement (tonnef)	LCB (m aft)	VCB (m)	LCF (m aft)	$\overline{BM}_L$ (m)
3.65	2245	2.30	2.25	6.61	288
3.50	2100	1.98	2.15	6.52	302
3.35	1970	1.72	2.07	6.33	313
3.20	1830	1.70	1.98	6.00	326

21. A new class of frigates has the following hydrostatic particulars at a mean draught of 4m as calculated for water of 1025kgjm^3: TPC = 7.20, MCT = 3000 tonnef/m, CF abaft amidships = 4 m, CB abaft amidships = I m, displacement = 2643 tonnef.

When one of the class was launched at Southampton, where the mass density of water is 1025kgjm^3, it took the water with a mean draught of 3.98 m and a stern trim of 0.6 m.

Supposing it to be in an exactly similar condition, how would you expect a sister ship to float after launch on the Clyde, where the water has a specific gravity of 1.006?

All draughts are at the perpendiculars 120m apart.

22. A ship of 5080 tonnef and 152.4m between perpendiculars, has its centre of gravity 1.5m abaft amidships, floats in salt water of 0.975 m^3 *jtonnef* and has draughts of 3.12 m at the forward perpendicular and 4.42 m at the after cut-up, which is 9.14m before the after perpendicular.

Estimate the draughts before docking in water of reciprocal weight density 0.995 m^3 *jtonnef.*

The rate of immersion is 20 tonnef per cm, MCT one metre is 7619 tonnef/m increasing at the rate of 2000 tonnef/m per metre increase in draught. The centre of flotation is 7.62 m abaft amidships.

23. A Great Lakes bulk carrier has hydrostatic curves drawn for water of reciprocal weight density I m^3 *jtonnef,* which show the following information:

Mean draught BP	8.53m	9.75m
Displacement (tonnef)	24,647	28,481
28,481 Trim by stern	30cm	30cm
TPC (tonnef/cm)	31.36	31.52
MCT BP (tonnef/m)	9447	9496
CF forward of midships	1.16m	0.67m
CB aft of midships	3.66m	2.44 m

The draughts forward and aft in river water of reciprocal weight density 0.994 m^3 *jtonnef* are respectively 9.296 m and 9.601 m. The length between perpendiculars is 176.8 m.

Calculate the draughts when the ship reaches the open sea.

24. The following are the half ordinates in m of a vessel for which the waterlines are 1.75 m apart and ordinates 25 m apart. Determine its volume and the position of the centre of volume below I WL.

	WLs				
Ords.	5	4	3	2	1
1	0	0	0	0	0
2	1	2	3	4	6
3	2	3	5	7	8
4	3	5	7	8	9
5	2	4	6	8	9
6	1	2	4	6	6
7	0	0	0	0	0

25. Given the breadths of the waterlines of a ship as shown below, work out the displacement of the main body up to the 1 WL. The WLs are 2m apart and the ordinates 24 m.

	Breadths in metres at ordinates				
	FP	2	3	4	AP
1 WL	1.1	21.3	30.2	27.6	4.4
2WL	0.8	20.7	29.9	26.7	1.1
3WL	0.5	18.9	29.0	24.9	1.8
4WL	0.0	16.2	26.6	21.0	2.9

26. It is desired to fit a gun mounting weighing 20.3 tonnef to a ship without altering the draught at the propellers which are 9.14 m forward of the after perpendicular. The draughts are 2.44m forward and 2.74m aft and other particulars of the ship before the gun is added are: **LBP** = 42.7 m, displacement = 264tonnef, MCT = 792tonnef/m, CF abaft amidships = 2.13m, TPC = 2.24.
   Where should the gun be mounted? What are the resulting draughts forward and aft?
27. Construct a metacentric diagram for a vessel of rectangular cross-section.
28. Construct metacentric diagrams for prisms, apex downwards,

   (a) having a parabolic cross-section 6 m deep by 3 m wide at the top;
   (b) having a section of an isosceles triangle, half apex angle 15 degrees and height 8 m.

4 Stability

The term stability refers to the tendency of a body or system to return to its original state after it has suffered a small disturbance. In this chapter we are concerned at first with a very specific example of static stability whereby a ship floating upright is expected to return to the upright after it has been buffeted for example by wind or wave. If a floating body is very stable it will return quickly to the upright and may engender motion sickness; if it is just stable a disturbance which is not small may cause it to capsize. The stability therefore must be just right in the range of conditions in which a vessel may find itself during its operation and life, even damaged or mishandled.

EQUILIBRIUM AND STABILITY

A rigid body is said to be in a state of equilibrium when the resultant of all the forces acting on it is zero and the resultant moment of the forces is also zero. If a rigid body, subject to a small disturbance from a position of equilibrium, tends to return to that state it is said to possess positive stability or to be in a state of stable equilibrium. If, following the disturbance, the body remains in its new position, then it is said to be in a state of neutral equilibrium or to possess neutral stability. If, following the disturbance, the excursion from the equilibrium position tends to increase, then the body is said to be in a state of unstable equilibrium or to possess negative stability.

The ship is a complex structure and is not in the mathematical sense a rigid body. However, for the purpose of studying stability it is permissible so to regard it. Throughout this chapter the ship will be regarded as a rigid body in calm water and not underway. If the ship is in waves or is underway there are hydrodynamic forces acting on the ship which may affect the buoyancy forces. This problem is discussed briefly in later sections.

Nevertheless, for stability purposes, it is usual to ignore hydrodynamic forC4t" except for high-speed craft, including hydrofoils. For fast motor boats the hydrodynamic forces predominate in assessing stability.

For *equilibrium,* the buoyancy force and weight must be equal and the two forces must act along the same straight line. For a floating body this line must be vertical.

DISTURBANCE FROM STATE OF EQUILIBRIUM

Any small disturbance can be resolved into three components of translation and three of rotation with reference to the ship's body axes. The conventional axes are defined as in Fig. 4.1, the positive directions being indicated by the arrows. Consider each component of a disturbance in turn.

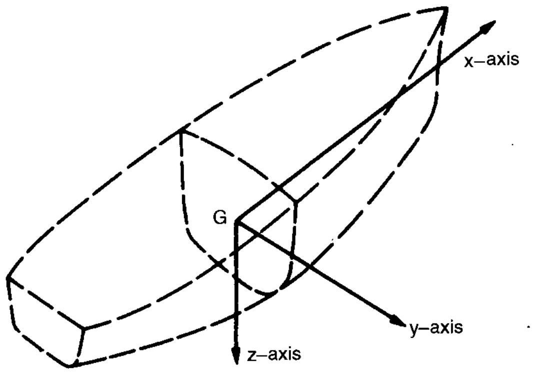


Fig. 4.1 Body axes

Translation along the x-axis, which is horizontal, leads to no resultant force so that the ship is in neutral equilibrium for this type of disturbance.

Translation along the y-axis also leads to no resultant force so that the ship is in neutral. equilibrium for this type of disturbance.

For a floating body, translation along the z-axis in the positive direction results in an augmented buoyancy force which will tend to move the ship in the negative sense of z, i.e. it tends to return the ship to its state of equilibrium and the ship is thus stable for this type of disturbance. The special case of a totally submerged body is considered later.

Rotation about the x-axis, heel, results in a moment acting on the ship about which no generalization is possible and the ship may display stable, neutral or unstable equilibrium.

Rotation about the y-axis, trim, leads to a condition similar to that for heel.

Rotation about the z-axis, yaw, results in no resultant force or moment so that the ship is in neutral equilibrium for this type of disturbance.

The above results are summarized in Table 4.1.

H is clear from Table 4.1 that the only disturbances which herein demand study are the rotations about the *x*- and y-axes. In principle, there is no distinction between these two but, for the ship application, the waterplane characteristics are such that it is convenient to study the two separately. Also, for small rotational disturbances at constant displacement any skew disturbance can be regarded as compounded of

(a) the stability exhibited by the ship for rotations about the x-axis-referred to as its transverse stability;
(b) the stability exhibited by the ship for rotations about the y-axis-referred to as its longitudinal stability.

Note that, because the shape of the heeled waterplane is not usually symmetrical, pure rotation in the *yz* plane must cause a tendency to a small trim. While this is usually of a second order and safely ignored, at large angles it becomes important.

Table 4.1
Summary of equilibrium conditions

Axis	Equilibrium condition for	
	Translation along:	Rotation about:
x	Neutral	Stable. neutral or unstable
y	Neutral	Stable. neutral or unstable
z	Stable	Neutral

Initial stability

Consider the irregular shaped body shown in Fig. 4.2 floating in a state of equilibrium. In the equilibrium state, the centre of buoyancy, Bo, and the centre of gravity, G, must lie on the same vertical line. Ifthe body is now subjected to a rotational disturbance, by turning it through a small angle (3at constant displacement, the centre of buoyancy will move to some new position, B. For convenience, in Fig. 4.2 the angular disturbance is shown as being caused by a pure moment about G, but it will be realized that the condition of constant displacement will, in general, require translational movements of G by forces as well.

The weight and buoyancy forces continue to act vertically after rotation but, in general, are separated so that the body is subject to a moment ~ -G-Z where Z is the foot of the normal from G on to the line of action of the buoyancy force. As drawn, this moment tends to restore the body to the original position. The couple is termed the *righting moment* and -G-Z is termed the *righting lever.*

Another way of defining the line of action of the buoyancy force is to use its point of intersection, M, with the z-axis. As the angle (3is indefinitely diminished M tends to a limiting position termed the *metacentre.* For small values of (3it follows that

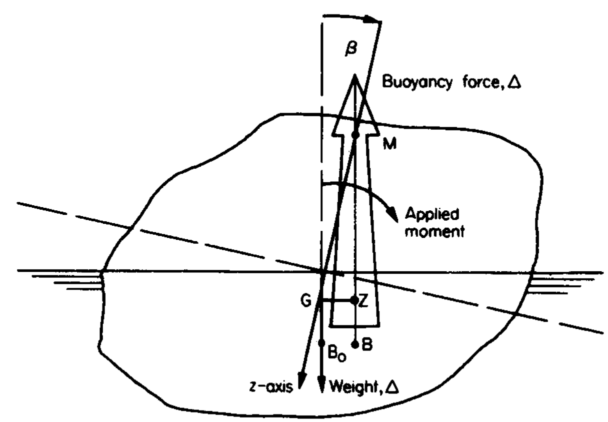


Fig. 4.2 Action of buoyancy force and weight for small rotational disturbance

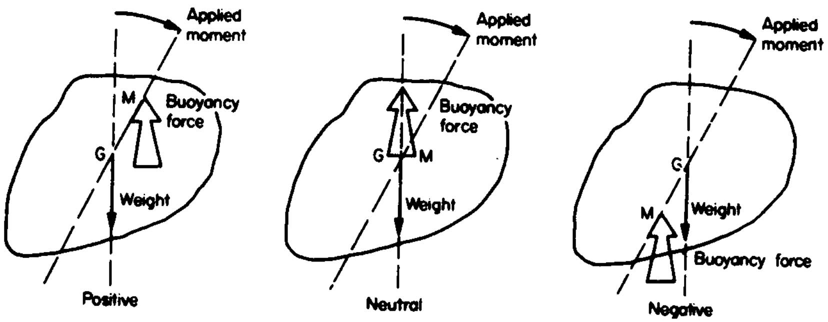


Fig. 4.3 The three stability conditions

$$\overline{GZ} = \overline{GM}\sin\beta$$
$$\simeq \overline{GM}\beta$$

The distance $\overline{GM}$ is termed the *metacentric height* and is said to be positive when M lies above G. This is the condition of stable equilibrium, for should M lie below G the moment acting on the body tends to increase β and the body is unstable. If M and G coincide the equilibrium is neutral (see Fig. 4.3).

ADJUSTMENT OF TRANSVERSE METACENTRIC HEIGHT BY SMALL CHANGES OF DIMENSIONS

The ship's form can be altered by increasing ordinates in the x, y and z directions such that all dimensions in any one direction are altered in the same ratio. This ratio need not be the same for all directions. This method of form changing has the advantage of maintaining the coefficients of fineness unaltered and is often used in the early stages of ship design. For instance, it may be necessary to decrease draught at the expense of the beam to ensure that the ship will be able to negotiate canals or harbour entrances. In this case,

$$\Delta = (\text{constant}) \times L \times B \times T$$

Hence

$$\log\Delta = \log(\text{constant}) + \log L + \log B + \log T$$

and differentiating

$$\frac{\mathrm{d}\Delta}{\Delta} = \frac{\mathrm{d}L}{L} + \frac{\mathrm{d}B}{B} + \frac{\mathrm{d}T}{T}$$

Thus, if the percentage changes in the main dimensions are small, their sum will provide the percentage change in the displacement.

Similarly as shown in Chapter 3,

$$\overline{BM} = \frac{I_T}{\nabla} = (\text{constant}) \times \frac{B^2}{T}$$

and by the same process

$$\frac{\mathrm{d}\,\overline{BM}}{\overline{BM}} = 2\frac{\mathrm{d}B}{B} - \frac{\mathrm{d}T}{T}$$

Also

$$\frac{\mathrm{d}\,\overline{BM}}{\overline{BM}} = \frac{\mathrm{d}\,\overline{BG} + \mathrm{d}\,\overline{GM}}{\overline{BM}} = \frac{\mathrm{d}\,\overline{BG} + \mathrm{d}\,\overline{GM}}{\overline{BG} + \overline{GM}}$$

Some special cases are worth considering.

(*a*) Where it is desired to change beam only, the increase in displacement is given by

$$\mathrm{d}\Delta = \Delta\frac{\mathrm{d}B}{B}$$

and the effect on $\overline{GM}$ is deduced as follows:

$$\frac{\mathrm{d}\,\overline{BG} + \mathrm{d}\,\overline{GM}}{\overline{BM}} = 2\frac{\mathrm{d}B}{B}$$

If it is assumed that $\overline{KG}$ remains unaltered then $\mathrm{d}\,\overline{BG} = 0$ and

$$\mathrm{d}\,\overline{GM} = 2\,\overline{BM}\cdot\frac{\mathrm{d}B}{B}$$

Since, in general, $\overline{BM}$ is greater than $\overline{GM}$ the percentage increase in metacentric height is more than twice that in beam.

(*b*) If it is desired to keep the displacement and draught unaltered, the length must be decreased in the same ratio as the beam is increased, that is

$$-\frac{\mathrm{d}L}{L} = \frac{\mathrm{d}B}{B}$$

Provided the assumption that $\overline{KG}$ is unchanged remains true the change in metacentric height is as above.

(*c*) To maintain constant displacement and length the draught must be changed in accord with

$$\frac{\mathrm{d}T}{T} = -\frac{\mathrm{d}B}{B}$$

This change in draught may, or may not, cause a change in $\overline{KG}$. There are a number of possibilities two of which are dealt with below.

(i) $\overline{KG} = (\text{constant}) \times T$. But $\overline{KB} = (\text{constant}) \times T$ also, i.e. $\overline{BG} = \overline{KG} - \overline{KB} = (\text{constant}) \times T$.

It should be noted that these constants of proportionality are not necessarily the same, hence

$$\frac{\mathrm{d}\,\overline{\mathrm{BG}}}{\overline{\mathrm{BG}}} = \frac{\mathrm{d}T}{T} = -\frac{\mathrm{d}B}{B}$$

It has been shown that

$$\frac{\mathrm{d}\,\overline{\mathrm{BG}} + \mathrm{d}\,\overline{\mathrm{GM}}}{\overline{\mathrm{BG}} + \overline{\mathrm{GM}}} = 2\frac{\mathrm{d}B}{B} - \frac{\mathrm{d}T}{T} = \frac{3\,\mathrm{d}B}{B}$$

Hence

$$-\overline{\mathrm{BG}}\frac{\mathrm{d}B}{B} + \mathrm{d}\,\overline{\mathrm{GM}} = 3(\overline{\mathrm{BG}} + \overline{\mathrm{GM}})\frac{\mathrm{d}B}{B}$$

and

$$\mathrm{d}\,\overline{\mathrm{GM}} = (4\overline{\mathrm{BG}} + 3\overline{\mathrm{GM}})\frac{\mathrm{d}B}{B}$$

(ii) $\overline{\mathrm{KG}}$ = constant. Since

$$\overline{\mathrm{BG}} = \overline{\mathrm{KG}} - \overline{\mathrm{KB}}$$

in this case, the change in $\overline{\mathrm{BG}} = \mathrm{d}\,\overline{\mathrm{BG}}$

$$= -(\text{change in } \overline{\mathrm{KB}})$$

$$= -\overline{\mathrm{KB}}\frac{\mathrm{d}T}{T} = \overline{\mathrm{KB}}\frac{\mathrm{d}B}{B}$$

As before

$$\mathrm{d}\,\overline{\mathrm{BG}} + \mathrm{d}\,\overline{\mathrm{GM}} = 3(\overline{\mathrm{BG}} + \overline{\mathrm{GM}})\frac{\mathrm{d}B}{B}$$

$$\mathrm{d}\,\overline{\mathrm{GM}} = (3\,\overline{\mathrm{BG}} + 3\,\overline{\mathrm{GM}} - \overline{\mathrm{KB}})\frac{\mathrm{d}B}{B}$$

$$\mathrm{d}\,\overline{\mathrm{GM}} = (4\,\overline{\mathrm{BG}} + 3\,\overline{\mathrm{GM}} - \overline{\mathrm{KG}})\frac{\mathrm{d}B}{B}$$

EXAMPLE 1. A ship has the following principal dimensions: $L = 360\,\mathrm{m}$, $B = 42\,\mathrm{m}$, $T = 12\,\mathrm{m}$, $\Delta = 1200\,\mathrm{MN}$, $\overline{\mathrm{KB}} = 7\,\mathrm{m}$, $\overline{\mathrm{KM}} = 21\,\mathrm{m}$, $\overline{\mathrm{KG}} = 18\,\mathrm{m}$.

It is desired to reduce draught to 11 m, keeping length and displacement constant and coefficients of fineness constant. Assuming $\overline{\mathrm{KG}}$ remains unaltered, calculate the new $\overline{\mathrm{GM}}$ and the new beam.

Solution:

$$\frac{\mathrm{d}\Delta}{\Delta} = 0 = \frac{\mathrm{d}L}{L} + \frac{\mathrm{d}B}{B} + \frac{\mathrm{d}T}{T}$$

Hence

$$\frac{\mathrm{d}B}{B} = \frac{1}{12} \quad \therefore \quad \mathrm{d}B = \frac{42}{12} = 3.5\,\mathrm{m}.$$

i.e.

new beam = 45.5 m.

$$\frac{\mathrm{d}\,\overline{\mathrm{BM}}}{\overline{\mathrm{BM}}} = 2\frac{\mathrm{d}B}{B} - \frac{\mathrm{d}T}{T} = 2 \times \frac{1}{12} + \frac{1}{12} = \frac{1}{4}$$

$$\mathrm{d}\,\overline{\mathrm{BM}} = 14 \times \frac{1}{4} = 3.5\,\mathrm{m}.$$

Because $\overline{\mathrm{KB}}$ is proportional to T

$$\frac{\mathrm{d}\,\overline{\mathrm{KB}}}{\overline{\mathrm{KB}}} = \frac{\mathrm{d}T}{T} = -\frac{1}{12}$$

and

$$\mathrm{d}\,\overline{\mathrm{KB}} = -\frac{7}{12}$$

Now

$$\overline{\mathrm{GM}} = \overline{\mathrm{KB}} + \overline{\mathrm{BM}} - \overline{\mathrm{KG}}$$

$$\therefore \quad \text{new } \overline{\mathrm{GM}} = 3 + \left(-\frac{7}{12}\right) + 3.5$$

$$= 5.92\,\mathrm{m}.$$

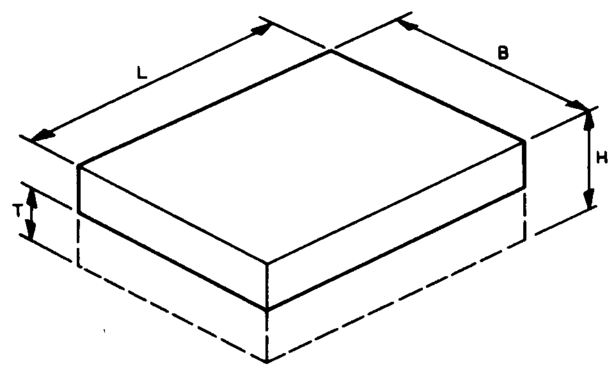


Fig. 4.4 Homogeneous block floating freely

i.e.

$$C = \frac{T}{H}$$

Now for a rectangular block

$$\overline{\mathrm{KB}} = \frac{T}{2}$$

$$\overline{\mathrm{BM}} = \frac{I}{\nabla} = \frac{1}{12}\frac{B^3L}{BLT} = \frac{B^2}{12T}$$

$$\overline{\mathrm{KG}} = \frac{H}{2}$$

and

$$\overline{\mathrm{GM}} = \overline{\mathrm{KB}} + \overline{\mathrm{BM}} - \overline{\mathrm{KG}} = \frac{T}{2} + \frac{B^2}{12T} - \frac{H}{2}$$

$$= \frac{CH}{2} + \frac{B^2}{12CH} - \frac{H}{2}$$

$$= \frac{H}{2}(C-1) + \frac{B^2}{12CH}$$

The condition for positive stability is that

$$\overline{\mathrm{GM}} > 0$$

i.e.

$$\frac{H}{2}(C-1) + \frac{B^2}{12CH} > 0$$

Similar expressions can be derived for homogeneous blocks of other cross-sectional shapes.

EXAMPLE 2. A rectangular block of homogeneous material is 20 m long, 5 m wide and 5 m deep. It is floating freely in fresh water with its longest dimension horizontal. Calculate the range of specific gravity over which the block will be unstable transversely.

Solution: As derived above, the block will be unstable if

$$\frac{H}{2}(C-1) + \frac{B^2}{12CH} < 0$$

i.e.

$$\frac{5}{2}(C-1) + \frac{25}{12C \times 5} < 0$$

$$C - 1 + \frac{1}{6C} < 0$$

$$6C^2 - 6C + 1 < 0$$

When $C = 0$, the block has zero draught and the inequality is not satisfied, i.e. the block is stable.

When $C = \frac{1}{2}$, the left-hand side $= \frac{6}{4} - 3 + 1$. This is less than zero so that the inequality is satisfied.

When $C = 1$, the other possible extreme value, the inequality is not satisfied.

The block is therefore stable at extreme values of C but unstable over some intermediate range. This range will be defined by the roots of the equation

$$6C^2 - 6C + 1 = 0$$

$$C = \frac{6 \pm \sqrt{(36-24)}}{12} = \frac{6 \pm \sqrt{12}}{12}$$

i.e.

$$C = 0.211 \text{ or } 0.789$$

Hence the block is unstable if the specific gravity lies within the range 0.211 to 0.789.

EFFECT OF FREE SURFACES OF LIQUIDS

It is quite a common experience to find it difficult to balance a shallow tray or tin containing water. Stability is essentially a problem of balance, and it is therefore necessary to investigate whether a tank of liquid in a ship affects the ship's initial stability. The problem is considered below in relation to transverse stability, but similar considerations would apply to the case of longitudinal stability.

In practice, a ship has tanks with several different liquids-fresh water for drinking or for boilers, salt water for ballast, fuels of various types and lubricating oils-so the problem must be examined with allowance made for the fact that the density of the liquid may not be the same as that of the sea water in which the ship is floating.

In the following, the subscript s is used to denote quantities associated with the ship or the sea water in which she floats and the subscript £ to denote the liquid or the tank in which it is contained.

Let the ship be floating initially at a waterline WL and let it be heeled through a small angle ¢ to a new waterline WILl. Since by definition the surface of the liquid in the tank is free, it will also change its surface inclinati<p relative to the tank, by the same angle ¢ (see Fig. 4.5).

In Chapter 3, it was shown that for small angles of inclination the transfer of buoyancy is given approximately by the expression

$$\Delta_s \frac{I_s}{\nabla_s}\phi = \rho_s I_s \phi g$$

It follows that, by similar arguments, the transfer of weight due to the movement of the liquid in the tank is approximately

$$\Delta_\ell \frac{I_\ell}{\nabla_\ell}\phi = \rho_\ell I_\ell \phi g$$

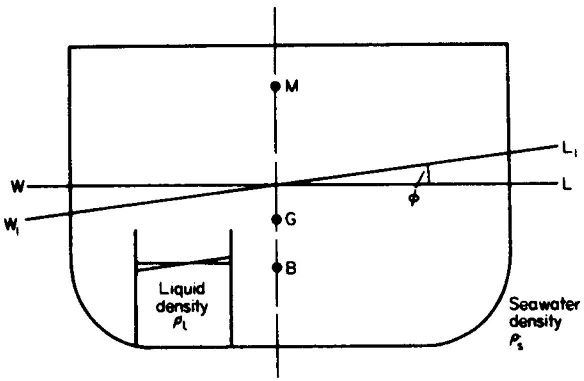


Fig. 4.5 Liquid free surface

This being a transfer of weight opposes the righting moment due to the transfer of buoyancy and results in the effective righting moment acting on the ship being reduced to

$$\Delta_s \overline{GM}_s \phi - \rho_\ell I_\ell \phi g$$

Thus, if $\overline{GM}_F$ is the effective metacentric height allowing for the action of the liquid free surface

$$\Delta_s \overline{GM}_F \phi = \Delta_s \overline{GM}_s \phi - \rho_\ell I_\ell \phi g$$

and therefore

$$\overline{GM}_F = \overline{GM}_s - \frac{\rho_\ell I_\ell}{\Delta_s} g$$

$$= \overline{GM}_s - \left(\frac{\rho_\ell}{\rho_s}\right) \frac{I_\ell}{\nabla_s}$$

It should be noted that the effect of the free surface is independent of the position of the tank in the ship; the tank can be at any height in the ship, at any position along its length and need not be on the middle-line. The effect is also independent of the amount of liquid in the tank provided the second moment of area of the free surface is substantially unchanged when inclined. A tank which is almost empty or almost full can suffer such a change in second moment of area and it is for this reason that if, during an inclining experiment (see later), tanks cannot be completely full or empty they are specified to be half full.

It is usual to regard the free surface effect as being a virtual rise of the centre of gravity of the ship, although it will be appreciated that this is merely a matter of convention.

A similar effect can arise from the movements of granular cargoes stowed in bulk such as grain. Here, however, there is no simple relation between the angle of inclination of the ship and the slope of the surface of the cargo. At small angles of heel the cargo is not likely to shift so there is no influence on initial stability.

The effect of a free liquid surface on stability at larger angles of heel is discussed later.

EFFECT OF FREELY SUSPENDED WEIGHTS

It is more difficult to balance a pole with a weight freely suspended from its top than it is to balance the same pole with the same weight lashed securely to the pole. It is of interest to the naval architect to know what effect weights suspended freely in a ship have on the stability of the ship. Such a practical situation arises when a ship unloads its cargo using its own derricks. Consider the moment the cargo load is lifted from the hold before any translation of the load has taken place. The effect on transverse stability will normally be more critical than that on longitudinal stability, although the principles involved are the same.

Referring to Fig. 4.6, let the weight W be suspended freely from a point S a distance h above the centroid of mass of the weight. Let the ship be floating at a waterline WL and consider it being heeled through a small angle ϕ to a new waterline WIL1. The weight, being freely suspended, will move until it is again vertically below S, i.e. its suspending wire will move through an angle ϕ. Thus, as far as the ship is aware the line of action of the weight of the cargo being lifted always passes through S. Hence, the effect on the ship is as though the weight W were placed at S.

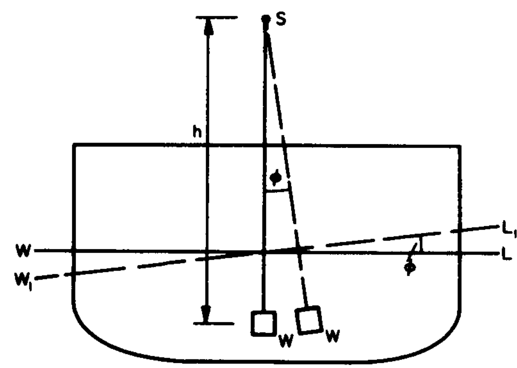


Fig. 4.6 Freely suspended weight

It will be noted that this result is independent of the magnitude of the angle of heel provided the movement of the suspended weight is not restricted in any way.

THE WALL-SIDED FORMULA

A ship is said to be wall-sided if, for the angles of inclination to be considered, those portions of the outer bottom covered or uncovered by the moving water-plane are vertical with the ship upright. No practical ships are truly wall-sided, but many may be regarded as such for small angles of inclination-perhaps up to about 10 degrees.

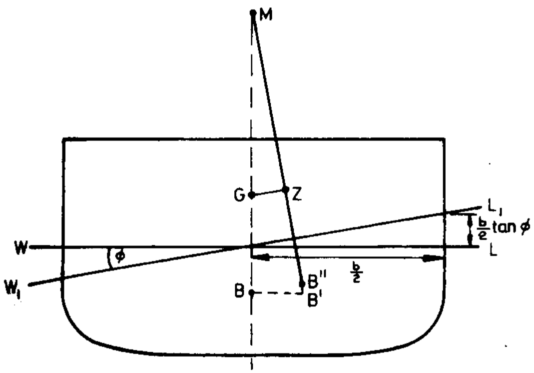


Fig. 4.7 Wall-sided formula

Referring to Fig. 4.7, let the ship be inclined from its initial waterline WL to a new waterline W_1L_1 by being heeled through a small angle ϕ. Since the vessel is wall-sided, WL and W_1L_1 must intersect on the centre line.

The volume transferred in an elemental wedge of length δL where the beam is b is

$$\delta L\left(\frac{1}{2}\times\frac{b}{2}\times\frac{b}{2}\tan\phi\right)=\left(\frac{b^2}{8}\tan\phi\right)\delta L$$

Moment of transfer of volume for this wedge in a direction parallel to WL

$$=\frac{b^2}{8}\tan\phi\frac{2b}{3}\delta L$$

Hence, for the whole ship, moment of transfer of volume is

$$\int_0^L\frac{b^3}{12}\tan\phi\,\mathrm{d}L$$

Hence, horizontal component of shift of B, $\overline{\mathrm{BB'}}$ is given by

$$\overline{\mathrm{BB'}}=\frac{1}{\nabla}\int_0^L\frac{b^3}{12}\tan\phi\,\mathrm{d}L$$

$$=\frac{I}{\nabla}\tan\phi=\overline{\mathrm{BM}}\tan\phi$$

Similarly, vertical shift $\overline{\mathrm{B'B''}}$ is given by

$$\overline{\mathrm{B'B''}}=\frac{1}{\nabla}\int_0^L\frac{b^2}{8}\tan\phi\frac{1}{3}b\tan\phi\,\mathrm{d}L$$

$$=\frac{I}{2\nabla}\tan^2\phi=\frac{\overline{\mathrm{BM}}}{2}\tan^2\phi$$

By projection on to a plane parallel to W_1L_1

$$\overline{\mathrm{GZ}}=\overline{\mathrm{BB'}}\cos\phi+\overline{\mathrm{B'B''}}\sin\phi-\overline{\mathrm{BG}}\sin\phi$$

$$=\overline{\mathrm{BM}}\left[\sin\phi+\frac{\tan^2\phi}{2}\sin\phi\right]-\overline{\mathrm{BG}}\sin\phi$$

$$=\sin\phi\left[\overline{\mathrm{BM}}-\overline{\mathrm{BG}}+\frac{\overline{\mathrm{BM}}}{2}\tan^2\phi\right]$$

i.e.

$$\overline{\mathrm{GZ}}=\sin\phi\left[\overline{\mathrm{GM}}+\frac{\overline{\mathrm{BM}}}{2}\tan^2\phi\right]$$

For a given ship if $\overline{\mathrm{GM}}$ and $\overline{\mathrm{BM}}$ are known, or can be calculated, $\overline{\mathrm{GZ}}$ can readily be calculated using this formula.

Wall-sided vessel containing a tank with vertical sides containing liquid

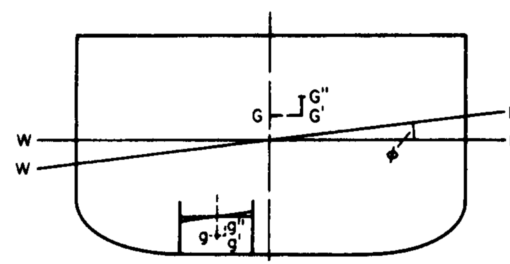


Fig. 4.8 Wall-sided ship with liquid contained in a wall-sided tank

Let the density of the liquid in the tank be ρ times that of the water in which the ship is floating.

Referring to Fig. 4.8, horizontal and vertical shifts of B will be as calculated for the ship without a tank. In addition, however, the centroid of mass of the liquid in the tank will suffer movements

$$\overline{\mathrm{gg'}}=\frac{I_\ell}{\nabla_\ell}\tan\phi$$

and

$$\overline{\mathrm{g'g''}}=\frac{I_\ell}{2\nabla_\ell}\tan^2\phi$$

Hence the centre of gravity of the ship, G, will suffer movements of

$$\overline{\mathrm{GG'}}=\frac{I_\ell}{\nabla_\ell}\tan\phi\times\rho\frac{\nabla_\ell}{\nabla_s}=\rho\frac{I_\ell}{\nabla_s}\tan\phi$$

and

$$\overline{\mathrm{G'G''}}=\rho\frac{I_\ell}{2\nabla_s}\tan^2\phi$$

Thus, the presence of the free surface will effectively reduce $\overline{GZ}$ to $\overline{GZ}_F$ where

$$\overline{GZ}_F = \sin\phi\left[\left(\overline{GM} - \rho\frac{I_\ell}{\nabla_s}\right) + \left(\overline{BM} - \rho\frac{I_\ell}{\nabla_s}\right)\frac{\tan^2\phi}{2}\right]$$

EXAMPLE 3. Calculate the ordinates of the curve of $\overline{GZ}$ for a wall-sided ship up to 15 degrees of heel given that $\overline{GM} = 3$ m and $\overline{BM} = 18$ m.

Solution:

$$\overline{GZ} = \left[\overline{GM} + \frac{\overline{BM}}{2}\tan^2\phi\right]\sin\phi$$

$$\overline{GZ} = [3 + 9\tan^2\phi]\sin\phi$$

Values of $\overline{GZ}$ for varying values of ϕ can be computed as in the table below.

ϕ (deg)	$\tan\phi$	$\tan^2\phi$	$9\tan^2\phi$	$3+9\tan^2\phi$	$\sin\phi$	$\overline{GZ}$ (m)
0	0	0	0	3	0	0
3	0.052	0.0027	0.0243	3.024	0.052	0.157
6	0.105	0.0110	0.0990	3.099	0.105	0.325
9	0.158	0.0250	0.2250	3.225	0.156	0.503
12	0.213	0.0454	0.4086	3.409	0.208	0.709
15	0.268	0.0718	0.6462	3.646	0.259	0.944

Complete stability

CROSS CURVES OF STABILITY

SO far, only stability at small angles of inclination has been discussed. In practice, it is necessary to have a knowledge of stability at large angles of inclination, particularly in the transverse plane. In the following sections, transverse stability at large angles is considered. In principle, the concepts are equally applicable to longitudinal stability but, in practice, they are not normally required because of the relatively small angles of trim a ship can accept for reasons other than stability.

Referring to Fig. 4.9, let the ship be floating initially at a waterline WL. Now let it be heeled through some angle ȸ to a new waterline WI L] such that the displacement remains constant. The buoyancy force b, will act through BI the new position of the centre of buoyancy, its line of action being perpendicular to WI L]. If Z] is the foot of the perpendicular dropped from G on to the line of action of the buoyancy force then the righting moment acting on the ship is given by b,-G-Z-].

This is similar to the concept applied to stability at small angles of heel but, in this case, it is not possible to make use of the concept of a metacentre as the buoyancy force does not intersect the ship's middle line plane in a fixed point.

It should be noted that, in general, a ship when heeled will also trim to maintain its longitudinal equilibrium. This can usually be ignored but where it

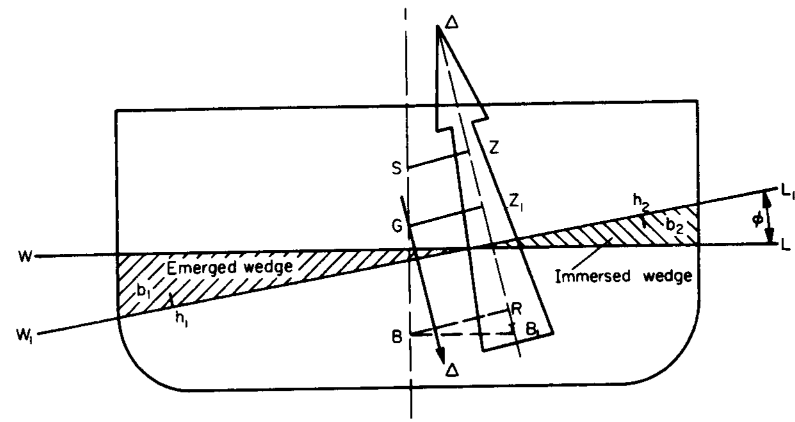


Fig. 4.9 Stability at large angles

is significant, the points shown in Fig. 4.9 must be regarded as projections of the true points on to a transverse plane of the ship.

Since the displacement to W_1L_1 is the same as that to WL, it follows that the volumes of the immersed and emerged wedges are equal.

Let

δ = buoyancy force associated with immersed wedge

b_1, b_2 = centroids of volume of the emerged and immersed wedges respectively

h_1, h_2 = feet of perpendiculars from b_1, b_2 on to W_1L_1

R = Foot of the perpendicular from B on to the line of action of the buoyancy force through B_1.

Then

$$\Delta\overline{BR} = \delta\overline{h_1h_2}$$

and righting moment acting on the ship is given by

$$\Delta\overline{GZ} = \Delta\overline{BR} - \Delta\overline{BG}\sin\phi$$
$$= \Delta\left[\frac{\delta}{\Delta}\overline{h_1h_2} - \overline{BG}\sin\phi\right]$$

This formula is sometimes known as *Atwood's Formula,* although it was known and used before Atwood's time.

Unfortunately, G depends upon the loading of the ship and is not a fixed position. It is therefore more convenient to think in terms of an arbitrary, but fixed, pole S and its perpendicular distance -SZ-from the line of action of the buoyancy force. -SZ-then depends only on the geometry of the ship and can be calculated for various angles of heel and for various values of displacement without reference to a particular loading condition. Since the position of S is

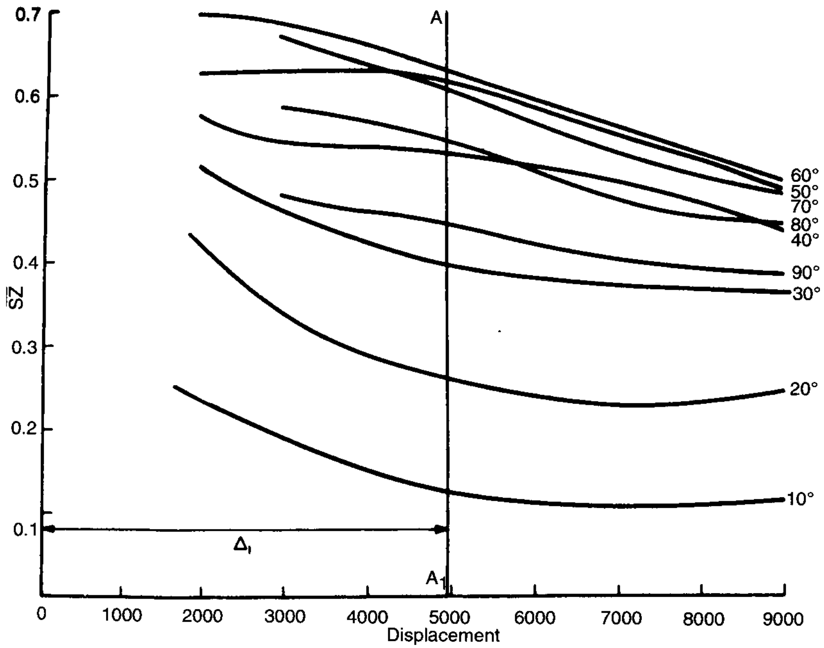


Fig. 4.10 Cross curves of stability

known, when G has been calculated for a particular condition of loading, $\overline{GZ}_1$ can always be calculated from the equation

$$\overline{GZ}_1 = \overline{SZ} + \overline{SG}\sin\phi \quad \text{(S above G)}$$

Where a whole range of ship conditions is required, it is usual to plot values of $\overline{SZ}$ against displacement for each of a number of angles of inclination as shown in Fig. 4.10. These curves are known as *Cross Curves of Stability.*

This form of plotting avoids the need to determine inclined waterplanes for precise displacements. It is convenient usually to adopt a set of waterlines which intersect at the same point on the ship's middle line. Then, if a plot of $\overline{SZ}$ against angle of inclination is required at a constant displacement Δ_1 this can be obtained by reading off the $\overline{SZ}$ values along a vertical line AAI in Fig. 4.10.

If a whole range of ship conditions is not required, the $\overline{SZ}$ curve for constant displacement can be obtained directly as follows:

Referring to Fig. 4.11, let WL be the upright waterline, WILl the inclined waterline at the same displacement and W2L2 the waterline at the inclination of WILl passing through the intersection ofWL with the centre line of the ship.

Let

8 = buoyancy of immersed or emerged wedge between WL and WILl

8_1 = emerged buoyancy between WL and W2L2

82 = immersed buoyancy between WL and W_2L_2.

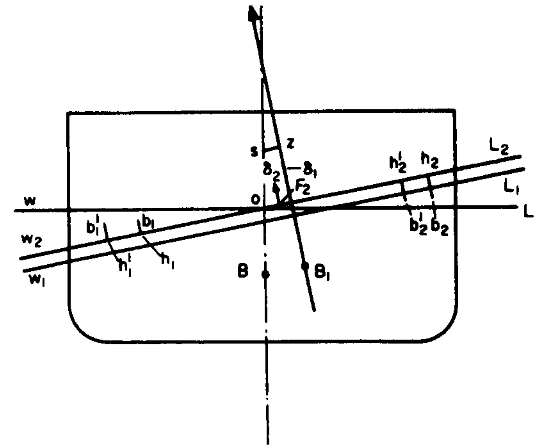


Fig. 4.11 Derivation of $\overline{SZ}$ values for constant displacement

Then, buoyancy of layer between W_1L_1 and W_2L_2 is $\delta_2 - \delta_1$ and let this force act through point F_2 in W_2L_2. If the layer is thin, then F_2 may be regarded as the centroid of the waterplane W_2L_2.

Let

b_1, b_2 be the centroids of the wedges between WL and W_1L_1

b_1', b_2' be the centroids of the wedges δ_1 and δ_2

h_1, h_2 projections of b_1, b_2 on to W_2L_2

h_1', h_2' projections of b_1', b_2' on to W_2L_2

Then

$$\delta\overline{h_1h_2} = \delta_1\overline{Oh_1'} + \delta_2\overline{Oh_2'} - (\delta_2 - \delta_1)\cdot\overline{OF_2}$$

DERIVATION OF CROSS CURVES OF STABILITY

It is now usual to derive data for the plotting of cross curves from a digital computer. A glimpse at manual methods and machines now to be found cp in museums is worthwhile because they illustrate the principles on which computer programs are written.

Integrator methods

The integrator was essentially a machine which measured the area lying within a closed curve and the first and second moments of that area about a datum line-the axis of the integrator. The integrator could be used in two ways.

The 'all-round' method

A body plan for the ship is prepared to as large a scale as is convenient for the integrator to work with. On this body plan is marked the position of the selected pole S, and radiating from S a series of lines representing the angles

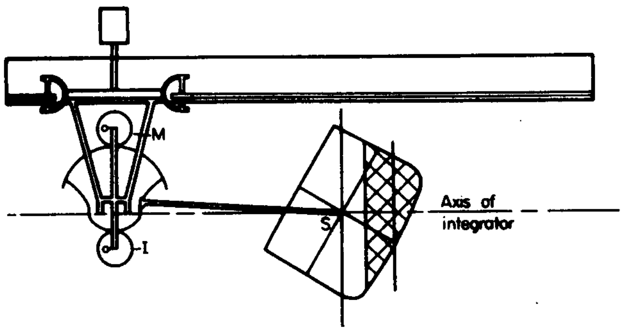


Fig. 4.12 Use of integrator in 'all-round' method

of heel for which SZ is to be measured. Then, on a separate sheet of tracing paper is marked off a series of parallel lines to represent waterlines, spaced so as to cover adequately the range of displacement over which it is desired to study the ship's stability. A line normal to the waterlines is drawn to represent the centre line of the ship when upright.

For each angle to be studied, the tracing paper is set up over the body plan with the 'centre line' passing through S and the 'waterlines' at the correct angle and covering the required range of draught. The integrator is then set up so that its axis lies along the 'centre line' drawn on the tracing paper-see Fig. 4.12.

Then, provided the scaling factors to be applied to the dial readings have been accurately assessed, the integrator can be used to determine the area shown cross-hatched in Fig. 4.12 and the moment of that area about the axis of the integrator. This can be repeated for each section of the ship. If a Simpson body plan is used, the areas and moments must be set out in tabular form, multiplied by the appropriate factor and summated to give the total volume of the ship up to the chosen waterline and the moment about the line through S. Displacement and SZ follow and the process can be repeated for each waterline and then for other angles of heel having adjusted the position of the body plan below the tracing paper. All the data required for plotting the cross curves of stability are now available except insofar as it may be necessary to allow for appendages.

If a Tchebycheff body plan is used then, since the multiplying factor is the same for all sections of the ship, it is possible to move the integrator round all sections in turn and note only the difference between the first and final readings on the dials. In the same way, the number of readings involved with a Simpson body plan can be reduced by dealing with all sections having the same multiplier without stopping.

The 'figure of eight' method

This method makes use of the fact that the only differences between the upright and an inclined condition are the immersed and emerged wedges. By passing the integrator point round these two wedges in opposite senses-i.e. in a figure

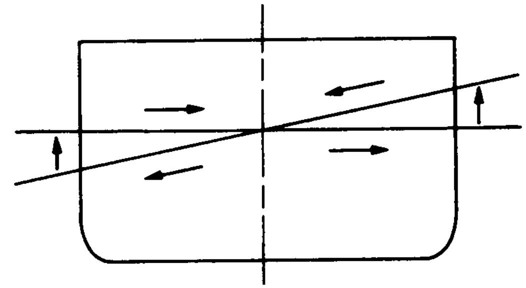


Fig. 4.13 'Figure of eight' method

of eight motion as in Fig. 4.13—the moment of transfer of buoyancy and the change in displacement are obtained.

Barnes's method

This method makes use of the general formula derived above for the transfer of buoyancy, viz.

$$\delta\overline{h_1 h_2} = \delta_1\overline{\mathrm{Oh}'_1} + \delta_2\overline{\mathrm{Oh}'_2} - (\delta_2 - \delta_1)\overline{\mathrm{OF}}_2$$

and the various terms are evaluated using radial integration.

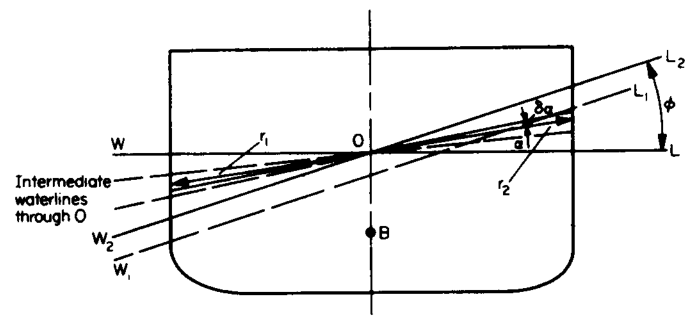


Fig. 4.14 Radial integration for Barnes's method of deriving statical stability data

By considering the element of angle $\delta\alpha$ in Fig. 4.14 and applying the principles of radial integration

$$\delta_1\overline{\mathrm{Oh}'_1} + \delta_2\overline{\mathrm{Oh}'_2} = \int_0^L \int_0^\phi \left[\frac{1}{2}r_1^2\,\mathrm{d}\alpha\frac{2r_1}{3} + \frac{1}{2}r_2^2\,\mathrm{d}\alpha\frac{2r_2}{3}\right]\cos(\phi - \alpha)\,\mathrm{d}x$$

$$= \int_0^L \int_0^\phi \left(\frac{r_1^3 + r_2^3}{3}\right)\cos(\phi - \alpha)\,\mathrm{d}x\,\mathrm{d}\alpha$$

This double integral can be evaluated by drawing intermediate radial waterlines at equal increments of heel and measuring the offsets r at the appropriate stations. The same process can be followed to evaluate

$$\delta_2 = \int_0^L \int_0^\phi \frac{1}{2} r_2^2 \,\mathrm{d}\alpha\,\mathrm{d}x$$

$$\delta_1 = \int_0^L \int_0^\phi \frac{1}{2} r_1^2 \,\mathrm{d}\alpha\,\mathrm{d}x$$

The area and first moment of area of waterplane W_2L_2 are given by the expressions

$$\text{Area} = \int_0^L r_\phi \mathrm{d}x$$

First moment of area about fore and aft axis through 0

$$= \int_0^L \frac{r_\phi^2}{2}\mathrm{d}x$$

By dividing the moment by the area the value of $\overline{OF_2}$ is found.
Substituting in Atwood's formula

$$\overline{SZ} = \left(\frac{\delta_1\overline{Oh'_1} + \delta_2\overline{Oh'_2} - (\delta_2 - \delta_1)\overline{OF_2}}{\Delta}\right) - \overline{BS}\sin\phi$$

Evaluating this expression for various values of ϕ gives the values of $\overline{SZ}$ at constant displacement.

An adaptation of Barnes's method, if a whole range of ship displacements are to be covéred, would be to find the displacement Δ_ϕ corresponding to waterline W_2L_2 through O and the corresponding $\overline{SZ}$, viz:

$$\Delta_\phi = \Delta + \delta_2 - \delta_1$$

$$\overline{SZ}_\phi = \left(\frac{\delta_1\overline{Oh'_1} + \delta_2\overline{Oh'_2}}{\Delta_\phi}\right) - \overline{BS}\sin\phi$$

These values correspond to those obtained by the integrator method.

Reech's method

Let B_α and M_α be the positions of the centre of buoyancy and metacentre corresponding to inclined waterplane $W_\alpha L_\alpha$. From the definition of pro-metacentre (see p. 20), if B_α moves to $B_{\alpha+\delta\alpha}$ for a very small additional inclination $\overline{B_\alpha B_{\alpha+\delta\alpha}} = \overline{B_\alpha M_\alpha}\delta_\alpha$, and $\overline{B_\alpha B_{\alpha+\delta\alpha}}$ is parallel to $W_\alpha L_\alpha$

$$\delta y = \overline{B_\alpha M_\alpha}\cos\alpha\,\delta\alpha$$

$$\delta z = -\overline{B_\alpha M_\alpha}\sin\alpha\,\delta\alpha$$

It follows by radial integration that from waterplane W_1L_1 the position of B_1 relative to B can be expressed as (y, z) where

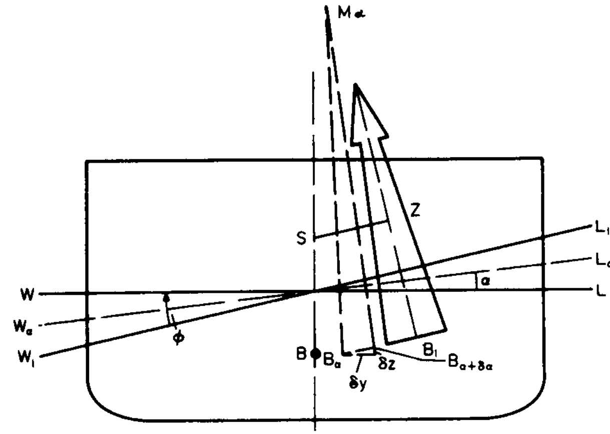


Fig. 4.15 Reech's method

$$y = \int_0^\phi \overline{B_\alpha M_\alpha}\cos\alpha\,\mathrm{d}\alpha$$

$$z = -\int_0^\phi \overline{B_\alpha M_\alpha}\sin\alpha\,\mathrm{d}\alpha$$

The negative sign for z signifies only that B_1 is 'higher' than B.
Hence

$$\overline{SZ} = y\cos\phi - z\sin\phi - \overline{BS}\sin\phi$$

$$= \cos\phi\int_0^\phi \overline{B_\alpha M_\alpha}\cos\alpha\,\mathrm{d}\alpha + \sin\phi\int_0^\phi \overline{B_\alpha M_\alpha}\sin\alpha\,\mathrm{d}\alpha - \overline{BS}\sin\phi$$

To evaluate this expression, it is necessary to determine a series of inclined waterlines at constant displacement and for each waterplane obtain the value of $\overline{BM}$ by finding the values of the second moment of area and dividing by the volume of displacement.

Use of model to obtain curve of statical stability

It has been noted that one limitation of the methods described above to determine the statical stability of a ship is the neglect of trim induced by the heeling of the ship. This can be overcome by finding the true equilibrium waterplane at each inclined waterplane, but this leads to considerable complication. One method of automatically allowing for the effects of trim is to use a scale model of the ship, applying to it known heeling moments and noting the resulting angles of heel.

Normally, the expense of making an accurately scaled model is not justified, but the method can be used where a model already exists for other purposes such as the conduct of ship tank tests. The method is particularly useful where there are large changes in waterplane shape for small changes in heel. **Such** changes can occur with forms which have large rise of floor, very rounded bilges, and/or long well-rounded cut-ups.

Prohaska's method

The stability lever $\overline{GZ}$ can be considered as composed of two parts, viz:

$$\overline{GZ} = \overline{GM}\sin\phi + \overline{MS}$$

The quantity $\overline{MS}$, which, in a projection on a transverse plane, is the distance from the upright metacentre to the line of action of the buoyancy force, is termed the residuary stability lever (see Fig. 4.16). For convenience in non-dimensional plotting, a coefficient C_{RS} is employed for which

$$C_{RS} = \frac{\overline{MS}}{\overline{BM}_{\text{upright}}}$$

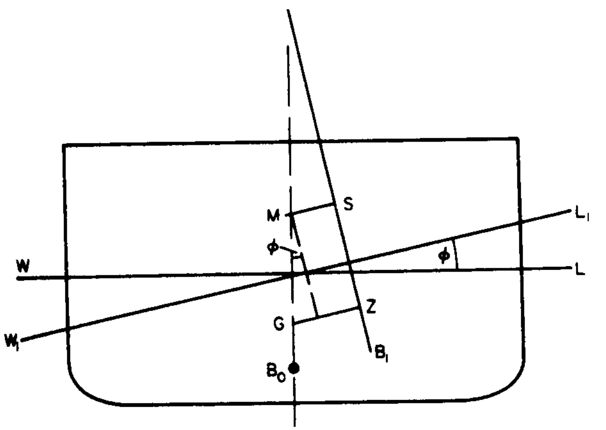


Fig. 4.16

By the use of displacement sheets, the co-ordinates of the centres of buoyancy for 90 and 180 degrees inclination can be calculated relative to those for the upright ship. Also, the meta centric radii for 0, 90 and 180 degrees can be calculated. Using this data and other geometric form data, the derivation of the residual stability curve can be reduced to a calculation in tabular form. The tables used are based on calculations performed on forty-two forms having form characteristics covering the usual merchant ship field, backed up by mathematical treatments assuming that the stability curve can be approximated by a trigonometric series. By its nature the method is approximate, but for general ship forms lying within the range of parameters considered, accuracy comparable with that achieved by the so-called 'exact' methods is obtained.

CURVES OF STATICAL STABILITY

It was explained earlier, that it is usually convenient to derive cross curves of stability from purely geometric considerations by making use of an arbitrary

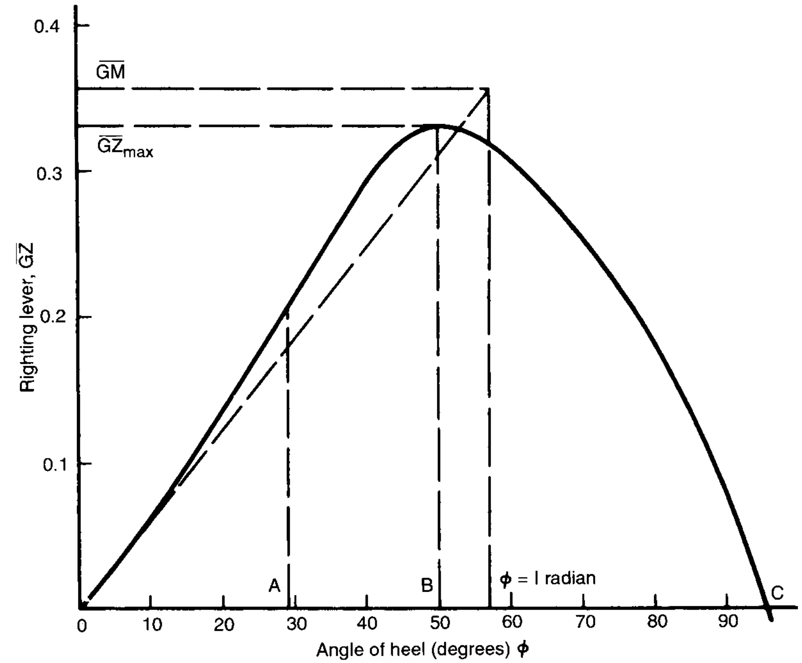


Fig. 4.17 Curve of statical stability

pole about which to measure righting moments. It was also seen, that it simplifies the work in deriving stability curves if displacement is allowed to vary as the ship is inclined and if both displacement and righting lever are calculated for each waterline.

For practical applications, however, it is necessary to present stability in the form of righting moments or levers about the centre of gravity, as the ship is heeled at constant displacement. Such a plot will in general appear as in Fig. 4.17 and is known as a *statical stability curve*.

The curve of statical stability, or $\overline{GZ}$ curve as it is commonly called, is derived from the cross curves of stability by setting up a vertical line such as AA_1 for displacement Δ_1 shown on Fig. 4.10. The $\overline{SZ}$ value corresponding to each angle of inclination plotted can be read off and then, using Fig. 4.9 and the $\overline{SG}$ known for the particular loading of the ship,

$$\overline{GZ} = \overline{SZ} - \overline{SG}\sin\phi$$

MAIN FEA TURES OF THE -G-ZCURVE

Certain features of the GZ curve are of particular significance and are useful parameters with which to define the stability possessed by a given design. They include:

(*a*) Slope at the origin. For small angles of heel, the righting lever is proportional to the angle of inclination, the metacentre being effectively a fixed point. It follows, that the tangent to the $\overline{GZ}$ curve at the origin represents the metacentric height.

(*b*) Maximum $\overline{GZ}$. This is proportional to the largest steady heeling moment that the ship can sustain without capsizing, and its value and the angle at which it occurs are both important.

(*c*) Range of stability. At some angle, often greater than 90 degrees, the $\overline{GZ}$ value reduces to zero and becomes negative for larger inclinations. This angle is known as the *angle of vanishing stability* and the range of angle (Oe in Fig. 4.17) for which $\overline{GZ}$ is positive is known as the *range of stability*. For angles less than this, a ship will return to the upright state when the heeling moment is removed.

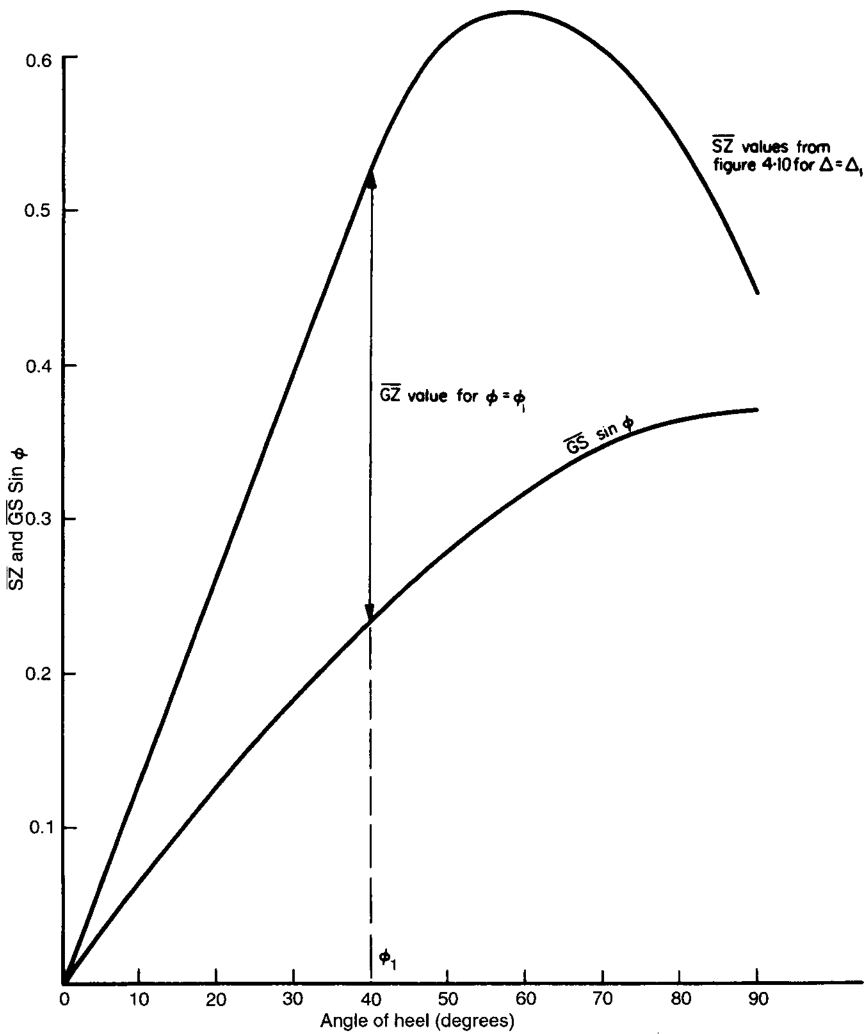


Fig. 4.18

(*d*) Angle of deck edge immersion. For most ship forms, there is a point of inflexion in the curve (shown in Fig. 4.17 as occurring at angle OA) corresponding roughly to the angle at which the deck edge becomes immersed. In general, of course, the angle at which the deck edge is immersed varies along the length, but is within a fairly narrow band for the larger sections amidships which exert most influence upon the stability. This point is not so much of interest in its own right as in the fact that it provides guidance to the designer upon the possible effect on stability of certain design changes.

(*e*) Area under the curve. The area under the curve represents the ability of the ship to absorb energy imparted to it by winds, waves or any other external agency. This concept is developed more fully later.

ANGLE OF LOLL

A special case arises when $\overline{GM}$ is negative but $\overline{GZ}$ becomes positive at some reasonable angle of heel. This is illustrated in Fig. 4.19 as ϕ_1. If the ship is momentarily at some angle of heel less than ϕ_1, the moment acting due to $\overline{GZ}$ tends to increase the heel. If the angle is greater than ϕ_1, the moment tends to reduce the heel. Thus the angle ϕ_1 is a position of stable equilibrium. Unfortunately, since the $\overline{GZ}$ curve is symmetrical about the origin, as ϕ_1 is decreased, the ship eventually passes through the upright condition and will then suddenly lurch over towards the angle ϕ_1 on the opposite side and overshoot this value before reaching a steady state. This causes an unpleasant rolling motion which is often the only direct indication that the heel to one side is due to a negative $\overline{GM}$ rather than to a positive heeling moment acting on the ship.

As a special case, consider a wall-sided vessel with negative $\overline{GM}$. In this case,

$$\overline{GZ} = \sin\phi\left(\overline{GM} + \tfrac{1}{2}\overline{BM}\tan^2\phi\right)$$

$\overline{GZ}$ is zero when $\sin\phi = 0$. This merely demonstrates that the upright condition is one of equilibrium. $\overline{GZ}$ is also zero when $\overline{GM} + \frac{1}{2}\overline{BM}\tan^2\phi = 0$. i.e. when

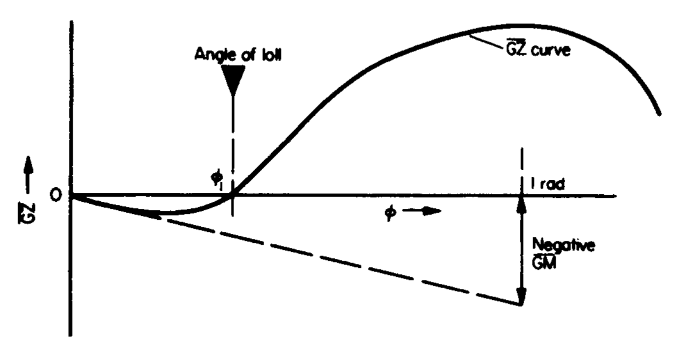


Fig. 4.19 Angle of loll ϕ_1

$$\tan\phi = \pm\sqrt{\left(-\frac{2\overline{GM}}{\overline{BM}}\right)}$$

Also, in this case, the slope of the $\overline{GZ}$ curve at ϕ_1 will be given by

$$\frac{d\overline{GZ}}{d\phi} = \cos\phi(\overline{GM} + \tfrac{1}{2}\overline{BM}\tan^2\phi) + \sin\phi\,\overline{BM}\tan\phi\sec^2\phi$$

$$= 0 + \overline{BM}\tan^2\phi_1/\cos\phi_1 \quad (\text{putting } \phi = \phi_1)$$

$$= -2\overline{GM}/\cos\phi_1$$

EFFECT OF FREE LIQUID SURFACES ON STABILITY AT LARGE ANGLES OF INCLINATION

It was shown earlier, that the effect of a free liquid surface on initial stability is equivalent to a reduction in the metacentric height of

$$\left(\frac{\rho_\ell}{\rho_s}\right)\frac{I_\ell}{\nabla_s}$$

This was derived by considering the transfer of weight of liquid which occurred in opposition to the righting moment induced by the transfer of buoyancy at the waterplane. The same principle can be applied to any form of tank at any angle of heel.

Referring to Fig. 4.20, let a ship initially floating at waterline WL be inclined, at constant displacement, through angle 1.Then if wl is the upright free surface of liquid in the tank shown, the surface when inclined becomes wIll also at angle 1 to wl.

Knowing the shape of the tank, the transfer of weight can be computed and translated into an effective reduction in the righting moment due to transfer of buoyancy in the wedges between WL and WILl. It follows, by analogy with the

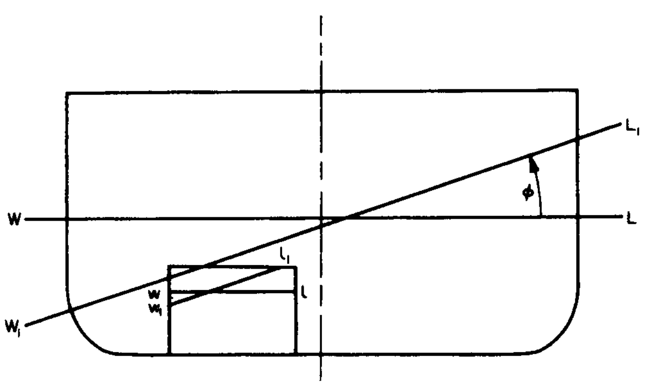


Fig. 4.20 Effect on stability of free surface at large angles of heel

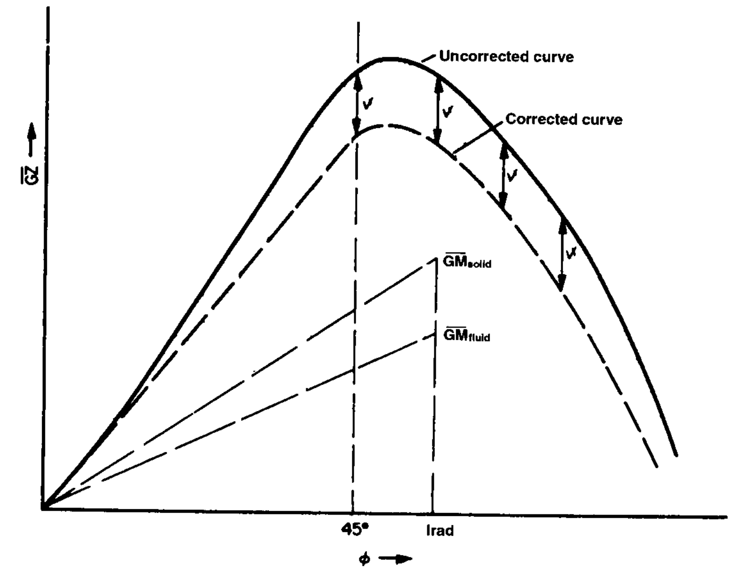


Fig. 4.21 $\overline{GZ}$ curve corrected for free surface

upright case, that the reduction in *slope* of the $\overline{GZ}$ curve at angle ϕ will be given by

$$\left(\frac{\rho_\ell}{\rho_s}\right)\frac{(I_\ell)_\phi}{\nabla_s}$$

where $(I_\ell)_\phi$ is the moment of inertia of the liquid surface at angle ϕ.

If the tank is not a simple geometric shape, the transfer of weight can be computed using an integrator or any of the other methods discussed earlier for the derivation of cross curves of stability. Even so, it is fairly tedious to evaluate for sufficient angles of inclination in order to define the effective $\overline{GZ}$ curve precisely, and it is common for some simplifying assumption to be made.

It is common practice to compute the metacentric height and the $\overline{GZ}$ value for 45 degrees heel allowing for the free surface. The effective $\overline{GZ}$ curve up to 45 degrees inclination is then constructed by drawing the curve through the 45 degree spot, following the general character of the uncorrected curve and fairing into the modified tangent at the origin. For angles greater than 45 degrees, the reduction of $\overline{GZ}$ at 45 degrees is applied as a constant correction. This method is illustrated in Fig. 4.21.

SURFACES OF B, M, F AND Z

When a body floats in water, there are unique positions of the centre of buoyancy, metacentre, centre of flotation of the waterplane and Z, the foot of the perpendicular from the e.g. on to the line of the buoyancy force. If the body is rotated at constant displacement, these points will, in general, move to new positions. Considering all possible inclinations B, M, F and Z trace out surfaces

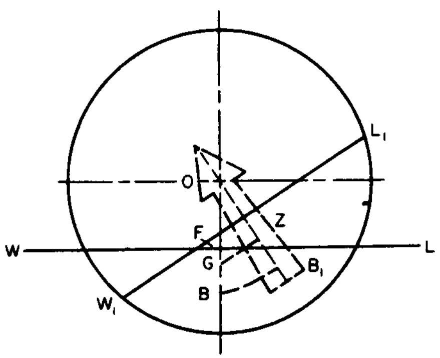


Fig. 4.22 Sphere floating in fluid at WL

which are fixed relative to the body. The simplest case to consider is that of a sphere as in Fig. 4.22.

Let the sphere float at a waterline WL as shown (the argument is unaffected if WL is above the centre of the sphere, O). Then, by arguments of symmetry it follows that for any inclination, OB will be constant, OF will be constant, F being the foot of the perpendicular from O on to WL, and M will always be at O. Thus, in this case, M is a single point and the surfaces traced out by B and F are spheres of radius OB and OF, respectively. As the sphere rotates, the angle OZG is always a right angle. Hence, the locus of Z is a sphere with $\overline{OG}$ as diameter.

Interest, as far as a ship is concerned, is usually centred on transverse stability, i.e. on rotation at constant displacement about a fore and aft axis. In this case, B, M, F and Z trace out lines across the surfaces discussed above. The projections of these traces on to a plane normal to the axis of rotation are termed the *curves of* B, M, F *and* Z and it is of interest to study certain of their characteristics. For a symmetrical ship form, of course, all curves will be symmetrical about the middle line of the ship.

B *curve or curve of buoyancy*

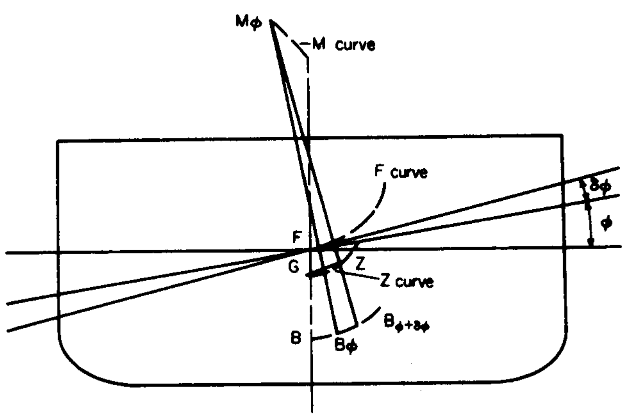


Fig. 4.23 Curves of B, M, F *and* Z

Let B_ϕ and $B_{\phi+\delta\phi}$ be the positions of the centre of buoyancy corresponding to inclinations ϕ and $\phi+\delta\phi$. Then $\overline{B_\phi B}_{\phi+\delta\phi}$ is parallel to the line joining the centroids of volume of the two wedges between the waterplanes at ϕ and $\phi+\delta\phi$. In the limit, $\overline{B_\phi B}_{\phi+\delta\phi}$ becomes the tangent to the curve of buoyancy and this tangent must be parallel to the corresponding waterplane.

The projections of the lines of action of buoyancy through B_ϕ and $B_{\phi+\delta\phi}$ intersect in a point which, in the limit, becomes M_ϕ, called the *prometacentre* and

$$\overline{B_\phi M}_\phi = \frac{I_\phi}{\nabla}$$

The co-ordinates of the point on the curve of buoyancy corresponding to any given inclination ϕ have already been shown to be

$$y = \int_0^\phi \overline{B_\alpha M}_\alpha \cos\alpha \, d\alpha$$

$$z = -\int_0^\phi \overline{B_\alpha M}_\alpha \sin\alpha \, d\alpha$$

The M *curve or metacentric curve*

The metacentric curve is the projection of the locus of pro-metacentres, for inclinations about a given axis, on to a plane normal to that axis.

Let W_1L_1 and W_2L_2 be two waterlines at relative inclination of $\delta\phi$ and let B_1 and B_2, M_1 and M_2 be the corresponding positions of the centre of buoyancy and pro-metacentre. Let C_M be the centre of curvature of the metacentric curve at M_1. Then $\overline{C_M M_1}$ is normal to $\overline{B_1 M_1}$ and therefore parallel to W_1L_1.

Referring to Fig. 4.24

$$\overline{M_1M_2} = \delta(\overline{B_\phi M}_\phi) = \overline{C_M M_1}\delta\phi$$

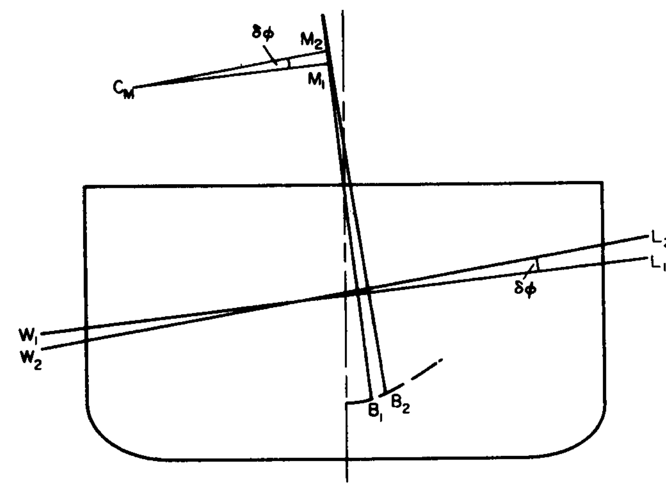


Fig. 4.24 Metacentric curve

and in the general case

$$\overline{C_M M_\phi} = \frac{d\overline{B_\phi M_\phi}}{d\phi} = \frac{1}{\nabla}\frac{dI_\phi}{d\phi}$$

The F *curve or curve of flotation*

For very small changes in inclination successive waterplanes intersect in a line through their centres of flotation. In the limit, therefore, the waterplane at any angle is tangential to the curve of flotation at the point corresponding to that inclination.

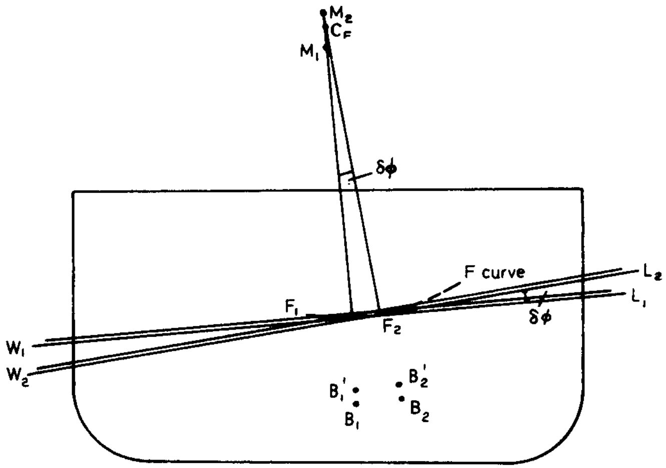


Fig. 4.25 Curve of flotation

Let W_1L_1 and W_2L_2 be waterplanes at a relative inclination $\delta\phi$ and B_1 and B_2, F_1 and F_2 and M_1 and M_2 their respective centres of buoyancy and flotation and pro-metacentres. Let the lines through F_1 and F_2 normal to W_1L_1 and W_2L_2 intersect in C_F which, in the limit, as $\delta\phi$ diminishes, will become the centre of curvature of the F curve.

Now, consider a small layer added to each waterline, the added volume of buoyancy in each case being $\delta\nabla$. Then let B_1B_2 take up new positions $B_1'B_2'$ and so on. Effectively, $\delta\nabla$ is added at F_1 and F_2 for the two inclinations and B_1, B_1' and F_1 will lie on a straight line as will B_2, B_2' and F_2.

By taking moments

$$\frac{\overline{B_1B'_1}}{\overline{B_1F_1}} = \frac{\delta\nabla}{\nabla + \delta\nabla} = \frac{\overline{B_2B'_2}}{\overline{B_2F_2}}$$

Also $\overline{B_1B_2}$, $\overline{B_1'B'_2}$ and $\overline{F_1F_2}$ are parallel, and

$$\overline{B_1B_2} = \overline{B_1M_1}\delta\phi = \frac{I_1}{\nabla}\delta\phi, \text{ if } \delta\phi \text{ is small.}$$

Hence

$$\frac{\overline{B_1F_1}}{\overline{B_1B_1'}} = \frac{\overline{B_1B_2} - \overline{F_1F_2}}{\overline{B_1B_2} - \overline{B_1'B_2'}}$$

By substitution, this becomes

$$\frac{\nabla + \delta\nabla}{\delta\nabla} = \frac{\dfrac{I_1}{\nabla}\delta\phi - \overline{C_F F_1}\delta\phi}{\dfrac{I_1}{\nabla}\delta\phi - \dfrac{I_1 + \delta I}{\nabla + \delta\nabla}\delta\phi}$$

From which it follows that, in the limit,

$$C_F F_1 = \frac{dI}{d\nabla}$$

This is known as Leclert's theorem.

The Z *curve*

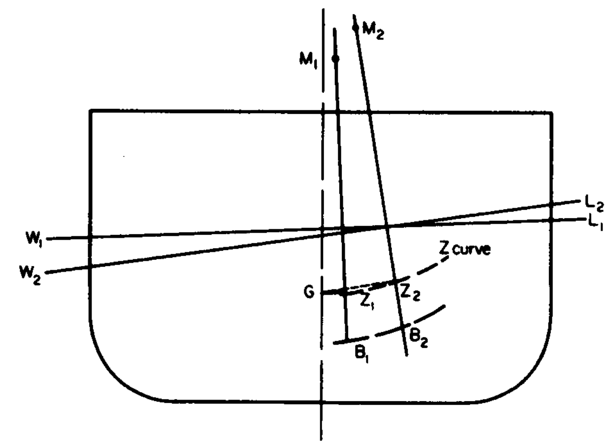


Fig. 4.26 Z *curve*

The Z curve is the projection on to a transverse plane of the foot of the perpendicular from G on to successive lines of action of the buoyancy force.

Curves of B, M, F *and* Z *for a typical ship*

The form of the B, M, F and Z curves is illustrated in Fig. 4.27 for a 67.5 m ship. The principal features of this design are:

Deep displacement, (MN)	6.78
L.B.P., (m)	67.5
Beam, (m)	7.58
Mean draught, (m)	2.23
Block coefficient	0.595
Prismatic coefficient	0.617
Waterplane coefficient	0.755

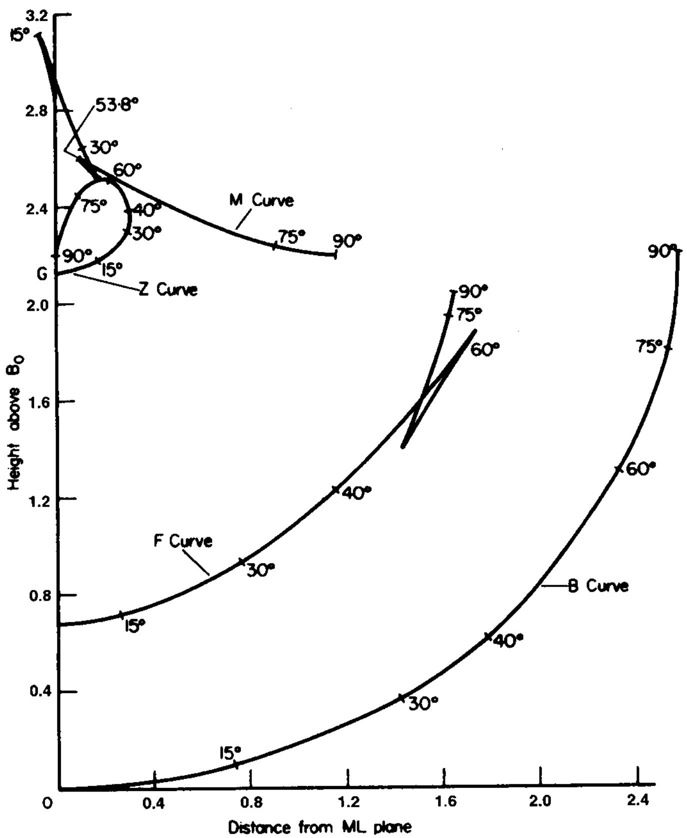


Fig. 4.27

INFLUENCE OF SHIP FORM ON STABILITY

It has been shown how a designer, faced with a given form and loading condition, can assess the stability characteristics of a ship. It is unlikely that the first choice of form will satisfy the requirements completely. Initial stability, maximum $\overline{GZ}$ and range may all be too great or too small. Perhaps one feature alone may need modification. How then is the designer to modify the form to produce the required result? The effect of small changes in principal dimensions on initial stability has already been discussed, and if only such a change is required this method might be adopted provided that the consequential changes in stability at large angles are acceptable.

In general, however, there must be a certain amount of 'trial and error' in the process. That is to say, the designer must make certain changes, re-calculate the stability and assess what additional changes may be necessary. Computers are used for these calculations as extensive stability data can be obtained in a very short time. Indeed, if the range of forms the designer has to deal with is limited,

the computer can produce a 'methodical series' of forms from which the designer can select a suitable design.

Although the designer cannot predict precisely the steps necessary to effect a desired change in stability characteristics, it is essential to know the general effect of changing the ship form in a specified manner. The following comments are intended as a general qualitative guide to the changes that can be expected. It will be appreciated that, in general, it is impossible to vary one parameter alone without some consequential change being required in some other factor.

Length

Iflength is increased in proportion to displacement, keeping beam and draught constant, the transverse $\overline{KB}$ and $\overline{BM}$ are unchanged. In general, increasing length increases $\overline{KG}$ so reducing initial stability although this will not be the case for a truly geometrically similar ship.

If length is increased at the expense of beam, there will be a reduction in stability over the full range. Iflength is increased at the expense of draught, there will, in general, be an increase in initial stability but a reduction at larger angles.

Beam

Of the form parameters which can be varied by the designer, the beam has the greatest influence on transverse stability. It has been shown that

$$\overline{\mathrm{BM}} \propto \frac{B^2}{T}$$

Thus $\overline{BM}$ will increase most rapidly if beam is increased at the expense of the draught. If freeboard remains constant, the angle of deck edge immersion will decrease and stability at larger angles will be reduced. If the total depth of ship remains constant, then stability will be reduced by the increase in $\overline{BG}$ and, although this is offset by the increased beam at small angles, the stability at larger angles will suffer.

Draught

Reduction in draught in proportion to a reduction in displacement increas' initial stability, leads to greater angle of deck immersion but reduces stability at large angles.

Displacement

Changes in length, beam and draught associated with displacement are discussed above. If these parameters are kept constant then displacement increases lead to a fatter ship. In general, it can be expected that the consequential filling out of the waterline will more than compensate for the increased volume of displacement and $\overline{BM}$ will increase. Fattening of the ship in this way is also likely to lead to a fall in G. These changes will also, in general, enhance stability at all angles.

Centre of gravity

The best way, but often the most difficult, of achieving improved stability at all angles is to reduce the height of the centre of gravity above the keel. This is clear from a consideration of the formulae used to derive $\overline{GZ}$ in earlier sections of this chapter. What is perhaps not so immediately obvious, but which cannot be too highly emphasized, is that too high a value of $\overline{KG}$ results in poor stability at larger angles no matter what practical form changes are made.

The effect of change in $\overline{KG}$, if not associated with any change of form, is given directly by the relationship

$$\overline{GZ} = \overline{SZ} - \overline{SG}\sin\phi$$

For a more thorough review of the influence of form changes, the student is advised to consult papers published in the various transactions of learned bodies. The broad conclusion is that the ratios of beam to draught and depth of ship have a comparatively greater influence on stability than that caused by the variation of the fullness parameters of the forms.

The results of applying regressional analysis to data from 31 ships to obtain expressions for C_{RS} in terms of other hull parameters showed that the following expressions give reasonable estimates at 30 degrees of heel

$$C_{RS} = 0.8566 - 1.2262\,\overline{KB}/T - 0.035\,B/T$$
$$C_{RS} = -0.1859 - 0.0315\,B/T + 0.3526\,C_M$$

STABILITY OF A COMPLETELY SUBMERGED BODY

As in the case of surface ships there will be a position of rotational equilibrium when B and G are in the same vertical line. Let us determine the condition for stable equilibrium for rotation about a horizontal axis.

Since the body is completely submerged, the buoyancy force always acts through B. Viewing this another way, in the absence of any waterplane $\overline{BM}$ must be zero so that B and M become coincident. If the body is inclined through an angle ϕ, the moment acting on it is given by

$$\Delta\overline{BG}\sin\phi$$

It is clear from Fig. 4.28, that this will tend to bring the body upright if B lies above G. This then is the condition for stable equilibrium. If B and G coincide, the body has neutral stability and if B lies below G it is unstable.

Both the transverse and longitudinal stability of a submerged body are the same and the $\overline{GZ}$ curve becomes a sine curve as in Fig. 4.29. A submarine is typical although the relative positions of B and G, and therefore the stability in the two directions, may be changed by flooding or emptying tanks.

Another mode of stability of a completely submerged body which is of interest is that associated with depth of submersion. If the body in Fig. 4.28 is initially in equilibrium, and is disturbed so as to increase its depth of submersion it is subject to greater hydrostatic pressure. This causes certain compression of the body which, if the body is more readily compressed

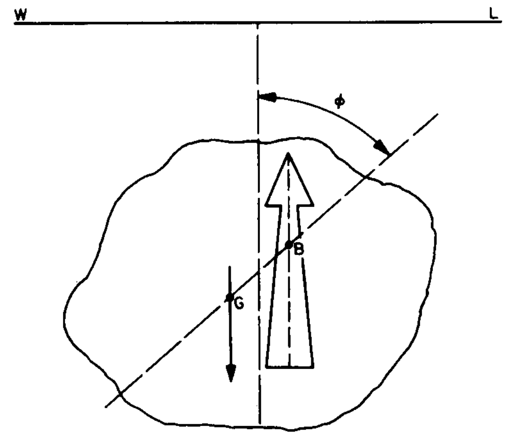


Fig. 4.28 Stability of completely submerged body

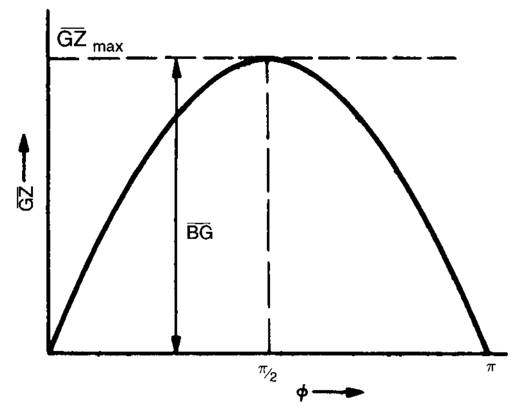


Fig. 4.29 Statical stability curve for submarine

than water, leads to a reduction in the buoyancy force. The weight of the body remains constant so a resultant downward force is created. This will tend to increase the depth of submersion still further. In these circumstances-applicable to a submarine-the body is unstable in depth maintenance.

Generally, in the case of a submarine the degree of instability is no great embarrassment and, in practice, the effect is usually masked by changes in water density associated with temperature or salinity as depth changes. It becomes important, however, with deep submergence vehicles.

Dynamical stability

The *dynamical stability* of a ship at a given angle of heel is defined as the work done in heeling the ship to that angle very slowly and at constant displacement, i.e., ignoring any work done against air or water resistance.

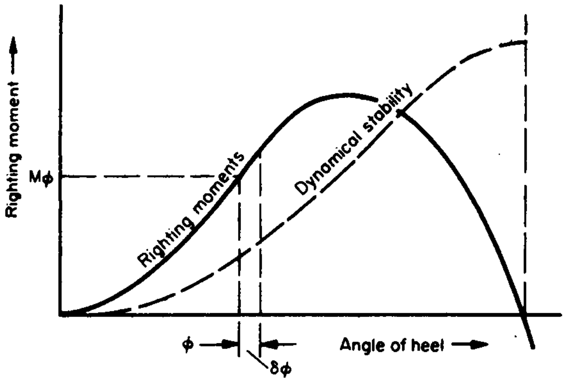


Fig. 4.30

Consider a ship with a righting moment curve as shown in Fig. 4.30.

Let the righting moment at an angle of heel ϕ be M_ϕ. Then, the work done in heeling the ship through an additional small angle $\delta\phi$ is given approximately by

$$M_\phi \delta\phi$$

Hence, the total work in heeling to an angle Φ is

$$\int_0^\Phi M_\phi \mathrm{d}\phi$$

$$= \int_0^\Phi \Delta \overline{\mathrm{GZ}}_\phi \mathrm{d}\phi$$

The dynamical stability at any angle, therefore, is proportional to the area under the statical stability curve up to that angle.

There is an alternative way of determining the work done in heeling a ship. Weight and displacement forces remain constant during the process but they are separated vertically, thereby raising the potential energy of the ship.

Referring to Fig. 4.9, let B move to B_1 when ship is heeled from waterline WL to waterline $\mathrm{W}_1\mathrm{L}_1$. Increased vertical separation $= \overline{\mathrm{B}_1\mathrm{Z}_1} - \overline{\mathrm{BG}}$. Hence, dynamical stability at angle ϕ is given by

$$\Delta(\overline{\mathrm{B}_1\mathrm{Z}_1} - \overline{\mathrm{BG}})$$

Let v be the volume of the immersed or emerged wedge, let their centroids of volume be at b_1 and b_2 and let h_1 and h_2 be the feet of perpendiculars dropped from b_1 and b_2 on to $\mathrm{W}_1\mathrm{L}_1$.
Then

$$\overline{\mathrm{B}_1\mathrm{R}} = \frac{v(\mathrm{b}_1\mathrm{h}_1 + \mathrm{b}_2\mathrm{h}_2)}{\nabla}$$

Now

$$\overline{\mathrm{B}_1\mathrm{Z}_1} = \overline{\mathrm{B}_1\mathrm{R}} + \overline{\mathrm{BG}} \cos\Phi$$

Therefore

$$\overline{\mathrm{B}_1\mathrm{Z}_1} = \frac{v(\mathrm{b}_1\mathrm{h}_1 + \mathrm{b}_2\mathrm{h}_2)}{\nabla} + \overline{\mathrm{BG}} \cos\Phi$$

and the dynamical stability is given by

$$\Delta\left[\frac{v(\mathrm{b}_1\mathrm{h}_1 + \mathrm{b}_2\mathrm{h}_2)}{\nabla} - \overline{\mathrm{BG}}(1 - \cos\Phi)\right]$$

This formula is known as *Moseley's Formula*. Although this expression can be evaluated for a series of angles of heel, and the curve of dynamical stability drawn, its application is laborious without a computer. Dynamical stability should not be confused with dynamic stability of course which is discussed in Chapter 13.

Stability assessment

STABILITY STANDARDS

Having shown how the stability of a ship can be defined and calculated, it remains to discuss the actual standard of stability to be aimed at for any particular ship bearing in mind its intended service. For conventional ships, the longitudinal stability is always high and need not be considered here. This may not be true for offshore drilling barges and other less conventional vessels.

What degree of simple static stability is needed? Unless a ship has positive stability in the upright condition, it will not remain upright because small forces-from wind, sea or movements within the ship-will disturb it. Even if it does not actually capsize, i.e. turn right over, it would be unpleasant to be in a ship which lolled to one side or the other of the upright. Thus one need for stability arises from the desire to have the ship float upright. Although, in theory, a very small metacentric height would be sufficient for this purpose, it must be adequate to cover all conditions of loading of the ship and growth in the ship during its life. If the ship is to operate in very cold climates, allowance will have to be made for the topweight due to ice forming on the hull and superstructure.

Next, it is necessary to consider circumstances during the life of the ship which will cause it to heel over. These include

(a) The action of the wind which will be most pronounced in ships with high freeboard or large superstructure;
(b) the action of waves in rolling the ship. This will be most important for those ships which have to operate in large ocean areas, particularly the North Atlantic;
(c) the action of the rudder and hull forces when the ship is manoeuvring;
(d) loading and unloading cargoes.

The ship may be subject to several of the above at the same time. In addition, the ship may, on occasion, suffer damage leading to flooding or she may suffer from shifting cargo in very rough weather. These aspects of the design are dealt with in more detail in the next chapter.

If a ship experiences a heeling moment in perfectly calm water it would be sufficient if the curve of statical stability had a lever in excess of that represented by the heeling moment. In practice, it is also necessary to allow a certain reserve of dynamical stability to enable the ship to absorb the energy imparted to it by waves or by a gusting wind.

Certain relaxations are made for ships operating in coastal waters and for ships in service, but for new design intact ships to operate in ocean areas the following criteria are specified.

(a) Beam winds and rolling

A wind heeling arm curve is calculated and is superimposed on the statical stability curve for the intact ship as in Fig. 4.31. A simple formula for calculating the heeling arm due to wind is

$$\text{Heeling arm} = 0.17 \times 10^{-6}\frac{AV^2l}{\Delta}\cos^2\phi\,\text{m}$$

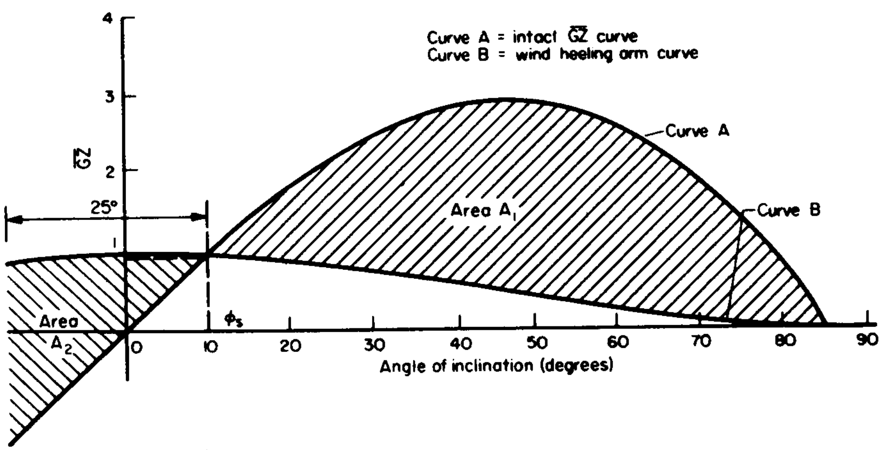


Fig. 4.31 Stability criteria for beam wind and rolling

where V = nominal wind velocity, knots (taken as 100 knots). A = projected area (m^2) of ship above waterline and l = lever arm from half draught to the centroid of the projected area. Δ = displacement in MN.
More accurately, the formula

$$\text{Heeling arm} = 0.19 \times 10^{-6}\frac{AV^2l}{\Delta}\cos^2\phi\,\text{m}$$

can be used when the wind speed V at each height above the sea surface is taken from Fig. 9.25 for a nominal wind velocity of 100 knots. In this case, it is necessary to divide A into a number of convenient horizontal strips and carry out a vertical integration.

The angle ϕ_s at which the two curves cross, is the steady heel angle the ship would take up if the wind were perfectly steady and there were no waves and

the corresponding righting arm is $\overline{GZ}_s$. To allow for the rolling of the ship the curves are extended back to a point 25 degrees to windward of ϕ_s and the areas A_1 and A_2 shown shaded in Fig. 4.31 are computed. For stability to be regarded as adequate the following conditions are typically aimed for:

$\overline{GZ}_s$ not more than $0.6 \times \overline{GZ}_{max}$
Area A_1 not less than $1.4 \times$ Area A_2.

(b) Lifting of heavy weights over the side

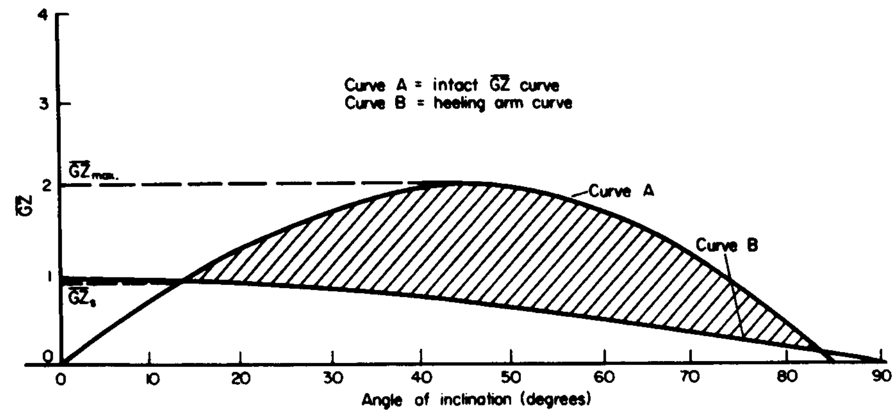


Fig. 4.32

Again, the heeling arm curve is superimposed on the statical stability curve (Fig. 4.32) the heeling arm being taken as

$$\frac{Wa}{\Delta}\cos\phi$$

where W = weight of lift, a = transverse distance from centre line to end of boom and Δ = displacement, including weight of lift.

The stability criteria applied by the USN are

$\overline{GZ}_s$ not more than $0.6 \times \overline{GZ}_{max}$
ϕ_s not greater than 15 degrees

Shaded area not less than 40 per cent of the total area under the statical stability curve.

(c) Crowding of passengers to one side

The same formula and criteria are used as for case (*b*), except that, in this case,

W = weight of passengers
a = distance from centre line of ship to c.g. of passengers.

(d) Heeling during a high speed turn

When a ship is turning steadily, it must be acted upon by a force directed towards the centre of the circle and of value

$$\frac{\Delta(\text{steady speed})^2}{g(\text{radius of turn})} = \frac{\Delta V^2}{gR}$$

This problem is discussed in more detail in Chapter 13, but, approximately, the heeling moment acting on the ship is given by

$$\frac{\Delta V^2}{gR}\left(\overline{\mathrm{KG}} - \frac{T}{2}\right)\cos\phi$$

The heeling arm curve is then

$$\frac{V^2}{gR}\left(\overline{\mathrm{KG}} - \frac{T}{2}\right)\cos\phi$$

and can be plotted as in Fig. 4.32. The stability criteria adopted are as for case (*b*) above.

So far, only minimum standards have been mentioned. Why not then make sure and design the ship to have a very large metacentric height? In general, this could only be achieved by considerably increasing the beam partially at the expense of draught. This will increase the resistance of the ship at a given speed, and may have adverse effects on stability at large angles of heel and following damage. It will also lead to more violent motion in a seaway which may be unpleasant to passengers and indeed dangerous to all personnel. In addition, the larger beam may make it impossible to negotiate certain canals or dock entrances. Hence the designer is faced with providing adequate but not excessive stability.

PASSENGER SHIP REGULATIONS

Standards for passenger ships in the UK are laid out in the Merchant Shipping (Passenger Ship Construction) Regulations. Typically they require

(*a*) The area under the $\overline{\mathrm{GZ}}$ curve shall not be less than

- (i) 0.055 m rad up to 30 degrees
- (ii) 0.09 m rad up to 40 degrees or up to the downflooding angle (the least angle where there are openings permitting flooding)
- (iii) 0.03 m rad between 30 degrees and angle (ii)

(*b*) $\overline{\mathrm{GZ}}$ must be greater than 0.20 m at 30 degrees
(*c*) Maximum $\overline{\mathrm{GZ}}$ must be at an angle greater than 30 degrees
(*d*) $\overline{\mathrm{GM}}$ must be at least 0.15 m.

THE INCLINING EXPERIMENT

It has been shown that, although much of the data used in stability calculations depends only on the geometry of the ship, the position of the centre of gravity must be known before the stability can be assessed finally for a given ship condition. Since $\overline{\mathrm{KG}}$ may be perhaps ten times as great as the metacentric height, it must be known very accurately if the metacentric height is to be assessed with reasonable accuracy.

$\overline{\mathrm{KG}}$ can be calculated for a variety of conditions provided it is accurately known for one precisely specified ship condition. It is to provide this basic knowledge that the *inclining experiment* is carried out. The term 'experiment' is a misnomer in the scientific sense but it is universally used. It would be better called a stability survey. The purposes of the inclining experiment are to determine the displacement and the position of the centre of gravity of the ship in an accurately known condition. It is usually carried out when the ship is as nearly complete as possible during initial building and, in the case of warships, may be repeated at intervals during her life as a check upon the changing armaments, etc. For instance, an additional inclining may be carried out following an extensive modernization or major refit.

The experiment is usually carried out by moving weights across the deck under controlled conditions and noting the resulting angle of heel. The angles are normally kept small so that they are proportional to the heeling moment, and the proportionality factor provides a measure of the metacentric height. The angle of heel is usually recorded using long pendulums rigged one forward and one aft in the ship, perhaps through a line of hatches. The pendulum bobs may be immersed in water or oil to damp their motion.

Conduct of an inclining experiment

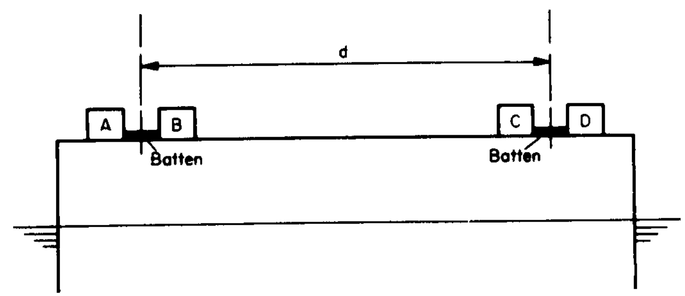


Fig. 4.33 Movement of weights during an inclining experiment

The principal steps in the carrying out of an inclining experiment are:

(*a*) The ship is surveyed to determine weights to be removed, to come on board or be moved for final completion.
(*b*) The state of all tanks is noted accurately.
(*c*) The draughts are accurately read at each set of draught marks, including amidships, on both sides of the ship.
(*d*) The density of the water in which the vessel is floating is measured at a number of positions and depths around the ship.
(*e*) Weights, arranged on the deck in four groups, shown in Fig. 4.33, are moved in the following sequence:
weights A to a position in line with weights C
weights B to a position in line with weights D
weights A and B returned to original positions

weights C to a position in line with weights A
weights D to a position in line with weights B
weights C and D returned to original positions
The weight groups are often made equal.

(*f*) The angle of heel is recorded by noting the pendulum positions before the first movement of weight and after each step given above.

Analysis of results

Due to the shift of a weight w through a distance d, G will move parallel to the movement of the weight to G' where

$$\overline{GG'} = \frac{wd}{\Delta}$$

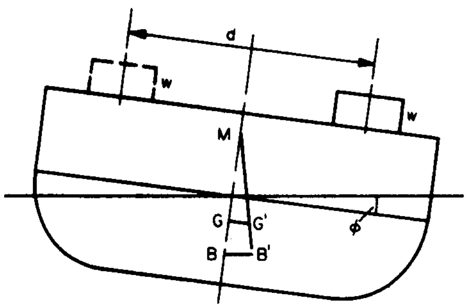


Fig. 4.34 Analysis of results

Since $\overline{GG'}$ is normal to the centre-line plane of the ship

$$\overline{GM} = \overline{GG'}/\tan\phi = \frac{wd}{\Delta\tan\phi}$$

If the movement of each pendulum, length l, is measured as y on a horizontal batten

$$\tan\phi = \frac{y}{l}$$

The displacement can be determined from the measurements of draught and water density. The latter are averaged and the former are used to determine the trim of the vessel. The amount of hog or sag present can be determined if the ship is provided with amidship draught marks. When hog or sag is detected it is usual to assume, in calculating the displacement, that the ship takes up a parabolic curve. The mean draught is calculated from the draughts at the ends of the ship and is increased or reduced by 70 per cent of the sag or hog respectively. Similarly, it may be possible to find the extent of any stern drop.

Hence, -G-M€an be calculated for each transfer of weight provided the metacentre remains fixed up to the maximum angle of heel used. Typically,

this should be of the order of 3 or 4 degrees and presents no problem for most ships. If equal weight transfers are made, this is readily confirmed if the angle changes are equal. If unequal weight transfers are used, an approximate check can be made by dividing the angles by the respective weights and comparing the results. If it is suspected that the relationship between angle and heeling moment is not linear, the results obtained must be plotted as a statical stability curve and the tangent at the origin determined.

The data can be analysed in various ways. One possible method is to produce a table of the asymmetric moment applied at each stage (m], m2, etc.) and the corresponding pendulum deflections (d], d2, etc.). The moment values are squared and the moments and deflections multiplied together as illustrated below:

Weight movement (Fig. 4.33)	Applied moment m	m^2	Measured deflection d	$m \times d$
A to C	m_1	m_1^2	d_1	$m_1 d_1$
B to D	m_2	m_2^2	d_2	$m_2 d_2$
A and B returned	0	0	d_3	0
C to A	$-m_4$	m_4^2	d_4	$-m_4 d_4$
D to B	$-m_5$	m_5^2	d_5	$-m_5 d_5$
C and D returned	0	0	d_6	0
		$\sum m^2$		$\sum md$

The average deflection per unit applied moment is given by $\sum md/\sum m^2$.

It will be noted that, in this method of analysis, the deflections measured when all weights are returned to their original positions play no part in the analysis. They are used at the time of the experiment to indicate whether any loose weights, or liquids in tanks have moved from one side to another.

The datum readings are used in a second method of analysis in which the deflection of the pendulum per unit applied moment is calculated for each shift of weights. This is illustrated below:

Weight moment (Fig. 4.33)	Applied moment, m	Measured deflection, d	Change in deflection per unit change of moment
A to C	m_1	d_1	d_1/m_1
B to D	m_2	d_2	$(d_2 - d_1)/(m_2 - m_1)$
A and B returned	0	d_3	$(d_3 - d_2)/(0 - m_2)$
C to A	$-m_4$	d_4	$(d_4 - d_3)/(-m_4)$
D to B	$-m_5$	d_5	$(d_5 - d_4)/(-m_5 + m_4)$
C and D returned	0	d_6	$(d_6 - d_5)/m_5$

The average deflection per unit change of moment is taken as the mean of the figures in the last column.

EXAMPLE 4. A ship of 6000 tonnef displacement is inclined. The ballast weights are arranged in four equal units of 10 tonnef and each is moved transversely through a distance of 10 m. The pendulum deflections recorded are 31, 63, 1, −30, −62 and 0 cm with a pendulum length of 7 m. Calculate the metacentric height at the time of the experiment.

Solution: Since the moment is applied in equal increments of 100 tonnef m, each such increment can be represented by unity in the table as below, using the first method of analysis:

Applied moment Units of 100 tonnef m m	m^2	Deflection d cm	$m \times d$
1	1	31	31
2	4	63	126
0	0	1	0
−1	1	−30	30
−2	4	−62	124
0	0	0	0
	$\sum m^2 = 10$		$\sum md = 311$

$\therefore$ Mean deflection per unit moment = 31.1 cm, i.e. the pendulum is deflected through 31.1 cm for each 100 tonnef m of applied moment

$$\text{Angle of heel} = \frac{\text{Pendulum deflection}}{\text{Pendulum length}}$$

$$= \frac{31.1}{700} = 0.0444 \text{ radians.}$$

For small angles

$$\text{Applied moment} = \Delta\overline{\mathrm{GM}}\phi$$

$$\therefore\ 100 = 6000\,\overline{\mathrm{GM}}\,0.0444$$

$$\therefore\ \overline{\mathrm{GM}} = 0.375\,\mathrm{m}$$

Applying the second method of analysis would have given:

Change of moment Units of 100 tonnef m	Change in pendulum deflection (cm)	Pendulum deflection per unit change of moment
1	31	31
1	32	32
−2	−62	31
−1	−31	31
−1	−32	32
2	62	31

$$\text{Hence mean pendulum deflection} = \frac{188}{6}$$

$$= 31.3\,\mathrm{cm}$$

This represents less than one per cent difference compared with the answer obtained by the first analysis. This is within the general limits of accuracy of the experiment.

Precautions to be taken

The importance of obtaining an accurate result has already been emphasized and certain precautions must be taken to ensure that accuracy is obtained. These are really a matter of common sense and include

(*a*) The ship must be floating upright and freely without restraint from ropes.
(*b*) There should be no wind on the beam.
(*c*) All loose weights should be secured.
(*d*) All cross-connections between tanks should be closed.
(*e*) Tanks should be empty or pressed full. If neither of these conditions is possible, the level of liquid in the tank should be such that the free surface effect is readily calculable and will remain sensibly constant throughout the experiment.
(j) The number of people on board should be kept to a minimum and they should go to specified positions for each reading of the pendulums.
(*g*) Any mobile equipment used to move the weights across the deck must return to a known position for each set of readings.

It should be noted that (*c*), (*d*), (f) and (*g*) will, to some extent, be checked by the accuracy with which the pendulums return to their zero readings following the return of weights to their original positions half way through the experiment and again at the end. If the pendulum reading fails to take up a steady value but slowly increases or diminishes, it is likely that liquid is passing between tanks through an open cross-connection.

PRECISION OF STABILITY STANDARDS AND CALCULATIONS

The stability characteristics of a given form can be derived with considerable precision and some authorities fall into the trap of quoting parameters such as initial metacentric height with great arithmetic accuracy. This shows a lack of appreciation of the true design concept of stability and the uncertainties surrounding it in the real world.

Even in the very limited case of the stability of a vessel at rest in calm water great precision is illusory since:

(*a*) Ships are not built precisely to the lines plan.
(*b*) Weights and -K-O of sister ships even in the unloaded state will differ. Apart from dimensional variations, plate thicknesses are nominal.
(*c*) The accuracy of -K-O- assessment is limited by the accuracy with which an inclining experiment can be conducted.
(*d*) Many calculations ignore the effects of trim.

However, these inaccuracies are relatively trivial compared with the circumstances appertaining when a vessel is on voyage in a rough sea.

By now the student should appreciate that stability is a balance between forces acting on the hull, trying on the one hand to overturn it and on the other to return it to the upright position. All the major elements in the equation are varymg, VIZ.:

(a) Weight of ship and its distribution depend upon initial lading and the way in which consumables are used up during a voyage. The presence or otherwise of free surfaces will be significant.
(b) Wind forces will depend upon speed and direction relative to ship. They will vary as the wind gusts and as the ship moves to present a different aspect to the wind, e.g. by rolling.
(c) Water forces will depend upon the nett wave pattern compounded by the ship's own pattern interacting with that of the surrounding ocean; also upon the ship's altitude and submergence as it responds to the forces acting on it. To take one example: the shipping of water and the ability of clearing ports to clear that water is important.

A ship's speed and direction relative to the predominant wave direction can be important. At relatively high speed in quartering seas a ship encounters waves of a wide range of lengths at almost exactly the same frequency. If this frequency happened to coincide with the ship's natural roll frequency, then very large roll amplitudes could build up quickly, possibly resulting in capsize.

Thus the concepts of static equilibrium and stability have many limitations in dealing with what is in reality a dynamic situation. To be realistic, a designer must recognise the varying environment and the variations in ship condition. Traditionally this has been dealt with by setting standards based loosely on realistic conditions and experience of previously successful design. Margins provide safety under all but extreme conditions which it would be uneconomical to design for. Whilst this approach can allow different standards for say ocean going or coastal shipping it has a number of drawbacks, viz.:

(a) A succession of successful ships may mean they have been overdesigned.
(b) The true risk factor is not quantified. Is it a 1 in 10^6 or 1 in 10^9 probability of a ship being lost?

In recent years more attention has been given to the problems of stability and capsizing as dynamic phenomena. Advances have been made in the understanding of the dynamic problem spurred on, unfortunately, by a number of marine disasters such as the loss of the trawler *Gaul.* This was a modern vessel which met IMO standards for the service in which it was used but which was nevertheless lost. In this case it was concluded that the *Gaul* was not lost because of inadequate intact stability or poor seakeeping qualities. It is surmised that it met severe wave and wind conditions at the same time as it suffered from some other unknown circumstance such as internal flooding.

The above remarks will have made the student more aware of the true nature of the problem of stability of ships at sea, and the limitations of the current static approach. However, there is, as yet, no agreed method of dealing with the true dynamic situation. The designer still uses the static stability criteria but

with an eye on the other issues. Quasi-static calculations of stability curves can be made with a ship balanced on a wave. Righting arms are reduced below the still water values when a wave crest is near amidships and increased when a trough is amidships. In the case of *Gaul* the variation in maximum $\overline{GZ}$ was about ±30 per cent. The changing roll stiffness due to the $\overline{GZ}$ variations as the ship passes through successive crests and troughs can result in a condition of parametric resonance leading to eventual capsize. The effect of the Smith correction on righting arm was found to be insignificant for angles up to 20 degrees but gave changes of 10 per cent at 50 degrees. Model experiments can be valuable for studies of stability in waves.

In another approach the hull shape was so configured as to retain a constant $\overline{GZ}$ curve in the changing circumstances between the 'as designed' and 'as built' conditions and in motion in a seaway. It discussed how to create a hull form in way of the design draught such that the metacentric height remains constant with changing displacement and trim and how to extend this to quasi-static sea conditions. Although the approach concentrated on initial stability it was shown that forms developed in accord with the principles exhibited sensibly constant $\overline{GZ}$ curves in waves, and for damaged stability and change of draught up to quite large heel angles. The shaping required did not conflict greatly with other design considerations.

Heeling trials

These trials are mentioned only because they are commonly confused with the inclining experiment. In fact, the two are quite unrelated. Heeling trials are carried out to prove that the ship, and the equipment within it, continue to function when the vessel is held at a steady angle of heel. For instance, pumps may fail to maintain suction, bearings of electric motors may become overloaded, boats may not be able to be lowered by the davits clear of the ship, etc.

For warships, the trials are usually carried out for angles of heel of 5, 10 and 15 degrees although not all equipment is required to operate up to the 15 degree angle.

~

This is another test carried out in warships which is quite unrelated to the inclining experiment but is sometimes confused with it. The tilt test is carried out to check the alignment of items of armament-gun turrets~ missile launchers, directors-with respect to each other and to a reference datum in the ship.

Problems

1. A small weight is added to a wall-sided ship on the centre line and at such a position as to leave trim unchanged. Show that the metacentric height is reduced if the weight is added at a height above keel greater than $T - \overline{GM}$, where T is the draught.

2. Prove that for a wall-sided ship, the vertical separation of B and G at angle of heel ϕ is given by

$$\cos\phi\left[\overline{\mathrm{BG}} + \tfrac{1}{2}\overline{\mathrm{BM}}\tan^2\phi\right]$$

where $\overline{\mathrm{BG}}$ and $\overline{\mathrm{BM}}$ relate to the upright condition. Show that when the ship has a free surface in a wall-sided tank the corresponding expression is

$$\overline{\mathrm{BG_v}} = \cos\phi\left[\overline{\mathrm{BG_S}} + \tfrac{1}{2}(\overline{\mathrm{BM}} - \overline{\mathrm{G_SG_F}})\tan^2\phi\right]$$

where subscripts S and F relate to the solid and fluid centres of gravity.

3. Show that the dynamical stability of a wall-sided vessel up to angle ϕ is given by the expression

$$\Delta\left[\overline{\mathrm{BG}}(\cos\phi - 1) + \tfrac{1}{2}\overline{\mathrm{BM}}\tan\phi\sin\phi\right]$$

4. Prove the wall-sided formula.

A rectangular homogeneous block 30 m long × 7 m wide × 3 m deep is half as dense as the water in which it is floating. Calculate the metacentric height and the $\overline{\mathrm{GZ}}$ at 15 degrees and 30 degrees of heel.

5. The statical stability curve for a cargo ship of 10,000 tonnef displacement is defined by

ϕ(degrees)	0	15	30	45	60	75	90
$\overline{\mathrm{GZ}}$ (m)	0	0.275	0.515	0.495	0.330	0.120	−0.100

Determine the ordinate of the dynamical stability curve at 60 degrees and the change in this figure and loss in range of stability if the c.g. of the ship is raised by 0.25 m.

6. A hollow triangular prism whose ratio of base to height is 1 : 2 floats vertex downwards. Mercury of density 13.6 is poured in to a depth of 1 m and water is then poured on top till the draught of the prism is 4 m. Calculate $\overline{\mathrm{GM}}$ solid and fluid assuming the prism weightless.

7. A ship has the following principal dimensions: length 120 m, beam 14 m, draught 4 m, displacement 30 MN. The centres of buoyancy and gravity are 2.5 m and 6 m above keel and the metacentre is 7 m above keel.

Calculate the new metacentric height and beam if draught is reduced to 3.6 m, keeping length, displacement, $\overline{\mathrm{KG}}$ and coefficients of fineness unaltered.

8. The model of a ship, 1.52 m long, is found to be unstable. Its waterplane has an area of 3548 cm^2 and a transverse MI of 228,930 cm^4. The draught is 11.68 cm and the volume of displacement is 36052 cm^3 in fresh water.

If G is originally 15.24 cm above the keel, how far above the keel must a mass of 9.1 kg be put to just restore stability? $\overline{\mathrm{KB}} = 6.35$ cm.

9. A hollow prism has a cross section in the form of an isosceles triangle. It floats apex down in fresh water with mercury, of specific gravity 13.5, dropped into it to make it float upright.

If the prism may be assumed weightless by comparison and has an included angle of 90 degrees, express the fluid metacentric height in terms of the draught T.

10. A ship is floating in fresh water at a draught of 4.27 m when a weight of 427 tonnef is placed 12.8 m above the keel and 4.27 m to port. Estimate the angle of heel.

In sea water, the hydrostatic curves show that at a draught of 4.27 m, TPC = 28.8; displacement = 36,576 tonnef; $\overline{\mathrm{KM}} = 7.92$ m; $\overline{\mathrm{KB}} = 2.44$ m. Before the weight is added $\overline{\mathrm{KG}} = 5.79$ m.

11. The raft shown has to support a weight which would produce a combined height of the centre of gravity above the keel of 3.5 m. The draught is then uniformly 1 m.

What is the minimum value of *d*—the distance apart of the centre lines of the baulks—if the $\overline{\mathrm{GM}}$ must not be less than 2 m?

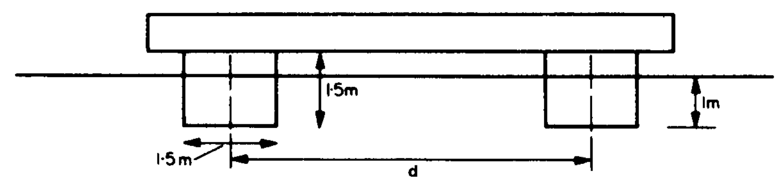


Fig. 4.35

12. A homogeneous solid is formed of a right circular cylinder and a right circular cone of the same altitude *h* on opposite sides of a circular base radius *r*. It floats with the axis vertical, the whole of the cone and half of the cylinder being immersed. Prove that the metacentric height is $3r^2/IOh$ - 2Ih/80, so that for equilibrium *r* must be greater than *0.935h*.

13. A ship of 50 MN (5100 tonnef) displacement has a metacentric height of 1 m. Suppose 1.0 MN of fuel to be shifted so that its centre of gravity moves 6 m transversely and 2 m vertically up, what would be the angle of heel~ the vessel, if it were upright before the movement? No free surface effects are produced.

14. A prismatic log of wood, of specific gravity 0.75, whose uniform transverse section is that of an isosceles triangle, floats in fresh water with the base of the section horizontal and vertex upwards. Find the least vertical angle of the section for these conditions to hold.

15. A pontoon raft 10m long is formed by two cylindrical pontoons 0.75m diameter spaced 2 m apart between centres and is planked over with wood forming a platform 10m x 3m.

When laden, the raft floats with the cylindrical pontoons half immersed in river water and its centre of gravity when laden is 1m above the waterline. Calculate the transverse and longitudinal metacentric heights.

16. Describe the procedure required to carry out an inclining experiment. The displacement of a vessel as inclined in sea water is 75 MN, 0.3 MN of ballast with a spread of 15 m causes a deviation of 0.2 m on a 4 m pendulum. The net weight to go on includes 7 MN of fuel (s.g. = 0.85) with a KG of 6 m. The transverse M.I. of the fuel surface is 1000 m^4 units. If the value of $\overline{KM}$ is 9 m and is kept at this value for the increased draught, find the final $\overline{GM}$.
17. Determine the transverse metacentric height from the following data obtained from an inclining experiment:

Length of pendulum	= 6.1 m
Mean deflection	= 21 cm
Shift of weight	= 20 tonnef through 17.34 m
Draughts at perpendiculars fwd.	= 5.42 m
aft.	= 5.82 m
Density of water	= 0.98 m^3/tonnef
Length BP	= 172 m

The hydrostatic particulars for normal trim of 0.61 m by the stern and water 0.975 m^3/tonnef are:

Draught (mean BP) (m)	Displacement (tonnef)	TPC	CF. aft of. ⊗ (m)
5.486	10,692	19.61	9.144
5.639	10,996	19.69	9.480

18. During an inclining experiment on a 60 MN guided missile destroyer weights in units of 0.05 MN are moved 10 m transversely. Deflections at the bottom of a pendulum 4 m long are defined by the following table:

Weight movement	Pendulum reading (m)
0	0
(*a*) 0.05 MN to port	0.023
(*b*) 0.05 MN to port	0.049
(*a*) & (*b*) returned	0.003
(*c*) 0.05 MN to starboard	−0.021
(*d*) 0.05 MN to starboard	−0.045
(*c*) & (*d*) returned	−0.001

Calculate the metacentric height as inclined.
19. A crane in a cargo liner, of 50 MN displacement and $\overline{GM}$ 1 m, lifts a weight of 0.2 MN from a cargo hold through a vertical height of 10 m. If the height of the crane's jib is 20 m above the hold and the radius of the jib is 10 m, calculate the reduction in $\overline{GM}$ caused by lifting the weight.
What is the approximate angle of heel caused by turning the crane through 30 degrees assuming it is initially aligned fore and aft?
20. A fire in the hangar of an aircraft carrier, floating in water of 0.975 m^3/tonnef is fought by dockside fire tenders using fresh water. When the fire is under control the hangar, which is 109.7 m long and 24.38 m broad, is found to be flooded to a depth of 0.91 m. If the hangar deck is 7.01 m above the original c.g. of the ship, what is the resultant $\overline{GM}$? The following particulars apply to the ship before the fire:

Displacement = 43,688 tonnef		$\overline{KB}$ = 5.79 m
Mean draught = 11.94 m		$\overline{BM}$ = 7.01 m
TPC = 47.2		$\overline{GM}$ = 3.51 m

21. The following particulars apply to a new cruiser design:

Deep displacement	12,500 tonnef
Length on WL	200 m
$\overline{KG}$	8 m
$\overline{GM}_T$	1.5 m
Area of waterplane	3000 m^2
Draught	6 m
Depth of hull amidships	14 m
Beam	21 m

A proposal is made to increase the armament by the addition of a weight of 200 tonnef, the c.g. 2 m above the upper deck. Find the amount by which the beam must be increased to avoid loss of transverse stability assuming other dimensions unchanged. Assume water of 0.975 m^3/tonne.
22. A pontoon has a constant cross-section in the form of a trapezium, width 6 m at the keel, 10 m at the deck and depth 5 m. At what draught will the centre of curvature of the curve of flotation be at the deck level?
23. Show that a wall-sided vessel with negative $\overline{GM}$ lolls to an angle given by

$$\tan\phi = \pm\left(\frac{2\overline{GM}}{\overline{BM}}\right)^{\frac{1}{2}}$$

A body of square cross-section of side 6 m has a metacentric height of −0.5 m when floating at a uniform draught of 3 m. Calculate the angle of loll.
24. Derive an expression for the radius of curvature of the metacentric locus. Prove that in a wall-sided vessel, the radius of curvature of the metacentric locus is given by $3\,\overline{BM}\sec^3\phi\tan\phi$ where $\overline{BM}$ relates to the upright condition.
25. The half-ordinates in metres of a waterplane for a ship of 5 MN, 56 m long are 0.05, 0.39, 0.75, 1.16, 1.63, 2.12, 2.66, 3.07, 3.38, 3.55, 3.60, 3.57, 3.46, 3.29. 3.08, 2.85, 2.57, 2.26, 1.89, 1.48, and 1.03. If $\overline{KB}$ is 1.04 m and $\overline{KG}$ is 2.2 m calculate the value of $\overline{GM}$ assuming the ship is in water of 0.975 m^3/Mg.
26. A computer output gives the following data for plotting cross curves of stability ($\overline{KS}$ = 5 m).

ϕ (deg)	Disp. (tonnef)	$\overline{SZ}$ (m)	Disp. (tonnef)	$\overline{SZ}$ (m)	Disp. (tonnef)	$\overline{SZ}$ (m)
10	1300	1.04	2350	0.69	3580	0.53
	4890	0.43	6200	0.37	7660	0.36
20	1410	1.63	2430	1.26	3660	1.02
	4950	0.87	6330	0.78	7830	0.77
30	1580	1.85	2600	1.59	3820	1.37
	5130	1.31	6550	1.26	8100	1.23
40	1810	1.95	2870	1.82	4120	1.79
	5500	1.72	6870	1.66	8350	1.53
50	2100	2.08	3270	2.10	4570	2.08
	5850	1.96	7150	1.83	8450	1.69
60	2530	2.30	3680	2.23	4950	2.10
	6200	1.96	7400	1.83	8600	1.70
70	2950	2.23	4030	2.12	5300	1.98
	6500	1.82	7620	1.71	8770	1.62
80	3250	1.94	4370	1.88	5530	1.75
	6680	1.61	7830	1.53	8970	1.47
90	3670	1.56	4630	1.50	5740	1.44
	6900	1.36	8050	1.30	9170	1.28

Plot the cross curves of stability and deduce the $\overline{GZ}$ curves for the following conditions:

(a) $\Delta = 3710$ tonnef; $\overline{GM} = 0.64$ m; $\overline{KG} = 7.28$ m
(b) $\Delta = 4275$ tonnef; $\overline{GM} = 0.93$ m; $\overline{KG} = 6.73$ m
(c) $\Delta = 4465$ tonnef; $\overline{GM} = 0.96$ m; $\overline{KG} = 6.63$ m
(d) $\Delta = 4975$ tonnef; $\overline{GM} = 1.19$ m; $\overline{KG} = 6.23$ m

Plot the GZ curves and find the angles of vanishing stability.

27. A ship of displacement 50 MN has a $\overline{KG}$ of 6.85 m. The $\overline{SZ}$ values read from a set of cross curves of stability are as below:

ϕ	10	20	30	40	50	60	70	80	90
$\overline{SZ}$ (m)	0.635	1.300	1.985	2.650	3.065	3.145	3.025	2.725	2.230

Assuming that $\overline{KS}$ for the curves is 5 m, calculate the ordinates of the $\overline{GZ}$ curve and calculate the dynamical stability up to 80 degrees.

28. A passenger liner has a length of 200 m, a beam of 25 m, a draught of 10 m and a metacentric height of 1 m. The metacentric height of a similar vessel having a length of 205 m, beam 24 m, draught of 10.5 m is also 1 m. Assuming that $\overline{KG}$ is the same for the two vessels, calculate the ratio of $\overline{KB}$ to $\overline{BM}$ in the original vessel. What is the ratio of the displacements of the two vessels?

29. A vessel 72 m long floats at 6 m draught and has 4.5 m freeboard, with sides above water vertical. Determine the $\overline{GZ}$ at 90°, assuming $\overline{KB} = 3.5$ m and $\overline{KG} = 5$ m with the vessel upright. The half-ordinates of the waterplane are 0.8, 3.3, 5.4, 6.5, 6.8, 6.3, 5.1, 2.8 and 0.6 m. The displacement is 3588 tonnef and the middle-line plane is rectangular in shape. Assume water is salt.

30. For a given height of the centre of gravity and a required standard of stability (i.e. $\overline{GM}$), what is the relationship between a ship's length, beam, draught and volume of displacement?

State your assumptions clearly.

A ship is designed to the following particulars:

Displacement 10,160 tonnef; draught 7.32 m; beam 17.07 m; $\overline{GM}$ 1.07 m; G above WL 1.22 m; TPC 19.2.

Alterations involve 101.6 tonnef extra displacement and 0.3 m rise of c.g. What increase in beam will be necessary to maintain $\overline{GM}$, assuming that the depth of the ship remains unaltered and the small alterations in dimensions will not further alter the position of the c.g. or weight?

Assume the ship to be wall-sided and that length, waterplane coefficients and block coefficients are unchanged.

31. A ship of 53,850 tonnef displacement carries 5390 tonnef of oil fuel in a rectangular tank 30 m long, 20 m wide and 20 m deep. If the oil is assumed solid the ship has a $\overline{GM}$ of 4.90 m and the following values of $\overline{GZ}$:

angle (deg)	15	30	45	60	75	90
$\overline{GZ}$ (m)	1.35	2.72	3.17	2.18	0.60	−1.13

Taking the density of the oil as 1.11 m^3/tonnef, find the values of $\overline{GZ}$ at 30, 45 and 75 degrees allowing for the mobility of the fuel, and state the initial $\overline{GM}$ (fluid) and the angle of vanishing stability.

32. Prove Atwood's formula for calculating $\overline{GZ}$ for a ship at an angle of heel ϕ.

A simple barge has a constant cross-section throughout its length, a beam B at the waterplane and straight sides which slope outwards at an angle α to the vertical (beam increasing with draught).

If the barge is heeled to an angle ϕ, prove that the heeled waterplane will cut the upright waterplane at a horizontal distance x from the centre line where:

$$x = \frac{B}{2\tan\phi\tan\alpha}\left[1 - \{1 - (\tan\phi\tan\alpha)^2\}^{\frac{1}{2}}\right]$$

If $B = 20$ m, $\alpha = 10°$ and $\phi = 30°$, calculate the value of x, and then, by calculation or drawing, obtain the $\overline{GZ}$ at 30° of heel if $\overline{B_0G} = 4.6$ m and the underwater section area = 140 m^2.

33. State the purposes of an inclining experiment. Explain the preparations at the ship before the experiment is performed.

Calculate the change in $\overline{GM}$ due to the addition of 152 tonnef, 11 m above the keel of a frigate whose particulars before the addition are given below:

Displacement = 2743 tonnef		$\overline{KG} = 6.49$ m
TPC	= 7.2	$\overline{KB} = 2.50$ m
Draught	= 4.27 m	$\overline{BM} = 5.64$ m

34. Derive an expression for the radius of curvature of the curve of flotation in terms of the waterplane characteristics and the volume of displacement (Leclert's theorem).

A homogeneous prism with cross-section as an isosceles triangle, base width B and height H, floats with its apex downwards in a condition of neutral equilibrium. Show that the radius of curvature of the curve of flotation in this condition is

$$B^2 H/(4H^2 + B^2)$$

5 Hazards and protection

The naval architect needs to be more precise than the old and loved hymn, and must ask why those on the sea are in peril and what can be done to help them. Hazard to those sailing in well designed ships arises through mishandling, misfortune or an enemy and these can cause flooding, fire, explosion, structural damage or a combination of these. It is the job of the naval architect to ensure that the effects do not result in immediate catastrophe and that they can be counteracted by equipment provided in the ship, at least up to a substantial level of damage. It is necessary to ensure that when the damage is such that the ship will be lost, workable equipment and time are adequate to save life. To this end, the naval architect must anticipate the sort of mishandling and misfortune possible and the probable intent of any enemy, so that the ship can be designed to mitigate the effects and render counter action readily available.

This chapter is concerned with this anticipation. It seeks to define the hazards in sufficiently precise terms and to describe the forms of protection and life saving possible. Some of these forms are demanded by international and national regulations; such regulations do not, however, remove the need to understand the problems.

Despite all that anticipation and regulation can achieve, losses at sea continue. In 1990, 230 ships were lost, representing about 0.3 per cent of the 76,000 ships at risk worldwide and 0.2 per cent of world tonnage. Typically about a third are lost by fire or explosion. Whilst not a large percentage, they remain an anxiety to naval architects as well as to masters, owners and safety authorities.

Flooding and collision

WATERTIGHT SUBDIVISION

Excluding loading in excess of the reserve of buoyancy, a ship can be sunk only by letting water in. Water may be let in by collision, grounding, by enemy action or by operation of a system open to the sea. A compartment which has been opened to the sea is said to have been *bilged.* However the water is let in, there is a need to isolate the flooded volume for the following reasons:

(a) to minimize the loss of transverse stability,
(b) to minimize damage to cargo,
(c) to prevent *plunge,* i.e. loss of longitudinal stability,
(d) to minimize the loss of reserve of buoyancy.

Ideally, a ship should sustain more and more flooding without loss of stability until it sinks bodily by loss of its reserve of buoyancy. This is *foundering.* The dangerous effect of asymmetric moment on the GZ curve was discussed in the last chapter; the ideal way to avoid such a possibility, is to subdivide the entire ship transversely and this is a major aim of the designer. It is not often wholly possible, but it is nowadays a major feature of most ships even at the expense of increasing the volume of water admitted. Free surface can also cause appreciable loss of transverse stability as discussed in Chapter 4. The effects of free surface can be minimized by longitudinal bulkheads which are undesirable and, better, by sills; because the loss of stability is proportional to the cube of the width of free surface, the effect of this type of subdivision is considerable (see Fig. 5.1). The sill reduces the effect of shallow flooding but cannot cure the effect of excessive transverse flooding.

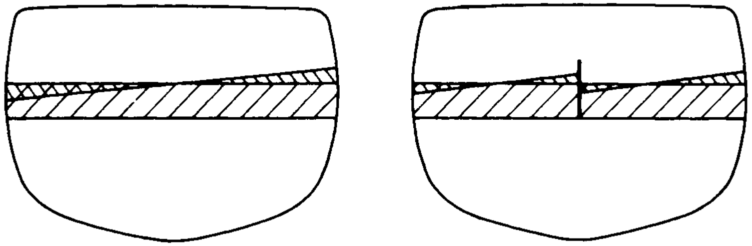


Fig. 5.1 *Effect ofa sill*

Extensive watertight subdivision is an inconvenience to everyone. The ship is larger, more expensive to build, access around the ship is inconvenient and cargo becomes more difficult to stow and more expensive to handle. A compromise between degree of safety and economics must be found. Generally, this results in watertight volumes larger in merchant ships than in warships, and a greater tendency in the former to plunge when damaged. Regulations to minimize this possibility in the behaviour of merchant ships are dealt with later in the chapter under 'Damage Safety Margins'.

Certain watertight subdivision is incorporated into a ship with the direct intention of isolating common and likely forms of damage. Collision, for example, is most likely to let water in at the bow. For this reason most passenger ships are required to have a collision bulkhead at 5 per cent of the length from the bow. A second bulkhead is required in ships longer than 100m. Also required are an afterpeak bulkhead and bulkheads dividing main and auxiliary machinery spaces from other spaces. They must be watertight up to the bulkhead deck and able to withstand the water pressures they might be subject to after damage. The collision bulkhead should not be pierced by more than one pipe below the margin line. A watertight double bottom is required and in ships over 76 m in length this must extend from the collision to afterpeak bulkhead. It provides useful tank capacity as well as protection against grounding and also some protection against non-contact underbottom explosions, although to be most effective it should be only partially filled with liquid (usually oil fuel), which causes large free surface losses. The effects of contact explosion by torpedo or mine are minimized in large warships by the provision at the sides of sandwich protection comprising watertight

compartments containing air or fluid (see Fig. 5.2). The basic principle is to absorb the energy of the explosion without allowing water to penetrate to the ship's vitals. Similar usable watertight space is efficacious in RoRo ferries against collision.

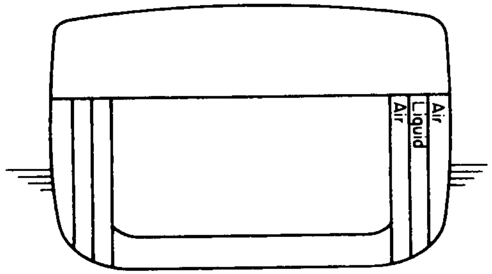


Fig. 5.2 Sandwich protection

Special subdivision is required in ships having a nuclear reactor. The reactor, primary circuit and associated equipment must be in a containment pressure vessel which should be so supported in the ship that it accepts no strain from the ship. It must be protected from damage by longitudinal bulkheads between it and the ship's side, by an extra deep double bottom and by cofferdam spaces longitudinally (Fig. 5.3).

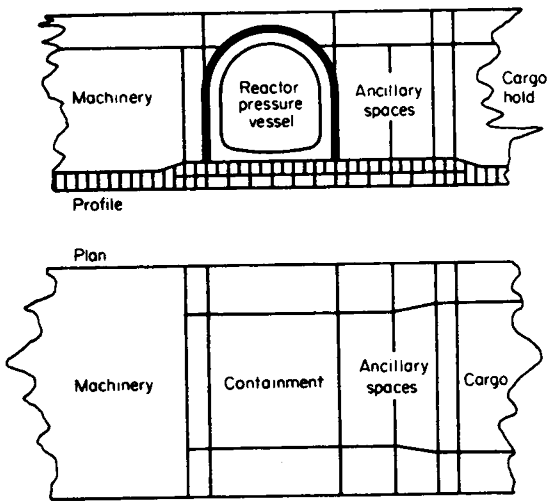


Fig. 5.3 Reactor protection

FLOTATION CALCULATIONS

In order to assess a ship's ability to withstand damage, it is first necessary in th design stages to define the degrees of flooding to be examined. A range examples will be chosen normally based, for a warship, on an expert assessme of likely weapon damage or, for a merchant ship, on statutory figures given in the United Kingdom in the Merchant Shipping (Construction) Rules. For each of these examples, it is necessary to discover

(*a*) the damaged waterline, heel and trim,
(*b*) the damaged stability for which minimum standards are laid down in the same Rules.

Consider, first, a central compartment of a rectangular vessel open to the sea (Fig. 5.4).

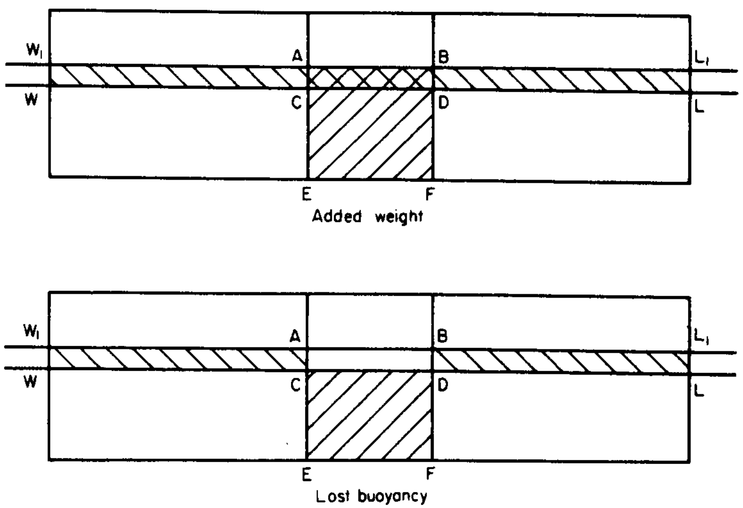


Fig. 5.4 A central compartment open to the sea

As a result of this flooding, the ship sinks from WL to WILl. The amount of weight added to the ship by the admission of water is represented by ABFE and the additional buoyancy required exactly to support it is represented by W1L1LW. To calculate the added weight to the depth AE, it is necessary to guess the new waterline and then verify that the guess is correct; trial and error will yield an answer. There is a better way to approach the problem. Instead of thinking of an increase in ship weight, the flooded portion CDFE must be thought of as a loss of buoyancy which must be made up by the buoyancies of W1ACW and BL1LD. The lost buoyancy CDFE can be calculated exactly because it is up to the original waterline only, and the additional buoyancy up to WILl can be calculated from the tonf parallel immersion of the waterplane excluding the portion AB. Weight and buoyancy of the portion ABDC cancel each other out. The ship's weight is unchanged. These two approaches are known generally as the *'added weight method'* and the *'lost buoyancy method'* respectively.

Compartments of ships open to the sea do not fill totally with water because some space is already occupied by structure, machinery or cargo. The ratio of

the volume which can be occupied by water to the total gross volume is called the *permeability*. Typical values are

Space	*Permeability* (per cent)
Watertight compartment	97 (warship) 95 (merchant ship)
Accommodation spaces	95
Machinery compartments	85
Dry cargo spaces	70
Coal bunkers, stores, cargo holds	60

Permeabilities should be calculated or assessed with some accuracy. Gross floodable volume should therefore be multiplied by the permeability to give the lost buoyancy or added weight. Formulae for the calculation of permeability for merchant ships are given in the Merchant Shipping (Construction) Rules.

Often, a watertight deck will limit flooding of a compartment below the level of the new waterline. But for a possible free surface effect due to entrapped air, the compartment may be considered pressed full and its volume calculated. In this case, the new waterline, heel and trim are best calculated by regarding the flooding as a known added weight whose effects are computed by the methods described in Chapters 3 and 4.

Where there is no such limitation and the space is free flooding, heel, trim and parallel sinkage are calculated by regarding the flooding as lost buoyancy in a manner similar to that already described for a central compartment. Now, because the ineffective area of waterplane is not conveniently central, the centre of flotation will move and the ship will not heel about the middle line. The procedure is therefore as follows:

(i) calculate permeable volume of compartment up to original waterline;
(ii) calculate TPI, longitudinal and lateral positions of CF for the waterplane with the damaged area removed;
(iii) calculate revised second moments of areas of the waterplane about the CF in the two directions and hence new $\overline{BM}$s;
(iv) calculate parallel sinkage and rise of CB due to the vertical transfer of buoyancy from the flooded compartment to the layer;
(v) calculate new $\overline{GM}$s
(vi) calculate angles of rotation due to the eccentricity of the loss of buoyancy from the new CFs.

This is best illustrated by an example.

EXAMPLE 1. A compartment having a plan area at the waterline of $100\,m^2$ and centroid 70 m before midships, 13 m to starboard is bilged. Up to the waterline obtaining before bilging, the compartment volume was $1000\,m^3$ with centres of volume 68.5 m before midships, 12 m to starboard and 5 m above keel. The permeability was 0.70.

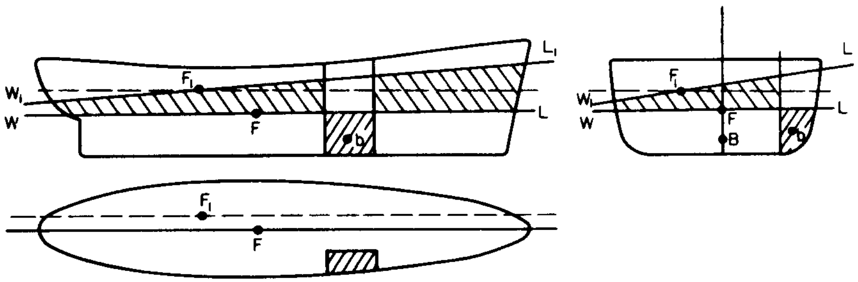


Fig. 5.5

Before the incident the ship was floating on an even draught of 10 m at which the following particulars obtained

Displacement mass, 30,000 tonnes	$\overline{KM}$ long, 170 m
$\overline{KG}$, 9.40 m	WP area, 4540 m^2
$\overline{KM}$ transverse, 11.40 m	CF, 1 m before midships
$\overline{KB}$, 5.25 m	LBP, 220 m

Calculate the heel and trim when the compartment is bilged.

Solution: Refer to Fig. 5.5. Use the lost buoyancy method.

(i) Permeable volume $= 0.70 \times 1000 = 700\,\text{m}^3$

(ii) Damaged WP area $= 4440\,\text{m}^2$

$$\text{Movement of CF aft} = \frac{100 \times (70 - 1)}{4440} = 1.55\,\text{m}$$

$$\text{Movement of CF to port} = \frac{100 \times 13}{4440} = 0.29\,\text{m}$$

(iii) Original transverse I

$$I = (11.40 - 5.25) \times 0.975 \times 30{,}000 = 179.889\,\text{m}^4$$

Damaged transverse I

$$I = 179.89 \times 10^3 - 100 \times 13^2 - 4440 \times (0.29)^2 = 162{,}620\,\text{m}^4$$

(ignoring I of the damage about its own axis)

$$\text{Damaged transverse } \overline{BM} = \frac{162{,}620}{0.975 \times 30{,}000} = 5.56\,\text{m}$$

Original long. I

$$I = (170.0 - 5.25) \times 0.975 \times 30{,}000 = 4.819 \times 10^6\,\text{m}^4$$

Damaged long. I

$$I = 4.819 \times 10^6 - 100\,(69)^2 - 4440\,(1.55)^2 = 4.332 \times 10^6\,\text{m}^4$$

Damaged long. $\overline{BM}$

$$\overline{BM} = \frac{4.332 \times 10^6}{0.975 \times 30{,}000} = 148.1\,\text{m}$$

(iv) Parallel sinkage $= \frac{700}{4440} = 0.16\,\text{m}$

$$\text{Rise of B} = \frac{700(10 + 0.08 - 5)}{0.975 \times 30{,}000} = 0.12\,\text{m}$$

(v) Damaged transverse $\overline{GM} = 5.25 + 0.12 + 5.56 - 9.40 = 1.53\,\text{m}$

Damaged long. $\overline{GM} = 5.25 + 0.12 + 148.1 - 9.40 = 144.1\,\text{m}$

(vi) $$\text{Angle of heel} = \frac{700 \times 12.29}{0.975 \times 30{,}000 \times 1.53} \times \frac{180}{\pi} = 11.0^\circ$$

$$\text{Angle of trim} = \frac{700 \times (68.5 - 1 + 1.55)}{0.975 \times 30{,}000 \times 144.1} = 0.01147\,\text{rads}$$

Change of trim $= 0.01147 \times 220 = 2.52\,\text{m}$ between perps.

Note that displacement and $\overline{KG}$ are unchanged.

The principal axes of the waterplane, as well as moving parallel to their original positions, also rotate. The effects of this rotation are not great until a substantial portion of the waterplane has been destroyed.

This approach to the flotation calculations is satisfactory for trim which does not intersect keel or deck and for heel up to about 10 degrees. Thereafter, it is necessary to adopt a trial and error approach and to calculate the vertical, athwartships and longitudinal shifts of the centre of buoyancy for several trial angles of waterplanes. That angle of waterplane which results in the new centre of buoyancy being perpendicularly under the centre of gravity is the correct one. This is illustrated for the transverse plane in Fig. 5.6, in which the trans[illegible]

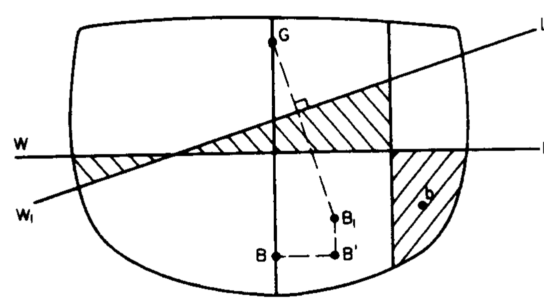


Fig. 5.6

of buoyancy from b to the two wedges (one negative) results in a move of the centre of buoyancy of the whole from B to $\mathrm{B_1}$. $\overline{\mathrm{B_1G}}$ must be perpendicular to $\mathrm{W_1L_1}$. To ensure that the displacement is unaltered, several waterlines parallel to $\mathrm{W_1L_1}$ will be necessary at each angle as described presently under 'Damaged Stability Calculations', Fig. 5.8.

DAMAGED STABILITY CALCULATIONS

Consider, first, the effect of a flooded compartment on initial stability. Let us show that, whether the flooding be considered an added weight or a lost buoyancy, the result will be the same.

Consider a ship of displacement Δ admitting a weight w having a free surface i as shown in Fig. 5.7. Regarding as an added weight,

$$\text{Rise of B} = \overline{\mathrm{BB_1}} = \frac{w\overline{\mathrm{Bb}}}{\Delta + w}$$

$$\text{Fall of G} = \overline{\mathrm{GG_1}} = \frac{w\overline{\mathrm{Gg}}}{\Delta + w}$$

$$\text{New } \overline{\mathrm{GM}} = \overline{\mathrm{B_1M_1}} - \overline{\mathrm{B_1G_1}} = \overline{\mathrm{B_1M_1}} - \overline{\mathrm{BG}} + \overline{\mathrm{BB_1}} + \overline{\mathrm{GG_1}}$$

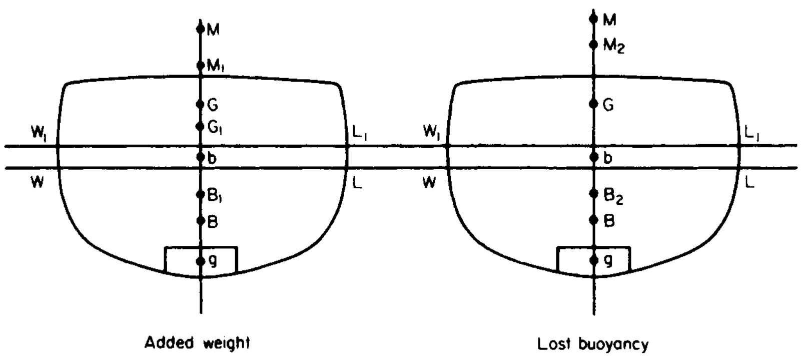


Fig. 5.7

i.e.

$$\overline{\mathrm{G_1M_1}} = \rho\frac{(\mathrm{I_1} - \mathrm{i})}{\Delta + w} - \overline{\mathrm{BG}} + \frac{w}{\Delta + w}(\overline{\mathrm{Bb}} + \overline{\mathrm{Gg}})$$

$$\text{Righting moment} = (\Delta + w)\,\overline{\mathrm{G_1M_1}}\phi = \{\rho(\mathrm{I_1} - \mathrm{i}) - \Delta\overline{\mathrm{BG}} - w\overline{\mathrm{BG}}$$

$$+ w(\overline{\mathrm{Bb}} + \overline{\mathrm{BG}} + \overline{\mathrm{Bg}})\}\phi$$

$$= \{\rho(\mathrm{I_1} - \mathrm{i}) - \Delta\overline{\mathrm{BG}} + w\overline{\mathrm{bg}}\}\phi$$

Regarding as a loss of buoyancy,

$$\text{Rise of B} = \overline{\mathrm{BB_2}} = \frac{w\overline{\mathrm{bg}}}{\Delta}$$

$$\text{New } \overline{\mathrm{GM}} = \overline{\mathrm{B_2M_2}} - \overline{\mathrm{B_2G}} = \overline{\mathrm{B_2M_2}} - \overline{\mathrm{BG}} + \overline{\mathrm{BB_2}}$$

$$\overline{\mathrm{GM_2}} = \rho\frac{(\mathrm{I_1} - \mathrm{i})}{\Delta} - \overline{\mathrm{BG}} + \frac{w\overline{\mathrm{bg}}}{\Delta}$$

$$\text{Righting moment} = \Delta\overline{\mathrm{GM_2}}\phi = \{\rho(\mathrm{I_1} - \mathrm{i}) - \Delta\overline{\mathrm{BG}} + w\overline{\mathrm{bg}}\}\phi \text{ as before.}$$

Thus, so far as righting moment is concerned (and this is what the ship actually experiences), it does not matter whether the flooding is regarded as added weight or lost buoyancy although $\overline{\mathrm{GM}}$ and $\overline{\mathrm{GZ}}$ values will not be the same. The lost buoyancy method is also called the *constant displacement method.*

It is probable that the approximate method of correcting the $\overline{\mathrm{GZ}}$ curve for contained liquid given in Chapter 4 will be insufficiently accurate for large volumes open to the sea. In this case, cross curves for the volume of flooding are constructed in the following manner:

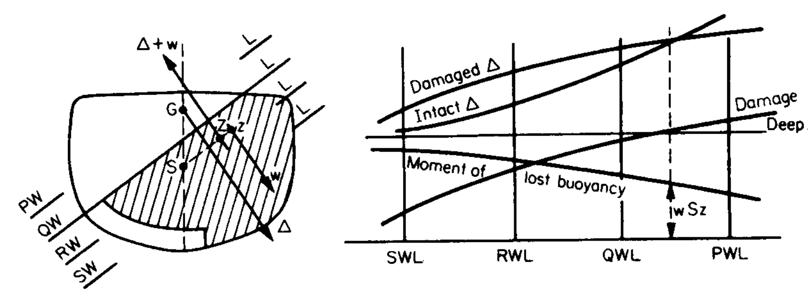


Fig. 5.8

For each angle of heel, a series of parallel waterlines P, Q, R, S, is drawn. For each of these waterlines, the weight of flooding and its lever $\overline{\mathrm{Sz}}$ from the pole of the ship's cross curves are calculated by integrator, computer or direct mensuration, i.e. by one of the methods used for the ship itself and described in Chapter 4. The weight of flooding up to each waterline is added to the deep displacement to give a damaged displacement which is plotted on a base of WL together with $w\overline{\mathrm{Sz}}$. Intact displacement to the same base will be available from the calculations which led to the cross curves for the ship (if not, they must be calculated). Where these two lines cross is the point of vertical equilibrium for that angle and $w\overline{\mathrm{Sz}}$ can be read off. Now the restoring lever $\overline{\mathrm{SZ}}$ for the intact ship at displacement $\Delta + w$ can be read directly from the cross curves. Then the damaged righting moment is given by

$$(\Delta + w)\overline{\mathrm{SZ}} - \Delta\overline{\mathrm{SG}}\sin\phi - w\overline{\mathrm{Sz}}$$

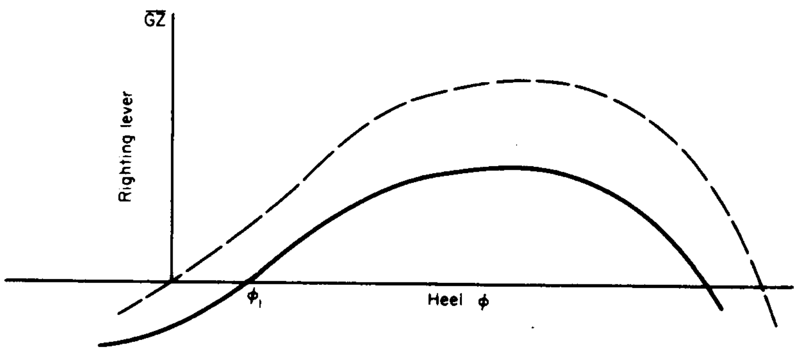


Fig. 5.9 Damaged $\overline{GZ}$ *curve*

A righting lever corresponding to the original displacement Δ is given by dividing this expression by Δ. This procedure is repeated for all angles. The damaged GZ curve will, in general, appear as in Fig. 5.9. The point at which the curve crosses the ϕ-axis, at angle ϕ_1, represents the position of rotational equilibrium; this will be zero when the flooding is symmetrical about the middle line.

With symmetrical flooding, the upright equilibrium position may not be one of stable equilibrium. In this case, the $\overline{GZ}$ curve will be as in Fig. 5.10 where ϕ_2 is the angle of loll. The significance of the angle of loll is discussed in Chapter 4. If the $\overline{GZ}$ curve does not rise above the ϕ-axis the ship will capsize; if little area is left above, capsizing will occur dynamically.

Definition of damaged stability is not easy. The Merchant Shipping Regulations require that the margin line shall not be submerged and a residual metacentric height of at least 0.05 m as calculated by the constant displacement method in the standard damaged condition be maintained. Adequacy of range, maximum $\overline{GZ}$ and angle of maximum stability are important, particularly for warships which do not conform to the margin line standards discussed presently. Trim in a warship can cause rapid loss of waterplane inertia and deterioration of damaged stability and this is one reason why the low quarter

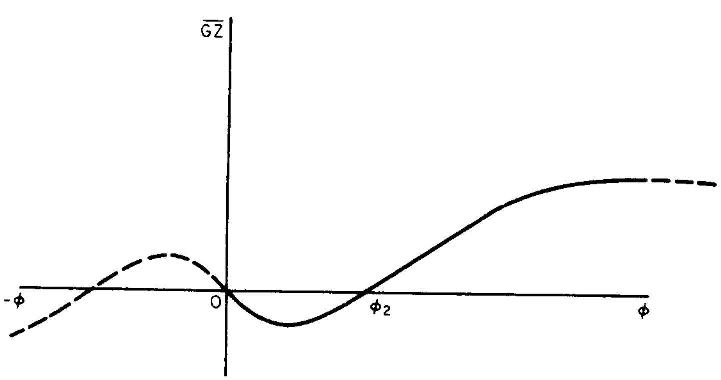


Fig. 5.10

deck destroyers common in the second world war have been replaced by modern flush deck, high freeboard frigates.

The fact that ships may take up large angles of heel following damage is recognized in the regulations governing ship construction, e.g. watertight doors must be able to be opened up to angles of heel of IS degrees.

DAMAGE SAFETY MARGINS

Minimum standards of behaviour of merchant ships under damaged conditions are agreed by international convention and enforced by national authority. For the United Kingdom this authority is the **DETR** who may delegate certain authority to specific agencies. The regulations which are the immediate concern of this chapter are those relating to damaged stability which has already been discussed, to *floodable length* and to *freeboard.* Floodable length calculations are required to ensure that there is sufficient effective longitudinal waterplane remaining in a damaged condition to prevent plunge, i.e. loss of longitudinal stability. Minimum permitted freeboard is allotted to ensure a sufficient reserve of buoyancy to accommodate damage. It is necessary, as always, to begin with clear definitions; while these embrace a majority of ships, interpretations for unusual vessels may have to be obtained from the full Rules, from the national authority or, even by litigation.

The *bulkhead deck* is the uppermost weathertight deck to which transverse watertight bulkheads are carried.

The *margin line* is a line at least 76 mm below the upper surface of bulkhead deck at side.

The *floodable length* at any point in the length of a ship is the length, with that point as centre, which can be flooded without immersing any part of the margin line when the ship has no list.

Formulae are provided in the Rules for the calculation of a *factor of subdivision* which must be applied to the floodable length calculations. This factor depends on the length of the ship and a *criterion numeral* which is intended to represent the criterion of service of the ship and is calculated from the volumes of the whole ship, the machinery spaces and the accommodation spaces and the number of passengers. Broadly, the factor of subdivision ensures that one, two or three compartments must be flooded before the margin line is immersed and ships which achieve these standards are called one-, two- or three-compartment ships. Compartment standard is an inverse of the factor of subdivision. As indicated in Fig. 5.11, very small ships would be expected to have a one-compartment and large passenger ships a three-compartment standard.

The product of the floodable length and the factor of subdivision gives the *permissible length.* With certain provisos concerning adjacent compartments, a compartment may not be longer than its permissible length. Flooding calculations either direct or aided by standard comparative parametric diagrams available in the United Kingdom from **DETR** can produce curves similar to those shown in Fig. 5.12 for floodable and permissible lengths and in general, a triangle erected from the corners of a compartment with height equal to base must have an apex below the permissible length curve. The base angle of this

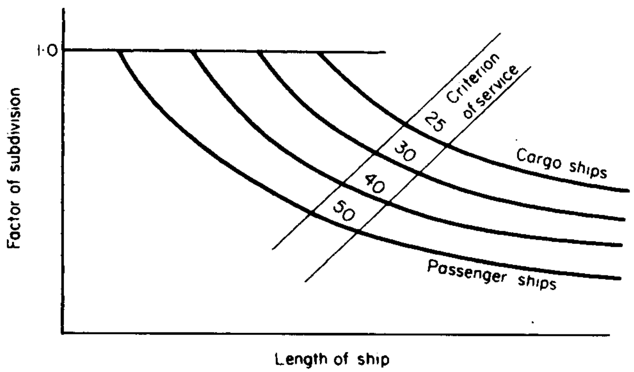


Fig. 5.11 Factor of subdivision

triangle is $\tan^{-1} 2$ if vertical and horizontal scales are the same. Lines at this angle, called the forward and after terminals, terminate the curves at the two ends of the ship. The steps in the curve are due to different permeabilities of compartments; in the initial determination of bulkhead spacing for the ship on this basis, it is necessary to complete sets of permissible length curves for a range of permeabilities.

Summarizing, the steps to be taken to carry out the floodable length calculations are as follows:

(*a*) Define bulkhead deck and margin line;
(*b*) calculate factor of subdivision;
(*c*) calculate permeabilities;
(*d*) assess floodable lengths;
(*e*) plot permissible lengths.

The second important insurance against damage in merchant ships is the allocation of a statutory freeboard. The rules governing the allocation of freeboard are laid down by an international Load Line Convention, of which one was held in 1966, and ratified nationally by each of the countries taking part. Amendments are introduced by IMO as necessary. Because the amount of

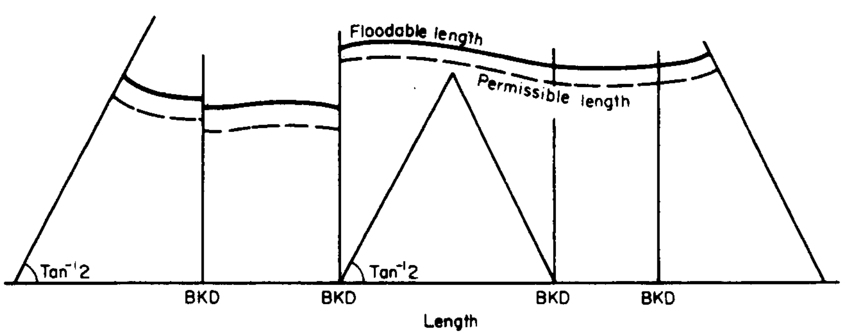


Fig. 5.12 Floodable and permissible length curves

cargo which can be carried is directly related to the minimum permissible freeboard, the authorities are anxious to be fair and exact and this tends to produce somewhat complicated rules. The intention, however, is clear enough. The intention is to provide a simple visual check that a laden ship has sufficient reserve of buoyancy and to relate, therefore, the watertight volume above the laden waterline to a height which can be readily measured. This clearly involves the geometry of the ship, and the statutory calculation of freeboard involves water density, ship's length, breadth, depth, sheer, size of watertight superstructures and other relevant geometrical features of the ship. The Rules require minimum standards of closure in watertight boundaries before allowing them to be counted in the freeboard calculation. Standards for hatch covers, crew protection, freeing port areas, ventilators and scuttles are enforced by these Rules.

The freeboard thus calculated results in a *load line* which is painted boldly on the ship's side in the form shown in Fig. 5.13. Because the density of water affects the reserve of buoyancy and weather expectation varies with season and location, different markings are required for Winter North Atlantic (WNA), Winter (W), Summer (S), Tropical (T), Fresh (F) and Tropical Fresh (TF).

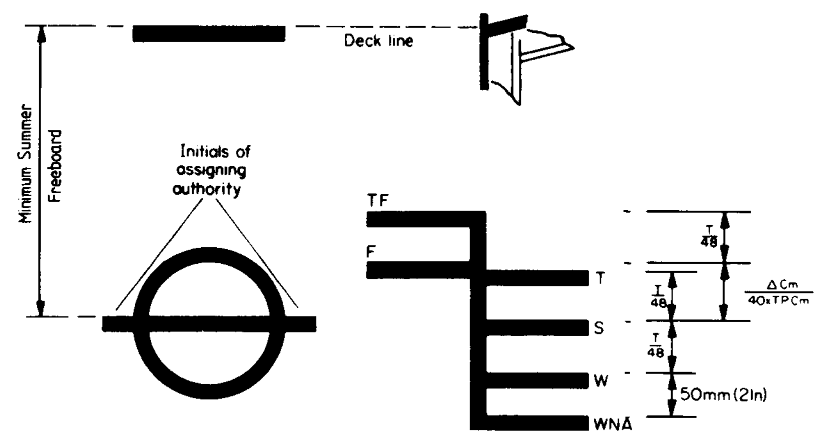


Fig. 5.13 Load line markings (min. freeboard differences shown)

DAMAGED STABILITY STANDARDS FOR PASSENGER SHIPS

Following the loss of the ferry *Herald of Free Enterprise* in 1987, IMO agreed revised standards to be incorporated into SOLAS which are now amendments to the Passenger Ship Construction Rules 1984. They require, *inter alia*, that after damage;

(*a*) the range of the residual $\overline{GZ}$ curve should be 15 degrees beyond the equilibrium point and that the area of the $\overline{GZ}$ curve beyond this point be 0.015 m rad up to the smaller of the downflooding angle or 22 degrees from the upright for one compartment flooding or 27 degrees for two compartment flooding.

(*b*) the residual $\overline{GZ}$ shall be obtained from the worst circumstance of all passengers (at 75 kg each, four per square metre) at the ship's side or all survival craft loaded and swung out or wind force at a pressure of $120\,N/m^2$ all calculated by:

$$\text{residual } \overline{GZ} = \frac{\text{heeling moment}}{\text{displacement}} + 0.04\,m \text{ but not less than } 0.10\,m$$

Intermediate stages of flooding must also be considered. The Master must determine and record the state of his ship before sailing and this is readily done with the help of a computer.

LOSS OF STABILITY ON GROUNDING

The upward force at the keel due to docking or grounding, whose magnitude was found in Chapter 3, causes a loss of stability. Let the force at the keel be w. Consider a slightly inclined vessel before and after the application of this force (Fig. 5.14).

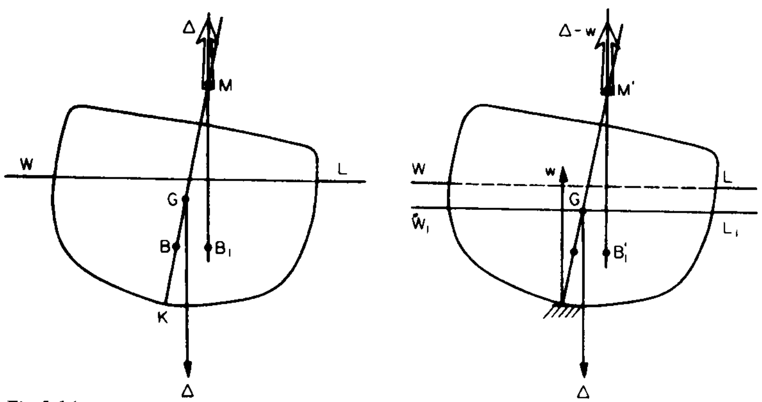


Fig 5.14

The righting moment at inclination ϕ before the application of w is, of course, $\Delta\overline{GM}\sin\phi$. After application, the righting moment is

$$(\Delta - w)\overline{GM'}\sin\phi - w\overline{KG}\sin\phi = \Delta\left(\overline{GM'} - \frac{w}{\Delta}\overline{KM'}\right)\sin\phi$$

The movement of M to M' is due, as described in Chapter 3 to (*a*), the fall of B due to the removal of the layer of buoyancy at the waterline, (*b*), the change in $\overline{BM}$ due to the differences in I and volume of displacement ∇.

BERTHING AND ICE NAVIGATION

A ship has no brakes and is slow to respond to the propulsive machinery; unexpected currents and gusts of wind can make a ship awkward to handle. Despite dexterous handling by the Master, there will be occasions when a ship comes alongside heavily and may be indented by fenders or catamarans or tugs. Local internal structural stiffening is often fitted in the vicinity of the waterline to afford some protection, and the waterline strake may be increased in thickness.

Similar stiffening is fitted in ships intended for navigation in ice and Lloyd's Register lays down five degrees of stiffening dependent on the type of ice to be negotiated, making a note against the ship to this effect in the Register Book. This is not to be confused with the stiffening required in an ice breaker which is much more formidable, since to perform its task, the ice breaker is driven to ride up on the ice, allowing its weight to break the ice.

Safety of life at sea

Representatives of over 150 seafaring nations contribute to the International Maritime Organization (IMO) which is an organization formed in 1959 under the auspices of the United Nations, intended to promote co-operation amongst all countries on the questions relating to ships. This organization promotes discussion on such topics as tonnage measurement, safety of life and standards of construction. It convened the Load Line Convention of 1966, the one previous having been held in 1930 and it has taken over the functions of the International Conferences on the Safety of Life at Sea (SOLAS) held in 1914 (following the *Titanic* disaster), in 1929, 1948, 1960, 1974 and 1990. The topics with which the SOLAS Conferences concerned themselves were standards of construction, watertight subdivision, damage behaviour, damage control, fire protection, life-saving appliances, dangerous cargoes, nuclear machinery protection, safety of navigation and many other aspects of safety. Following the 1960 SOLAS Conference, IMO set up a dozen or so sub committees dealing actively with stability and subdivision, fire protection, bulk cargoes, oil pollution, tonnage, signals and other aspects of safety under the auspices of a Maritime Safety Committee.

FIRE

In the design and construction of a ship, it is necessary to provide means to contain fire, the most feared, perhaps, of all the hazards which face the mariner. Large death tolls are to be expected in ship fires due mainly to the effects of smoke. In the passenger ship *Scandinavian Star* in 1990, 158 people died, all but six due to the effects of smoke and toxic fumes. When it is appreciated that these disasters occur in peacetime, with so many modern aids available, the seriousness of fire as a hazard can be seen. The causes of fires are manifold--electrical short circuits, chemical reactions, failure of insulation, ingress of hot particles. Whatever the cause, however, there are two factors about fire which give the clue to its containment.

(*a*) fire cannot be sustained below a certain temperature which depends upon the particular material;
(*b*) fire cannot be sustained without oxygen.

Thus, to put a fire out, it must be cooled sufficiently or it must be deprived of oxygen. A ship must be designed readily to permit both.

The principal method of cooling a fire is to apply water to it and this is most efficiently done by fine spray. Automatic sprinkler systems are often fitted, actuated by bulbs which burst at a temperature of about 80°C, permitting water to pass. Passenger vessels must embody arrangements whereby any point in the ship can be reached by two hoses delivering a specified quantity of water. Dual escape from every point is also needed.

Deprivation of oxygen is effected in a variety of ways. The watertight subdivision of the ship is conveniently adapted to provide vertical zone fire boundaries built to particular standards of fire resistance that can be closed off in the event of a fire. The fire itself can be smothered by gas, fog or foam provided by portable or fixed apparatus, especially in machinery spaces.

Regulations for merchant ships registered in the United Kingdom are contained in the Merchant Shipping (Fire Appliances) Rules and allied publications issued by Government bodies. These specify the constructional materials, method of closing the fire boundaries, the capacities of pumps and the provision of fire-fighting appliances. While the quantities depend upon the particular classes of ship, the principles of fire protection in merchant ships and warships are similar:

(a) the ship is subdivided so that areas may be closed off by fire boundaries; this also prevents the spread of smoke, which can asphyxiate and also hampers firefighters in finding the seat of the fire;
(b) a firemain of sufficient extent and capacity is provided, with sufficient cross connections to ensure supply under severe damage;
(c) fixed foam, or inert gas smothering units are provided in machinery spaces and in most cargo holds;
(d) sufficient small portable extinguishers are provided to put out a small fire before it spreads;
(e) fire control plans are supplied to the Master and a fire fighting headquarters provided.

Some cargoes contain or generate their own oxygen and are not therefore susceptible to smothering. Cooling by spraying or flooding is the only means of extinguishing such fires and any extensive flooding thus caused brings with it the added hazard of loss of stability.

LIFE-SAVING EQUIPMENT

Ships must conform to the safety regulations of the country with whom they are registered. Standards depend upon the class of ship and the service. Generally sufficient lifeboats and rafts are to be provided for all persons carried. They must be launch able with the ship listing up to 15 degrees and trimming up to 10 degrees. There have been significant advances in life saving appliances in recent years following SOLAS 74. Modern systems now include escape chutes similar to those in aircraft while rescue boats at rigs can withstand severe oil fires. Information provided to passengers, particularly in ferries, has been improved and drills for crews are now regularly enforced in all well run ships.

ANCHORING

Lives of the crew and the value of ship and cargo are often entrusted totally to the anchor-an item of safety equipment much taken for granted. Bower anchors carried by ships need to hold the ship against the pull of wind and tide. The forces due to wind and tide enable the naval architect to select the size of cable and an anchor of sufficient holding pull. Classification societies define an equipment number to represent the windage and tide forces and relate this to the cable and to approved anchor designs.

A good anchor should bite quickly and hold well in all types of sea bed. It must be stable (i.e. not rotate out), sufficiently strong and easy to weigh.

Efficiency of the anchor itself is measured by the ratio of holding pull to anchor weight which may nowadays exceed 10 in most sea beds. However, the efficiency of the mooring depends also upon the length of cable veered and the depth of water, the ratio of these being the scope. The scope must be sufficient to ensure a horizontal pull on the anchor shank. Figure 5.15.

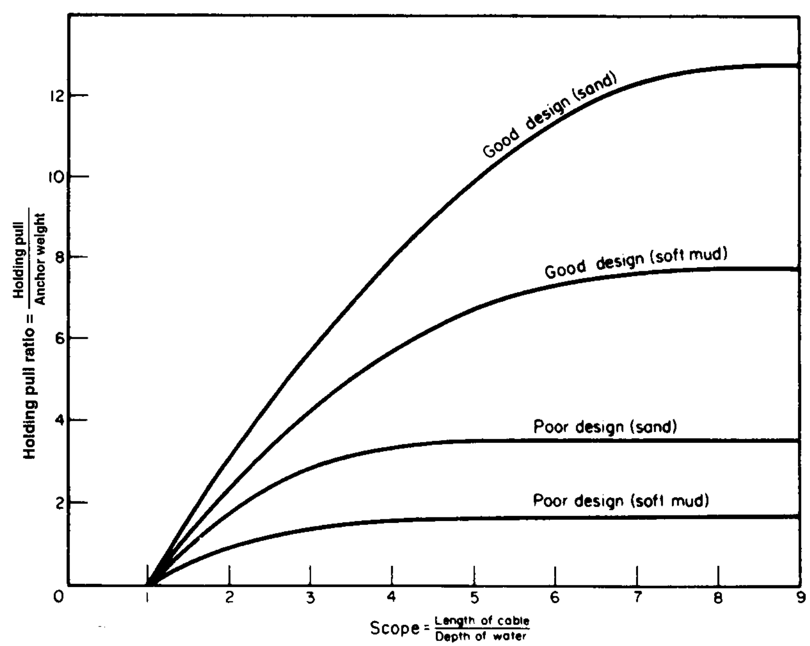


Fig. 5.15 Characteristics of anchoring

Ideally, the holding pull of the anchor should be slightly less than its own proof load and slightly more than the total wind and tide drag on the ship, while the proof load of the cable should be slightly greater than both.

The above applies to anchors which provide horizontal restraint and are used in ships and moorings. A different type is the uplift resisting anchor used for permanent moorings. Modern types depend upon being buried deep in the sea bed. This is often achieved using mechanical aids-water jetting, vibration or explosives.

DAMAGE CONTROL

The importance of a damage control organization is recognized in both merchant shipping and warship circles. Because of its greater complexity and its need to defend itself from nuclear, chemical and bacteriological attack as well as explosion and flooding, the organization in a warship is more complex. However, the Master of a merchant ship is required to be familiar with the damage control characteristics of that ship, to institute and drill an effective organization and to ensure that damage control plans are kept displayed.

The aim is to organize the ship in order rapidly to ascertain the damage, to isolate the incident and to take corrective action. The ship is constructed to make this as easy as possible; watertight subdivision is arranged as already discussed; watertight doors and hatches must be operable at 15 degrees of heel and are marked in levels of importance; minimum standards of damaged stability are met; pumping, flooding and fire-fighting systems are arranged to facilitate action under conditions of damage; electrical supplies are arranged with a view to providing sufficient breakdown capacity and duplication. All important systems are cross connected to ensure continuity of supplies following damage.

In a warship, information and control is concentrated in a central compartment with several subsidiary control posts each controlling one of the sections of the ship into which it is divided for purposes of damage control. Displayed in the headquarters is the flooding board which shows the watertight subdivision of the ship and the approximate heel and trim which would result from the flooding of any watertight compartment. Also displayed is information on the pumping, electrical, ventilation and other systems. A comprehensive communication network is available in the headquarters. With this organization, the damage control officer is able to appreciate the position and order corrective action.

Because a ship is most easily sunk by loss of transverse stability, the most urgent corrective action, once flooding has been contained by the closure of boundaries, is to reduce the angle of heel. This is most effectively done by counterflooding, i.e. flooding a compartment on the side opposite the damage and the extent necessary is indicated by the flooding board. In merchant ships, various regulations govern the limiting angle of heel. Merchant Shipping Regulations stipulate that after corrective action, heel should not exceed 7 degrees or the margin line immersed. Asymmetric flooding should be avoided if possible and cross-flooding systems capable of bringing heel under control in 15 minutes. Following corrective measures, the damage control officer may order reinforcement of bulkheads by shoring or the repair of leaks by wedges, boxes and concrete and the slower time repair of damage and restoration of supplies as far as is possible.

If A is the area of a hole and d is its distance below the waterline, the velocity of inflow of water, v, is given approximately by *v = 4.5y1dmjs*, d being in metres. The volume of water admitted by the hole is KvA, where K is a coefficient of contraction at the hole. Assuming a value of 0.625 for K,

$$\text{inflow} = \frac{11}{4} A\sqrt{d}\,\mathrm{m}^3/\mathrm{s}, \text{ with } A \text{ in } \mathrm{m}^2$$

Thus, a 15 cm diameter hole, 16 m below the waterline would admit

$$\frac{11}{4} \times \frac{\pi}{4} \times 0.0225 \times 4 \times 1.025 \times 60 = 12 \text{ tonnes of salt water per minute.}$$

UNCOMFORTABLE CARGOES

Grain, oil, explosive, hygroscopic materials, liquid gas and radioactive materials are some of the cargoes which may provide an intrinsic hazard to a ship and for which a large network of rules exists. These cannot be examined in a book of this size beyond a brief mention of their existence.

In the nineteenth century, when loading was not supervised by port surveyors, grain ships were lost because voids were left under the decks. Grain and similar cargoes may shift when the ship rolls at sea, providing an upsetting moment which, if greater than the ship's maximum righting moment, causes it to capsize; if it is less, the ship heels, which may cause a further shift of the grain, causing a larger angle of heel, which may immerse openings or capsize the ship. Protection for deep sea ships is now provided as follows:

(*a*) In special bulk carriers, either upper wing tanks are fitted thus making the holds 'self-trimming' and reducing the free surface in them when full, or at least two longitudinal curtain bulkheads are used to break up the free surface.

(*b*) In general cargo carriers (mostly two deckers), shifting boards are fitted in the upper parts of the ship, usually on the centre line in conjunction with large feeder boxes. These feeders supply grain for settlement to those voids which fill in bad weather while limiting the slack surface produced higher up. When the ship's stability is adequate, feeders may be replaced by deep saucers of bagged cargo plugging the hatches.

(In both types of ship, when it is necessary for compartments to be partly filled, the shift of grain can be restricted by overstowing with bagged grain, or by strapping down with dunnage platforms and wires, or by carrying the shifting boards down through the grain.)

(*c*) In both classes of ship, it is now customary to carry grain stability information so that for bulk carriers the angle of heel after an assumed shift of grain can be found before loading and kept under 12 degrees, and in a general cargo ship, the stability required for the method of loading can be predetermined. In both cases, the loading is then planned to meet the stability requirements. To provide what is probably the most effective protection, the principal grain exporting countries appoint surveyors to check fittings and stability before loading. Later, they supervise the stowing of the grain so that voids are reduced as far as is practicable.

Oils and petrols are hazardous if they are volatile, i.e. if, at normal temperatures they emit a vapour that can accumulate until an explosive mixture with air is created. The vapour may be ignited at any mixture if it is raised to a critical and well defined temperature known as the flashpoint. For heavy and crude oils, this temperature is quite high, often above the temperature of a cigarette. Cleaning of empty crude oil tanks by steam hoses has led to several disastrous explosions due to the discharge of electrostatic build-up in the swirling fluid. International codes cover the use of inert gas systems and precautions against electrostatic generation in tankers. During crude oil washing oxygen content must not exceed 8 per cent by volume.

Regulations, in general, provide for:

(a) total isolation of a dangerous oil cargo where possible;
(b) the removal of any possibility of creating ignition or overheating;
(c) adequate ventilation to prevent a build up of vapour to an explosive mixture;
(d) a comprehensive fire warning and fighting system.

Cofferdams to create an air boundary to oil spaces are a common requirement.

Explosives may be carried by ships which comply with the Merchant Shipping (Dangerous Goods) Rules. Because they must carry explosives where there is, in wartime, considerable risk of ignition by an enemy, the carriage of explosives in warships requires especial study. General regulations are contained in Naval Magazines and Explosive Regulations but these must be reviewed in the light of every new weapon development. The carriage of cordite in silk bags which were ignited by flash in the *Queen Mary* at Jutland and, probably, *Hood* in the second world war has been long discontinued; fixed robust ammunition is now carried alongside delicate guided weapons and torpedoes, some with exotic fuels. No ship construction can prevent the ingress of all offensive weapons, although the probability of causing catastrophic damage can be reduced. Structure can be arranged to minimize the effects of any incident and to this end the magazines are:

(a) isolated and provided with flashtight boundaries;
(b) kept cool by adequate insulation, ventilation and the prohibition of any electrical and mechanical devices which can be a source of unacceptable heat;
(c) provided with vent plates which permit an excessive pressure caused by burning to be vented to atmosphere;
(d) provided with fire detection and fighting facilities, often automatic.

Each dangerous or uncomfortable cargo must be examined on its own merits. Hygroscopic materials such as sisal may absorb much moisture and become so heavy that safe limits of deadweight or stability are broken; they may also swell and cause structural damage. Liquid gas is carried at very low temperatures and, if it leaks, may render the steel with which it comes in contact so brittle that it fractures instantly. Radioactive materials must, of course, have their radiations suitably insulated. Hydrogen peroxide combusts in contact with grease. Even some manure gives off dangerous organic gases.

NUCLEAR MACHINERY

A nuclear reactor is simply a type of boiler. Instead of producing heat by burning oil, a reactor produces heat by controlled decay of radioactive material. The heat is removed by circulation in a primary circuit, whence it is transferred at a heat exchanger to a secondary circuit to turn water into steam for driving the turbines. Whatever else can go wrong with the reactor there is no possibility whatever of a nuclear explosion, i.e. an explosion involving nuclear fission. There are, nevertheless, some unpleasant hazards associated with a reactor, viz.:

(a) under normal operation, dangerous gamma-rays are emitted from materials, activated by neutron bombardment. All personnel serving in the ship, who would be damaged by such rays must be protected by shielding of lead or other dense material. In the first nuclear merchant ship, the *Savannah,* there were nearly 2000 tonnef of shielding and there is similar shielding in nuclear submarines. Such heavy concentrated weight gives strength problems. Embrittlement of some materials by this irradiation is a further hazard;
(b) by mishandling or by accident, precise control of decay may be lost and radioactive fission products may be released. To contain such products, the reactor must be surrounded by a containment vessel which is made with scrupulous care to withstand the pressure build up due to the most serious credible accident. Safety devices are also built into the automatic control equipment which regulates the behaviour of the reactor. The subdivision of the ship referred to earlier is aimed at reducing the chances of loss of control by collision or grounding damage.

The SOLAS Conferences have made a series of recommendations for the safety of ships with nuclear reactors and these have now been incorporated into national law and classification society rules. In addition to the cofferdam protection discussed earlier, two-compartment standards are required. The Merchant Shipping Acts and the classification society rules give further details.

Other hazards

VULNERABILITY OF WARSHIPS

It is not enough to design a ship for normal operations. All ships can have accidents and warships (and sometimes merchant ships) must withstand action damage. The ability of a ship to survive in battle depends on its *susceptibilit~* being hit and its *vulnerability* to the effects of a delivered weapon. Susceptibility can be reduced by reducing the various ship signatures to make it harder to detect, using jammers to defeat an enemy's detection systems, using decoys to seduce weapons that get through these defences and by hard kill weapons to destroy the incoming projectile or weapon carrier. The details of these defence systems must be suited to the weapons' characteristics. Various conventional weapons are illustrated in Fig. 5.16. They can cause structural failure, flooding, fire, blast, shock and fragment damage. To combat these requires efficient structural design, separation, zoning, redundancy, protection and containment.

An underwater explosion is likely to provide the most serious damage. Its effects are shown in Fig. 5.17. The pulsating bubble of gaseous explosion

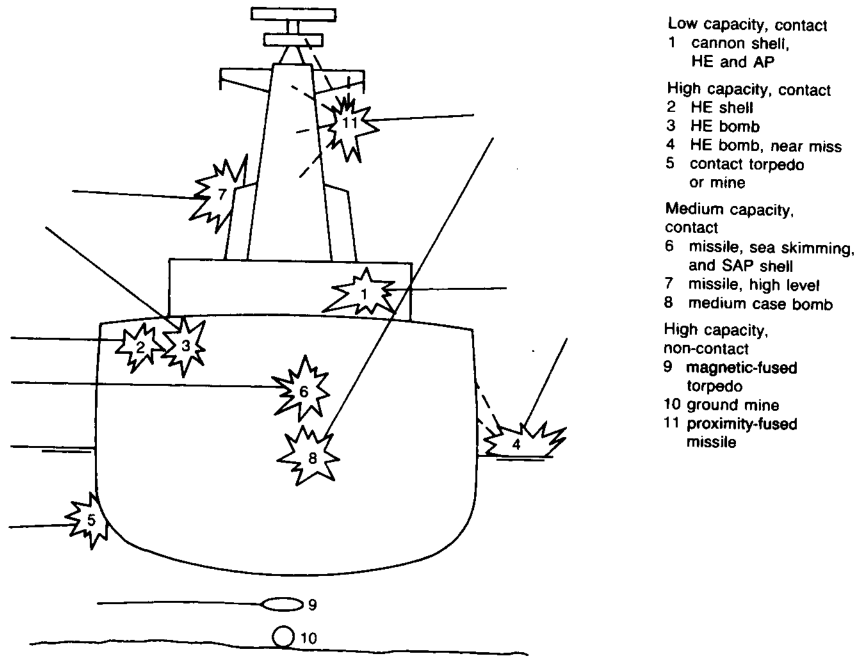


Fig. 5.16 Conventional weapon attack

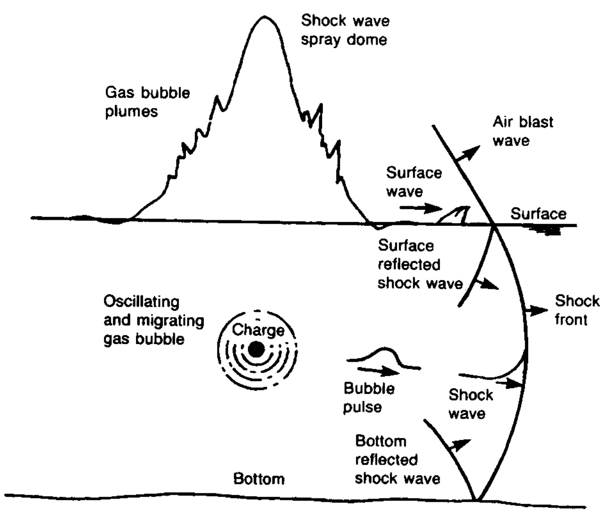


Fig. 5.17 Effects of an underwater explosion

products contains about half the energy of the explosion and causes pressure waves which impact upon the hull. The frequency of these waves is close to the fundamental hull modes of vibration of small ships and the effects are most

severe when the explosion occurs beneath the hull and the ship whips. These motions can lead to buckling and loss of girder strength.

The other major feature of an underwater explosion is the shock wave which typically contains about a third of the energy of the explosion. It is transmitted through the water, into and through the ship's structure causing shock and possibly hull rupture. The intensity of shock experienced by the ship will depend upon the size, distance and orientation of the explosion relative to the ship. These factors are combined in various ways to produce a shock factor. Since several formulations exist care is needed in their use. The intensity of shock experienced by an item of equipment will depend additionally upon its weight, rigidity, position in the ship and method of mounting. For critical systems (e.g. a nuclear power installation) it may be necessary to deduce the shock likely at a specific position in a given design. This can be done by calculation and/or model experiment using methods validated by full scale trials. More usually equipments will be fitted to several designs and in different positions, so they must be able to cope with a range of conditions. The approach is to design to a generalized shock grade curve. The overall design can be made more robust by providing shock isolation mounts for sensitive items and by siting system elements in positions where the structure offers more shock attenuation. This has the advantages that the item itself does not have to be so strong and the mounts can attenuate any noise the equipment produces, thus reducing its contribution to the underwater noise signature. The dynamic response of an item to shock will depend upon its flexibility and this must be allowed for in calculating its ability to survive and function.

As a guide, designers should avoid cantilevered components, avoid brittle materials, mount flexibly and ensure that resultant movements are not impeded by pipe or cable connections and do not cause impact with hard structure. It is important to allow for the behaviour of materials used when subject to high rates of strain. Some, notably mild steel, exhibit an increase in yield point by a factor up to two under these conditions. In warships essential equipment is designed to remain operable up to a level of shock at which the ship is likely to be lost by hull rupture. The overall design is tested by exploding large charges (up to 500 kg) fairly close to the hull of the first ship of a class.

In order to assess a design's vulnerability, and to highlight any weak elements for rectification, each new design of warship is the subject of a vulnerability assessment. This assesses the probability of various types of attack on the ship, allowing for its susceptibility. The ability of a ship to withstand each attack and retain various degrees of fighting capability, and finally survive, is computed. Essentially, the contribution of each element of the ship and its systems to each fighting capability (e.g. to detect and destroy an enemy submarine), is established. For each form of attack the probability of the individual elements being rendered non-operative is assessed using a blend of calculation, model and full scale data. If one element is particularly sensitive, or especially important, it can be duplicated (or perhaps given special protection) to reduce the overall vulnerability. The modelling for these calculations is very similar to that adopted for reliability assessments. Having made the assessments for each form of attack these can be combined, allowing for the probability of each form, to give an overall vulnerability for the design. The computations can become quite lengthy. There are also a number of difficulties which mean that any results must be carefully interpreted. These include the fact that reduced general services (electricity or chilled water, say) may be adequate to support some but not all fighting capabilities. What then happens in battle will depend upon which capabilities the command needs to deploy. This can be overcome by setting the vulnerability results in the context of various engagement scenarios. Also, in many cases, the full consequences of an attack will depend upon the actions taken by the crew in damage limitation. For instance, how effectively will they deal with fire, how rapidly will they close doors and valves to limit flooding? Recourse must be made to exercise data and statistical allowances made.

Offensive agencies may not cause damage to material but may be none the less lethal. Gas, bacteria and radioactive particles must be excluded by the crash closing of an airtight boundary. Deposits are washed away by a pre-wetting system which drenches the whole ship in salt water spray. A large air burst explosion such as that from a nuclear weapon causes a blast effect followed by a slight suction, each of which may cause structural damage against which the ship may be designed.

The effects of a ship's weapons on the ship itself must be catered for; blast, shock, heating, scouring, chemical deposit, recoil loading, noise and physical obstruction. High intensity transmissions by modern radio and radar can be hazardous to personnel and can trigger weapons which have not been properly shielded. This type of radiation hazard is called radhaz.

A good example of protection by dispersion is afforded by the unitization of machinery in large warships. Each unit, which may comprise more than one watertight compartment, is self-contained, e.g. in a steam turbine ship it would include boiler, steam turbine, gearing and auxiliaries. The total electrical generating capacity of the ship would also be divided amongst the various units. A ship designed for about 30 knots on two units could attain more than 25 knots on one unit. Thus, keeping one unit available after damage, affords considerable mobility to the ship. A ship unable to move would be a sitting duck for further attack.

SHIP SIGNATURES

As discussed in the previous section, the ability of a ship to survive depends, in part, on its susceptibility to detection and attack. Many ways of reducing the susceptibility lie in the province of the weapons engineers although the naval architect will be directly concerned with integrating them into the design. However, the designer will have a direct responsibility for, and influence on signatures. If these can be kept low, the enemy will have difficulty in detecting and classifying the target and decoys are more likely to be effective. Each signature brings its own problems for the designer and they must all be

considered. To reduce all but one would leave the ship susceptible in that respect. The signatures of most concern are:

(a) the ship has a natural magnetism which can be used to trip the fuse of a magnetic mine; such a signature can be counteracted by passing a current through electric cables wound round the inside of the ship. This is known as degaussing. Such magnetism can also be caused by induced eddy currents in a ship, particularly an aluminium alloy ship, rolling and thereby cutting the earth's magnetic lines of force;

(b) a point at the bottom of the sea experiences a pressure variation as a ship passes over it which can be used to trip a fuse in a pressure mine;

(c) the heat given off by a ship produces an infra-red radiation which can be used to home a guided weapon;

(d) radio and radar emissions can be used to detect the position of a ship and to home a guided weapon;

(e) underwater noise gives away a ship's position and can be used for homing;

(f) detritis and hydrodynamic disturbance allows the ship's track to be detected many hours after its passage.

GENERAL VULNERABILITY OF SHIPS

Ships are vulnerable to accident at any time but especially in storm conditions. The statistics of ship casualties over many years enables us to depict the frequency of events which has occurred in the past and to use that pattern to predict the likelihood of a particular event in the future. For example, it would not be surprising to discover that all of the lengths of damage due to collision and grounding which had occurred in the past, showed a frequency density as in Fig. 5.18. The area of any strip $8x$ represents the fraction of the total number of ships to have suffered a damage length lying between x_1 and $x_1 + 8x$ of ship length.

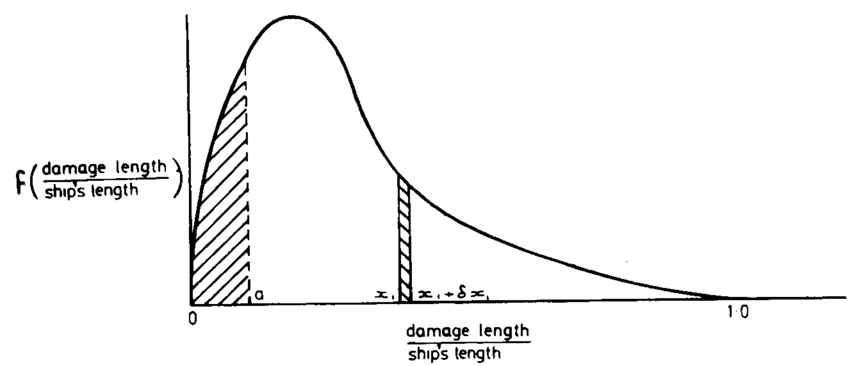


Fig. 5.18

This information could be used to predict that the probability of a damage length equal to a fraction *a* of ship length or less occurring to any ship in the future is given by the area shaded up to *a*. This is the basis for approach to ship vulnerability promulgated for merchant ships by IMO in 1973. The probability of two or more independent events is found by multiplying together their separate probabilities. Thus, the probability that a ship in collision will suffer a damage length l and a penetration from the side d and at the same time be in a condition with a permeability μ is given by

$$p(l) \times p(d) \times p(\mu)$$

Moreover, this must be compounded with the probability of collision at a certain position along the length of the ship. Probability of collision or grounding occurring at all is clearly dependent upon the routeing of the ship which is affected by the weather, sea state and traffic density met; it will be high in the English Channel or St Lawrence and low in the Indian Ocean. It must also depend upon the number and competence of the crew.

The effects upon the ship of a stated set of damage can be found in terms of trim, flotation and stability for which permissible values have already been declared. Thus, the vulnerability of the ship—that is to say the likelihood of unacceptable damage occurring—can be determined from the architecture of the ship and the probability density functions for: position of damage; length of damage; penetration of damage; permeability at the time of damage; occurrence of an incident; crew competence.

'Standard' shapes for the first four of these have been proposed by IMO who have also described how they should be applied. There are so many variables that computer programs have been written for the calculation. Similar standard functions have been in use by various navy departments for some while to determine the vulnerabilities of warships which are subjected to weapons with different homing devices and explosive content.

A steady change to probabilistic standards, begun in Code A265, is IMO's aim. Both passenger ship vulnerability and, from 1992, cargo ship safety are examined in this way. These standards should not be applied blindly or to the exclusion of an examination of compartment standard or, indeed, from a complete safety survey to be considered in Chapter 15.

The probabilistic standards are formulated upon a required subdivision index, R, such that:

$R = (0.002 + 0.009L_s)^{1/3}$ to be compared with an achieved subdivision index, A,

$A = \sum p_i S_i$ where $p =$ the probability of flooding a compartment i and $S =$ the probability of survival after flooding compartment i,

$S = C\sqrt{0.5 GZ_{max}(\text{range})}$ and $C = \sqrt{(30 - \theta_e)/5}$ or $= 1$ if $\theta_e \leq 25°$ or $= 0$ if $\theta_e \geq 30°$

$\theta_e =$ equilibrium angle in degrees.

There are many caveats to these basic requirements known commonly (but incorrectly) as SOLAS 90. Note that there is no direct relationship between the value of A obtained from these crude formulae and the actual survivability standards of a ship which owners would wish to know as part of their safety survey.

It is not easy, once it is acknowledged that nothing is absolutely safe or absolutely reliable, to declare a probability of catastrophe which, however tiny, is nevertheless acceptable. Actuaries have had to make such judgments for many years of course in settling the levels of premium for insurance involving human life. Such a cold declaration in monetary terms does not commend itself to the public, whose outrage at a particular catastrophe may be whipped up by the press. Nevertheless, a standard of acceptable probability must be declared and this will fall to national authorities and insurers, generally on the basis of what has been found acceptable in the past.

Abnormal Waves

Abnormal waves can be created by a combination of winds, currents and seabed topography. A ship may be heading into waves 8m high when suddenly the bow falls into a long, sloping trough so that in effect it is steaming downhill. At the bottom it may meet a steep wall of water, perhaps 18m high and about to break, bearing down on it at 30 knots.

Vessels can suffer severe damage. In 1973 the 12,000 grt *Bencruachan* had the whole bow section forward of the break of the 36.6m long forecastle bent downwards at 7 degrees by the impact of a 15m wave. A 15,000 dwt freighter was broken in two by a freak wave on its maiden voyage. The VLCC *Athene* met a 30m wave which knocked out all the windows of the crows nest, nearly 18m above the laden sea level. Even very large ships can be overwhelmed by exceptional sea conditions.

Mariners have always been aware that 'freak' waves could be experienced but for many years it was thought that they could not be predicted or quantified, and so could not be taken into account during the design process. By basing a new ship on a previously successful one it was hoped it could survive in other than very abnormal conditions. It did not follow, however, that the basis ship had met the most extreme waves that might be encountered in service.

As with so many other aspects of naval architecture this is a matter where statistics can be employed although our knowledge is still imperfect. The so-called freak waves are not unexplained quirks of nature and oceanographers can calculate their probability of occurrence, assisted by extensive wave d~ from buoys, ships and satellites. Buckley and Faulkner have done a lot of wk in this area. They support retaining the present design methods for dealing with the normal operations of ships. They suggest these methods should be supplemented with procedures intended to give a ship at least some low safety capability to survive critical operational conditions under abnormal waves, unlikely as the ship may be to encounter these waves. To do this requires establishing the critical design conditions, associated seaway criteria and analytical methods and criteria.

Critical design conditions would include failure of main propulsion or the steering system, loss of rudder and so on. That is the designer should allow for the ship suffering a range of defects which degrade its capabilities. They should also, as far as may be possible, make allowance for human error. That is they should try to foresee the consequences of human actions which might not be the best in the circumstances. Whilst human error cannot be eliminated it may be possible to reduce the consequences by making the design more tolerant of such errors.

As to waves, opinions are hardening up as more data becomes available. Students should refer to the latest ideas, for instance, the work of Hogben and oceanographers. Longuet-Higgins has suggested that the extreme wave height during the chosen survival design storm is given by

$$H_s[0.5(\ln N - \ln(-\ln(1 - p_e)))]^{0.5}$$

where p_e is a small but acceptable probability of H_s being exceeded, perhaps less than 5% during N wave encounters. N is given by dividing the length of time considered by the wave period T_p. Separately it has been proposed that, in metric units, the range of waves considered should embrace

T_p^2 from 13 to 30 times the significant wave height.

Amongst other things, the naval architect must design for the impact loads arising from the severe waves both on near vertical structures such as bridge fronts and on decks and hatch covers.

Environmental pollution

Besides ensuring that the ship can meet the hazards imposed on it by the environment, it is important to ensure that the ship itself does not hazard the environment. Under one of the fundamental conventions of IMO, MARPOL, requirements are laid down to restrict or totally prevent pollution in certain areas of sea particularly those close to land and in land-locked seas such as the Mediterranean. These deal with oil, sewage, food disposal and packaging materials. Ships must either store waste products or process them on board. Sewage treatment plants can be fitted to turn raw sewage into an inoffensive effluent that can be pumped overboard; incinerators can burn many packaging materials; crushers can compact waste so that it can be stored more conveniently. Standards for any effluent pumped overboard and the composition gases emanating from incinerators are kept under regular review. Suitable means for disposing stored material must be available at ports of call.

Modern society is very concerned with all aspects of pollution and the constraints upon the seafarer are gradually becoming more severe. For instance, the application of hull anti-fouling coatings containing tributyltin (TBT) are prohibited after January 2003. Fortunately TBT-free coatings are now available from industry.

Problems

1. A water lighter is in the form of a rectangular barge, 42.7 m long by 5.5 m beam, floating at an even draught of 2.44 m in sea water. Its centre of gravity is 1.83 m above the keel. What will be the new draughts at the four corners if 20 m^3 of fresh water are pumped out from an amidships tank whose centre of volume is 0.3 m from the port side and 0.91 m above the keel? Ignore free surface effects.

2. A box-form vessel, 150 m long, 50 m wide and 20 m deep floats on an even keel with a draught of 6 m in salt water. A transverse watertight bulkhead is fitted 25 m from the forward end. If the compartment thus formed is open to the sea, estimate the new draughts forward and aft to the nearest centimetre.
3. A methane carrier, displacing 6000 m^3 floats at draughts of 5.1 m forward and 5.8 m aft, measured at marks which are 80.5 m forward and 67.5 m aft amidships. The vessel grounds, on a falling tide, on a rock at a position 28 m forward of amidships. Calculate the force on the ship's bottom and the new draughts after the tide has fallen 25 cm.

   The rock then ruptures the bottom and opens to the sea, a compartment between two transverse bulkheads 14.5 m and 28.5 m forward of amidships and between two longitudinal bulkheads 5.2 m each side of the centre line. Estimate the draughts of the ship when the tide has risen to float the ship free.

   The hydrostatic curves show for about these draughts:

   WP area = 1850 m^2 Length BP = 166 m

   CF abaft amidships = 6 m

   $\overline{GM}_L = 550$ m
4. A catamaran is made up of two hulls, each 80 m long and with centres 30 m apart. The constant cross-section of each hull has the form of an equilateral triangle, each side being of 10 m length. The draught of the craft is 4 m and its $\overline{KG}$ is 10 m.

   Calculate the heeling couple and the angle of heel necessary to bring one hull just clear of the water, assuming that the broadside wind force does not effectively increase the displacement.

   Each of the hulls has main bulkheads 10 m each side of amidships. If these central compartments were both open to the sea, calculate the virtual $\overline{GM}$ of the craft in the bilged condition, comparing the added weight and lost buoyancy methods.
5. A ship displacing 2255 tonnef and 110 m in length has grounded on a rock at a point 33.5 m forward of amidships. At low water the draughts forward and aft are 2.72 m and 4.04 m respectively. Given the following hydrostatic particulars of the ship in level trim and taking the $\overline{KG}$ as 5.04 m, calculate; (i) the force on the rock at low tide; (ii) the virtual $\overline{GM}$ at low tide; (iii) the rise in tide necessary to refloat the ship.

Mean draught (m)	Displacement (tonnef)	LCB (m aft)	VCB (m)	LCF (m aft)	$\overline{BM}_L$ (m)	$\overline{BM}_T$ (m)
3.66	2235	2.32	2.26	6.97	287.4	3.71
3.35	1970	1.72	2.08	6.33	312.7	4.14

6. A ship, for which the hydrostatic particulars of the preceding question apply at level trim, has a length of 110m and displaces 2067 tonnef at level trim and with a $\overline{KG}$ of 5.08 m. A collision causes uncontrollable flooding of the auxiliary machinery space. Calculate *(a)* the final draughts at the forward and aft perpendiculars; *(b)* the virtual $\overline{GM}$ in the bilged condition. Particulars of the flooded compartment are as follows: after bulkhead, 6.1 m forward of amidships, length of compartment, 6.1 m, beam above 3.05 m waterline assumed constant at 11.58 m, area of section assumed constant at 29.7m^2 up to 3.05m waterline, centroid of section above keel, 2.01 m. Hydrostatic data may be linearly extrapolated when required.
7. A rectangular pontoon, 300m x 12m x 4m draught, is divided by a longitudinal bulkhead at the middle line and by four equally spaced transverse bulkheads. The metacentric height is 2 m. Find the draughts at the four corners when one corner compartment is bilged. Ignore rotation of the principal axes of the waterplane.
8. A box-shaped vessel 100m long and 20 m broad, floats at a draught of 6 m forward and 10m aft, the metacentric height being 2.25 m. Find the virtual metacentric height when the keel just touches level blocks throughout its length.
9. A vessel of box form, 150m long and 25 m broad, floats at an even draught of 8 m, and has a watertight deck 8.5 m above the keel. If a central compartment, 30 m long, bounded by two transverse bulkheads extending up to the deck, is bilged, what will be *(a)* the new draught of the vessel, *(b)* the alteration in metacentric height if the water admitted is regarded as lost buoyancy.
10. A ship 120m long and floating at a level draught of 4.0 m has a displacement vol = 3140m^3. Its centre of gravity is 5.1 m above the keel and, at the 4m waterline, the TPM = 1200, MCT = 8000 tonnef m/m, $\overline{KB}$ = 2.2m and the centre of flotation is 5.1 m abaft amidships.

    After striking a rock at a point 2 m abaft the fore perpendicular, the foremost 7 m are flooded up to the waterline. The displacement lost up to the 4 m waterline is 40 m^3 with its centre of buoyancy, 4.5 m abaft the FP and 2 m above the keel. Twenty square metres of waterplane are lost, with the centroid 4.5 m abaft the FP

    Making reasonable assumptions, estimate: *(a)* the force on the rock immediately after the accident; *(b)* the force on the rock when the tide has fallen 30cm; *(c)* the rise of tide necessary to lift the bow just clear of the rock.
11. The following particulars refer to a 30,500 tonnef aircraft carrier:

length, 213m	KB, 4.88m
beam, 27.4m	$\overline{KG}$, 9.14m
draught, 8.53 m	TPC, 40 tonnef/cm
$\overline{GM}_T$, 1.83m	

    The design is modified by increasing all athwart ships dimensions by 3 per cent, and all longitudinal dimensions by 2 per cent. The weight of the flight deck armour is adjusted to limit the increase in displacement to 914 tonnef.

    It should be assumed that the height of the flight deck above keel remains constant at 21.3 m and that the weight of the ship varies directly as the length and beam.

    In compiling the flooding board for the ship, a compartment 18.3 m long by 6.1 m wide is assumed flooded, and open to the sea. If the compartment extends the full depth of the ship, with its centre 12.2 m to starboard of the middle line, estimate the resulting angle of heel. The effects of changing trim should be ignored.
12. A vessel of length 120m between perpendiculars floats at a uniform draught of 5.0 m with the following particulars:

    Displacement, 3000 tonnef
    WP area, 1300 m^2
    CF abaft amidships, 3 m
    CB above keel, 2.70 m
    CG above keel, 4.52 m
    Transverse metacentre above keel, 6.62 m
    MCT BP, 7300 tonnef m/m

    It may be assumed to be wall-sided in the region of the waterplane.

    The vessel is involved in an accident in which 60 tonnef outer bottom (with centre of gravity at keel level and 3.2m from the middle line) are torn away and a compartment 3.1 m wide and 7.9 m long is opened to the sea.

    Estimate the resulting angle of heel, assuming that the compartment extends from keel level to above the new waterplane and neglecting any change of trim. The centroids of volume and plan area of the flooded compartment are 3.0 m from the middle line.
13. A ship 120m BP for which hydrostatic and other data are given in question 12, floats at a uniform draught of 5.0m.

    The vessel grounds on a rock at keel level, 20 m forward of amidships and 3.0 m from the middle line. Making reasonable assumptions, calculate the force on the rock and the angle of heel when the tide has fallen 15cm.
14. A rectangular pontoon 3.66m deep, 10.97m wide and 32m long is divided into three equal sections by two transverse bulkheads. The centre compartment is further subdivided into three equal compartments by two lon~ tudinal bulkheads. All compartments extend the full depth of the pontoon. The centre of gravity is 1.83 m above the keel.

    The pontoon is damaged in such a way that the three central compartments are open to the sea at keel level. Of the three compartments, the centre one is vented at deck level and the wing compartments are airtight above keel level. The draught after damage is 2.19 m and the depth of water in the wing tanks is 0.52 m. The position of the centre of gravity of the pontoon may be assumed unaffected.

    Calculate the virtual metacentric height in the damaged condition, assuming that the height of the sea water barometer is 10m.
15. A ship floating at a draught of 2 m has a hole of area 0.3 m^2 in the bottom at the keel, giving access to a rectangular compartment right across the

ship. The compartment is 10m long and has an even permeability of 0.80 throughout. This ship has initially WP area = 300 m^2. Its length is 70 m, beam 8 m and it does not trim due to the damage. Calculate how long it will be before the maximum flooded draught occurs.

16. Calculate the probability of losing half and total power of worked example 1, assuming that there was no 15m separation of the machinery units.

6 The ship girder

Few who have been to sea in rough weather can doubt that the structure of a ship is subject to strain. Water surges and crashes against the vessel which responds with groans and shudders and creaks; the bow is one moment surging skywards, the next buried beneath green seas; the fat middle of the ship is one moment comfortably supported by a wave and the next moment abandoned to a hollow. The whole constitutes probably the most formidable and complex of all structural engineering problems in both the following aspects:

(a) the determination of the loading
(b) the response of the structure.

As with most complex problems, it is necessary to reduce it to a series of unit problems which can be dealt with individually and superimposed. The smallest units of structure which have to be considered are the panels of plating and single stiffeners which are supported at their extremities by items which are very stiff in comparison; they are subject to normal and edge loads under the action of which their dishing, bowing and buckling behaviour relative to the supports may be assessed. Many of these small units together constitute large flat or curved surfaces of plating and sets of stiffeners called grillages, supported at their edges by bulkheads or deck edges which are very stiff in comparison; they are subject to normal and edge loading and their dishing and buckling behaviour as a unit relative to their supports may be assessed. Finally, many bulkheads, grillages and decks, together constitute a complete hollow box whose behaviour as a box girder may be assessed. It is to this last unit, the whole ship girder, that this chapter is confined, leaving the smaller units for later consideration.

Excluding inertia loads due to ship motion, the loading on a ship derives from only two sources, gravity and water pressure. It is impossible to conceive a state of the sea whereby the loads due to gravity and water pressure exac~ cancel out along the ship's length. Even in still water, this is exceedingTy

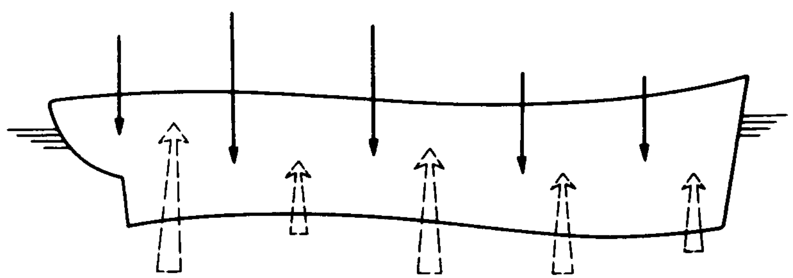


Fig. 6.1

unlikely but in a seaway where the loading is changing continuously, it is inconceivable. There is therefore an uneven loading along the ship and, because it is an elastic structure, it bends. It bends as a whole unit, like a girder on an elastic foundation and is called the *ship girder.* The ship will be examined as a floating beam subject to the laws deduced in other textbooks for the behaviour of beams.

In still water, the loading due to gravity and water pressure are, of course, weight and buoyancy. The distribution of buoyancy along the length follows the curve of areas while the weight is conveniently assessed in unit lengths and might, typically, result in the block diagram of Fig. 6.2. (Clearly, the areas representing total weight and total buoyancy must be equal.) This figure would

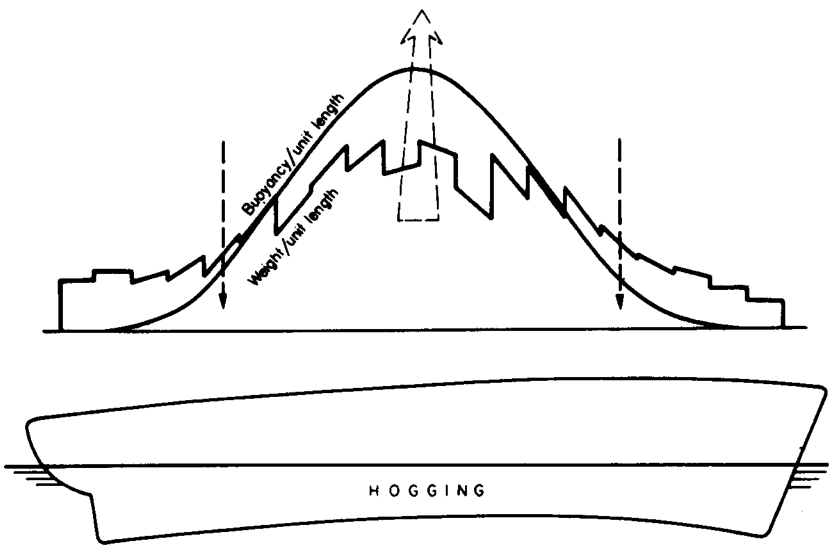


Fig. 6.2 *Still water hogging*

give the resultants dotted which would make the ship bend concave downwards or *hog.* The reverse condition is known as *sagging.* Because it is not difficult to make some of the longer cargo ships break their backs when badly loaded, consideration of the still water hogging or sagging is vital in assessing a suitable cargo disposition. It is the first mate's yardstick of structural strength.

It is not difficult to imagine that the hog or sag of a ship could be much increased by waves. A long wave with a crest amidships would increase the upward force there at the expense of the ends and the hogging of the ship would be increased. **If** there were a hollow amidships and crests towards the ends sagging would be increased (Fig. 6.3). The loads to which the complete **hull** girder is subject are, in fact:

(a) those due to the differing longitudinal distribution of the downward forces of weight and the upward forces of buoyancy, the ship considered at rest in still water;

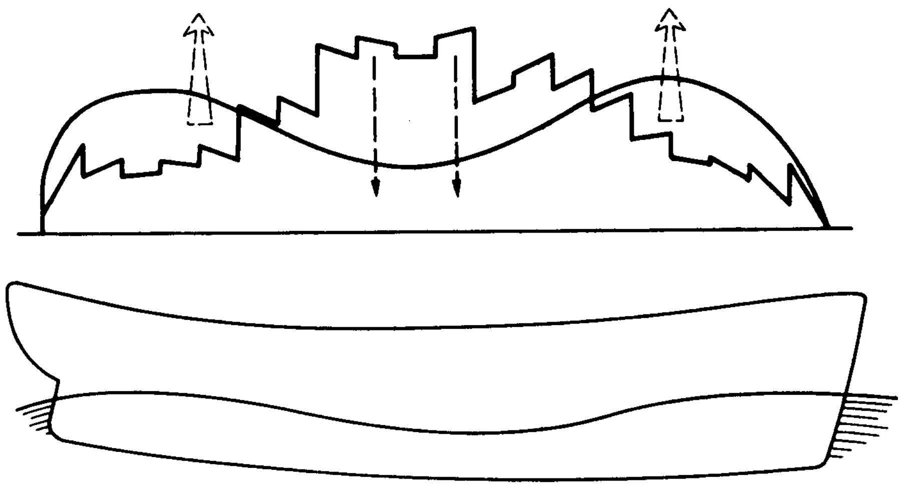


Fig. 6.3 *Sagging on a wave*

(b) the additional loads due to the passage of a train of waves, the ship remaining at rest;
(c) loads due to the superposition on the train of waves of the waves caused by the motion of the ship itself through still water;
(d) the variations of the weight distribution due to the accelerations caused by ship motion.

Consideration of the worst likely loading effected by *(a)* and *(b)* is the basis of the standard calculation. The effects of *(c)* and *(d)* are smaller and are not usually taken into account except partly by the statistical approach outlined later.

The standard calculation

This is a simple approach but one that has stood the test of time. It relies on a comparison of a new design with previous successful design. The calcula~ stresses are purely notional and based on those caused by a single wavof length equal to the ship's length, crest normal to the middle line plane and with

(a) a crest amidships and a hollow at each end causing maximum hogging, and
(b) a hollow amidships and a crest at each end causing maximum sagging.

The ship is assumed to be momentarily still, balanced on the wave with zero velocity and acceleration and the response of the sea is assumed to be that appropriate to static water. **In** this condition, the curves of weight and buoyancy are deduced. Subtracted one from the other, the curves give a curve of net loading p'. Now, a fundamental relationship at a point in an elastic beam is

$$p' = \frac{\mathrm{d}S}{\mathrm{d}x} = \frac{\mathrm{d}^2 M}{\mathrm{d}x^2}$$

where p' is the load per unit length, S is the shearing force, M is the bending moment, and x defines the position along the beam.

$$\therefore S = \int p' \, dx \quad \text{and} \quad M = \int S \, dx$$

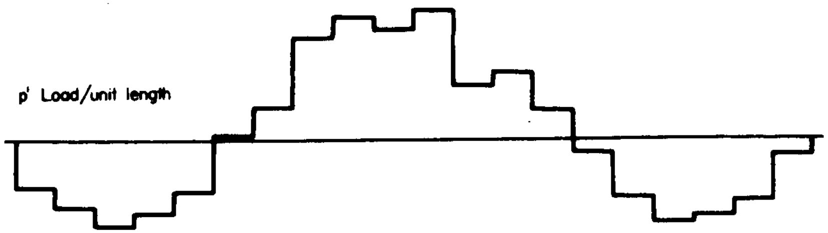


Fig. 6.4 Loading curve, p'

Thus, the curve of loading p' must be integrated along its length to give a curve of shearing force and the curve of shearing force must be integrated along its length to give the bending moment curve. From the maximum bending moment, a figure of stress can be obtained,

$$\text{stress } \sigma = \frac{M}{I} y$$

I/y being the modulus of the effective structural section.

A closer look at each of the constituents of this brief summary is now needed.

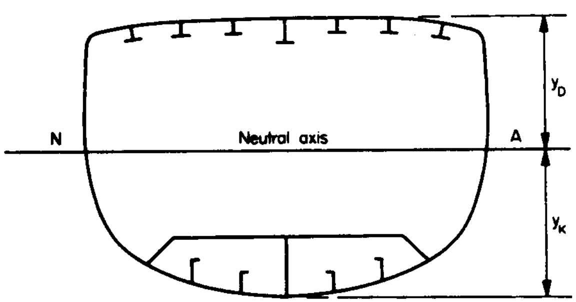


Fig. 6.5 Simplified structural section

THE WAVE

Whole books have been written about ocean waves. Nevertheless, there is no universally accepted standard ocean wave which may be assumed for the standard longitudinal strength calculation. While the shape is agreed to be trochoidal, the observed ratios of length to height are so scattered that many 'standard' lines can be drawn through them. Fortunately, this is not of primary importance; while the calculation is to be regarded as comparative, provided that the same type of wave is assumed throughout for design and type ships, the comparison is valid.

A trochoid is a curve produced by a point at radius r within a circle of radius R rolling on a flat base. The equation to a trochoid with respect to the axes shown in Fig. 6.6, is

$$x = R\theta - r\sin\theta$$
$$z = r(1 - \cos\theta)$$

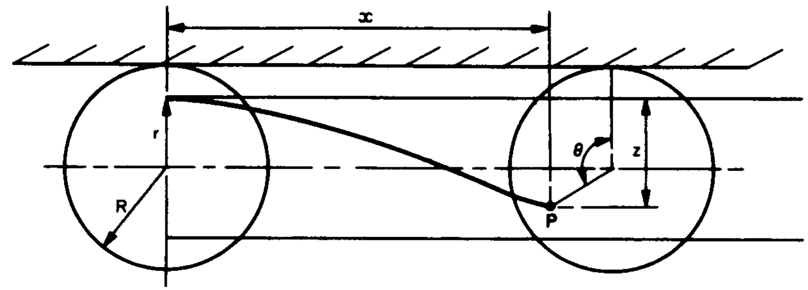


Fig. 6.6 Construction of a trochoid

One accepted standard wave is that having a height from trough to crest of one twentieth of its length from crest to crest. In this case, $L = 2\pi R$ and $r = h/2 = L/40$ and the equation to the wave is

$$x = \frac{L}{2\pi}\theta - \frac{L}{40}\sin\theta$$

$$z = \frac{L}{40}(1 - \cos\theta)$$

The co-ordinates of this wave at equal intervals of x are:

$\frac{x}{L/20}$	0	1	2	3	4	5	6	7	8	9	10
$\frac{z}{L/20}$	0	0.034	0.128	0.266	0.421	0.577	0.720	0.839	0.927	0.982	1.0

Research has shown that the $L/20$ wave is somewhat optimistic for wavelengths from 90 m up to about 150 m in length. Above 150 m, the $L/20$ wave becomes progressively more unsatisfactory and at 300 m is probably so exaggerated in height that it is no longer a satisfactory criterion of comparison. This has resulted in the adoption of a trochoidal wave of height $0.607\sqrt{L}$ as a standard wave in the comparative longitudinal strength calculation. This wave has the equation

$$x = \frac{L}{2\pi}\theta - \frac{0.607\sqrt{L}}{2}\sin\theta \qquad x, z \text{ and } L \text{ in metres}$$

$$z = \frac{0.607\sqrt{L}}{2}(1 - \cos\theta)$$

The $0.607\sqrt{L}$ wave has the slight disadvantage that it is not non-dimensional, and units must be checked with care when using this wave and the formulae derived from it. Co-ordinates for plotting are conveniently calculated from the equations above for equal intervals of θ.

The length of the wave is, strictly, taken to be the length of the ship on the load waterline; in practice, because data is more readily available for the displacement stations, the length is often taken between perpendiculars (if this is different) without making an appreciable difference to the bending moment. Waves of length slightly greater than the ship's length can, in fact, produce theoretical bending moments slightly in excess of those for waves equal to the ship's length, but this has not been an important factor while the calculation continued to be comparative.

Waves steeper than *L/7* cannot remain stable. Standard waves of size *L/9* are used not uncommonly for the smaller coastal vessels. It is then a somewhat more realistic basis of comparison.

WEIGHT DISTRIBUTION

Consumable weights are assumed removed from those parts of the ship where this aggravates the particular condition under investigation; in the sagging condition, they are removed from positions near the ends and, in the hogging condition, they are removed amidships. The *influence lines* of a similar design should be consulted before deciding where weights should be removed, and the decision should be verified when the influence lines for the design have been calculated (see later), provided that the weights are small enough.

The longitudinal distribution of the weight is assessed by dividing the ship into a large number of intervals. Twenty displacement intervals are usually adequate. The weight falling within each interval is assessed for each item or group in the schedule of weights and tabulated. Totals for each interval divided by the length give mean weights per unit length. It is important that the centre of gravity of the ship divided up in this way should be in the correct position. To ensure this, the centre of gravity of each individual item should be checked after it has been distributed.

One of the major items of weight requiring distribution is the hull itself, and this will sometimes be required before detailed structural design of the hull has been completed. A useful first approximation to the hull weight distribution is obtained by assuming that two-thirds of its weight follows the still water buoyancy curve and the remaining one-third is distributed in the form of a trapezium, so arranged, that the centre of gravity of the whole hull is in its correct position (Fig. 6.7).

Having obtained the mean weight per unit length for each interval, it is plotted as the mid-ordinate of the interval and a straight line is drawn through it, parallel to the chord of the buoyancy curve. This is a device for simplifying later stages of the calculation which introduces usually insignificant inaccuracies. A sawtooth distribution of weight per unit length, as shown in Fig. 6.3, results.

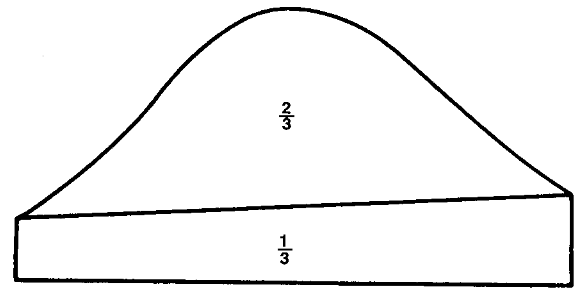


Fig. 6.7 Approximate hull weight distribution

BUOYANCY AND BALANCE

If the standard calculation is performed by hand, the wave is drawn on tracing paper and placed over the contracted profile of the ship on which the Bonjean curve has been drawn at each ordinate. Figure 6.8 shows one such ordinate. It is necessary for equilibrium to place the wave at a draught and trim such that

(a) the displacement equals the weight and
(b) the centre of buoyancy lies in the same vertical plane as the centre of gravity.

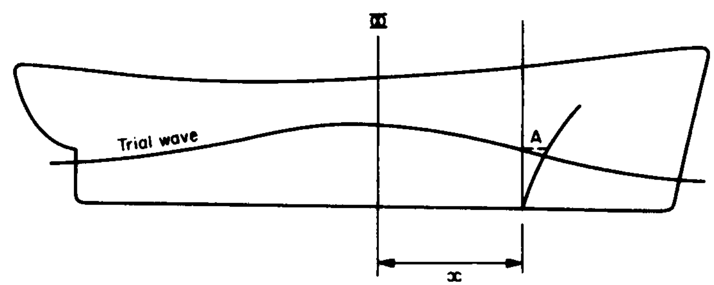


Fig. 6.8

Simple hydrostatic pressure is assumed; the areas immersed at each ordinate can be read from the Bonjean curves where the wave profile cuts the ordinate, and these areas are subjected to normal approximate integration to give displacement and LCB position. The position of the wave to meet the two conditions can be found by trial and error. It is usual to begin at a draught to the midships point of the trochoid about 85 per cent of still water draught for the hogging condition and 120 per cent for the sagging wave. A more positive way of achieving balance involves calculating certain new tools. Consider a typical transverse section of the ship, *x* m forward of amidships, at which the Bonjean curve shows the area immersed by the first trial wave surface to be $A\,m^2$.

If $\nabla\,\mathrm{m}^3$ and $M\,\mathrm{m}^4$ are the volume of displacement and the moment of this volume before amidships for the trial wave surface, then, over the immersed length

$$\nabla = \int A\,\mathrm{d}x \text{ and } M = \int Ax\,\mathrm{d}x$$

Suppose that $\nabla_0\,\mathrm{m}^3$ and $M_0\,\mathrm{m}^4$ are the volume of displacement and moment figures for equilibrium in still water and that ∇ is less than ∇_0 and M is less than M_0. The adjustment to be made to the trial wave waterline is therefore a parallel sinkage and a trim by the bow in order to make $\nabla = \nabla_0$ and $M = M_0$. Let this parallel sinkage at amidships be z_0 and the change of trim be m radians, then the increased immersion at the typical section is

$$z = (z_0 + mx)\,\mathrm{m}$$

Let the slope of the Bonjean curve be $s = \mathrm{d}A/\mathrm{d}z\,\mathrm{m}$. Assuming that the Bonjean curve is straight over this distance, then

$$A + zs = A + (z_0 + mx)s\,\mathrm{m}^2$$

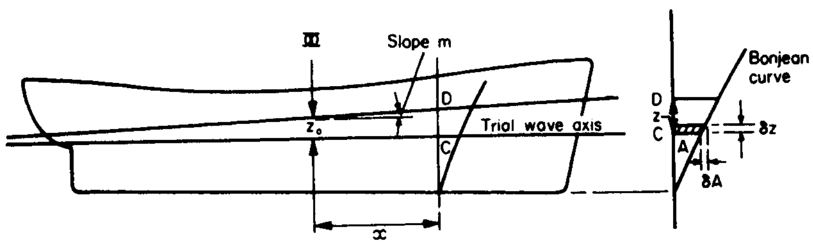


Fig. 6.9

In order to satisfy the two conditions, $\nabla = \nabla_0$ and $M = M_0$, therefore,

$$\int [A + (z_0 + mx)s]\,\mathrm{d}x = \nabla_0$$

and

$$\int [A + (z_0 + mx)s]x\,\mathrm{d}x = M_0$$

Now put

$$\int s\,\mathrm{d}x = \delta\nabla\,\mathrm{m}^2$$

$$\int sx\,\mathrm{d}x = \delta M\,\mathrm{m}^3$$

$$\int sx^2\,\mathrm{d}x = \delta I\,\mathrm{m}^4$$

Then

$$z_0 \cdot \delta\nabla + m\delta M = \nabla_0 - \nabla$$

and

$$z_0 \cdot \delta M + m\delta I = M_0 - M$$

$\delta\nabla$, δM and δI can all be calculated by approximate integration from measurement of the Bonjean curves. From the pair of simultaneous equations above the required sinkage z_0 and trim m can be calculated. A check that this wave position in fact gives the required equilibrium should now be made. If the trial waterplane was a poor choice, the process may have to be repeated. Negative values of z_0 and m, mean that a parallel rise and a trim by the stern are required.

Having achieved a balance, the curve of buoyancy per metre can be drawn as a smooth curve. It will have the form shown in Fig. 6.2 for the hogging calculation and that of Fig. 6.3 for the sagging condition.

LOADING, SHEARING FORCE AND BENDING MOMENT

Because the weight curve has been drawn parallel to the buoyancy curve, the difference between the two, which represents the net loading p' for each interval, will comprise a series of rectangular blocks. Using now the relationship $S = \int p'\,\mathrm{d}x$, the loading curve is integrated to obtain the distribution of shearing force along the length of the ship. The integration is a simple cumulative addition starting from one end and the shearing force at the finishing end should, of course, be zero; in practice, due to the small inaccuracies of the preceding steps, it will probably have a small value. This is usually corrected by canting the base line, i.e. applying a correction at each section in proportion to its distance from the starting point.

The curve of shearing force obtained is a series of straight lines. This curve is now integrated in accordance with the relationship $M = \int S\,\mathrm{d}x$ to obtain the distribution of bending moment M. Integration is again a cumulative addition of the areas of each trapezium and the inevitable final error, which should be small, is distributed in the same way as is the shearing force error. If the error is large, the calculations must be repeated using smaller intervals for the weight distribution.

The integrations are performed in a methodical, tabular fashion as shown in the example below. In plotting the curves, there are several important features which arise from the expression

$$p' = \frac{\mathrm{d}S}{\mathrm{d}x} = \frac{\mathrm{d}^2M}{\mathrm{d}x^2}$$

which will assist and act as checks. These are

(*a*) when p' is zero, S is a maximum or a minimum and a point of inflexion occurs in the M curve.
(*b*) when p' is a maximum, a point of inflexion occurs in the S curve,
(*c*) when S is zero, M is a maximum or minimum.

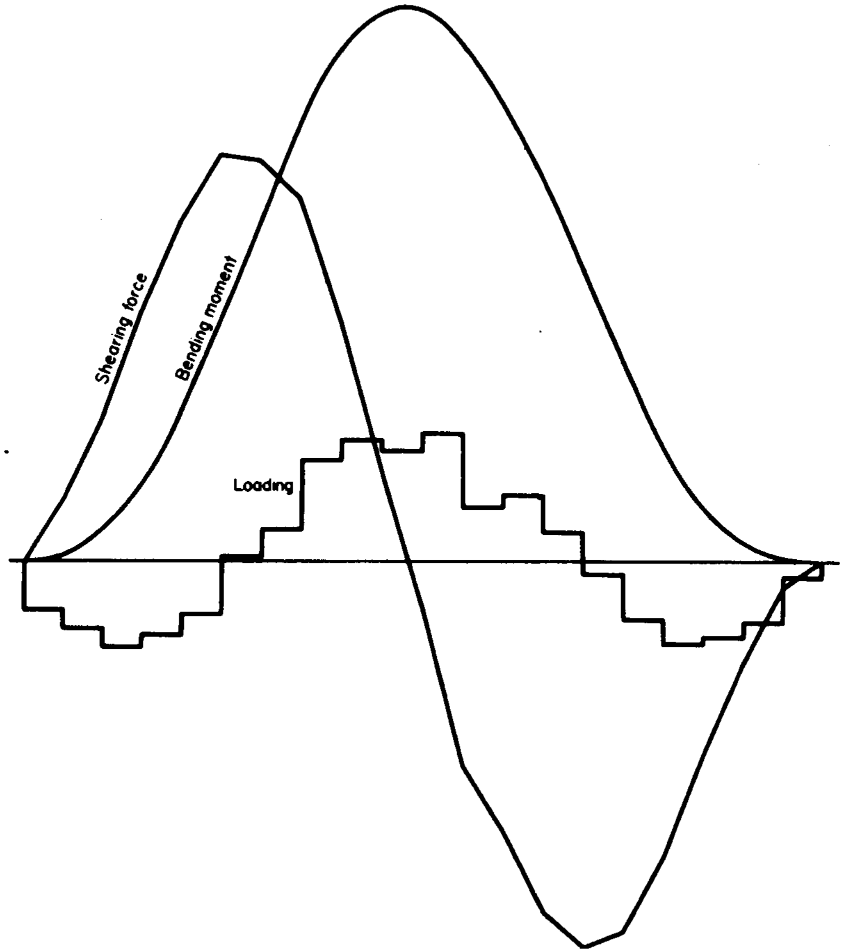


Fig. 6.10 Typical loading, shearing force and bending moment curves

A typical set of curves is shown in Fig. 6.10.

EXAMPLE 1. The mean immersed cross-sectional areas between ordinates of a ship, 300 m in length, balanced on a hogging wave, as read from the Bonjean curves are given below, together with the mass distribution. The second moment of area of the midship section is 752 m^4 and the neutral axis is 9.30m from the keel and 9.70m from the deck.

Calculate the maximum direct stresses given by the comparative calculation and the maximum shearing force.

Making the assumption that the second moment of area is constant along the length, estimate the difference in slope between ordinates number 4 and 10 and the deflection of the middle relative to the ends.

Ordinate	11	10	9	8	7	6	5	4	3	2	1
Area A. m^2	10.5	49.5	243	490	633	565	302	71	2.5	0.5	
Mass M, Mg	5030	6790	10,670	9900	9370	7830	9260	6400	4920	2690	

Solution: Integrate in tabular form using trapezoidal rule

Ordinate spacing = 30 m

$$\text{Weight per metre} = \frac{M\,\text{Mg}}{30\ \text{m}} \cdot \frac{9.81\,\text{m}}{s^2} \cdot \frac{1000\,\text{kg}}{\text{Mg}} = 327\,M, \text{N/m}$$

$$\text{Buoyancy per metre} = A\,\text{m}^3 \cdot \frac{1.026\,\text{Mg}}{\text{m}^3} \cdot \frac{9.81\,\text{m}}{\text{s}^2} \cdot \frac{1000\,\text{kg}}{\text{Mg}} = 10{,}060\,A, \text{N/m}$$

a Ord.	b wt./m $\frac{327\,M}{10^6}$ (MN)	c buoy./m $\frac{10{,}060A}{10^6}$ (MN)	d load/m b − c (MN)	e δSF 30d (MN)	f SF Σe (MN)	g mid SF (MN)	h δBM 30 g (MN)	i BM Σh (MN m)	j corrn.	k BM (MN m)
1					0			0		0
	0.880	0.005	0.875	26.25		13.13	394			
2					26.25			394	−4	390
	1.609	0.025	1.584	47.52		50.01	1500			
3					73.77			1894	−8	1886
	2.093	0.714	1.379	41.37		94.46	2834			
4					115.14			4728	−12	4716
	3.028	3.038	−0.010	−0.30		114.99	3450			
5					114.84			8178	−16	8162
	2.560	5.684	−3.124	−93.72		67.98	2039			
6					21.12			10,217	−20	10,197
	3.064	6.368	−3.304	−99.12		−28.44	−853			
7					−78.00			9364	−24	9340
	3.237	4.929	−1.692	−50.76		−103.38	−3101			
8					−128.76			6263	−28	6235
	3.489	2.445	1.044	31.32		−113.10	−3393			
9					−97.44			2870	−32	2838
	2.220	0.498	1.722	51.66		−71.61	−2148			
10					−45.78			722	−36	686
	1.645	0.106	1.539	46.17		−22.70	−681			
11					0.39			41	−40	1

Maximum shearing force = 128.76 MN
Maximum bending moment = 10.220 MN m (by plotting column k)

$$\text{Keel stress} = \frac{10.220\,\text{MN m}}{752\,\text{m}^4} \times 9.3\,\text{m} = 126.5\,\text{N/mm}^2\ (8.2\,\text{tonf/in}^2)$$

$$\text{Deck stress} = \frac{10{,}220}{752} \times 9.7 = 132\,\text{N/mm}^2\ (8.6\,\text{tonf/in}^2)$$

$$\text{Slope} = \frac{1}{EI}\int M\,\text{d}x = \frac{\text{d}y}{\text{d}x} = \frac{10^{-6}}{0.208 \times 752}\int M\,\text{d}x \text{ and } y = \int \frac{\text{d}y}{\text{d}x}\cdot \text{d}x$$

Columns e and h are not strictly necessary to this table since the multiplier 30 could have been applied at the end; they have been included merely to give the student an idea of the values expressed in meganewtons.

a Ord	b BM (MN m)	c mid BM	d Σc (MN m^2)	e mid d	f Σe	g $y = 5.75$f ($\times 10^{-6}$)	h chord	i h − g
1	0		0		0	0	0	
		195		98				
2	390		195		98			
		1138		764				
3	1886		1333		862			
		3301		2984				
4	4716		4634		3846			
		6439		7854				
5	8162		11,073		11,700	0.066	0.484	0.418
		9180		15,663				
6	10,197		20,253		27,363	0.154	0.605	0.451
		9769		25,138				
7	9340		30,022		52,501	0.296	0.726	0.430
		7788		33,916				
8	6235		37,810		86,417			
		4537		40,079				
9	2838		42,347		126,496			
		1762		43,228				
10	686		44,109		169,724			
		343		44,280				
11	1		44,452		214,004	1.209	1.209	

$$\text{Difference in slope 4–10 Ords.} = \frac{10^{-6}}{0.208 \times 752}(44{,}109 - 4634) \times 30 = 0.0076 \text{ radians}$$

$$= 0.0076 \times \frac{180}{\pi} \times 60 = 26 \text{ minutes of arc}$$

$$\text{Deflection } y = \frac{30 \times 30 \times 10^{-6}}{0.208 \times 752}\text{f} = 5.75\text{f} \times 10^{-6}$$

$$\text{Central deflection} = 0.451 \text{ m}$$

Column h is necessary because, as shown in Fig. 6.11 by integrating from no. 1 ord., zero slope has been assumed there; to obtain deflection relative to the ends, the chord deflection must be subtracted.

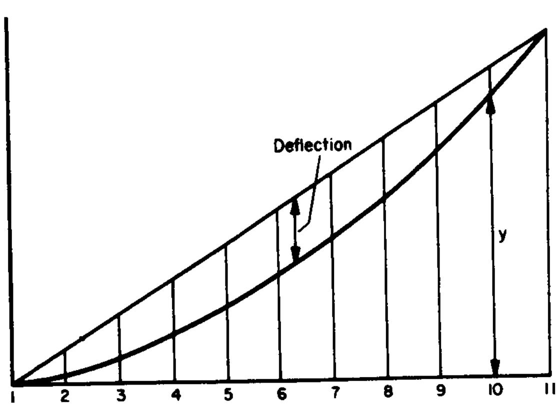


Fig. 6.11

SECOND MOMENT OF AREA

It is now necessary to calculate the second moment of area, I, of the section or the ship girder. For a simple girder this is a straightforward matter. For a ship girder composed of plates and sections which may not extend far longitudinally or which might buckle under compressive load or which may be composed of differing materials, a certain amount of adjustment may be judged necessary. Unless there is some good reason to the contrary (such as inadequate jointing in a wood sheathed or armoured deck), it is assumed here that local structural design has been sufficient to prevent buckling or tripping or other forms of load shirking. **In** fact, load shirking by panels having a width/thickness ratio in excess of seventy is likely and plating contribution should be limited to seventy times the thickness (see Chapter 7). No material is assumed to contribute to the section modulus unless it is structurally continuous for at least half the length of the ship amidships. Differing materials are allowed for in the manner described on the next page. **In** deciding finally on which parts of the cross section to include, reference should be made to the corresponding assumptions made for the designs with which final stress comparisons are made. Similar comparisons are made also about members which shirk their load.

Having decided which material to include, the section modulus is calculated in a methodical, tabular form. An *assumed neutral axis* (ANA) is first taken near the mid-depth. Positions and dimensions of each item forming the structural mid-section are then measured and inserted into a table of the following type:

Table 6.1
Modulus calculation

1	2	3	4	5	6	7
Item	A (cm^2)	h (m)	Ah (cm^2 m)	Ah^2 (cm^2 m^2)	k^2 (m^2)	Ak^2 (cm^2 m^2)
Each item above ANA						
Totals above ANA	$\sum A_1$		$\sum A_1h_1$	$\sum A_1h_1^2$		$\sum A_1k_1^2$
Each item below ANA						
Totals below ANA	$\sum A_2$		$\sum A_2h_2$	$\sum A_2h_2^2$		$\sum A_2k_2^2$

where A = cross-sectional area of item, h = distance from ANA, k = radius of gyration of the structural element about its own NA.
Subscript 1 is used to denote material above the ANA and subscript 2 for material below.

$$\text{Distance of true NA above ANA} = \frac{\sum A_1h_1 - \sum A_2h_2}{\sum A_1 + \sum A_2} = d$$

Second moment of area about true NA

$$= \sum A_1h_1^2 + \sum A_2h_2^2 + \sum A_1k_1^2 + \sum A_2k_2^2 - \left(\sum A_1 + \sum A_2\right)d^2 = I$$

Lever above true neutral axis from NA to deck at centre = y_D
Lever below true neutral axis from NA to keel = y_K

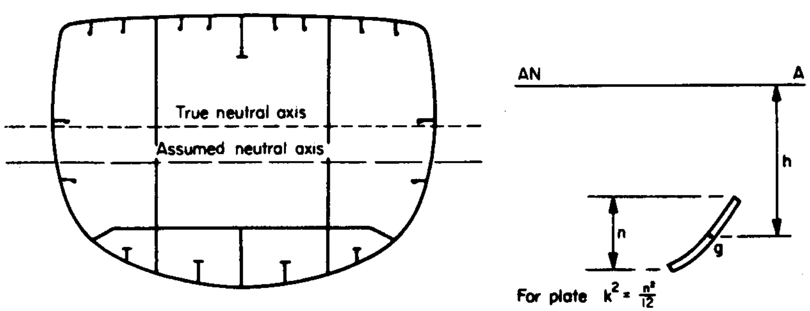


Fig. 6.12

. The strength section of some ships is composed of different materials, steel, light alloy, wood or plastic. How is the second moment of area calculated for these composite sections? Consider a simple beam composed of two materials, suffixes a and b. From the theory of beams, it is known that the stress is directly proportional to the distance from the neutral axis and that, if R is the radius of curvature of the neutral axis and E is the elastic modulus,

$$\text{stress } \sigma = \frac{E}{R}h$$

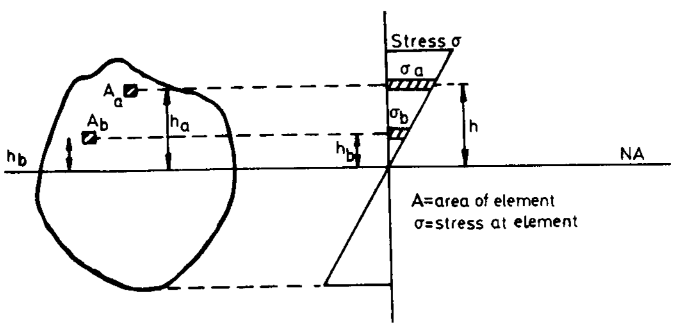


Fig. 6.13 Composite section

Consider the typical element of area A of Fig. 6.13. For equilibrium of the cross-section, the net force must be zero, therefore

$$\Sigma(\sigma_a A_a + \sigma_b A_b) = 0$$

$$\Sigma\left(\frac{E_a}{R}A_a h_a + \frac{E_b}{R}A_b h_b\right) = 0$$

i.e.

$$\Sigma\left(A_a h_a + \frac{E_b}{E_a}A_b h_b\right) = 0$$

Now the contribution to the bending moment by the element A is the force σA multiplied by the distance from the neutral axis, h. For the whole section,

$$\begin{aligned} M &= \Sigma(\sigma_a A_a h_a + \sigma_b A_b h_b) \\ &= \frac{E_a}{R}\Sigma\left(A_a h_a^2 + \frac{E_b}{E_a}A_b h_b^2\right) \\ &= \frac{E_a}{R}\cdot I_{\text{eff}}. \end{aligned}$$

where I_{eff} is the effective second moment of area.

It follows from the equations above that the composite section may be assumed composed wholly of material a, provided that an effective area of material b, $(E_b/E_a)A_b$, be used instead of its actual area. The areas of material b used in columns 2, 4, 5 and 7 of Table 6.1 must then be the actual areas multiplied by the ratio of the elastic moduli. The ratio E_b/E_a for different steels is approximately unity; for wood/steel it is about -&, in compression and 15 in tension. For aluminium alloy/steel it is about! and for glass reinforced plastic/steel it is between -ts and -t0. The figures vary with the precise alloys or mixtures and should be checked.

BENDING STRESSES

Each of the constituents of the equation, $\sigma = (M/I)y$, has now been calculated, maximum bending moment M, second moment of area, I, and the levers y for two separate conditions, hogging and sagging. Values of the maximum direct stress at deck and keel arising from the standard comparative longitudinal strength calculation can thus be found.

Direct stresses occurring in composite sections are, following the work of the previous section

It is at this stage that the comparative nature of the calculation is apparent, since it is now necessary to decide whether the stresses obtained are acceptable or whether the strength section needs to be modified. It would, in fact, be rare good fortune if this was unnecessary, although it is not often that the balance has to be repeated in consequence. The judgment on the acceptable level of stress is based on a comparison with similar ships in similar service for which a similar calculation has previously been performed. This last point needs careful checking to ensure that the same wave has been used and that the same assumptions regarding inclusion of material in weight and modulus calculations are made. For example, a device was adopted many years ago to allow for the presence of rivet holes, either by decreasing the effective area of section on the tension side by -fi or by increasing the tensile stress in the ratio ~ Although this had long been known to be an erroneous procedure, some authorities continued to use it in order to maintain the basis for comparison for many years.

Stresses different from those found acceptable for the type ships may be considered on the following bases:

(a) Length of ship. An increase of acceptable stress with length is customary on the grounds that standard waves at greater lengths are less likely to be met. This is more necessary with the $L/20$ wave than the $0.607\sqrt{L}$ wave for which the probability varies less with length. If a standard thickness is allowed for corrosion, it will constitute a smaller proportion of the modulus for larger ships.

(b) Life of ship. Corrosion allowance is an important and hidden factor. Classification societies demand extensive renewals when survey shows the plating thicknesses and modulus of section to be appreciably reduced. It could well be economical to accept low initial stresses to postpone the likely time of renewal. Comparison with type ship ought to be made before corrosion allowance is added and the latter assessed by examining the performance of modern paints and anti-corrosive systems (see Chapter 14).

(c) Conditions of service. Classification societies permit a reduced modulus of section for service in the Great Lakes or in coastal waters. Warship authorities must consider the likelihood of action damage by future weapons and the allowance to be made in consequence. Warships are not, of course, restricted to an owner's route.

(d) Local structural design. An improvement in the buckling behaviour or design at discontinuities may enable higher overall stress to be accepted.

(e) Material. Modern high grade steels permit higher working stresses and the classification societies encourage their use, subject to certain provisos.

(f) Progress. A designer is never satisfied; a structural design which is entirely successful suggests that it was not entirely efficient in the use of materials and he is tempted to permit higher stresses next time. Such progress must necessarily be cautiously slow.

A fuller discussion of the nature of failure and the aim of the designer of the future occurs later in the chapter. Approximate values of total stress which have been found satisfactory in the past are given below:

Ships	Wave	Design stresses Deck	Keel
		N/mm^2	N/mm^2
100 m frigate	$L/20$	110	90
150 m destroyer	$L/20$	125	110
200 m general cargo vessel	$0.607\sqrt{L}$	110	90
250 m aircraft carrier	$L/20$	140	125
300 m oil tanker	$0.607\sqrt{L}$	140	125

SHEAR STRESSES

The shearing force at any position of the ship's length is that force which tends to move one part of the ship vertically relative to the adjacent portion. It tends to distort square areas of the sides into rhomboids. The force is distributed over the section, each piece of material contributing to the total. It is convenient to consider shear stress, the force divided by the area and this is divided over the cross-section of a simple beam according to the expression

$$\tau = \frac{SA\bar{y}}{Ib}$$

$A\bar{y}$ is the moment about the NA of that part of the cross-section above section PP where the shear stress τ is required. I is the second moment of area of the whole cross-section and b is the total width of material at section P (Fig. 6.15). The distribution of shear stress over a typical cross-section of a ship is shown in Fig. 6.14. The maximum shear stress occurs at the neutral axis at those points along the length where the shearing force is a maximum. A more accurate distribution would be given by shear flow theory applied to a hollow box girder, discussed presently.

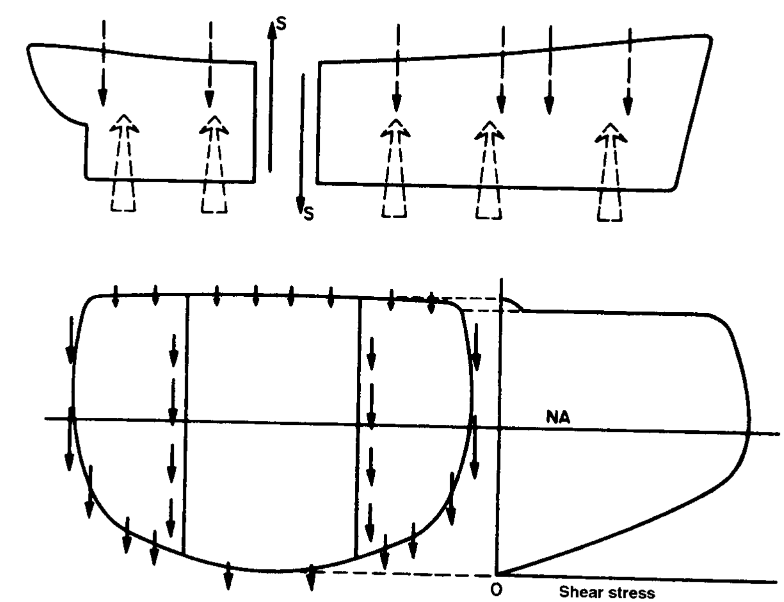


Fig. 6.14 Shear loading on a ship girder

Acceptable values of shear stress depend on the particular type of side construction. Failure under the action of shear stress would normally comprise wrinkling of panels of plating diagonally. The stress at which this occurs depends on the panel dimensions, so while the shear stress arising from the standard longitudinal strength calculation clearly affects side plating thickness

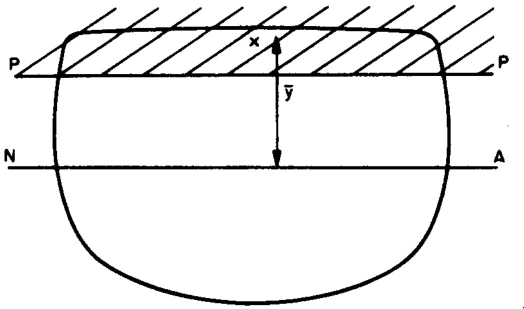


Fig. 6.15

and stiffener spacings, it does not have a profound effect on the structural cross-section of the ship.

Already, there is a tendency to assume that the stresses obtained from the standard comparative calculation actually occur in practice, so that the design of local structure may be effected using these stresses. Because local structure affected by the ship girder stresses cannot otherwise be designed, there is no choice. Strictly, this makes the local structure comparative. In fact, the stresses obtained by the comparative calculation for a given wave profile have been shown by full scale measurements to err on the safe side so that their use involves a small safety factor. Often, other local loading is more critical. This is discussed more fully under 'Criterion of failure.'

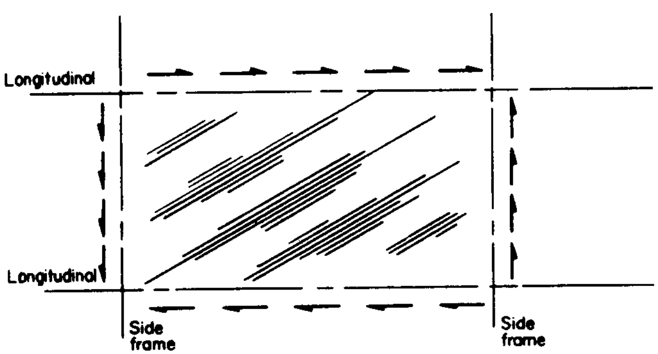


Fig. 6.16

INFLUENCE LINES

The ship will not often, even approximately, be in the condition assumed for the standard calculation. It is important for designers and operators to know at a glance, the effect of the addition or removal of weight on the longitudinal strength. Having completed the standard calculation, the effects of small additions of weight are plotted as influence lines in much the same way as for bridges and buildings (see Fig. 6.17).

An influence line shows the effect on the *maximum* bending moment of the addition of a unit weight at any position along the length. The height of the line

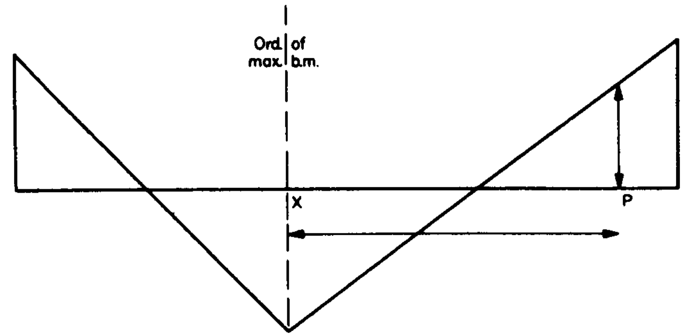


Fig. 6.17

at P represents the effect on the maximum bending moment at X of the addition of a unit weight at P. Two influence lines are normally drawn, one for the hogging and one for the sagging condition. Influence lines could, of course, be drawn to show the effect of additions on sections other than that of the maximum BM but are not generally of much interest.

Consider the addition of a weight P, x aft of amidships, to a ship for which the hogging calculation has been performed as shown in Fig. 6.18. It will cause a parallel sinkage s and a change of trim t over the total length L. If A and I are the area and least longitudinal second moment of area of the curved waterplane and the ship is in salt water of reciprocal weight density u, then, approximately,

$$\text{sinkage } s = \frac{uP}{A} \quad \text{and} \quad \text{trim } t = \frac{P(x-f)uL}{I}$$

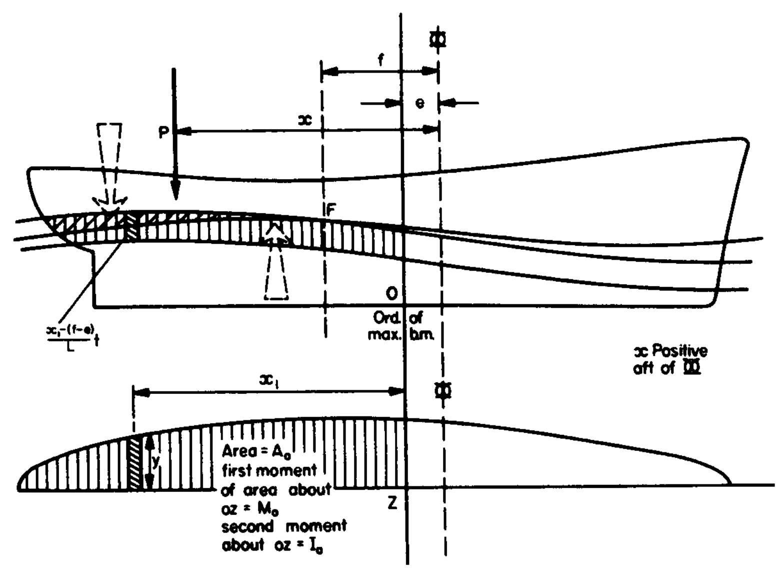


Fig. 6.18

The increase in bending moment at OZ is due to the moment of added weight less the moment of buoyancy of the wedges aft of OZ less the moment of buoyancy of the parallel sinkage aft of OZ.

Moment of buoyancy of parallel sinkage about OZ $= \dfrac{sM_a}{u} = \dfrac{PM_a}{A}$

Moment of buoyancy of wedges

$$= \frac{1}{u}\int 2yx_1 \times \frac{(x_1 - \overline{f - e})}{L} t\, dx_1$$

$$= \frac{I_a t}{uL} - \frac{M_a(f-e)t}{uL} = \{I_a - M_a(f-e)\}\frac{P(x-f)}{I}$$

Moment of weight which is included only if the weight is aft of OZ and therefore only if positive

$$= P[x - e], \text{ positive values only.}$$

Increase in BM for the addition of a unit weight,

$$\frac{\delta M}{P} = -\{I_a - M_a(f-e)\}\frac{(x-f)}{I} - \frac{M_a}{A} + [x-e]$$

Note that this expression is suitable for negative values of x (i.e. for P forward of amidships), provided that the expression in square brackets, [ ], is discarded if negative. A discontinuity occurs at OZ, the ordinate of maximum bending moment. The influence lines are straight lines which cut the axis at points about 0.2–0.25 of the length from amidships. It is within this length, therefore, that weights should be removed to aggravate the hogging condition and outside this length that they should be removed to aggravate the sagging condition.

Certain simplifying assumptions are sometimes made to produce a simpler form of this equation. There is little point in doing this because no less work results. However, if $(f - e)$ is assumed negligible, n put equal to I_a/I and F put equal to M_a/A

$$\frac{\delta M}{P} = -n(x-f) - F + [x-e]$$

This gives the results shown in Fig. 6.19.

A note of caution is necessary in the use of influence lines. They are intended for small weight changes and are quite unsuitable for large changes such as might occur with cargo, for example. In general, weight changes large enough to make a substantial change in stress are too large for the accuracy of influence lines.

CHANGES TO SECTION MODULUS

Being a trial and error process, the standard calculation will rarely yield a suitable solution first time. It will almost always be necessary to return to the structural section of the ship to add or subtract material in order to adjust the resulting stress. The effect of such an addition, on the modulus I/y, is by no

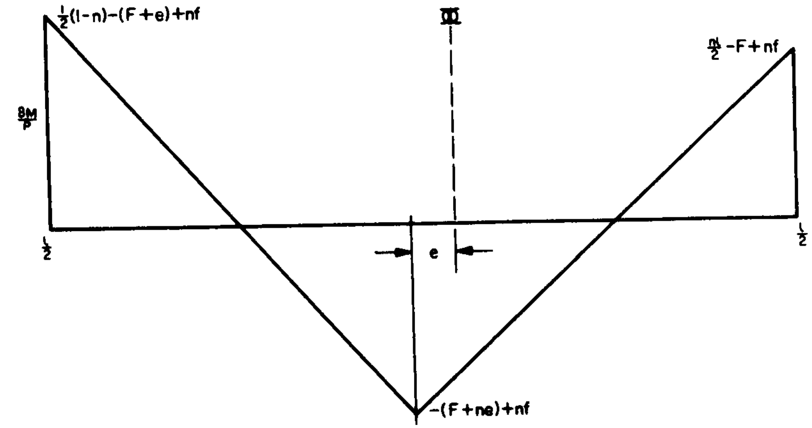


Fig. 6.19

means obvious. Moreover, it is not obvious in the early stages, whether the superstructure should be included in the strength section or whether the construction should be such as to discourage a contribution from the superstructure. Some measure of the effect of adding material to the strength section is needed if we are to avoid calculating a new modulus for each trial addition.

In order to provide such a measure, consider (Fig. 6.20) the addition of an area a at a height z above the neutral axis of a structural section whose second moment of area has been calculated to be Ak^2, area A, radius of gyration k and levers y_1 to the keel and y_2 to the deck. Now

$$\text{stress } \sigma = \frac{M}{I/y}$$

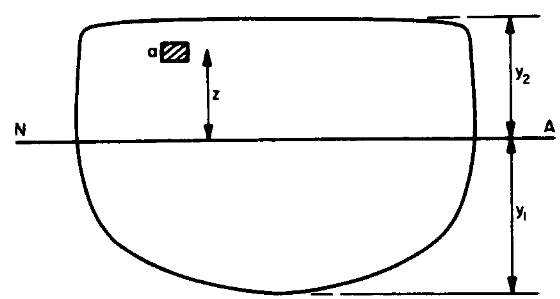


Fig. 6.20

By considering changes to I and y it can be shown that for a given bending moment, M, stress σ will obviously be reduced at the deck and will be reduced at the keel if $z > k^2/y_1$, when material is added within the section, $z < y_2$.

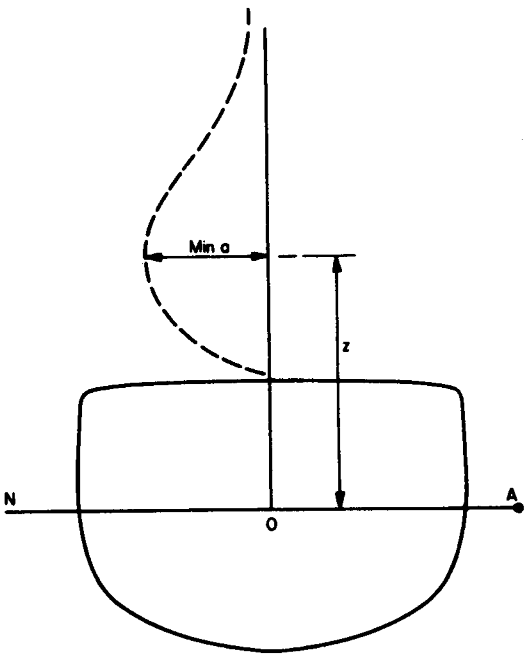


Fig. 6.21

If the material is added above the deck, $z > y_2$ then the maximum stress occurs in the new material and there will be a reduction in stress above the neutral axis (Fig. 6.21) if

$$a > \frac{A(z/y_2 - 1)}{(z^2/k^2 + 1)}$$

The position is not quite so simple if the added material is a superstructure. See later section on superstructures.

EXAMPLE 2. In converting a steel survey ship, it is proposed to extend the short forecastle for the whole length of the ship and to arrange the structure so that it contributes 100 per cent to the hull girder. The new structure is wholly of light alloy. Estimate the new nominal stresses due to the change in section modulus, assuming that the bending moment remains unchanged.

Before conversion: BM = 7742 tonnef/m, $I = 23{,}970\,\text{cm}^2\,\text{m}^2$, $A = 4520\,\text{cm}^2$, y, deck = 2.9 m; y, keel = 3.05 m

Added structure: side plating 2.3 m × 10 mm stiffened by one 26 cm² girder at mid-height; deck plating 11 m × 10 mm stiffened by five 26 cm² girders with centre of area 8 cm below the deck.

E, light alloy: 69 GPa
E, steel: 207 GPa

Solution:
Above upper deck

Item	A (cm²)	h (m)	Ah (cm² m)	Ah^2 (cm² m²)	k^2 (m²)	Ak^2 (cm² m²)
two sides	460	1.15	529	608	0.44	202
two side girders	52	1.15	60	69	0	0
deck	1100	2.30	2530	5819	0	0
five deck girders	130	2.22	289	642	0	0
	1742	1.96	3408	7138		202

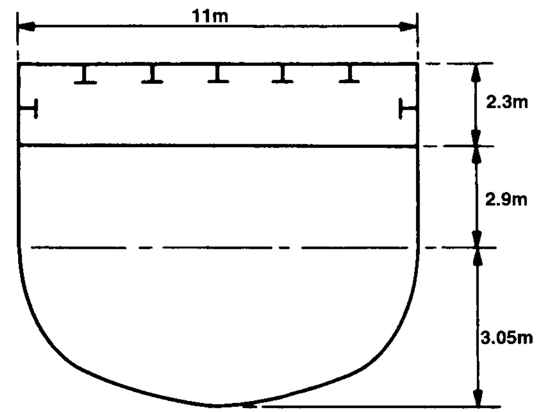


Fig. 6.22

$$\text{New effective area} = 4520 + \tfrac{1}{3}(1742) = 5{,}100\,\text{cm}^2$$

$$\text{Movement of neutral axis} = \frac{1}{3}\frac{(1742)(2.9 + 1.96)}{5100} = 0.553\,\text{m}$$

$$\begin{aligned} I \text{ about old NA} &= 23{,}970 + \tfrac{1}{3}\Big\{7138 + 202 - 1742\,(1.96)^2 \\ &\qquad + 1742(1.96 + 2.9)^2\Big\} \\ &= 37{,}900\,\text{cm}^2\,\text{m}^2 \end{aligned}$$

$$I \text{ about new NA} = 37{,}900 - 5{,}100\,(0.553)^2 = 36{,}340\,\text{cm}^2\,\text{m}^2$$

$$y\text{, new deck} \quad = 5.2 - 0.553 = 4.65\,\text{m}$$

$$y\text{, keel} \quad = 3.05 + 0.553 = 3.60\,\text{m}$$

$$\text{Old stresses were, } \sigma\text{, keel} = \frac{7742}{23{,}970} \times 3.05 = 0.99\,\text{tonnef/cm}^2 \text{ (steel)}$$

$$\sigma\text{, deck} = \frac{7742}{23{,}970} \times 2.9 = 0.94\,\text{tonnef/cm}^2 \text{ (steel)}$$

$$\text{New stresses are, } \sigma, \text{ keel} = \frac{7742}{36,340} \times 3.60 = 0.77 \text{ tonnef/cm}^2 \text{ (steel)}$$

$$\sigma, \text{ deck} = \frac{1}{3} \times \frac{7742}{36,340} \times 4.65 = 0.33 \text{ tonnef/cm}^2 \text{ (alloy)}$$

$$\sigma, \text{ deck} = \frac{7742}{36,340} \times 2.35 = 0.50 \text{ tonnef/cm}^2 \text{ (steel)}$$

SLOPES AND DEFLECTIONS

The bending moment on the ship girder has been found by integrating first the loading p' with respect to length to give shearing force S, and then by integrating the shearing force S with respect to length to give M. There is a further relationship with which students will already be familiar.

$$M = EI \frac{d^2y}{dx^2}$$

or

$$\frac{dy}{dx} = \frac{1}{E} \int \frac{M}{I} dx \quad \text{and} \quad y = \int \frac{dy}{dx} dx$$

Thus, the slope at all points along the hull can be found by integrating the M/EI curve and the bent shape of the hull profile can be found by integrating the slope curve. It is not surprising that the errors involved in integrating approximate data four times can be quite large. Moreover, the calculation of the second moment of area of all sections throughout the length is laborious and the standard calculation is extended this far only rarely—it might be thus extended, for example, to give a first estimate of the distortion of the hull between a master datum level and a radar aerial some distance away. It may also be done to obtain a deflected profile in vibration studies.

HORIZONTAL FLEXURE

Flexure perpendicular to the plane with which we are normally concerned may be caused by vibration, flutter, uneven lateral forces or bending while rolling. Vibrational modes are discussed in Chapter 9. Unsymmetrical bending can be resolved into bending about the two principal axes of the cross-section. The only point in the ship at which the maxima of the two effects combine is the deck edge and if the ship were to be balanced on a standard wave which gave a bending moment M, the stress at the deck edge would be

$$\sigma = \frac{Mz}{I_{yy}} \cos\theta + \frac{My}{I_{zz}} \sin\theta$$

However, it is not quite so easy as this; if the ship were heading directly into a wave train, it would not be rolling. Only in quartering or bow seas will bending and rolling be combined and, in such a case, the maximum bending moments in the two planes would not be in phase, so that the deck edge effect will be less than that obtained by superposition of the two maxima. Some limited research

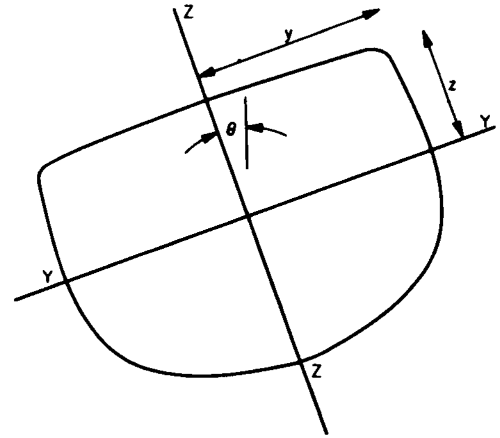


Fig. 6.23

into this problem indicates that horizontal bending moment maxima are likely to be of the order of 40 per cent of the vertical bending moment maxima and the ratio of the stresses is likely to be about 35 per cent in common ship shapes. The increase in deck edge stress over that obtained with head seas is thought to be of the order of 20-25 per cent. This is an excellent reason to avoid stress concentrations in this area.

BEHAVIOUR OF A HOLLOW BOX GIRDER

The simple theory of bending of beams assumes that plane sections of the beam remain plane and that the direct stress is directly proportional to the distance from the neutral axis. A deck parallel to the neutral axis would therefore be expected to exhibit constant stress across its width. In fact, the upper flange of a hollow box girder like a ship's hull can receive its load only by shear at the deck edge. The diffusion of this shear into a plane deck to create the direct stresses is a difficult problem mathematically; the diffusion from the edge elements across the deck is such that the plane sections do not remain plane, and the mathematics shows that the direct stresses reduce towards the middle of the dedit' This has been borne out by observations in practice. The effect is known as *shear lag* and its magnitude depends on the type of loading and dimensions of the ship; it is more pronounced, for example, under concentrated loads.

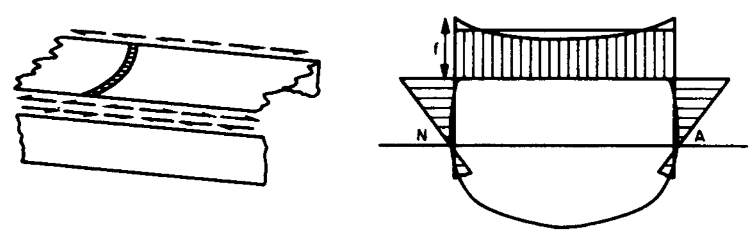


Fig. 6.24 Direct stresses in practice

In common ship shapes, it accounts for a difference in stress level at mid-deck of only a few per cent. It is more important in the consideration of the effects of superstructures and of the effective breadth of plating in local strength problems which will be discussed later.

WAVE PRESSURE CORRECTION

This is usually known as the *Smith Correction.* In calculating the buoyancy per unit length at a section of the ship, the area of the section given by the Bonjean curve cut by the wave surface was taken. This assumes that the pressure at a point P on the section is proportional to the depth h of the point below the wave surface and the buoyancy $= w\int h\,\mathrm{d}B = w \times$ area immersed (Fig. 6.25).

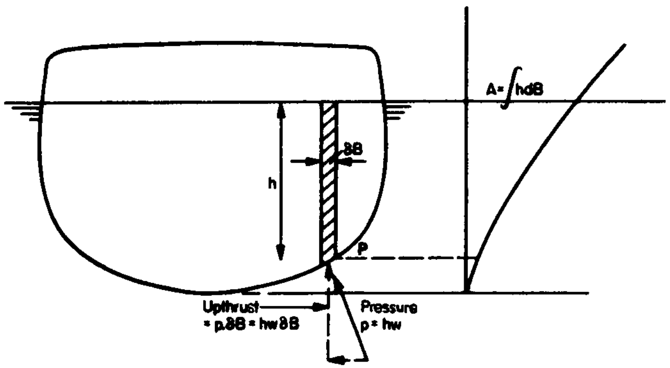


Fig. 6.25

It is shown in Chapter 9, that this simple hydrostatic law is not true in a wave because the wave is caused by the orbital motion of water particles. It can be shown that the pressure at a point P in a wave at a depth h below the wave surface is the same as the hydrostatic pressure at a depth h', where h' is the distance between the mean (or still water) axis of the surface trochoid and the

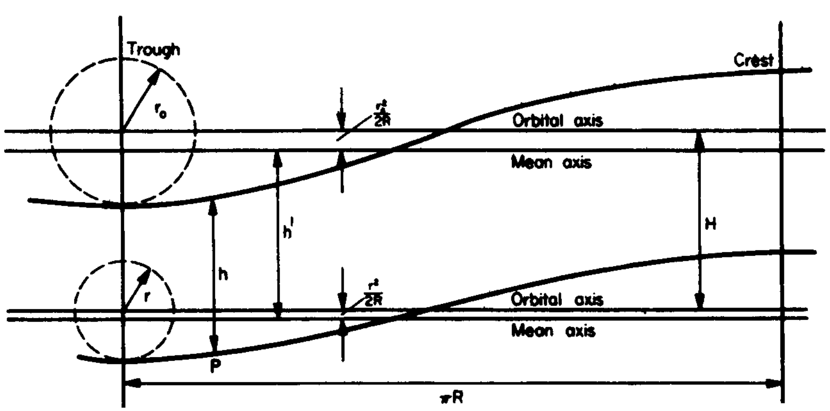


Fig. 6.26

subsurface trochoid through P (Fig. 6.26). Thus, the buoyancy at a section of a ship is

$$w\int h'\,\mathrm{d}B = w \times (\text{effective area immersed}).$$

Setting up h' from the sub-surface trochoid through P and similar heights from a series of sub-surface trochoids drawn on the wave profile, the effective immersed area up to W′L is obtained. New Bonjean curves can thus be plotted (Fig. 6.27). The formula for h' is

$$h' = H - \frac{r_0^2}{2R}\left\{1 - \exp\left(-\frac{2H}{R}\right)\right\}$$

An approximation to the Smith correction is:

$$\text{corrected wave BM} = \text{wave BM} \times \exp\left(-\frac{nT}{L}\right)$$

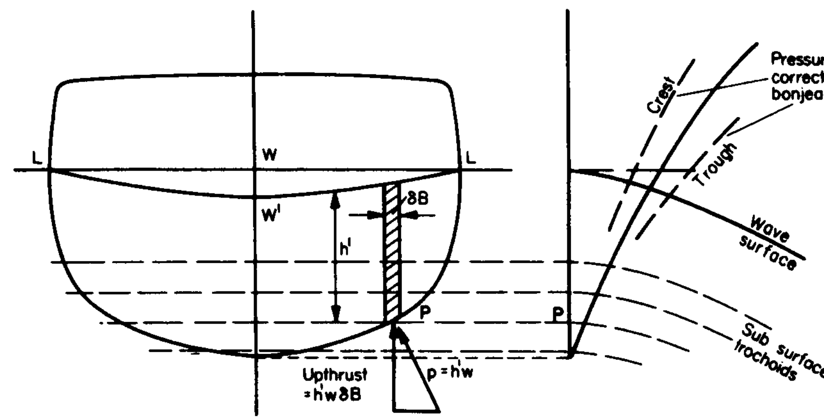


Fig. 6.27

where T is the draught, L the length and n a constant given by:

Block coeff. C_B	n, sagging	n, hogging
0.80	5.5	6.0
0.60	5.0	5.3

LONGITUDINAL STRENGTH STANDARDS BY RULE

Until 1960 the Classification Societies prescribed the structure of merchant ships through tables of dimensions. They then changed to the definition of applied load and structural resistance by formulae. In the 1990s the major Classification Societies under the auspices of the International Association of Classification Societies (IACS) agreed a common minimum standard for the longitudinal strength of ships supported by the statistics of structural failure.

There is today widespread acceptance of the principle that there is a very remote probability that load will exceed strength during the whole lifetime of a ship. This probability may be as low as 10^{-8} at which level the IACS requirement is slightly more conservative than almost every Classification Society standard.

Loading on a merchant ship is separated into two parts:

(a) the bending moment and shear force due to the weight of the ship and the buoyancy in still water,
(b) the additional effects induced by waves.

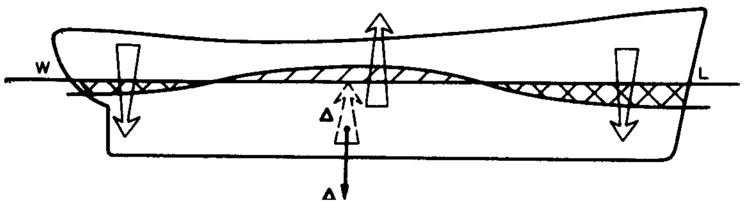


Fig. 6.28 Wave induced bending forces

Still water loading is calculated by the simple methods described at the beginning of this chapter without, of course, the wave which is replaced by the straight waterline of interest. Several such waterlines will usually be of concern. Stresses caused by such loading may be as much as 40 per cent of the total stresses allowed and incorrect acceptance of cargo or unloading of cargo has caused spectacular failures of the ship girder. With such ships as bulk carriers in particular, Masters must follow the sequences of loading and unloading recommended by their Classification Societies with scrupulous care. Not only may the ship break its back by bending but failure could be caused by the high shear forces that occur between full and empty holds. Indeed, a simple desktop computer to assess changes as they are contemplated has become common since high capacity cargo handling has evolved.

Wave induced bending moment (WIBM) is now accepted to be represented by the formulae:

$$\text{Hogging WIBM} = 0.19\,MCL^2BC_b\,\text{kN m}$$

$$\text{Sagging WIBM} = -0.11\,MCL^2B(C_b + 0.7)\,\text{kN m}$$

where L and B are in metres and $C_b \geq 0.6$

$$\text{and } C = 10.75 - \left(\frac{300 - L}{100}\right)^{1.5} \quad \text{for } 90 \leq L \leq 300\,\text{m}$$

$$= 10.75 \quad \text{for } 300 < L < 350\,\text{m}$$

$$= 10.75 - \left(\frac{L - 350}{150}\right)^{1.5} \quad \text{for } 350 \leq L$$

M is a distribution factor along the length of the ship.

$$M = 1.0 \text{ between } 0.4L \text{ and } 0.65L \text{ from the stern}$$

$$= 2.5x/L \text{ at } x \text{ metres from the stern up to } 0.4L$$

$$= 1.0 - \frac{x - 0.65L}{0.35L} \text{ at } x \text{ metres from the stern between } 0.65L \text{ and } L.$$

Wave induced shear force is given by IACS as

$$\text{Hogging condition } S = 0.3\,F_1CLB(C_b + 0.7)\,\text{kN}$$

$$\text{Sagging condition } S = -0.3\,F_2CLB(C_b + 0.7)\,\text{kN}$$

where L, B and C_b are as given above and F_1 and F_2 by Fig. 6.29.

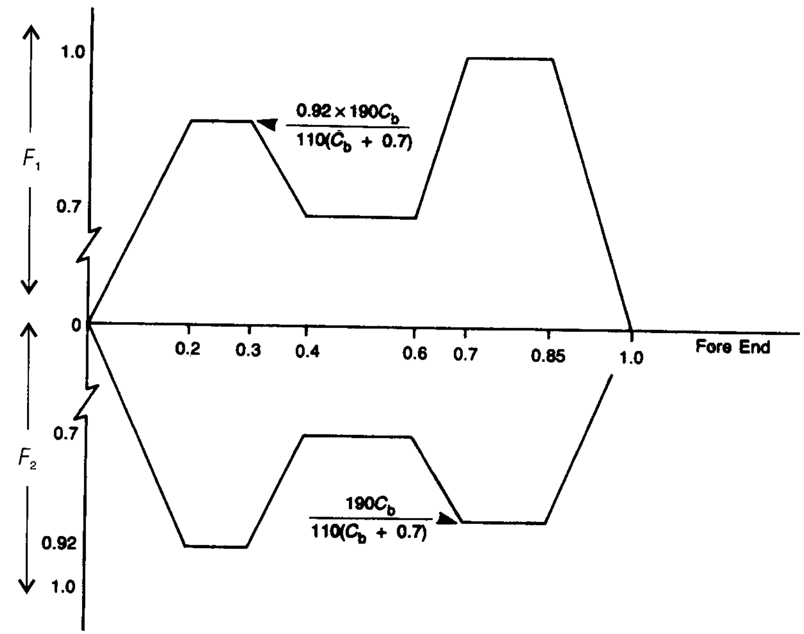


Fig. 6.29 Factors F_1 and F_2

The modulus of cross section amidships must be such that bending stress caused by combining still water and wave induced BM is less than 175 N/mm^2. There is the further proviso that, despite the waters in which the vessel may sail, this midship section modulus shall not be less than:

$$CL^2B(C_b + 0.7)\,\text{cm}^3$$

which represents a stress of 110 N/mm^2.

The maximum permissible total shear stress, wave induced plus still water is also 110 N/mm^2.

The formulae given above embrace most common ship types. Special consideration must be given to vessels with large deck openings, large flare, heated cargoes or which fall outside limits when:

$L \geq 500$ m $\quad L/B \leq 5$

$C_b < 0.60 \quad B/D \geq 2.5$

Classification Societies now rely upon computer programs based on the methods presently to be described in this chapter. Fundamental to this is the representation of the wave loading by wave spectra and by strip theory. Strip theory is linear and, in essence, assumes the vessel to be wallsided. Thus, wave induced loading would be the same in both hogging and sagging conditions which is known in practice not to be so. Sagging stresses are greater by about 10 per cent than hogging stresses due in part to the support at the end of the ship provided by flare. This accounts for the different coefficients given in the formulae above, providing some degree of non-linearity in the treatment.

This gratifying conformity among Classification Societies is, of course, reviewed periodically. It may be reviewed if statistics show that standards have become inadequate or if oceanographers demonstrate that world wave patterns are changing with climate. Variations remain among the Classification Societies in the more detailed aspects of structural design and in allowances made for high quality steels. Variations also occur in the stresses permitted among different ship types and with restricted service conditions.

FULL SCALE TRIALS

Full scale trials can be carried out for a variety of purposes.

(a) To load a hull, possibly to destruction, noting the (ultimate) load and the failure mechanisms.

(b) To measure the strains in a ship at sea, over a short period, in a seaway which is itself measured.

(c) To measure strains in ships at sea over a prolonged period of service with the crew noting any signs of failure. This is the only trial which is likely to throw light on fatigue failure.

There have been several attempts to measure the behaviour of the ship girder by loading ships and recording their behaviour at various positions by strain gauges and measurements of deflection. *Neverita* and *Newcombia, Clan Alpine* and *Ocean Vulcan* were subject to variations in loading by filling different tanks while they were afloat. *Wolf, Preston, Bruce* and *Albuera* were supported at the middle or the ends in dry dock and weight added until major structural failure occurred.

Short duration sea trials pose difficulty in finding rough weather and in measuring the wave system accurately. In one successful trial two frigates of significantly different design were operated in close company in waves with

significant heights up to 8 metres. Electrical resistance gauges were used to record stress variations with time and the seas measured using wave buoys. Of particular interest was the slamming which excites the hull girder whipping modes. It was found that the slam transient increased the sagging bending moment much more than the hogging. In frigates whipping oscillations are damped out quite quickly but they can persist longer in some vessels.

Long term trials are of great value. They provide data that can be used in the more representative calculations discussed later. For many years now a number of RN frigates have been fitted with automatic mechanical or electrical strain gauges recording the maximum compressive and tensile deck stresses in each four hour period. Ship's position, speed and sea conditions are taken from the log. Maximum bending stresses have exceeded the *L/20* standard values by a factor of almost two in some cases. This is not significant while the standard calculation is treated as purely comparative. It does show the difficulties of trying to extrapolate from past experience to designs using other constructional materials (e.g. reinforced plastic or concrete) or with a distinctly different operating pattern.

THE NATURE OF FAILURE

Stress has never caused any material to fail. Stress is simply a convenient measure of the material behaviour which may 'fail' in many different ways. 'Failure' of a structure might mean permanent strain, cracking, unacceptable deflection, instability, a short life or even a resonant vibration. Some of these criteria are conveniently measured in terms of stress. In defining an acceptable level of stress for a ship, what 'failure' do we have in mind?

Structural failure of the ship girder may be due to one or a combination of *(a)* Cracking, *(b)* Fatigue failure, *(c)* Instability.

It is a fact that acceptable stress levels are at present determined entirely by experience of previous ships in which there have been a large number of cracks in service. The fact that these might have been due to poor local design is at present largely disregarded, suggesting that some poor local design or workmanship somewhere in the important parts of the hull girder is inevitable.

However good the design of local structure and details might be, there is one important influence on the determination of acceptable levels of stress. The material built into the ship will have been rolled and, finally, welded. These processes necessarily distort the material so that high stresses are already built in to the structure before the cargo or sea impose any loads at all. Very little is known about these built-in stresses. Some may yield out, i.e. local, perhaps molecular, straining may take place which relieves the area at the expense of other areas. The built-in stress clearly affects fatigue life to an unknown degree. It may also cause premature buckling. Thus, with many unknowns still remaining, changes to the present practice of a stress level determined by a proliferation of cracks in previous ships, must be slow and cautious. Let us now examine this progress.

REALISTIC ASSESSMENT OF LONGITUDINAL STRENGTH

Study of the simple standard longitudinal strength calculations so far described has been necessary for several good reasons. First, it has conveyed an initial look at the problem and the many assumptions which have had to be made to derive a solution which was within the capabilities of the tools available to naval architects for almost a hundred years. Moreover, the standard has been adequate on the whole for the production of safe ship designs, provided that it was coupled with conventions and experience with similar structure which was known to have been safe. As a comparative calculation it has had a long record of success. Second, it remains a successful method for those without ready access to modern tools or, for those who do enjoy such access, it remains an extremely useful starting point for a process which like any design activity is iterative. That is to say, the structural arrangement is guessed, analysed, tested against standards of adequacy, refined, reanalysed and so on until it is found to be adequate. Analysis by the standard calculation is a useful start to a process which might be prolonged. Third, much of the argument which has been considered up to now remains valid for the new concepts which must now be presented.

Of course, the standard calculation can be performed more readily now with the help of computers. The computer, however, has permitted application of mathematics and concepts of behaviour which have not been possible to apply before. It has permitted an entirely new set of standards to replace the static wave balance and to eradicate many of the dubious assumptions on which it was based. Indeed, it is no longer necessary even to assume that the ship is statically balanced. The basis of the new methods is one of realism; of a moving ship in a seaway which is continuously changing.

During a day at sea, a ship will suffer as many as 10,000 reversals of strain and the waves causing them will be of all shapes and sizes. At any moment, the ship will be subject not to a single *L/20* wave but a composite of many different waves. Their distribution by size can be represented by a histogram of the numbers occurring within each range of wave lengths. **In** other words, the waves can be represented statistically. Now the statistical distribution of waves is unlikely to remain constant for more than an hour or two by which time wind, weather and sea state will modify the statistics. Over the life of a ship such sea state changes will have evened out in some way and there will be a lifetime statistical description of the sea which will be rather different from the short-term expectation.

This provides the clue to the new approaches to longitudinal strength. What is now sought as a measure is the likelihood that particular bending moments that the sea can impose upon the ship will be exceeded. This is called the probability of exceedence and it will be different if it is assessed over one hour, four hours, one day or 25 years. What the new standard does is to ensure that there is a comfortably small probability of exceedence of that bending moment which would cause the ship to fail during its lifetime. With some 30 million or so strain reversals during a lifetime, the probability of exceedence of the ship's

strength needs to be very small indeed. If the frequency distribution of applied bending moment is that given in Fig. 6.30 and the ship's strength is S, then the probability of failure is

$$p(\text{failure}) = \int_S^\infty f(\text{BM})\,\text{d BM}$$

At this stage we have some difficulty in proceeding. While it is absolutely necessary for the student to be aware of these revised methods, much of the mathematics is well beyond the scope of this book, although the concept is not. As a first step, the student is advised to break off from study of this chapter and proceed to consider Chapter 7 on the behaviour of elements of the structure and Chapter 9 for an introduction to the statistics of waves. Chapter 12 will provide greater detail in due course. We can then presume upon such understanding and ask only that the reader take on trust a general description of some of the other mathematics which will probably not yet be familiar. Thus armed we may consider longitudinal strength in terms somewhat more realistic than the single overwhelming trochoidal wave:

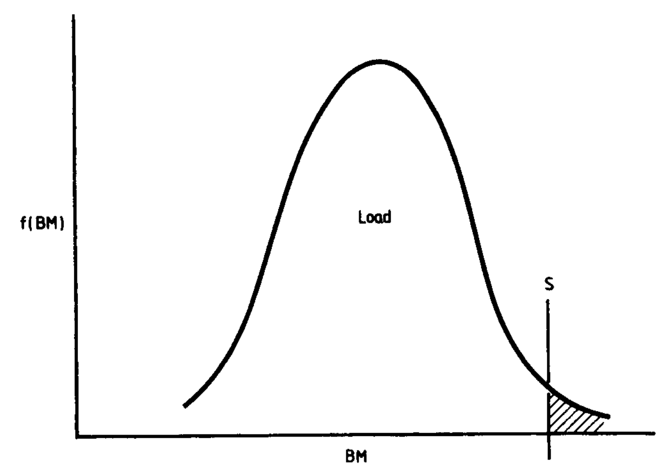


Fig. 6.30

(a) loading imposed upon the ship;
(b) response of the longitudinal structure;
(c) assessment of structural safety.

REALISTIC ASSESSMENT OF LOADING LONGITUDINALLY

It is necessary first to return to the effect of a single wave, not a trochoid which is too inconvenient mathematically but to a sinewave of a stated height and length or frequency. We need to examine the effect of this wave, not upon a rock-like ship presumed to be static, but upon a ship which will move, in particular to heave and to pitch. Movement *q(t)* of any system with one degree of freedom subject to an excitation $Q(t)$ is governed by the differential equation.

$$\ddot{q}(t) + 2k\,\omega_0 \dot{q}(t) + \omega_0 q = Q(t)$$

where k is the damping factor and ω_0 is the natural frequency. If now the input to the system $Q(t)$ is sinusoidal $= x\cos\omega t$, the solution to the equation, or the output, is

$$q(t) = HQ(t - \varepsilon)$$

where ε is a phase angle and H is called the response amplitude operator (RAO)

$$\text{RAO} = \frac{1}{\{(\omega_0^2 - \omega^2)^2 + 4k^2\omega_0^2\}^{1/2}}$$
$$= \frac{\text{amplitude of output}}{\text{amplitude of input}}$$

Both Hand ε are functions of the damping factor k and the tuning factor w/wo·

This form of solution is a fairly general one and applies when the input is expressed in terms of wave height and the output is the bending moment amidships. The multiplier H is of course more complex but depends upon damping, wave frequency and ship shape, heave position and pitch angle. It is called the bending moment response amplitude operator and may be calculated for a range of wave frequencies (Fig. 6.31).

The calculation of the RAOs-and indeed the heave and pitch of the ship subject to a particular wave-is performed by standard computer programs using some applied mathematics which is known generally as strip theory.

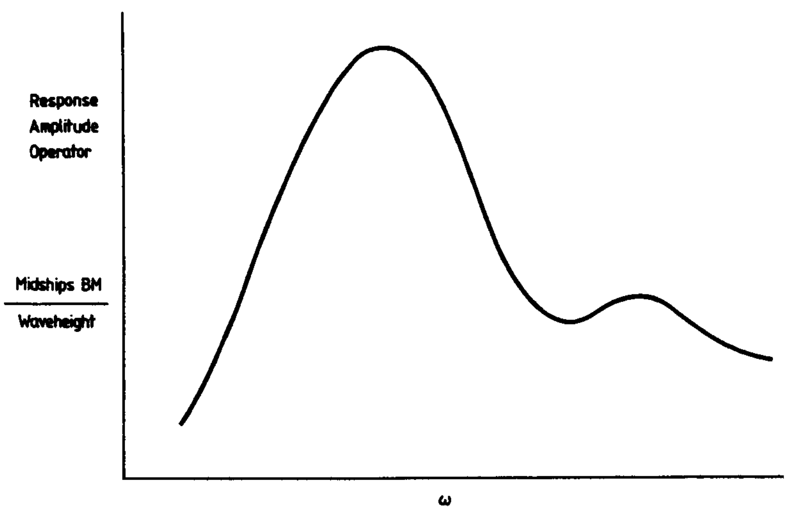


Fig. 6.31

One aspect of theoretical hydrodynamics is the study of the shape of the streamlines of a perfect fluid flowing in some constrained manner. The mathematics may hypothesize a source or a sink of energy in the fluid represented by a potential, like the broadcast of radio waves from a transmitter. A mathematical function may represent the size of the potential and streamlines, velocities of flow and pressures may be calculated, showing the effects of such sources and sinks upon the remaining fluid. This is part of potential flow theory.

An array of sources and sinks can be devised to represent the flow past a prismatic body even when there is an air/water interface. Flow past the body, the forces on the body, the consequent movement of the body and the elevation or depression of the interface can all be calculated. A ship can be represented by a series of short prismatic bodies or strips, all joined together and the total forces and bending moments upon such a hull calculated, the different elevations representing the self-generated waves. Waves imposed upon the ship may also be represented in the same way. The potential functions needed for this representation are pulsating ones and interference occurs between waves created and imposed. These are affected by the beam of the ship relative to the wave length of the self-generated waves and this has resulted in several different approaches to strip theory using slightly different but important assumptions. There are other important assumptions concerned with the boundaries. The general term strip theory embraces all such studies, including slender body theory and systematic perturbation analysis.

The mathematics is based upon several assumptions, the most important of which has been that the relationship of bending moment and wave height is linear, i.e. bending moment proportional to waveheight. Another effect of the assumptions is that the mathematics makes the ship wallsided. Despite these obviously incorrect assumptions, the results from strip theory are remarkably accurate and with steady refinements to the theory will improve further.

It is now necessary to combine the effect of all of those waves which constitute the sea as oceanographers have statistically described. For a short time ahead they have found that each maximum hog or sag bending moment can be well represented by a Rayleigh distribution

$$p(\text{BM}_{\max}) = \frac{\text{BM}_{\max}}{m_0}\exp\left(-\frac{\text{BM}_{\max}^2}{2m_0}\right)$$

where m_0 is the total energy in the ship response which is the mean square of the response

$$m_0 = \int_0^{\infty}\int_{-\pi}^{\pi} \text{RAO}^2\, S(\omega, \theta)\,\mathrm{d}\theta\,\mathrm{d}\omega$$

S being the total energy of the waves in the direction θ. The probability that $\text{BM}_{\max}$ will exceed a value B in any one cycle is given by

$$p(\text{BM}_{\max} > B) = \exp\left(-\frac{B^2}{2m_0}\right)$$

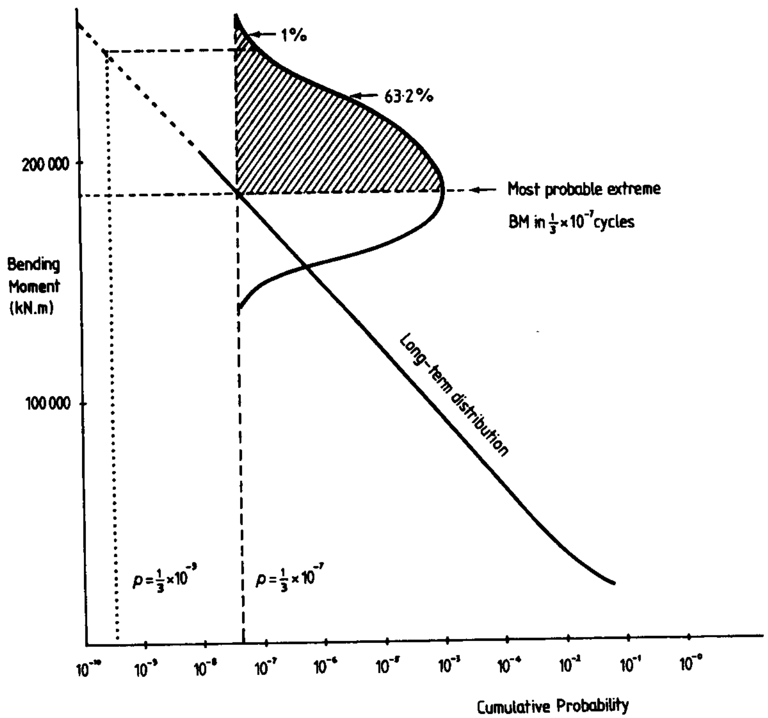


Fig. 6.32 Cumulative probability

or the probability of exceedence in n cycles

$$p(\mathrm{BM}_{\max} > B, n) = 1 - \{1 - \exp(-B^2/2m_0)\}^n$$

This illustrates the general approach to the problem. It has been necessary to define Sew, e), the spectrum of wave energy which the ship may meet in the succeeding hour or two. Over the entire life of the ship, it may be expected to meet every possible combination of wave height and frequency coming from every direction. Such long-term statistics are described by a two-parameter spectrum agreed by the ISSC which varies slightly for different regions of the world. The procedure for determining the probability of exceedence is a little more complicated than described above but the presentation of the results is similar.

Figure 6.32 shows such a result. At any given probability of exceedence the short time ahead is predicted by a Rayleigh or a Gaussian distribution based on the mean value of the sea characteristics pertaining.

Other approaches to the problem are possible. Some authorities, for example, use as a standard the likelihood that a particular bending moment will be exceeded during any period of one hour (or four hours) during the life of the ship. This has the advantage of being conveniently compared with measurement of statistical strain gauges installed in ships which record the maximum experienced every hour (or four hours).

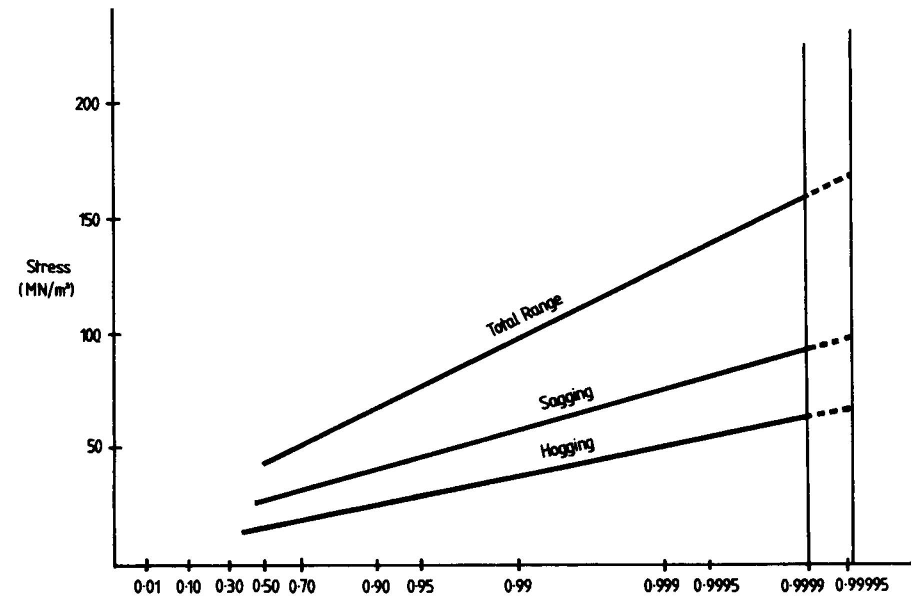


Fig. 6.33 Probability of non-exceedence in a 4-hour period

Figure 6.33 is such a plot on a log logscale which shows linear results that can be easily extrapolated.

We may now capitalize upon the study we made of the simple trochoidal wave. The effective wave height H_e is defined as that trochoidal wave of ship length which by the static standard wave calculation (without the Smith correction) gives the same wave bending moment as the worst that the ship would experience during its lifetime. That which appears to fit frigates very well is

$$H_e = 2.2\,L^{0.3} \text{ metres}$$

Figure 6.34 shows various effective wave height formulae compared with frigates at probability of once in a lifetime of 3×10^7 reversals and with merchant ships at 10^8.

Comparisons can show that the $L/20$ wave gives 50–70 per cent of the remote expectation for frigates while $0.607\sqrt{L}$ runs through the 10^8 spots, overestimating the load for very large ships. As a first estimate of required longitudinal strength therefore a designer may safely use the static standard strength calculation associated with a wave height of $L/9$ up to 50 m, $2.2\,L^{0.3}$ around 100 m and $0.607\sqrt{L}$ m up to about 300 m.

Let us leave the problems of determining the load for the time being and take a look at the ability of a ship to withstand longitudinal bending.

REALISTIC STRUCTURAL RESPONSE

The totally elastic response of the ship treated as if it were a simple beam has already been considered. Small variations to simple beam theory to account for

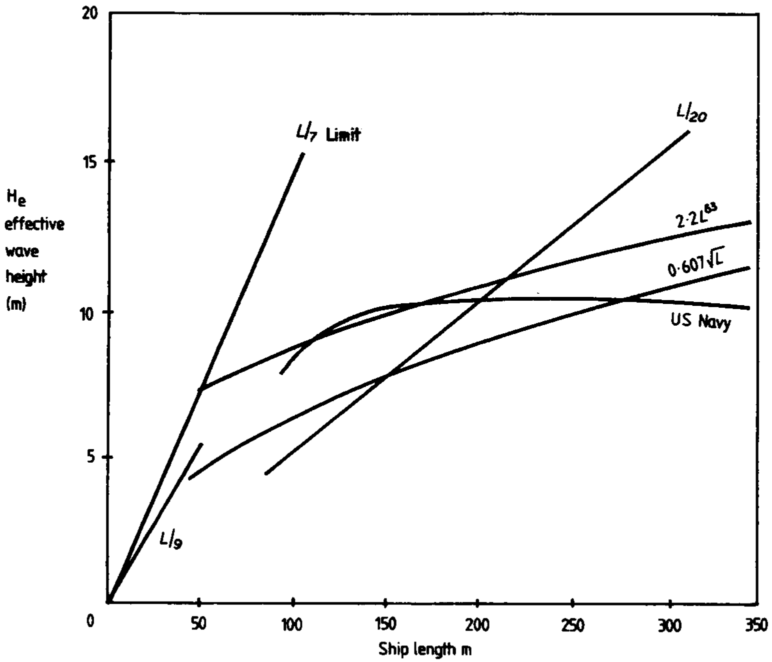


Fig. 6.34

shear lag were also touched upon. There are some other reasons why a hollow box girder like a ship may not behave like a simple beam. Strain locked in during manufacture, local yielding and local buckling will cause redistribution of the load-bearing capability of each part and its contribution to the overall cross section. If extreme loads are to be just contained, it is necessary to know the ultimate load-bearing ability of the cross section.

Let us first go to one extreme, and it is again necessary to draw upon some results of Chapter 7. It is there explained that the ultimate resistance to bending of a beam after the load has been increased to the point when the beam becomes totally plastic is

where S* is the sum of the first moments of area of cross section each side of the neutral axis. This surely is the ultimate strength of a perfect beam which is turned into a plastic hinge. Common sense tells us that such a state of affairs is most unlikely in the large hollow box girder which is the ship. While the tension side might conceivably become totally plastic, the compression side is likely to buckle long before that. This is taken into account in the concept of the ultimate longitudinal resistance to load which forms the basis on which

*This is a different S from that in the previous section.

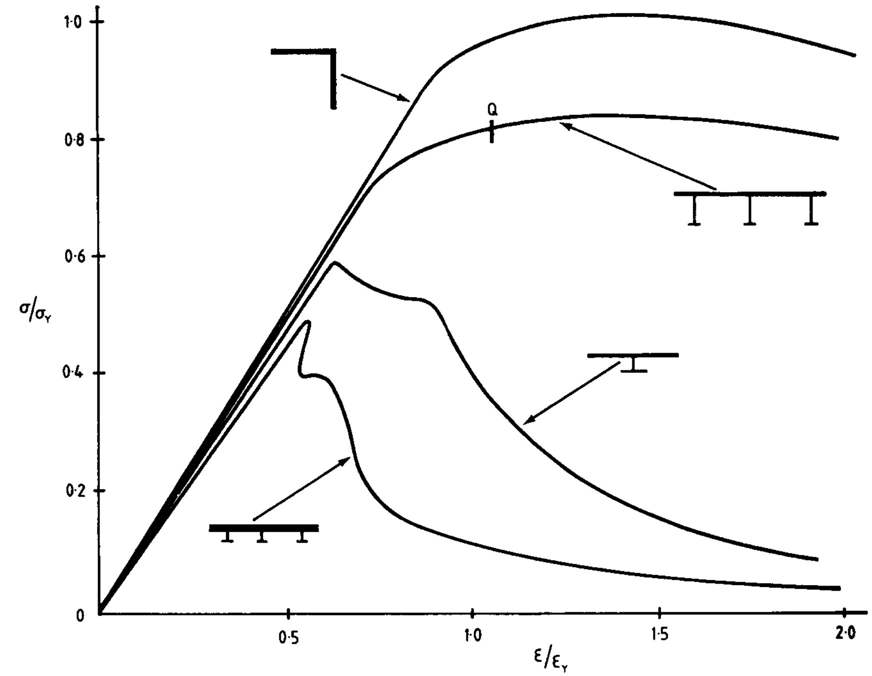


Fig. 6.35

ultimate strength is now assessed. The essence of the method is to build up the total resistance to bending from a summation of the contribution of each element. It is not difficult to imagine that as compressive load in a deck is increased there comes a time when the panels will buckle, shirk their load and throw an increased burden upon their adjacent longitudinal stiffeners. These in turn may buckle as the load is further increased, shirk their contribution and throw an extra burden on 'hard' areas like corners of decks and junctions with longitudinal bulkheads.

This is how a realistic assessment of cross sectional resistance should be made, a method which is now in general use. Every element which constitutes the effective cross section is first examined and a stress-strain curve is plotted. When the buckling behaviour of the element is embraced, the curves are called load shortening curves and are usually plotted in non-dimensional form as a family of curves with a range of initial assumed imperfections (Fig. 6.35).

The cross section of the ship girder is then assumed to bend with plane sections remaining plane to take up a radius of curvature R, the cross section rotating through an angle θ (Fig. 6.36). At any distance h above the neutral axis an element n of area A_n will be strained by an amount ε_n. Its load-bearing capacity can be picked off the relevant load shortening curve, say, at Q (Fig. 6.35). All such elements, so calculated, will lead to a bending resistance of the cross section

$$M = \sum_n \sigma_n A_n h_n$$

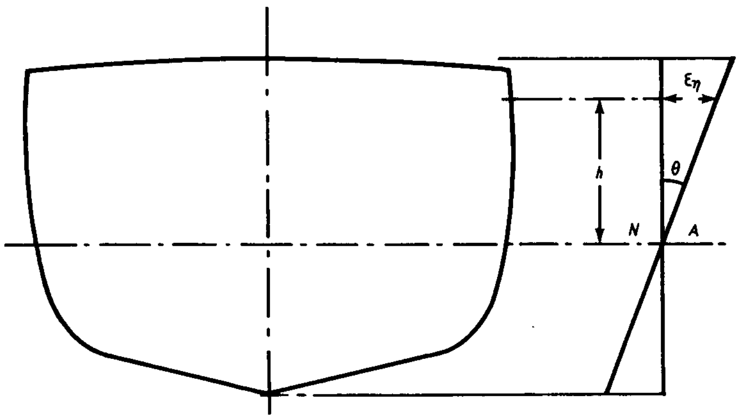


Fig. 6.36

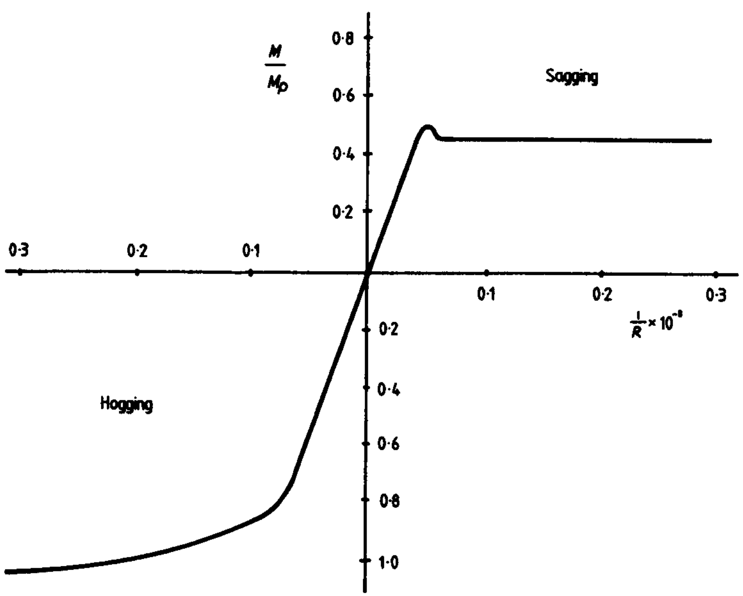


Fig. 6.37

Carried out for a range of values of () or R, a load shortening curve for the whole cross section can be calculated to provide a good indication of the collapse bending moment (Fig. 6.37). With the heavy structure of the bottom in compression when the ship hogs, the ultimate BM approaches the plastic moment M_p but a ratio of 0.6 in the sagging condition is not unusual.

In the calculation of M, the neutral axis does not stay still except when the behaviour is wholly elastic so that the additional condition for equilibrium of cross section must be imposed

$$\sum_n \sigma_n A_n = 0$$

For this reason, it is convenient to consider changes to () in increments whil.:h are successively summed and the process is called incremental analysis. Load shortening curves for the elements may embrace any of the various forms of buckling and indeed plastic behaviour whether it be assisted by locked-in manufacturing stresses or not.

While there are some obvious approximations to this method and some simplifications, it is undoubtedly the most realistic assessment of the collapse strength of the ship girder yet. The near horizontal part of Fig. 6.37 clearly represents the failure load, or collapse strength or ultimate strength, whatever it may be called.

ASSESSMENT OF STRUCTURAL SAFETY

At its simplest, failure will occur when the applied load exceeds the collapse load. It is first necessary to decide what probability of failure is acceptable, morally, socially or economically. History has shown that a comfortable level is a probability of 0.01 or 1 per cent that a single wave encounter will load the ship beyond its strength. With 30 million such encounters during a lifetime---every 10 seconds or so for 25 years-this represents a probability of 0.33×10^{-9}. Some argue that this is altogether too remote and the ship could be weaker. In fact, noting the logarithmic scale, there is not a great deal of difference in the applied load at these very remote occurrences (Fig. 6.32). Because the statistical method is still essentially comparative, all that is necessary is to decide upon one standard and use it to compare new designs with known successful practice or, rarely, failure. This is what all major authorities now do. It is worth taking a glimpse at further developments.

From the previous section, we have been able to declare a single value of the strength S which the loading must exceed for failure to occur. S may also be assumed to vary statistically because it may itself possess a variability. The strength, for example, will be affected by the thickness of plating rolled to within specified tolerances because that is what happens in practice. It will be dependent upon slightly different properties of the steel around the ship, upon ship production variations, upon defects, upon welding procedures and initial distortion which differs slightly. All such small variations cannot be precisel~ quantified at the design stage but the range over which they are likely can r Instead of a single figure S for the strength, a probability density function can be constructed. Figure 6.38 more properly shows the realistic shapes of loading and strength. For bending moment M, the shaded area under the load curve to the right of M represents the probability that the load will exceed M. The shaded area under the strength curve to the left of M represents the probability that the ship will not be strong enough to withstand M.

We are thus concerned with the shaded overlap at the bottom. If we are studying extreme loads whose probability of occurrence is remote we have to define with some precision the shapes of the very tiny tails of the distributions.

This direct approach to structural reliability is still in its infancy. Instead, most authorities match the two distributions to known mathematical shapes and allow the mathematics to take care of the tails. The most usual shapes to be

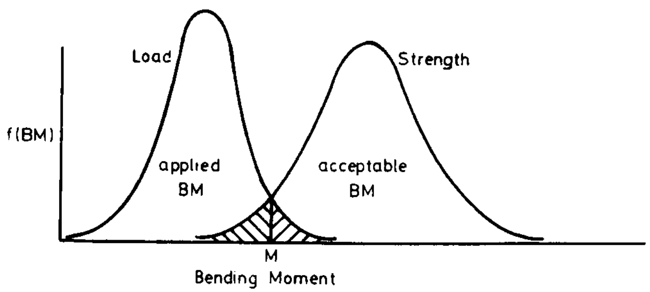


Fig. 6.38

assumed are Rayleigh or Gaussian which are defined by two properties only, the mean and the root mean square (or variance). This leads to a relationship between load P and strength S

$$S = \gamma_p \gamma_s P$$

where γ_p is the partial safety factor deduced from the mean and variance of the applied loading distribution and γ_s is the partial safety factor assessed for the mean and variance of the strength variability. A further partial safety factor is often added, γ_c, which assesses subjectively (i.e. as a matter of judgement) the gravity of failure. This partial safety factor approach is not yet in widespread use, except in building codes for civil engineering.

HYDROELASTIC ANAL YSIS

The ship in a seaway is an elastic body which enjoys bodily movements in all six directions and distorts also about and along all three axes. Its distortions affect the load applied by the sea and both structural and hydrodynamic damping affect the problem. The sea itself is a random process.

Until recently a reasonably complete mathematical description and solution of this formidable problem had never been achieved. It has now been possible to describe the dynamic behaviour of the ship in terms of modes of distortion which also embrace such solid body movements as pitch and heave, thus bringing together seakeeping and structural theories. Superposition of each element of behaviour in accordance with the excitation characteristics of the sea enables the total behaviour to be predicted, including even slamming and twisting of the hull.

The mathematics is wildly beyond the scope of this book and much of it has yet to be transformed into readily usable tools. Computer programs are available but require a great deal of data on mass and stiffness distribution which is not available until very late in the design.

This powerful analytical approach has been used to examine the overall stress distribution in large ships, showing that important problems emerge at sections of the ship other than amidships. Combination of shear force and bending moment cause principal stresses much higher than had been suspected previously. Areas of particular concern are those about 20 per cent of the ship's

length from the stern or from the bow, where slamming may further exacerbate matters. A lack of vigilance in the detail design or the production of the structure in these areas could, it has been suggested, have been responsible for some bulk carrier and VLCC fractures.

SLAMMING (see also Chapter 12)

One hydroelastic phenomenon which has been known for many years as slamming has now succumbed to theoretical treatment. When flat areas of plating, usually forward, are brought into violent contact with the water at a very acute angle, there is a loud bang and the ship shudders. The momentum of the ship receives a check and energy is imparted to the ship girder to make it vibrate. Strain records show that vibration occurs in the first mode of flexural vibration imposing a higher frequency variation upon the strain fluctuations due to wave motion. Amplitudes of strain are readily augmented by at least 30 per cent and sometimes much more, so that the phenomenon is an important one.

The designer can do a certain amount to avoid excessive slamming simply by looking at the lines 30-40 per cent of the length from the bow and also right aft to imagine where acute impact might occur. The seaman can also minimize slamming by changes of speed and direction relative to the wave fronts. In severe seas the ship must slow down.

Extreme values of bending moment acceptable by the methods described in this chapter already embrace the augmentation due to slamming. This is because the relationships established between full scale measurements and the theory adopted make such allowance.

Material considerations

A nail can be broken easily by notching it at the desired fracture point and bending it. The notch introduces a stress concentration which, if severe enough, will lead to a bending stress greater than the ultimate and the nail breaks on first bending. If several bends are needed failure is by fatigue, albeit, low cycle fatigue.

A stress concentration is a localized area in a structure at which the stress.~ significantly higher than in the surrounding material. It can convenientlye conceived as a disturbance or a discontinuity in the smooth flow of the lines of stress such as a stick placed in a fast flowing stream would cause in the water flow. There are two types of discontinuity causing stress concentrations in ships:

(a) discontinuities built into the ship unintentionally by the methods of construction, e.g. rolling, welding, casting, etc.;
(b) discontinuities deliberately introduced into the structural design for reasons of architecture, use, access, e.g. hatch openings, superstructures, door openings, etc.

Stress concentrations cannot be totally avoided either by good design or high standards of workmanship. Their effects, however, can be minimized by attention to both and it is important to recognize the effects of stress concentrations on the ship girder. Many ships and men were lost because these effects were not recognized in the early Liberty ships of the 1940s.

In general, stress concentrations may cause yield, brittle fracture or buckling. There is a certain amount of theory which can guide the designer, but a general understanding of how they arise is more important in their recognition and treatment because it is at the detail design stage that many can be avoided or minimized.

GEOMETRICAL DISCONTINUTIES

The classical mathematical theory of elasticity has produced certain results for holes and notches in laminae. The stress concentration factors at A and B of Fig. 6.39 of an elliptical hole in an infinite plate under uniform tension in the direction of the b-axis are given by

$$j_A = \frac{\text{stress}_A}{\sigma} = -1$$

$$j_B = \frac{\text{stress}_B}{\sigma} = 1 + \frac{2a}{b}$$

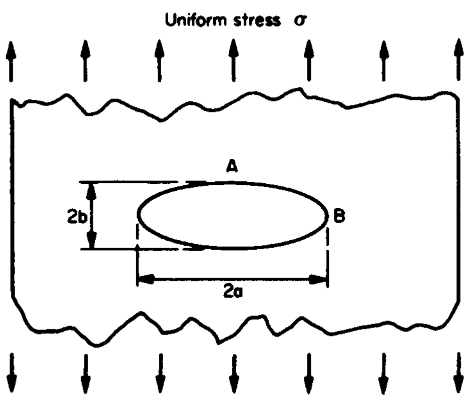


Fig. 6.39

For the particular case of a circular hole, $a =$ band $iB = 3$, i.e. the stress at the sides of a circular hole is three times the general tensile stress level in the plate, while at top and bottom there is a compressive stress equal to the general stress level. **If** a crack is thought of, ideally, as a long thin ellipse, the equation above gives some idea of the level of stress concentration at the ends; a crack twenty times as long as its width, for example, lying across the direction of loading would cause a stress, at the ends, forty-one times the general stress level and yielding or propagation of the crack is likely for very modest values ofa.

A square hole with radiused corners might be represented for this examination by two ellipses at right angles to each other and at 45 degrees to the

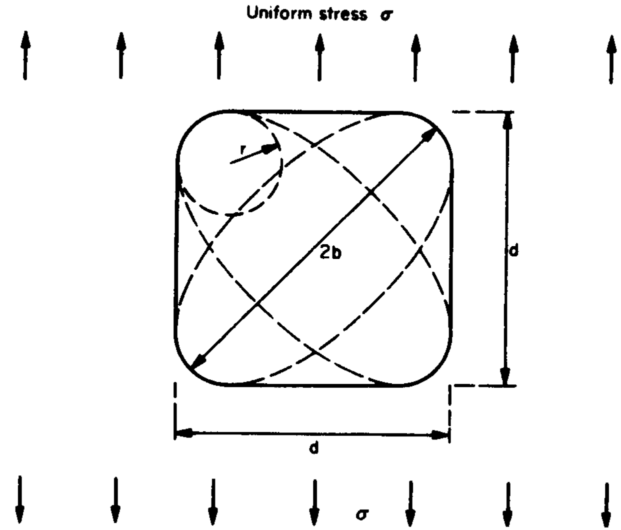


Fig. 6.40

direction of load. For the dimensions given in Fig. 6.40 the maximum stress concentration factor at the corners is given approximately by

$$j = \frac{1}{2}\sqrt{\frac{b}{r}}\left\{1 + \frac{\sqrt{(2b + 2r)}}{\sqrt{b} - \sqrt{r}}\right\}$$

Figure 6.41 shows the effect of the variation of corner radius r on length of side b for a square hole with a side parallel to the direction of stress and for a square hole at 45 degrees. The figure shows

(*a*) that there may be a penalty of up to 25 per cent in stress in failing to align a square hole with rounded corners with the direction of stress;
(*b*) that there is not much advantage in giving a corner radius greater than about one-sixth of the side;
(*c*) that the penalty of corner radii of less than about one-twentieth of the side is severe. Rim reinforcement to the hole can alleviate the situation.

These results are suitable for large hatches. With the dimensions as given in Fig. 6.42, the maximum stress concentration factor can be found with good accuracy from the expression,

$$j_{\max} = \frac{2 - 0.4^{B/b}}{2 - 0.4^{l/B}}\left\{1 + \frac{0.926}{1.348 - 0.826^{20r/B}}\left[0.577 - \left(\frac{b}{B} - 0.24\right)^2\right]\right\}$$

It is of importance that the maximum stress occurs always about 5–10 degrees around the corner and the zero stress 50–70 degrees round. Butts in plating should be made at this latter point. Figure 6.43 shows the results for a hole with $l = B$. Note that the concentration factor is referred not to the stress in the clear plate but to the stress at the reduced section.

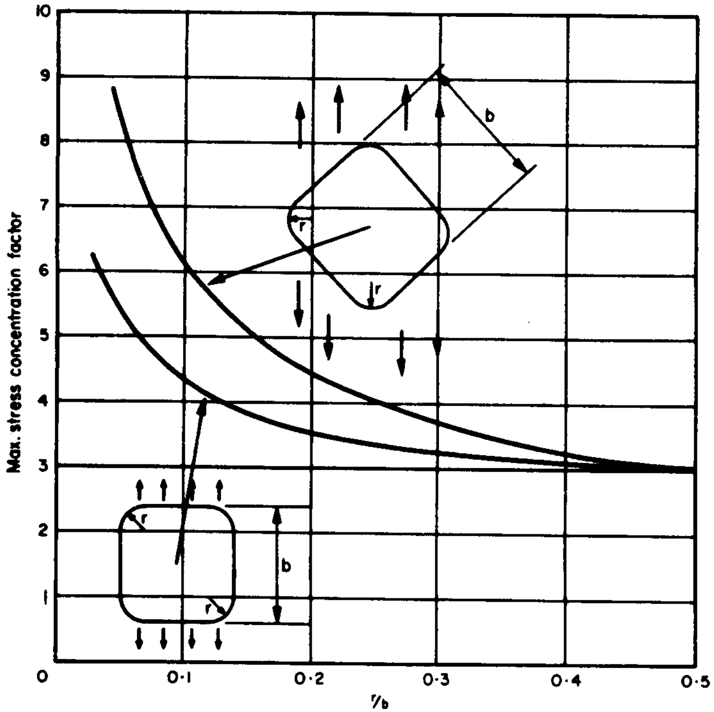


Fig. 6.41

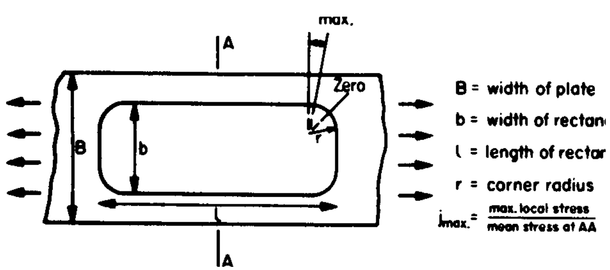


Fig. 6.42 *Rectangular hole in finite plate*

BUILT-IN STRESS CONCENTRA nONS

The violent treatment afforded a plate of mild steel during its manufacture, prevents the formation of a totally unstrained plate. Uneven rolling or contraction, especially if cooling is rapid, may cause areas of strain, even before the plate is selected for working into a ship. These are often called 'built in' or 'locked up' stresses (more accurately, strains). Furthermore, during the processes of moulding and welding, more strains are built in by uneven cooling. However careful the welders, there will be some, perhaps minute, holes, cracks and slag inclusions, lack of penetration and undercutting in a weld deposit.

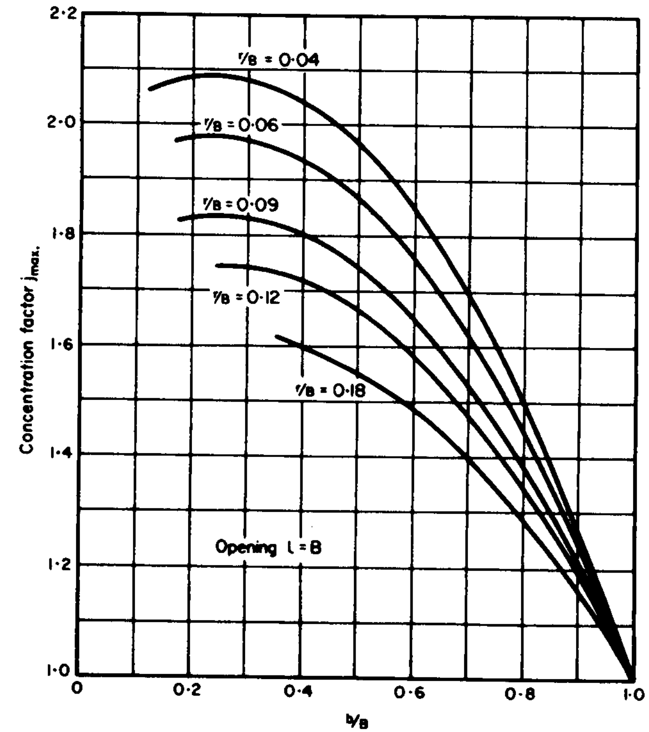


Fig. 6.43 Stress concentration factor for a hole with $l = B$

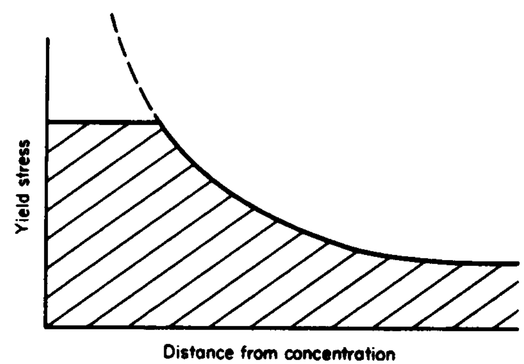


Fig. 6.44 *Yielding out*

Inspection of important parts of the structure will minimize the number of visual defects occurring, and radiography can show up those below the surface, but lack of homogeneity cannot be observed by normal inspection procedures, even though the 'built-in' stress may exceed yield. What then happens to them when the ship is subjected to strain?

In ductile materials, most of the concentrations 'yield out', i.e. the concentration reaches the yield point, shirks further load and causes are-distribution of stress in the surrounding material. If the concentration is a crack, it may propagate to an area of reduced stress level and stop. If it becomes visible, a temporary repair is often made at sea by drilling a circular hole at its end, reducing the concentration factor. This is a common first aid treatment. There is considerably more anxiety if the material is not so ductile as mild steel, since it does not have so much capacity for 'yielding out'; furthermore, a high yield steel is often employed in places where a general high stress level is expected so that cracks are less able to propagate to areas of reduced stress level and stop. A further anxiety in all materials is the possibility of fatigue, since concentrations at which there is, locally, a high stress level will be able to withstand few reversals.

CRACK EXTENSION, BRITTLE FRACTURE

Cracks then, cannot be prevented but can be minimized. It is important that they are observed and rectified before they cause catastrophic failure. They can extend under the action of fatigue or due to brittle fracture. Even in heavy storms fatigue cracks are only likely to increase in length at a rate measured in mm/s. A brittle fracture, however, can propogate at around 500 m/s. Thus brittle fracture is of much greater concern. The loss of Liberty ships has already been mentioned. More recent examples have been the MV *Kurdistan* which broke in two in 1979 and the MV *Tyne Bridge* which experienced a 4 m long crack in 1982. Some RN frigates damaged in collision in the 'Cod War' in the 1970s exhibited brittle fracture showing that thin plates are not necessarily exempt from this type of failure as had been generally thought up to that time.

The critical factors in determining whether brittle fracture will occur are stress level, length of crack and material toughness, this last being dependent upon temperature and strain rate. The stress level includes the effects of stress concentrations and residual stresses due to the fabrication processes. The latter are difficult to establish but, as an illustration, in frigates a compressive stress of about 50 MPa is introduced in hull plating by welding the longitudinals, balanced by local regions in the vicinity of the weld where tensile stresses are at yield point.

At low temperatures fracture of structural steels and welds is by cleavage. Once the threshold toughness for crack initiation is exceeded, the energy required for crack extension is so low that it can be provided by the release of stored elastic energy in the system. Unless fracture initiation is avoided structural failure is catastrophic. At higher temperatures fracture initiation is by growth and coalescence of voids. Subsequent crack extension is sustained only by increased load or displacement. The temperature marking the transition in fracture mode is termed the transition temperature. This temperature is a function of loading rate, structural thickness, notch acuity and material microstructure.

Ideally one would like a simple test that would show whether a steel would behave in a 'notch ductile' manner at a given temperature and stress level. This does not exist because the behaviour of the steel depends upon the geometry and method of loading. For instance, cleavage fracture is favoured by high

triaxial stresses and these are promoted by increasing plate thickness. The choice then, is between a simple test like the Charpy test (used extensively in quality control) or a more expensive test which attempts to create more representative conditions (e.g. the Wells Wide Plate or Robertson Crack Arrest tests). More recently the development of linear elastic fracture mechanics based on stress intensity factor, K, has been followed by usable elastic-plastic methodologies based on crack tip opening displacement. CTOD or δ, and the J contour integral, has in principle made it possible to combine the virtues of both types of test in one procedure.

For a through thickness crack of length $2a$ subject to an area of uniform stress, σ, remote from stress concentration the elastic stress intensity factor is given by

$$K = \sigma(\pi a)^{1/2}$$

The value at which fracture occurs is K_c and it has been proposed that a $K_c = 125\,\text{MPa(m)}^{1/2}$ would provide a high assurance that brittle fracture initiation could be avoided. A fracture parameter, J_c, can be viewed as extending K_c into the elastic–plastic regime, with results presented in terms of K_{Jc} which has the same units as K_c. Approximate equivalents are

$$K_{J_c} = [J_c E]^{1/2} = [2\delta_c \sigma_Y E]^{1/2}$$

It would be unwise to assume that cracks will never be initiated in a steel structure. For example, a running crack may emerge from a weld or heat affected zone unless the crack initiation toughness of the weld procedures meets that of the parent plate. It is prudent, therefore, to use steels which have the ability to arrest cracks. It is recommended that a crack arrest toughness of the material of between 150 and 200 MPa $(m)^{1/2}$ provides a level of crack arrest performance to cover most situations of interest in ship structures.

Recommendations are:

(*a*) To provide a high level of assurance that brittle fracture will not initiate, a steel with a Charpy crystallinity less than 70 per cent at 0 °C be chosen.
(*b*) To provide a high level of crack arrest capability together with virtually guaranteed fracture initiation avoidance, a steel with Charpy crystallinity less than 50 per cent at 0 °C be chosen.
(*c*) For crack arrest strakes a steel with 100 per cent fibrous Charpy fracture appearance at 0 °C be chosen.

If the ship is to operate in ice, or must be capable of withstanding shock or collision without excessive damage, steels with higher toughness would be appropriate.

FATIGUE

Provided ships are inspected regularly for cracking, the relatively slow rate of fatigue crack growth means that fatigue is not a cause for major concern in relation to ship safety. If, however, cracks go undetected their rate of growth will increase as they become larger and they may reach a size that triggers brittle fracture. Also water entering, or oil leaking, through cracks can cause problems and repair can be costly. Fatigue, then, is of concern particularly as most cracks occurring in ship structures are likely to be fatigue related. It is also important to remember that fatigue behaviour is not significantly affected by the yield strength of the steel. The introduction of higher strength steels and acceptance of higher nominal stress levels (besides the greater difficulty of welding these steels) means that fatigue may become more prevalent. Thus it is important that fatigue is taken into account in design as far as is possible.

Design for fatigue is not easy-some would say impossible. However, there are certain steps a designer can, and should, take. Experience, and considerable testing, show that incorrect design of detail is the main cause of cracking. The situation may be summarized by saying that design for fatigue is a matter of detail design and especially a matter of design of welded connections. Methods used rely very heavily on experimental data. The most common to date has been one using the concept of a nominal stress. Typically for steel the fatigue characteristics are given by a log/log plot of stress range against number of cycles to failure. This *S-N* curve as it is termed takes the form of a straight line with life increasing with decreasing stress range until a value below which the metal does not fatigue. As a complication there is some evidence that in a corrosive atmosphere there is no lower limit. However, in laboratories, tests of welded joints lead to a series of S - *N* lines of common slope. The various joints are classified by number, the number being the stress range (N/mm2) at 20 million cycles based on a mean test value less two standard deviations which corresponds to a survival probability of 97.7 per cent. As an example, a cruciform joint, K butt weld with fillet welded ends is in Class 71 (Fig. 6.45).

These data relate to constant amplitude loading and they are not too sensitive to mean stress level. However, a ship at sea experiences a varying load depending upon the conditions of sea and loading under which it operates. This is usually thought of in terms of a spectrum of loading and a transfer factor must be used to relate the stress range under spectrum loading to the data for constant amplitude. Testing at Hamburg suggests that a transfer factor of 4 is appropriate for the range of notch cases existing in ship structures, assuming 20 million cycles as typical of the average merchant ship life. It also recommends a safety factor of 4/3. For a Class 71 detail this gives a permissible stress range of 71 x 4 x 3/4 = 213. This must be checked against the figures derived from the longitudinal strength calculation.

As is discussed in the next chapter, many structures are now analysed by finite element methods. These are capable of analysing local detail on which fatigue strength depends but interpretation of the results is made difficult by the influence of the mesh size used. The smaller the mesh, and the closer one approaches a discontinuity, the higher the stress calculated. The usual 'engineering' solution is to use a relatively coarse mesh and compare the results with the figures accepted in the nominal stress approach described above. The best idea of acceptable mesh size is obtained by testing details which have been analysed by finite element methods and comparing the data for varying mesh size.

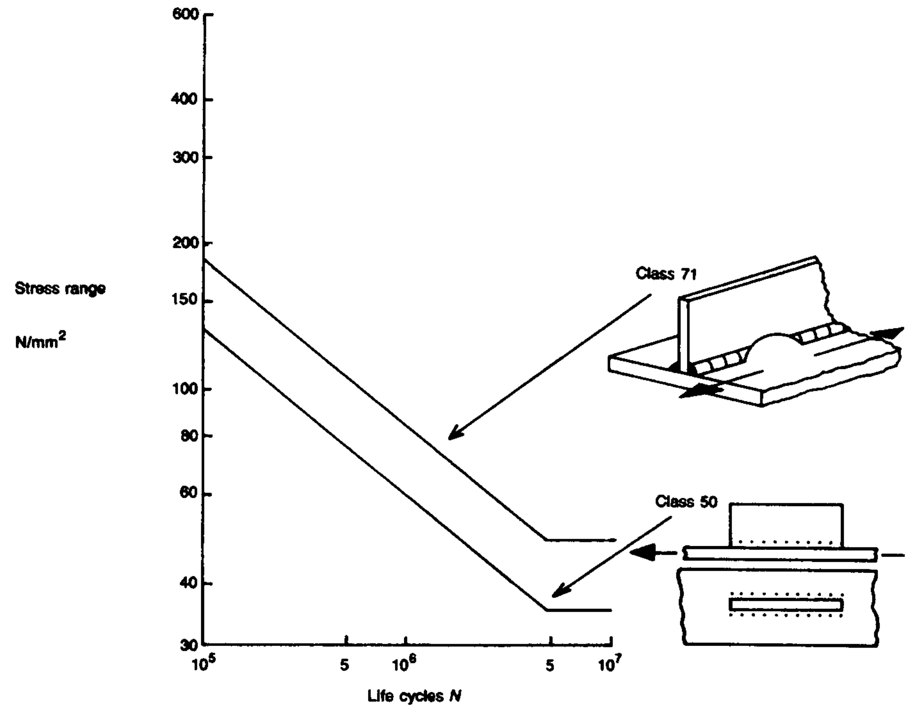


Fig. 6.45 *S-N curves (based on Hamburg work)*

Other methods are available but there is not space in a volume such as this to describe them.

DISCONTINUITIES IN STRUCTURAL DESIGN

In the second category of stress concentrations, are those deliberately introduced by the designer. Theory may assist where practical cases approximate to the assumptions made, as, for example, in the case of a side light or port hole. Here, in a large plate, stress concentration factors of about three may *JtIt!* expected. At hatches too, the effects of corner radii may be judged from Figs. 6.41 and 6.43. Finite element techniques outlined in Chapter 7 have extended the degree to which theory may be relied upon to predict the effect of large openings and other major stress concentrations.

Often, the concentrations will be judged unacceptably high and reinforcement must be fitted to reduce the values. One of the most effective ways to do this is by fitting a rim to the hole or curved edge. A thicker insert plate may also help, but the fitting of a doubling plate is unlikely to be effective unless means of creating a good connection are devised.

Designers and practitioners at all levels must be constantly on the watch for the discontinuity, the rapid change of structural pattern and other forms of stress raiser. It is so often bad local design which starts the major failure. Above all, the superposition of one concentration on another must be avoided-but the reinforcement rim where the concentration is lowest, avoid stud welds and fittings at structural discontinuities, give adequate room between holes, grind the profile smooth at the concentration!

SUPERSTRUCTURES AND DECKHOUSES

These can constitute severe discontinuities in the ship girder. They can contribute to the longitudinal strength but are unlikely to be fully effective. Their effectiveness can be improved by making them long, minimizing changes in plan and profile, extending them the full width of the hull and paying careful attention to their connections to the hull.

The only contact between the upper deck and the superstructure is along the bottom of the superstructure sides through which the strains and forces must be transmitted. Because the upper deck stretches, so also must the lower edge of the superstructure sides thus causing shear forces which tend to distort the superstructure into a shape opposite to that of the hull (Fig. 6.46). The two are held together, however, and there must be normal forces on the superstructure having the opposite effect. The degree of these normal forces must depend upon the flexibility of the deck beams and main transverse bulkheads if the superstructure is set in from the ship's sides. The effect of the shear forces will depend on the manner in which the shear is diffused into the superstructure, and the shear lag effects are likely to be more appreciable there than in the main hull girder.

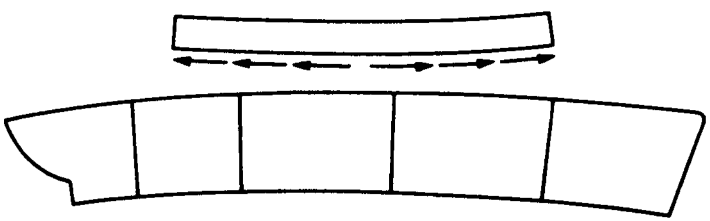


Fig. 6.46

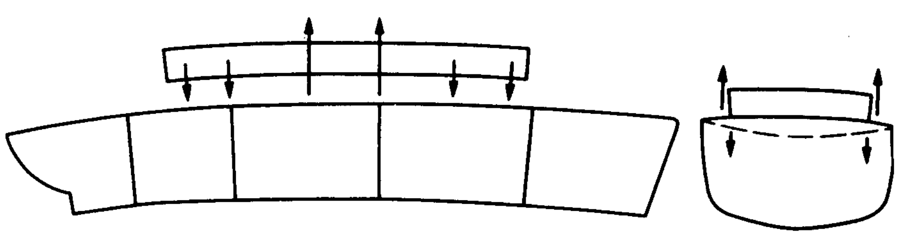


Fig. 6.47

If the effects of shear lag are ignored, the results are suitable for the middle portions of long superstructures. This approach embraces the effects of shear diffusion but ignores the concentrated forces from main transverse bulkheads, and it is suitable for short superstructures or for those which extend out to the ship's side. An efficiency of superstructure is defined as

$$\eta = \frac{\sigma_0 - \sigma}{\sigma_0 - \sigma_1}$$

where σ_0 is the upper deck stress which would occur if there were no superstructure present, σ is the upper deck stress calculated and σ_1 is the upper deck stress with a fully effective superstructure. Curves are supplied from which the factors leading to the efficiency may be calculated. As might be expected, the efficiency depends much on the ratio of superstructure length to its transverse dimensions.

The square ends of the superstructure constitute major discontinuities, and may be expected to cause large stress concentrations. They must not be superimposed on other stress concentrations and should be avoided amidships or, if unavoidable, carefully reinforced. For this reason, expansion joints are to be used with caution; while they relieve the superstructure of some of its stress by reducing its efficiency, they introduce stress concentrations which may more than restore the stress level locally. To assist the normal forces, superstructure ends should coincide with main transverse bulkheads.

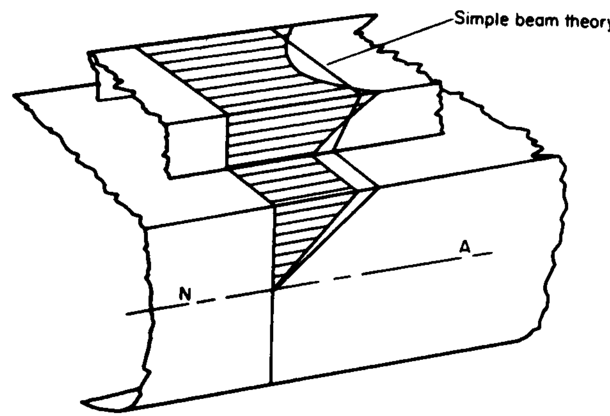


Fig. 6.48 Direct stress in a superstructure

Unwanted hull-superstructure interactions can be avoided by using low modulus material in the superstructure such as reinforced plastic which offers tensile and compressive strengths comparable to the yield strength of mild steel with an E-value less than a tenth that of steel. In this case the superstructure will not make any significant contribution to longitudinal strength.

Conclusions

The student may be forgiven if he began this chapter safely assured by the static calculation (sometimes called Reed's method) and ended it aware of some considerable uncertainty. As the proceedings of the International Ship Structures Congress bear witness, this is precisely the current situation. Naval architects are rapidly adopting the statistical approach to ship girder loading and are preparing to marry it to structural reliability when sufficient information about strength variability is to hand. They continue to use the results of the long process of data collection at sea to refine their standards of acceptability.

Problems

1. A landing craft of length 61 m may be assumed to have a rectangular cross-section of beam of 8.53 m. When empty, draughts are 0.46 m forward and 1.07 m aft and its weight is made up of general structure weight assumed evenly distributed throughout the length, and machinery and superstructure weight spread evenly over the last 12.2 m. A load of 101.6 tonnef can be carried evenly distributed over the first 48.8 m, but to keep the forward draught at a maximum of 0.61 m, ballast water must be added evenly to the after 12.2 m.

   Draw the load, shearing force and bending moment diagrams for the loaded craft when in still water of specific volume 0.975 m^3/tonnef. Determine the position and value of the maximum bending moment acting on the craft.
2. State the maximum bending moment and shearing force in terms of the weight and length of a vessel having the weight uniformly distributed and the curve of buoyancy parabolic and quote deck and keel moduli. Where do the maximum shearing force and bending moment occur?
3. The barge shown in the figure floats at a uniform draught of 1 m in sea water when empty.

   A heavy weight, uniformly distributed over the middle 5 m of the barge, increases the draught to 2 m. It may be assumed that the buoyancy curves for the barge (loaded and unloaded) and the weight distribution of the unloaded

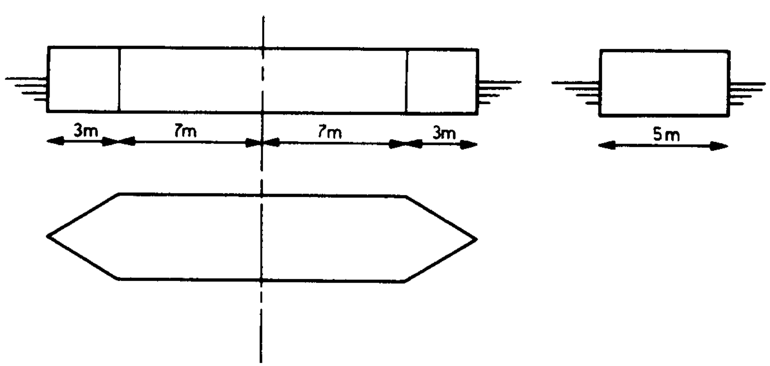


Fig. 6.49

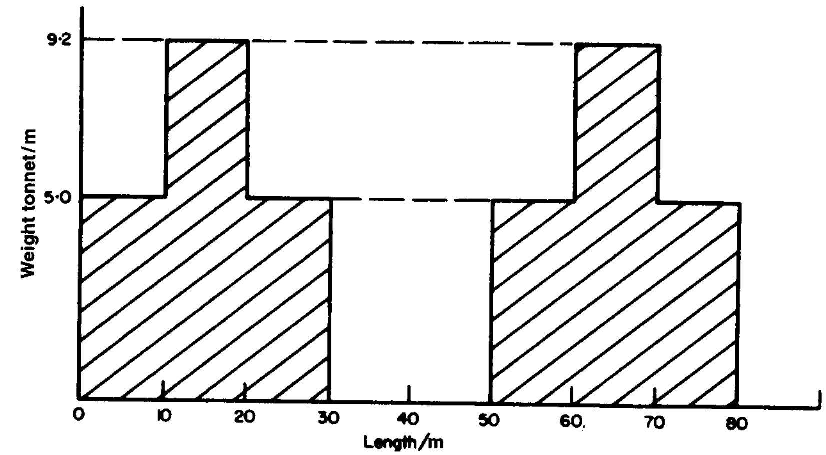


Fig. 6.50

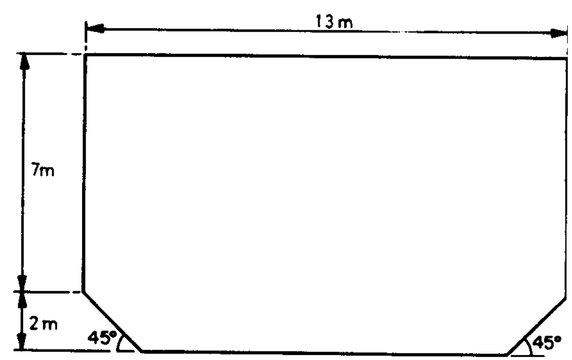


Fig. 6.51

barge are constant over the parallel length of the barge, decreasing linearly to zero at the two ends.

   Draw curves of buoyancy, weight, loading, shear force and bending moment for the barge loaded and at rest in salt water.
4. A rectangular barge is 80m long and has a beam of 14m. The weight distribution of the partially loaded barge is shown in Fig. 6.50. The barge is floating at rest in still water. Draw curves of loading, shearing force and bending moment, stating the maximum shearing force and bending moment and the positions where they act.
5. A barge, 70m long, has the cross-section shown in Fig. 6.51. The empty weight is 420 tonnef evenly distributed over the whole length, whereas the load is spread evenly over the middle 42 m of length. Draw the curves of load, shearing force and bending moment for the barge, when it is loaded

and balanced on a sea wave whose profile gives draughts, symmetrical about amidships, as follows:

Section	FE	2	3	4	5	Amidships
Draught (m)	5.00	4.55	3.61	2.60	2.10	2.00

6. A steel barge, shown in Fig. 6.52, of constant rectangular section, length 72 m, floats at a draught of 5 m when loaded. The weight curve of the loaded barge may be regarded as linear, from zero at the two ends to a maximum at the mid-length. The structural section is shown below. If the stress in the deck in still fresh water is not to exceed 123 MPa, estimate the thickness of the plating if this is assumed to be constant throughout.
7. In a calculation of the longitudinal strength for the sagging condition, the following mean ordinates in tonnef/m were found for sectional lengths of a ship each 12 m long, starting from forward:

Section	1	2	3	4	5	6	7	8	9	10	11
Weight	8.3	12.6	24.2	48.2	66.2	70.0	65.1	40.7	23.3	13.0	6.0
Buoyancy	24.8	40.6	39.2	33.6	28.2	30.0	39.6	48.7	47.4	36.0	9.5

Draw the shearing force and bending moment diagrams and state the positions and values of the maxima.

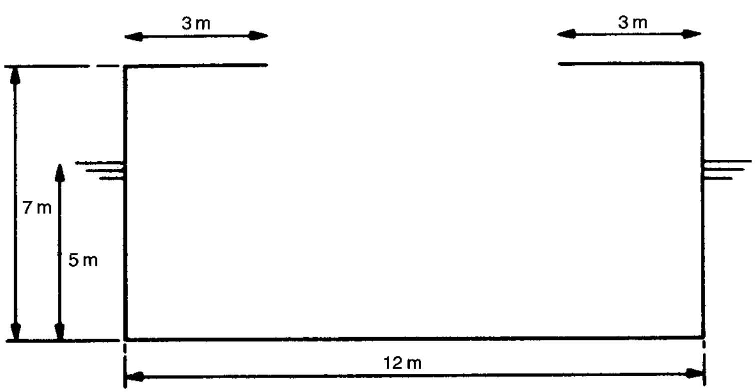


Fig. 6.52

8. A dumb lighter is completely wall-sided, 100m long, 20 m maximum beam and it floats at a draught of 6 m in still water. The waterline is parabolic and symmetrical about amidships with zero ordinates at the ends. The weight is evenly distributed along its length.

   If the vessel is 15m deep, has a sectional second mt. = 30 m^4 and has a neutral axis 6 m from the keel, find the maximum hogging stress when balanced on an *L/20* wave.

The coordinates of a trochoid are:

$\frac{x}{L/20}$	0	1	2	3	4	5	6	7	8	9	10
$\frac{z}{L/20}$	0	0.018	0.073	0.161	0.280	0.423	0.579	0.734	0.872	0.964	1.0

9. The following data apply to the half-section modulus of a ship girder 13.7 m deep with an assumed neutral axis 6.1 m above the keel:

	Area (cm^2)	1st moment about ANA (m^3)	2nd moment about ANA (m^4)
Above ANA	6,000	3.15	19.88
Below ANA	6,916	2.91	14.59

   Find the stresses in the keel and upper deck if the bending moment is 61,935 tonnef/m.

10. The data given below applies to a new frigate design. Draw the curves of loading, shearing force and bending moment for the frigate and calculate the maximum stresses in the keel and deck given that:

$$\text{2nd moment of area of section about NA} = 5.54\,\text{m}^4$$
$$\text{total depth} = 10.52\,\text{m}$$
$$\text{NA above keel} = 3.96\,\text{m}$$
$$\text{length} = 110\,\text{m}$$

Ordinate	1	2	3	4	5	6	7	8	9	10	11
Buoyancy (tonnef/m)	3.0	26.83	33.87	30.87	22.67	19.33	20.50	24.87	29.70	29.63	21.07
Weight (tonnef/m)	12.67	18.67	19.33	39.67	34.00	33.00	34.67	33.00	17.67	12.00	

11. The particulars of a structural strength section 13 m deep, taken about mid-depth, are as follows:

	Area (m^2)	1st moment (m^3)	2nd moment (m^4)
Above mid-depth	0.33	1.82	12.75
Below mid-depth	0.41	2.16	14.35

   On calculation, the maximum stress in the upper deck, for the hogging condition, is found to be 122 N/mm^2. Determine how many longitudinal girders must be welded to the upper deck to reduce the stress to 100 N/mm^2 under the same loading conditions, given that

   (*a*) area of cross-section of each girder = 0.002 m^2
   (*b*) c.g. of each girder can be considered to be at the height of the upper deck.
   (*c*) second moment of each girder about its own c.g. is negligible.

12. Calculate the deck and keel moduli of the structural section given below. Calculate the maximum stresses due to a sagging bending moment of 11,576 tonnef/m.

The centre of area of each girder and longitudinal may be assumed to lie in the plating to which it is attached. Second moments of girders about their own c.gs are negligible.

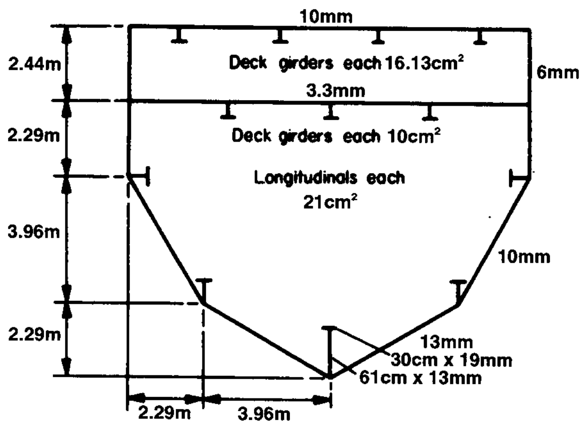


Fig. 6.53

13. The forecastle of a destroyer is to be extended in aluminium alloy, over the midships area by the structure shown below. Details of the existing steel structural midship section are as follows:

Depth, keel to upper deck	6.1 m
Neutral axis below upper deck	3.84 m
Total area of section	4839 cm^2
Second moment of area about NA	4.85 m^4
Calculated upper deck hogging stress	106 MPa

Calculate the stress in the deck of the alloy extension and the new keel stress for the same bending moment.

The value of E for alloy should be taken as $\frac{1}{3}$ that for steel

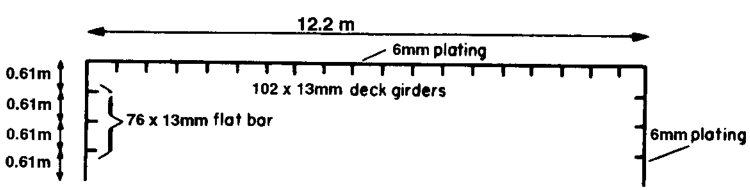


Fig. 6.54

14. An aluminium superstructure made up of a deck 10 m wide supported on two sides 3 m deep, is to be included for strength purposes with the steel structural section whose particulars are as follows:

total area of section	0.545 m^2
second moment of area	10.29 m^4

depth of section	14 m
height of NA above keel	7.20 m

Since the design stresses are already large, it is proposed to replace the steel upper deck by one of aluminium of the same cross-sectional area of 0.09 m^2.

Making reasonable first assumptions and assuming that stiffeners can be taken with the plating, determine the area/metre run of the aluminium superstructure deck and sides (assumed constant for deck and sides), to maintain the same keel stress as before.

Take E for aluminium to be $\frac{1}{3}$ that for steel.

15. A rectangular vessel is 15 m broad and 10 m deep and has deck plating 5 mm thick and sides and bottom 9 mm thick. At a certain section, it is subjected to a hogging bending moment of 65 MN m and a shearing force of 3.15 MN. Calculate, in accordance with classical beam theory,

(*a*) Maximum tensile and compressive stresses,
(*b*) Maximum shearing stress.

16. If, in the previous example, the deck and bottom were 9 mm thick and the sides 5 mm thick, what would then be the stresses?

17. A barge has a rectangular waterline 61 m long with a beam of 5.34 m. When it is floating in calm sea water, the position of the maximum longitudinal bending moment is 6.1 m abaft amidships.

Deduce, from first principles, the influence lines, showing the effect on the maximum still water BM of the addition of a unit concentrated weight at any position along the length of the barge. What is the effect of an addition (*a*) at the extreme after end, (*b*) at the point of maximum BM?

Check that the effect of a load uniformly distributed along the length is zero.

8. The particulars of a structural midship section, 12.2 m deep, taken about mid-depth, are as follows:

	Area (cm^2)	Moment (m^3)	2nd moment (m^4)
above mid-depth	3548	1.77	10.5
below mid-depth	4194	2.24	13.2

Displacement increases would cause deck stresses of 124MPa to arise in the standard sagging condition.

It is proposed to stiffen up the section by the addition of ten girders to the upper deck and five similar longitudinals to the flat bottom. If these stiffeners are to be cut down versions of T bars having a cross-sectional area of 29 cm^2, determine the actual cross-sectional area required to reduce deck design stress to 116 MPa.

19. A barge is 60 m long and completely symmetrical about amidships. It is constructed of steel.

The following table gives the mean net loading between ordinates and the second moments of area of sections at regular intervals between the forward perpendicular and amidships:

	Amidships										FP
Ordinate	6		5		4		3		2		1
Load (tonnef/m)		−19		−8		7		15		5	
Second moment of area (m^4)	1.77		1.77		1.77		1.60		1.28		0.42

Estimate the breakage due to this loading, which is symmetrical about amidships. $E = 209\,GN/m^2$.

20. The following table gives the loading diagram which was obtained after balancing a ship of length 213 m in the standard longitudinal strength calculation:

	AE		10		9		8		7		6		5		4		3		2		FP
Load (tonnef. m)		30		57		60		−60		−70		−90		−60		13		80		40	
$I/(m^4)$	3.6		4.5		5.4		8.1		9.0		9.0		8.1		5.4		4.5		3.6		2.7

The neutral axis is 7.32 m below the strength deck. Find the stress in this deck amidships.

The table also gives the MI of section at the ordinates. Estimate the relative angular movement between a missile launcher at 3 ordinate and a guidance radar amidships ($E = 207$ GPa).

21. A ship of length 105 m is 9 m deep. It is moored in the effluent of a power station on a cold night, which results in the keel increasing in temperature by 15° Celsius and the upper deck falling 15° Celsius.

Estimate the breakage resulting. The coefficient of linear expansion of steel is 0.000012 in Celsius units.

7 Structural design and analysis

It is not possible within the confines of one chapter-or even a book of this size-to present the naval architect with all that is needed to design and analyse a ship's structure. The object of this chapter is to provide an understanding of the structural behaviour of the ship and a recognition of the unit problems. Applied mechanics and mathematics will have provided the tools; now they must be applied to specific problems. The scope and the limitations of different theories must be known if they are to be used with success and the student needs to be aware of the various works of reference. Recognition of the problem and knowledge of the existence of a theory suitable for its solution are exceedingly valuable to the practising engineer. There is rarely time to indulge an advanced and elegant theory when a simple approach provides an answer giving an accuracy compatible with the loading or the need. On the other hand, if a simple approach is inadequate, it must be discarded. It is therefore understanding and recognition which this chapter seeks to provide.

Optimum design is often assumed to mean the minimum weight structure capable of performing the required service. While weight is always significant, cost, ease of fabrication and ease of maintenance are also important. Cost can increase rapidly if non-standard sections or special quality materials are used; fabrication is more difficult with some materials and, again, machining is expensive. This chapter discusses methods for assessing the minimum requirements to provide against failure. The actual structure decided upon must reflect all aspects of the problem.

The whole ship girder provides the background and the boundaries for local structural design. The needs of the hull girder for areas at deck, keel and sides must be met. Its breakdown into plating and stiffener must be determined; there is also much structure required which is not associated with longitudin~ strength; finally, there are many particular fittings which require individ~ design. An essential preliminary to the analysis of any structure or fitting, however, is the assessment of the loading and criterion of failure.

LOADING AND FAILURE

Because it is partly the sea which causes the loading, some of the difficulties arising in Chapter 6 in defining the loading apply here also. The sea imposes on areas of the ship impact loads which have not yet been extensively measured, although the compilation of a statistical distribution of such loading continues. While more realistic information is steadily coming to hand, a loading which is likely to provide a suitable basis for comparison must be decided upon and used to compare the behaviour of previous successful and unsuccessful elements. **For** example, in designing a panel of plating in the outer bottom, hydrostatic pressure due to draught might provide a suitable basis of comparison, and examination of previous ships might indicate that when the ratio of permanent set to thickness exceeded a certain percentage of the breadth to thickness ratio, extensive cracking occurred.

Some loads to which parts of the structure are subject are known with some accuracy. Test water pressure applied during building (to tanks for example), often provides maximum loads to which the structure is subject during its life; bulkheads cannot be subjected to a head greater than that to the uppermost watertight deck, unless surging of liquid or shift of cargo is allowed for; forces applied by machinery are generally known with some accuracy; acceleration loads due to ship motion may be known statistically. More precise analytical methods are warranted in these cases.

Having decided on the loading, the next step is to decide on the ultimate behaviour, which, for brevity, will be called failure. From the point of view of structural analysis, there are four possible ways offailing, by *(a)* direct fracture, *(b)* fatigue fracture, *(c)* instability, and *(d)* unacceptable deformation.

(a) *Direct fracture* may be caused by a part of the structure reaching the ultimate tensile, compressive, shear or crushing strengths. If metallurgical or geometrical factors inhibit an otherwise ductile material, it may fail in a brittle manner before the normally expected ultimate strength. It should be noted, that yield does not by itself cause fracture and cannot therefore be classed as failure in this context;

(b) *Fatigue fracture.* The elastic fatigue lives of test specimens of materials are fairly well documented. The elastic fatigue lives of complex structures are not divinable except by test, although published works of tests on similar structural items may give a lead. The history of reversals in practical structures at sea also requires definition. Corrosion fatigue is a special case of accelerated failure under fatigue when the material is in a corrosive element such as sea water. The 'bent nail' fatigue is another special case, in which yield is exceeded with each reversal and the material withstands very few reversals;

(c) *Instability.* In a strut, buckling causes an excessive lateral deflection; in a plating panel, it may cause load shirking by the panel or wrinkling; in a cylinder under radial pressure, instability may cause the circumference to corrugate; in a plating stiffener it may cause torsional tripping. Most of these types of instability failure are characterized by a relatively rapid increase in deflection for a small increase in load and would generally be regarded as failure if related to the whole structure; where only part of the structure shirks its load, as sometimes happens, for example, with panels of plating in the hull girder section, overall 'failure' does not necessarily occur.

(d) *Deformation.* A particular deflection may cause a physical foul with machinery or may merely cause alarm to passengers, even though there is no danger. Alignment of machinery may be upset by excessive deflection.

Such deformations may be in the elastic or the elasto-plastic range. The stiffness of a structure may cause an excessive amplitude of vibration at a well used frequency. Any of these could also constitute failure.

For each unit of structure in a ship, first the loading must be decided and then the various ways in which it would be judged to have failed must be listed and examined. What are these units of structure?

STRUCTURAL UNITS OF A SHIP

There are four basic types of structure with which the ship designer must deal: *(a)* plating-stiffener combinations, *(b)* panels of plating, *(c)* frameworks, *(d)* fittings.

(a) *Plating-stiffener combinations.* The simplest form of this is a single simple beam attached to a plate. Many parallel beams supporting plating constitute a grillage with unidirectional stiffening. Beams intersecting at right angles constitute an orthogonally stiffened grillage. These various units may be initially flat or curved, loaded in any plane and possess a variety of shapes and boundaries.

(b) *Panels of plating.* These are normally rectangular and supported at the four edges, subject to normal or in-plane loads. Initially, they may be nominally flat or dished;

(c) *Frameworks.* These may be portals of one or more storeys. Frameworks may be constituted by the transverse rings of side frames and deck beams or the longitudinal ring of deck girder, bulkhead stiffeners and longitudinal. They may be circular as in a submarine. Loading may be distributed or concentrated in their planes or normal to their planes.

(d) *Fittings.* There is a great variety of fittings in ships the adequacy of whose strength must be checked. Particular ones include control surfaces such

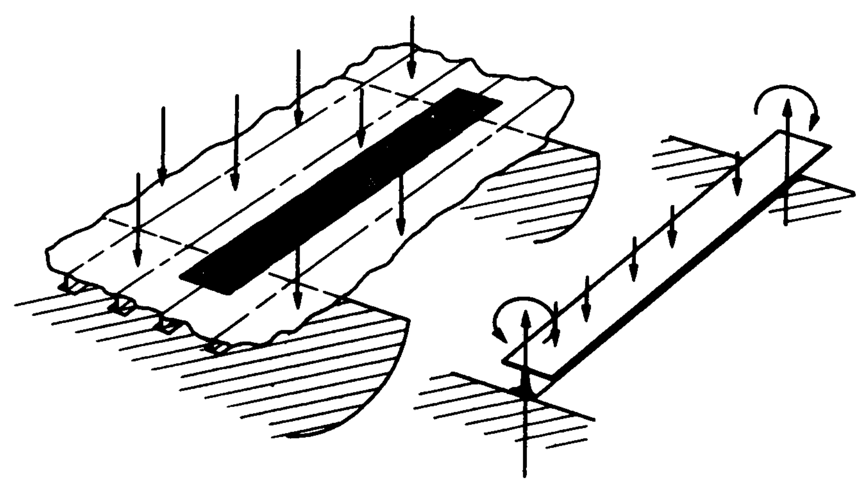


Fig. 7.1

as rudders and hydroplanes, shaft brackets and spectacle plates, masts, derricks, davits and machinery seatings.

Let us examine the methods of analysis available for each of these four basic structural units.

Stiffened plating

SIMPLE BEAMS

Very many of the local strength problems in a ship can be solved adequately by the application of simple beam theory to a single stiffener-plating combination. This is permissible if the boundaries of the unit so isolated are truly represented by forces and moments that adjacent units apply to it. Frequently, when there is a series of similarly loaded units, the influence of adjacent units on the edges will be zero; similarity longitudinally might also indicate that the end slopes are zero. These edge constraints have a large influence and in many cases will not be so easily determined. It is important that the deflection of the supporting structure is negligible compared with the deflection of the isolated beam, if the unit is to be correctly isolated; this is likely to be true if the supports are bulkheads but not if they are orthogonal beams. A summary of results for common problems in simple beams is given in Fig. 7.2.

According to the Bernoulli-Euler hypothesis from which the simple theory of bending is deduced, sections plane before bending remain plane afterwards and

$$\sigma = \frac{M}{I}y$$

For many joists and girders this is very closely accurate. Wide flanged beams and box girders, however, do not obey this law precisely because of the manner in which shear is diffused from the webs into the flanges and across the flanges. The direct stresses resulting from this diffusion do not quite follow this law but vary from these values because sections do not remain plane. Distribution of stress across stiffened plating under bending load is as shown by the wavy lines in Fig. 7.3, and this effect is known as the shear lag effect. While the wavy distribution of stress cannot be found without some advanced mathematics, the maximum stress can still be found by simple beam theory if, instead of assuming that all of the plating is partially effective, it is assumed that part of the plating is wholly effective. This effective breadth of flange, >.(Fig. 7.3), is used to calculate the effective second moment of area of cross-section. It is dependent on the type ofloading and the geometry of the structure. Because it is quite close to the neutral axis, the effective breadth of plating is not very influential and a figure of thirty thicknesses of plating is commonly used and sufficiently accurate; otherwise>. = *BI2* is used.

There remains, in the investigation of the single stiffener-plating combination, the problem of behaviour under end load. Classical Euler theories assume perfect struts and axial loads which never occur in practice. Many designers use these or the Rankine-Gordon formula which embraces the overriding case of

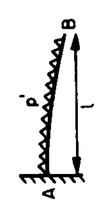
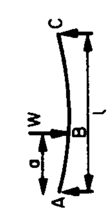
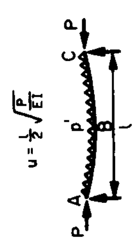
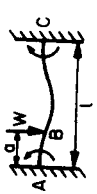
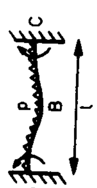
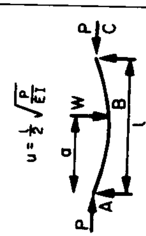
Problem	Bending moments	Deflections
	$BM_A = Wl$	$\delta_B = \frac{Wl^3}{3EI}$
	$BM_A = \frac{p'l^2}{2}$	$\delta_B = \frac{p'l^4}{8EI}$
	$BM_B = Wa\left(\frac{l-a}{l}\right)$	$\delta = \frac{Wx}{6EI}\left(\frac{l-a}{l}\right)(2al - a^2 - x^2) + \frac{W}{6EI}[x-a]^3$* if $a = \frac{l}{2}$, $\delta_B = \frac{Wl^3}{48EI}$ * Disregard last term if $x < a$.
	$BM_B = \frac{p'l^2}{8} \cdot \frac{2(1-\cos u)}{u^2 \cos u}$	$\delta = \frac{p'l^2}{4Pu^2}\left\{\frac{\cos u(1-2x/l)}{\cos u} - 1\right\} - \frac{p'x}{2P}(l-x)$ if $P = 0$, $\delta_B = \frac{5p'l^4}{384EI}$

Fig. 7.2

Fig. 7.2 (cont.)

Problem	Bending moments	Deflections
	$BM_C = Wa\left(\frac{l-a}{l}\right)^2$ $BM_A = W(l-a)\left(\frac{a}{l}\right)^2$ if $a = \frac{l}{2}$, $BM_A = BM_C = \frac{Wl}{8}$	$\delta_{max} = \frac{2Wa^3}{3EI}\left(\frac{l-a}{l+2a}\right)^2$ if $a = \frac{l}{2}$, $\delta_B = \frac{Wl^3}{192EI}$
	$BM_A = BM_C = \frac{p'l^2}{12}$	$\delta = \frac{p'x^2(l-x)^2}{24EI}$ $\delta_B = \frac{p'l^4}{384EI}$
	For AB $BM = \frac{Wl}{2u}\cdot\frac{\sin\frac{2ux}{l}\sin 2u\left(1-\frac{a}{l}\right)}{\sin 2u}$ max at $x = \frac{\pi l}{4u}$ or a For BC $BM = \frac{Wl}{2u}\cdot\frac{\sin\frac{2au}{l}\sin 2u\left(1-\frac{x}{l}\right)}{\sin 2u}$ max at $x = l\left(1-\frac{\pi}{4u}\right)$ or a	For AB $\delta = \frac{Wl}{2uP}\left\{\frac{\sin 2u\left(1-\frac{a}{l}\right)\sin\frac{2ux}{l}}{\sin 2u} - \frac{2xu}{l}\left(1-\frac{a}{l}\right)\right\}$ For BC $\delta = \frac{Wl}{2uP}\left\{\frac{\sin\frac{2au}{l}\sin 2u\left(1-\frac{x}{l}\right)}{\sin 2u} - \frac{2au}{l}\left(1-\frac{x}{l}\right)\right\}$

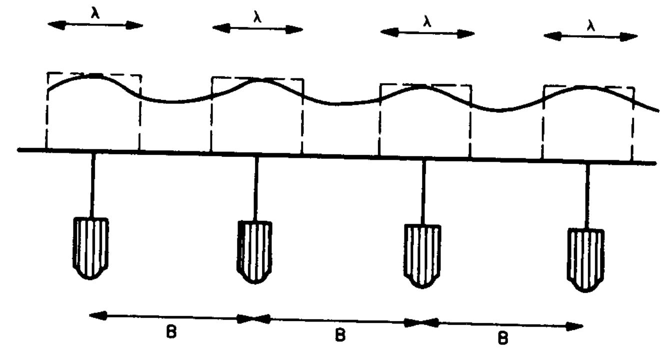


Fig. 7.3 Shear lag effects

yield, together with a factor of safety often as high as twenty or thirty. This is not a satisfactory approach, since it disguises the actual behaviour of the member. Very often, in ship structures there will be a lateral load, which transforms the problem from one of elastic instability into one of bending with end load. Typical solutions appear in Fig. 7.2. The lateral load might be sea pressure, wind, concentrated weights, personnel load, cargo or flooding pressure. However, there will, occasionally, arise problems where there is no lateral load in the worst design case. How should the designer proceed then?

Practical structures are always, unintentionally, manufactured with an initial bow, due to welding distortion, their own weight, rough handling or processes of manufacture. It can be shown that the deflection of a strut with an initial simple bow y_0 is given by:

$$y = \frac{P_E}{P_E - P}y_0 \quad (y \text{ is the total deflection, including } y_0).$$

$P_E/(P_E - P)$ is called the *exaggeration factor*. P_E is the classical Euler collapse load, $P_E = \pi^2 EI/l^2$, l being the effective length (Fig. 7.4).

The designer must therefore decide first what initial bow is likely; while some measurements have been taken of these in practical ship structures, the designer will frequently have to make a common-sense estimate. Having decided the value, and calculated the Euler load, the designer can find the maximum deflection from the equation above.

The maximum bending moment for a member in which end rotation is not constrained is, of course,

$$\text{Max BM} = Py_{max}$$

In general, so far as end loading is concerned, the assumption of simple support is safe.

GRILLAGES

Consider the effect of a concentrated weight W on two simply-supported beams at right angles to each other as shown in Fig. 7.5. This is the simplest form of

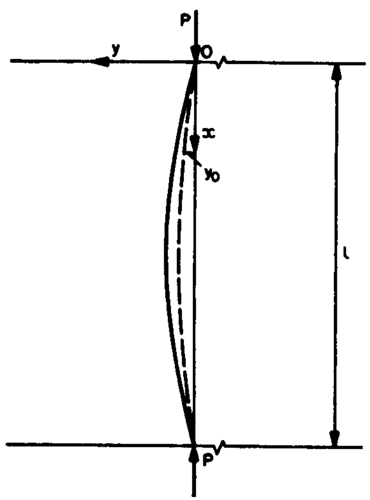


Fig. 7.4

grillage. Assume that the beams, defined by suffixes 1 and 2, intersect each other in the middle and that each is simply supported. What is not immediately obvious is how much the flexure of beam 1 contributes to supporting W and how much is contributed by the flexure of beam 2. Let the division of W at the middles be R_1 and R_2, then

$$R_1 + R_2 = W$$

Examining each beam separately, the central deflections are given by

$$\delta_1 = \frac{R_1 l_1^3}{48EI_1}$$

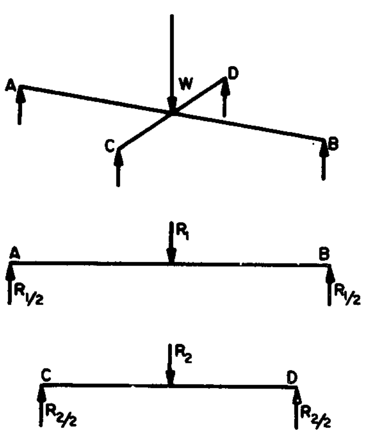


Fig. 7.5

and

$$\delta_2 = \frac{R_2 l_2^3}{48EI_2}$$

But, if the beams do not part company, $\delta_1 = \delta_2$

$$\therefore \quad \frac{R_1 l_1^3}{I_1} = \frac{R_2 l_2^3}{I_2}$$

Since $R_1 + R_2 = W$,

$$\delta = \frac{W l_1^3}{48EI_1} \times \frac{1}{\left(1 + \dfrac{I_2 l_1^3}{I_1 l_2^3}\right)}$$

and the maximum bending moments in the two beams are

$$\text{Max BM}_1 = \frac{R_1 l_1}{4} = \frac{W l_1}{4\left(1 + \dfrac{I_2 l_1^3}{I_1 l_2^3}\right)}$$

$$\text{Max BM}_2 = \frac{R_2 l_2}{4} = \frac{W l_2}{4\left(1 + \dfrac{I_1 l_2^3}{I_2 l_1^3}\right)}$$

This has been an exceedingly simple problem to solve. It is not difficult to see, however, that computation of this sort could very quickly become laborious. Three beams in each direction, unaided by symmetry could give rise to nine unknowns solved by nine simultaneous equations. Moreover, a degree of fixity at the edges introduces twelve unknown moments while moments at the intersections cause twist in the orthogonal beams. Edge restraint, uneven spacing of stiffeners, differing stiffeners, contribution from plating, shear deflection afflB other factors all further contribute to making the problem very difficult indeed.

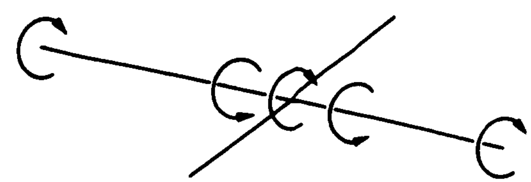


Fig. 7.6

Mathematical theories have evolved to solve a great range of such problems whose solutions are available through computer programs and, in some cases, by data sheets.

SWEDGED PLATING

Fabrication costs can be reduced in the construction of surfaces which need not be plane, by omitting stiffeners altogether and creating the necessary flexural rigidity by corrugated or swedged plating. Main transverse bulkheads in a large oil tanker, for example, may have a depth of swedge of 25 cm. Such plating is incorporated into the ship to accept end load in the direction of the swedges as well as lateral loading. It tends to create difficulties of structural discontinuity where the swedge meets conventional stiffeners, for example where the main transverse bulkhead swedges meet longitudinal deck girders.

Buckling of some faces of the plating is possible if the swedges are not properly proportioned and it is this difficulty, together with the shear diffusion in the plating, which makes the application of simple beam theory inadequate. Properly designed, corrugated plating is highly efficient.

COMPREHENSIVE TREATMENT OF STIFFENED PLATING

As will be discussed presently, panels or stiffened plating may shirk their duty by buckling so that they do not make their expected contribution to the overall ship's sectional modulus. This shirking, or load shortening, is illustrated in Fig. 7.7. This shows that the load shortening depends upon:

(*a*) the imperfections of the stiffeners in the form of a bow
(*b*) plating panel slenderness ratio $\beta = (b/t)\sqrt{\sigma_Y/E}$
(*c*) the ratio of stiffener cross section A_s to the overall cross section $A(A_s/A = 0.2$ average imperfections of stiffener)
(*d*) the stiffener slenderness ratio

$$\lambda = \frac{l}{\pi k}\sqrt{\frac{\sigma_Y}{E}}$$

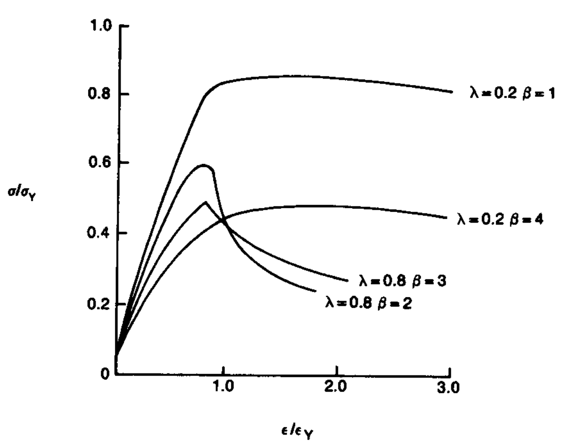


Fig. 7.7 Load shortening curves for panels with tee bar stiffeners

where k is the radius of gyration of the plating/stiffener combination and l is the stiffener length.

Note from Fig. 7.7 the sudden nature of the change, occurring especially for high values of A_s/A is also found to be very significant as might be expected. Readers are referred to published papers for a more complete appraisal and also for load shortening curves for plating panels alone like that shown in Fig. 7.37.

Panels of plating

Knowledge of the behaviour of panels of plating under lateral pressure (sometimes called plate elements) has advanced rapidly in the last 40 years. It is important for the designer to understand the actual behaviour of plates under this loading, so that the theory or results most suitable for the application can be selected. To do this, consider how a panel behaves as the pressure is increased.

BEHAVIOUR OF PANELS UNDER LATERAL LOADING

Consider, at first, the behaviour of a rectangular panel with its four edges clamped, and unable to move towards each other. As soon as pressure is applied to one side, elements in the plate develop a flexural resistance much like the elements of a simple beam but in orthogonal directions. Theories relating the pressure to the elastic flexural resistance of the plate alone are called *small deflection theories*. As the pressure is increased and deflection of the same order as the plate thickness results, the resistance of the plate to the pressure stiffens because of the influence of membrane tension. This influence is dependent upon the deflection, since the resistance is due to the resolute of the tension against the direction of the pressure. Clearly, it has but a small influence when the deflection is small. Figure 7.9 shows a typical section through the plate; the orthogonal section would be similar. Elastic theories which take into account both flexural rigidity and membrane tension are called *large deflection theories*.

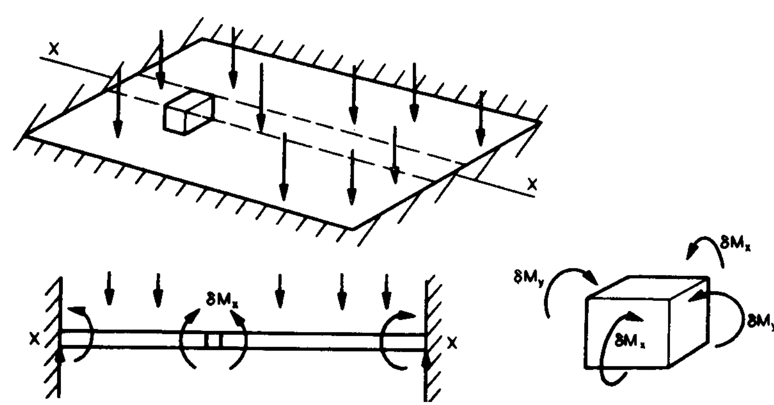


Fig. 7.8 Flexural resistance of a plate

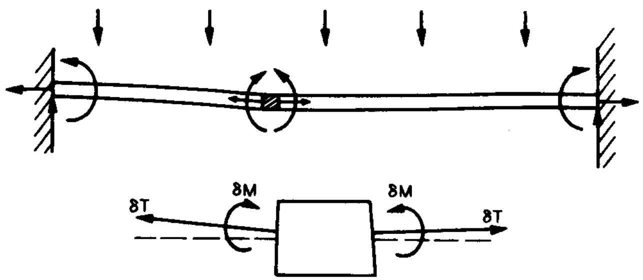


Fig. 7.9 Membrane tension effects in a plate

With a further increase in pressure, yield sets in, deflection increases more rapidly and the effect of membrane tension becomes predominant. The plate is partly elastic and partly plastic and theories relating to the behaviour following the onset of yield are called *elasto-plastic theories.* Yield is first reached at the middles of the two longer sides on the pressure side of the plate; soon afterwards, yield is reached on the other side of the plate and this area of plasticity spreads towards the corners. Plasticity spreads with increase in pressure in the stages illustrated in Fig. 7.10. The plastic areas, once they have developed through the thickness, are called hinges since they offer constant resistance to rotation. Finally, once the hinges have joined to form a figure of eight, distortion is rapid and the plate distends like a football bladder until the ultimate tensile strength is reached.

The precise pattern of behaviour depends on the dimensions of the panel but this description is typical. There can be no doubt that after the first onset of yield, a great deal of strength remains. Unless there are good reasons not to do so, the designer would be foolish not to take advantage of this strength to effect an economical design. Once again, this brings us to an examination of 'failure'. As far as a panel under lateral pressure is concerned, failure is likely to be either fatigue fracture or unacceptable deflection. Deflection considered unacceptable for reasons of appearance, to avoid the 'starved horse' look, might nowadays be thought an uneconomical criterion. In considering fatigue fracture, it must be remembered that any yield will cause some permanent set; removal of the load and reapplication of any lesser load will not increase the permanent set and the plate will behave elastically. A plate designed to yield under a load met very rarely, will behave elastically for all of its life save for the one loading which causes the maximum permanent set. Indeed, initial permanent set caused by welding distortion will permit the plate to behave elastically, thereafter, if this is taken as the maximum acceptable permanent set.

In ship's structures, small deflection elastic theories would be used for plates with high fluctuating loading such as those opposite the propeller blades in the outer bottom and where no permanent set can be tolerated in the flat keel, or around sonar domes, for example, in the outer bottom. Large deflection elastic theory is applicable where deflection exceeds about a half the thickness, as is likely in thinner panels. Elastoplastic theories are appropriate for large areas of the shell, for decks, bulkheads and tanks.

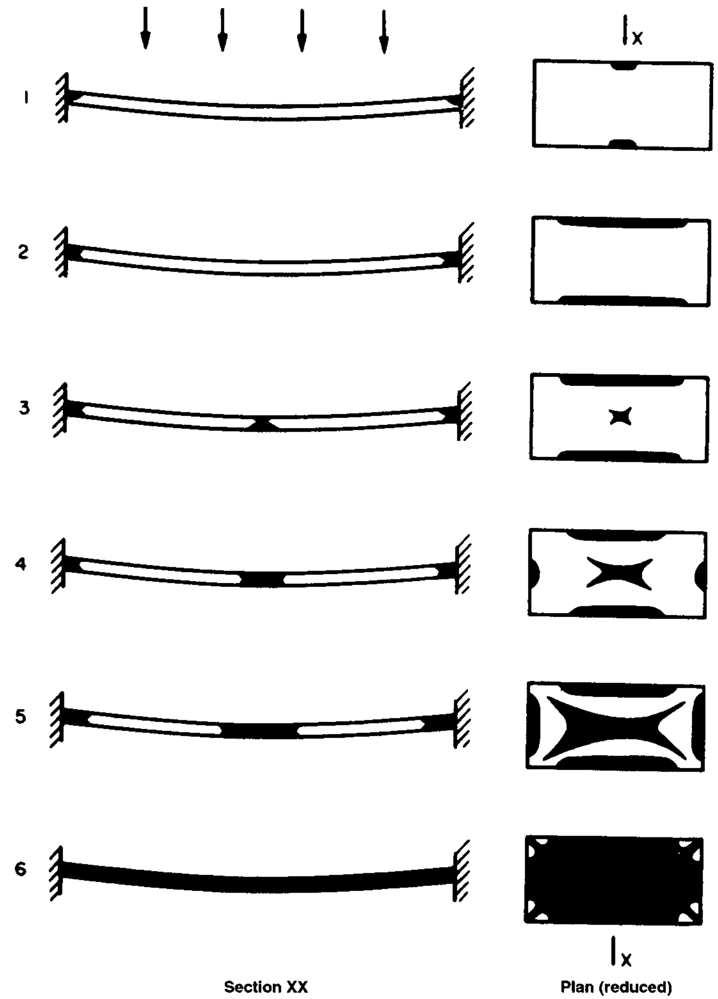


Fig. 7.10 Onset of plasticity in a plate

AVAILABLE RESULTS FOR FLAT PLATES UNDER LATERAL PRESSURE

The three ranges covered by the different theories are illustrated in Fig. 7.11. It is clear that unless the correct theory is chosen, large errors can result.

For plates with totally clamped edges (which are rare):

$$\text{Central deflection} = k_1 \cdot \frac{pb^4}{384D}$$

$$\text{Maximum stress} = k_2 \cdot \frac{p}{2}\left(\frac{b}{t}\right)^2$$

where

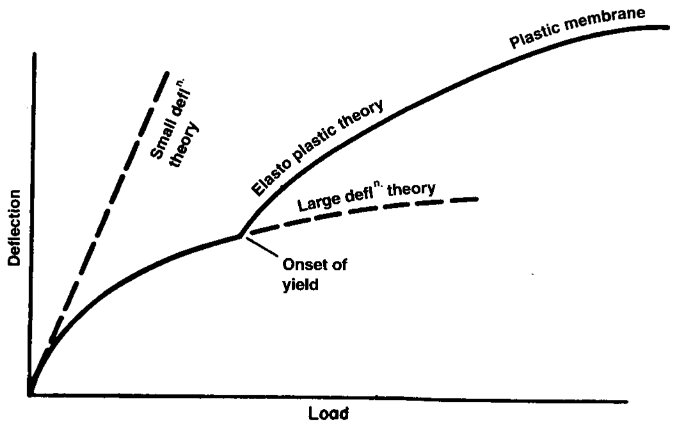


Fig. 7.11 Comparison of plate theories

$$D = \frac{Et^3}{12(1 - \nu^2)}$$

k_1 and k_2 are non-dimensional and the units should be consistent.

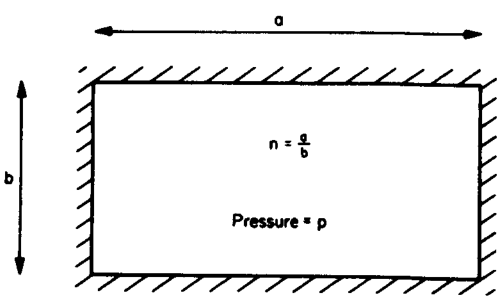


Fig. 7.12

n	1	1.25	1.50	1.75	2	∞
k_1	0.486	0.701	0.845	0.928	0.976	1.0
k_2	0.616	0.796	0.906	0.968	0.994	1.0

Large deflection elastic theory gives results as shown in Fig. 7.13.

Elasto-plastic results are based upon the maximum allowable pressure defined, arbitrarily, as the lesser pressure which will cause either

(a) a central plastic hinge in a long plate or 1.25 times the pressure which causes yield at the centre of a square plate; or

(b) the membrane tension to be two-thirds the yield stress.

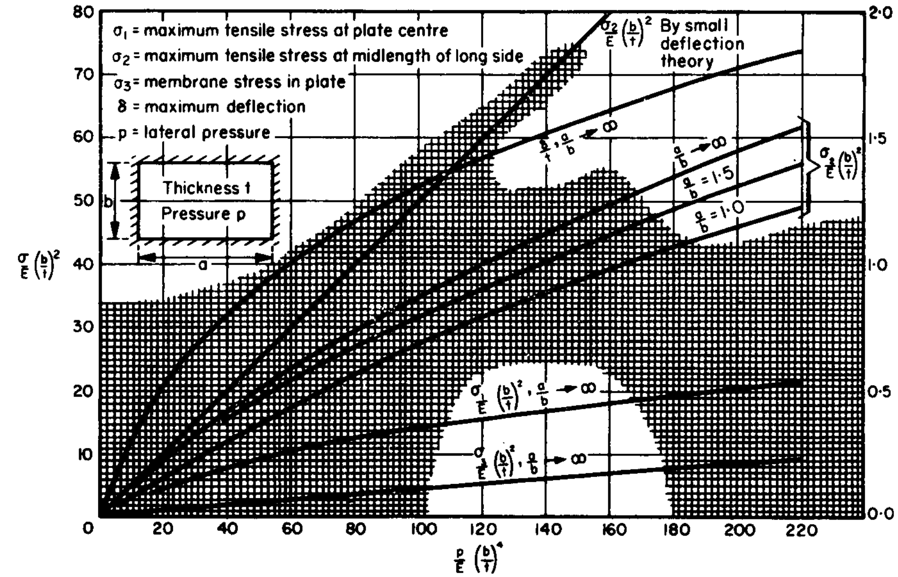


Fig. 7.13 Large deflection results

The first criterion applies to thick plates and the latter to thin plates. Design curves giving maximum permissible pressure, deflection and permanent set based on these criteria are shown in Fig. 7.14. Results from an important extension to plate theory, in which pressures have been calculated which will not permit any increase in an initial permanent set, i.e. the plate, after an initial permanent set (caused perhaps by welding) behaves elastically, are presented in Fig. 7.15 for long plates.

In considering the real behaviour of panels forming part of a grillage it has been shown that the pull-in at the edges has an appreciable effect on panel behaviour, and that a panel in a grillage does not have the edge constraint necessary to ensure behaviour in the manner of Fig. 7.14. The edge constraint which can give rise to membrane tension arises from the hoop effects in the plane of the boundary. Figure 7.16 gives design curves assuming that the edges of the panel are free to move inwards.

EXAMPLE 1. Compare the design pressures suitable for a steel plate O.56m x 1.26m = 6mm yielding at 278 N/mm^2 assuming, (a) the edges fixed and (b) the edges free to move inwards. Calculate also the pressure which will not cause additional permanent set above an initial bow of 2.8 mm due to welding. Had the plate been initially flat, what pressure would have first caused yield?

Solution:

$$\frac{a}{b} = \frac{1.26}{0.56} = 2.25; \; \frac{b}{t}\left(\frac{\sigma_y}{E}\right)^{\frac{1}{2}} = \frac{0.56}{0.006}\left(\frac{278}{209,000}\right)^{\frac{1}{2}} = 3.41$$

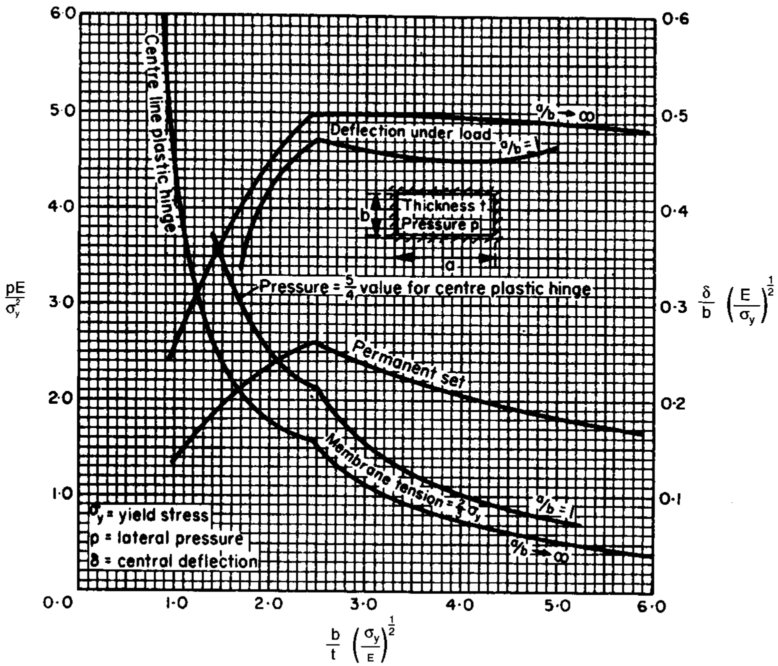


Fig. 7.14 Elasto-plastic results

for which Fig. 7.14 gives

$$\frac{pE}{\sigma_y^2} = 1.00 \text{ approx.} \quad \text{and} \quad \frac{\delta}{b}\left(\frac{E}{\sigma_y}\right)^{\frac{1}{2}} = 0.224$$

Hence,

$$\text{design pressure} = \frac{1.00 \times 278^2}{209{,}000} = 0.37\,\text{N/mm}^2$$

Using the same criterion of failure, viz. a permanent set coefficient of 0.224 and interpolating between the two diagrams of Fig. 7.16,

$$\frac{pE}{\sigma_\nu^2} = 0.57 \text{ approx.}$$

Hence,

$$\text{design pressure} = \frac{0.57}{1.00} \times 0.37 = 0.21\,\text{N/mm}^2$$

From Fig. 7.15, with a permanent deflection coefficient of

$$\frac{0.0028}{0.56}\left(\frac{209{,}000}{278}\right)^{\frac{1}{2}} = 0.137, \quad \text{and} \quad \frac{b}{t}\left(\frac{\sigma_y}{E}\right)^{\frac{1}{2}} = 3.41,$$

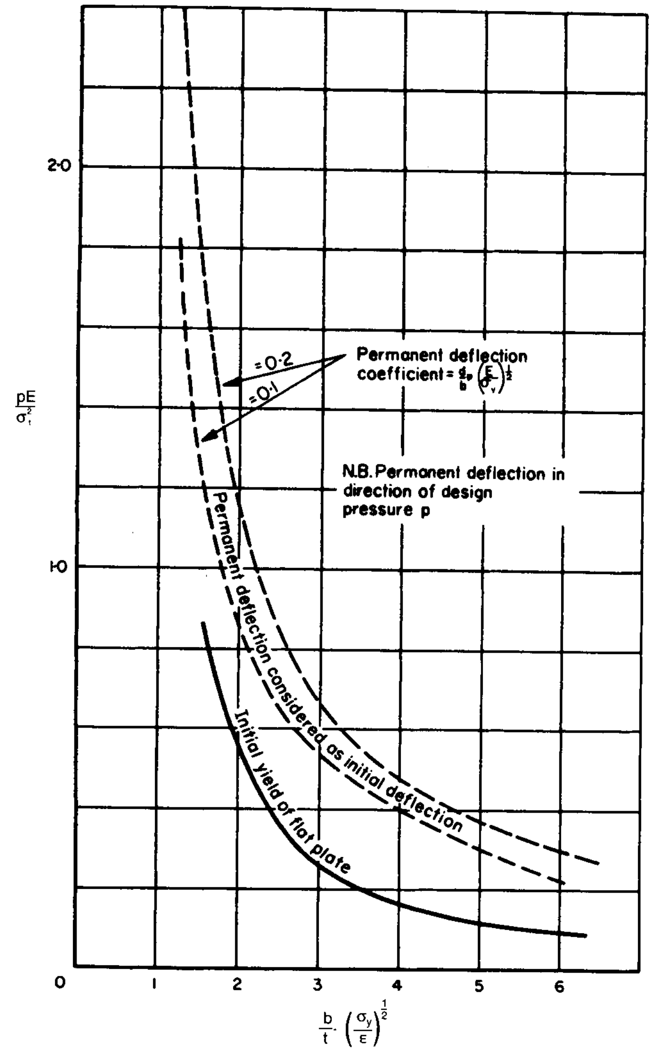


Fig. 7.15

the pressure which will not cause additional permanent set is obtained from $pE/\sigma_y^2 = 0.50$, whence

$$p = \frac{0.50}{1.00} \times 0.37 = 0.185\,\text{N/mm}^2$$

To find the pressure at which this plate begins first to yield, try first small deflection theory:

$$\frac{a}{b} = 2.25 \quad \text{so that} \quad k_2 = 1.0 \text{ approximately}$$

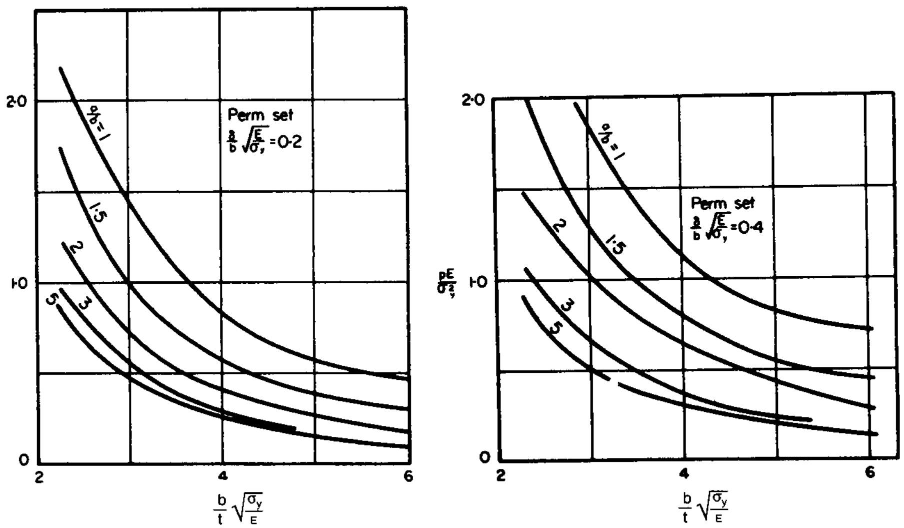


Fig. 7.16 Design curves for panels with edges free to move inwards

Then

$$278\ \mathrm{N/mm^2} = \frac{p}{2}\left(\frac{0.56}{0.006}\right)^2$$

whence

$$p = 0.0638\ \mathrm{N/mm^2}$$

Check the deflection at first yield:

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{209(6)^3 10^{-6}}{12(1-0.09)} = 0.00413\ \mathrm{MN\,m} \text{ and } k_1 = 0.98$$

$$\text{deflection} = \frac{0.98 \times 0.0638(0.56)^4}{384 \times 0.00413} = 3.9\ \mathrm{mm}$$

Small deflection theory thus gives a deflection about 65 per cent of the thickness and is probably adequate. However, let us take this example to illustrate the use of large deflection theory. Because the stress is divided into two parts, direct calculation of the pressure to cause yield is not possible, and two trial pressures must be taken: referring to Fig. 7.13,

$$\frac{\sigma}{E}\left(\frac{b}{t}\right)^2 = \frac{278}{209{,}000}\left(\frac{0.56}{0.006}\right)^2 = 11.6$$

Now, for

$$p = 0.06\ \mathrm{N/mm^2} \quad \frac{p}{E}\left(\frac{b}{t}\right)^4 = \frac{0.06}{209{,}000}\left(\frac{0.56}{0.006}\right)^4 = 21.8$$

whence, from Fig. 7.13,

$$\frac{\sigma_3}{E}\left(\frac{b}{t}\right)^2 = 1.2 \text{ and } \frac{f_2}{E}\left(\frac{b}{t}\right)^2 = 10.1$$

thus, $\sigma_3 = 28.8$ and $\sigma_2 = 242.3$ so that the total stress is $271\ \mathrm{N/mm^2}$. Similarly, for $p = 0.07\ \mathrm{N/mm^2}$

$$\frac{p}{E}\left(\frac{b}{t}\right)^4 = \frac{0.07}{0.06} \times 21.8 = 25.4$$

$$\frac{\sigma_3}{E}\left(\frac{b}{t}\right)^2 = 1.3 \quad \text{and} \quad \frac{f_2}{E}\left(\frac{b}{t}\right)^2 = 11.2$$

so that $\sigma = 31.2 + 268.7 = 300\ \mathrm{N/mm^2}$

By interpolation, the pressure first to cause yield of $278\ \mathrm{N/mm^2}$ is $0.062\ \mathrm{N/mm^2}$. This example typifies the use of Fig. 7.13 for large deflection elastic theory; in this particular case, it confirms that small deflection theory was, in fact, sufficiently accurate.

BUCKLING OF PANELS

Buckling of panels in the direction of the applied lateral pressure is known as snap through buckling. This is likely where the initial permanent set is $1\frac{1}{2} - 3$ times the thickness.

Buckling due to edge loading has been dealt with on a theoretical basis. For a panel simply supported at its edges, the critical buckling stress is given by

$$\sigma_c = \frac{k\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2$$

where k is given by Fig. 7.17. When a plate does buckle in this way, it may not be obvious that it has occurred; in fact, the middle part of the panel shirks its load which is thrown on to the edge stiffeners. It is common practice to examine only the buckling behaviour of these stiffeners associated with a width of plating equal to thirty times its thickness.

Buckling due to shear in the plane of the plate causes wrinkling in the plate at about 45 degrees. Such a failure has been observed in the side plating of small ships at the sections of maximum shear. The critical shear stress is given by

$$\tau_c = kE\left(\frac{t}{b}\right)^2$$

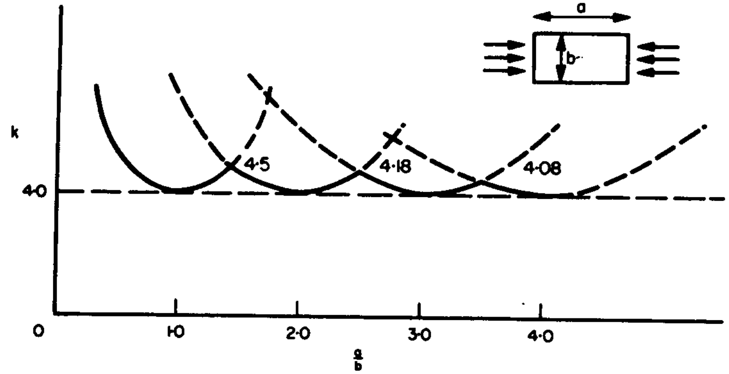


Fig. 7.17

where $k = 4.8 + 3.6(b/a)$ for edges simply supported or $k = 8.1 + 5.1(b/a)^2$ for edges clamped. A more truly representative examination of panel behaviour under biaxial compression and lateral pressure is given in the comprehensive work by C. S. Smith, *et al.*, published by the RINA. This important culmination of many years of research enables a designer to determine optimum panel shapes and to take into account initial strains and imperfections in the plating.

Frameworks

Analysis of the three-dimensional curved shape of the hull between main bulkheads is the correct approach to the determination of its strength. Finite element techniques can be used for this but, for the student, the reduction of the problem to two-dimensional strips or frames remains important. In reducing the problem to two-dimensional frameworks, it is necessary to be aware that approximations are being made....

There are in general three types of plane framework with which the ship designer is concerned:

(a) orthogonal portals
(b) ship-shape rings
(c) circular rings.

Portals arise in the consideration of deckhouses, superstructures and similar structures and may have one or more storeys. If the loading in the plane of the portal is concentrated, the effect of the structure perpendicular to the plane of the portal is likely to be one of assistance to its strength. If the loading is spread over many portals, one of which is being isolated for analysis, the effects of the structure perpendicular to its plane will be small, from considerations of symmetry unless there is sidesway when the in-plane stiffness of the plating will be appreciable. Thus, the reduction of the problems to two-dimensional frameworks is, in general, pessimistic and safe, although each problem should be examined on its merits.

Ship-shape rings arise by isolating a transverse slice of the ship, comprising bottom structure, side frames and deck beams, together with their associated plating. Treatment of the complete curved shell in this manner is likely to be highly pessimistic because the effect of longitudinal structure in keeping this ring to shape must be considerable. These longitudinals connect the ring to transverse bulkheads with enormous rigidity in their own planes. The calculations are, nevertheless, performed to detect the likely bending moment distribution around the ring so that material may be distributed to meet it. Transverse strength calculations are, therefore, generally comparative in nature except, perhaps, in ships framed predominantly transversely.

Circular rings occur in submarines and other pressure vessels, such as those containing nuclear reactors.

METHODS OF ANALYSIS

There are many good textbooks available, which explain the various analytical tools suitable for frameworks in more detail than is possible here. Standard textbooks on structural analysis treat the more common framed structures met in ship design. A brief summary of four methods of particular use to the naval architect may, however, be worthwhile. These methods are: (a) moment distribution, (b) slope-deflection, (c) energy methods, and (d) limit design methods.

(a) The *moment distribution* of Hardy Cross is particularly suitable for portal problems where members are straight and perpendicular to each other. Bending moment distribution is obtained very readily by this method but slopes and deflection are not obtained without the somewhat more general relaxation methods of R. V. Southwell.

The moment distribution process is one of iteration in methodical sequence as follows:

(i) all joints of the framework are assumed frozen in space, the loading affecting each beam as if it were totally encastre, with end fixing moments;
(ii) one joint is allowed to rotate, the total moment at the joint being distributed amongst all the members forming the joint according to the formula $(I/l)/(\Sigma I/l)$; the application of such a moment to a beam causes a carry-over of one-half this value (if encastre at the other end) to the far end, which is part of another joint;
(iii) this joint is then frozen and a half of the applied moment is carried over to each remote end (sometimes, the carry-over factor is less than one-half indeed when the remote end is pinned it is zero);
(iv) the process is repeated at successive joints throughout the framework until the total moments at each joint are in balance.

This process prevents sidesway of the framework which occurs unless there is complete symmetry. This is detected by an out-of-balance moment on the overall framework. As a second cycle of operations therefore, sufficient side force is applied to liquidate this out-of-balance without allowing joint rotation, thus causing new fixing moments at the joints. These are then relaxed by repeating the first cycle of operations and so on. A simple example of a moment distribution is given presently, but for a full treatment, the student is referred to standard textbooks on the analysis of frameworks.

(*b*) *Slope deflection analysis* is based upon the fundamental equation

$$M = EI\frac{\mathrm{d}^2y}{\mathrm{d}x^2}$$

Thence, the area of the M/EI-curve, $= \int \frac{M}{EI}\,\mathrm{d}x$, gives the change of slope, $\mathrm{d}y/\mathrm{d}x$. Integrated between two points in a beam, $\int \frac{M}{EI}\mathrm{d}x$ gives the difference in the slopes of the tangents at the two points. Further, $\int \frac{Mx}{EI}\mathrm{d}x$ between A and B, i.e. the moment of the M/EI-diagram about a point A gives the distance AD between the tangent to B and the deflected shape as shown in Fig. 7.18. These two properties of the M/EI-diagram are used to determine the distribution of bending moment round a framework.

Consider the application of the second principle to a single beam AB subjected to an external loading and end fixing moments M_{AB} and M_{BA} at which the slopes are θ_{AB} and θ_{BA}, positive in the direction shown in Fig. 7.18.

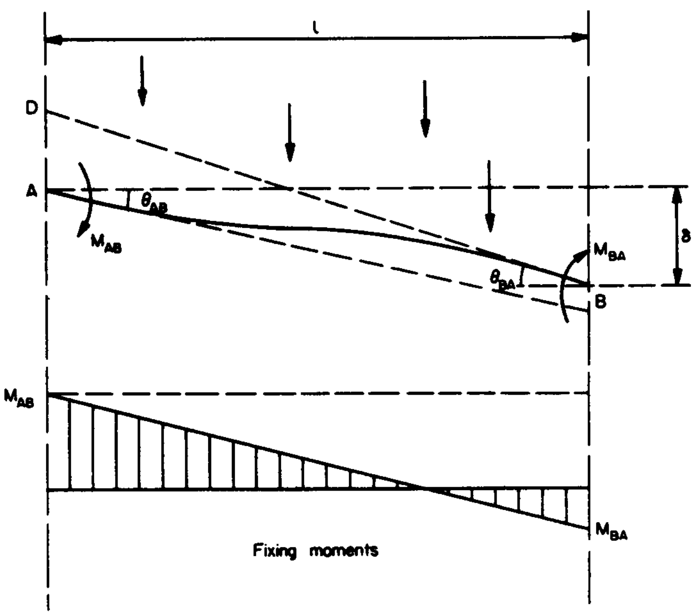


Fig. 7.18

Let the first moment of the free bending moment diagram (i.e. that due to the external loading assuming the ends to be pinned) about the ends A and B be respectively m_A and m_B. Then, taking moments of the M/EI-diagram about A,

$$\frac{M_{AB}}{EI}\frac{l^2}{2} - \frac{(M_{AB}+M_{BA})}{2EI}\frac{2l^2}{3} + \frac{m_A}{EI} = -l\theta_{BA} + \delta$$

i.e.

$$\theta_{BA} = -\frac{l}{6EI}\left(M_{AB} - 2M_{BA} + \frac{6m_A}{l^2}\right) + \frac{\delta}{l}$$

Taking moments about B,

$$\frac{M_{AB}}{EI}\frac{l^2}{2} - \frac{(M_{AB}+M_{BA})}{2EI}\frac{l^2}{3} + \frac{m_B}{EI} = l\theta_{AB} - \delta$$

i.e.

$$\theta_{AB} = \frac{l}{6EI}\left(2M_{AB} - M_{BA} + \frac{6m_B}{l^2}\right) + \frac{\delta}{l}$$

These expressions are fundamental to the slope deflection analysis of frameworks. They may be applied, for example, to the simple portal ABCD of Fig. 7.19, expressions being obtained for the six slopes, two of which will be zero and two pairs of which (if C and B are rigid joints) will be equated. On eliminating all these slopes there will remain five equations from which the five unknowns, M_{AB}, M_{BA}, M_{CD}, M_{DC} and δ can be found. This method has the advantage over moment distribution of supplying distortions, but it becomes arithmetically difficult when there are several bays. It is suitable for computation by computer where repetitive calculations render a program worth writing. Sign conventions are important.

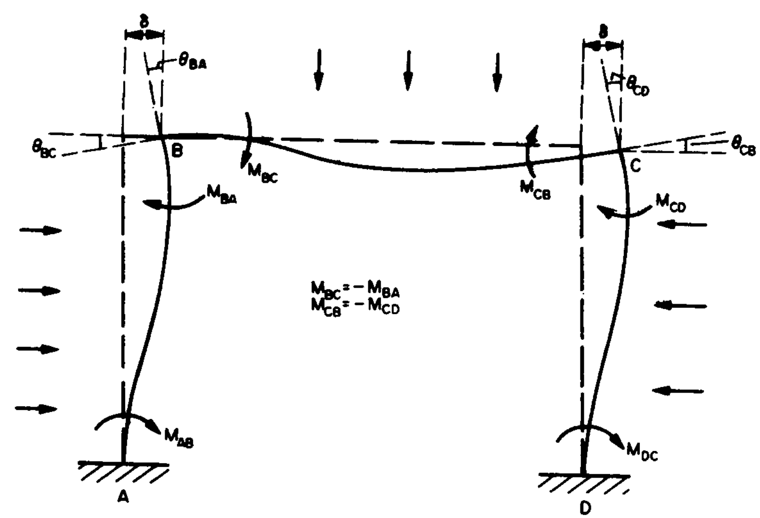


Fig. 7.19

(*c*) The *energy method* most useful in the context of this chapter is based on a theorem of Castigliano. This states that the partial derivative of the total strain energy U with respect to each applied load is equal to the displacement of the structure at the point of application in the direction of the load:

$$\frac{\partial U}{\partial P} = \delta_p$$

The expressions for strain energies U due to direct load, pure bending, torsion and shear are:

Direct load P, member of cross-sectional area A,

$$U = \int \frac{P^2}{2AE}\,dx$$

Bending moment M, curved beam of second moment I,

$$U = \int \frac{M^2}{2EI}\,ds$$

Torque T, member of polar second moment J,

$$U = \int \frac{T^2}{2CJ}\,dx$$

Shearing force S, element of cross-sectional area A, (Fig. 7.20)

$$U = \int \frac{S^2}{2AC}\,dx = \int \frac{\tau^2 A\,dx}{2C}$$

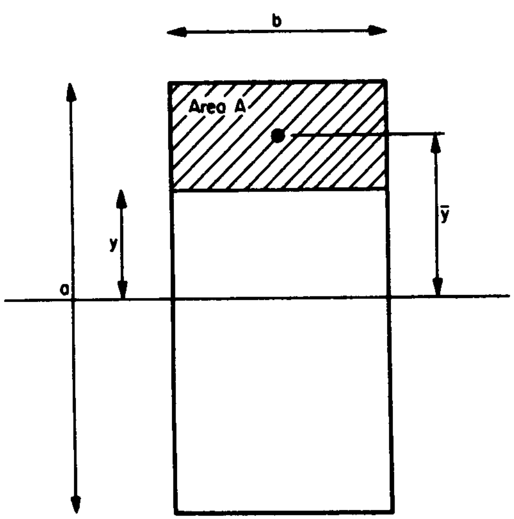


Fig. 7.20

The strain energy in a cantilever of rectangular cross-section $a \times b$, for example, with an end load W is given by

$$U = \int_0^l \frac{M^2}{2EI}\,dx + \int_0^{ab}\int_0^l \frac{\tau^2}{2C}\,dA\,dx$$

Now shear stress τ varies over a cross-section according to the expression $\tau = (SA/Ib)\bar{y}$. For the rectangular cantilever then,

$$\tau = \frac{6W}{a^3 b}\left(\frac{a^2}{4} - y^2\right) \quad \text{and} \quad dA = b\,dy.$$

Then

$$U = \int_0^l \frac{W^2 x^2\,dx}{2EI} + 2\int_0^{a/2}\int_0^l \frac{18W^2}{a^6 bC}\left(\frac{a^4}{16} - \frac{a^2 y^2}{2} + y^4\right) dy\,dx$$

$$= \frac{W^2 l^3}{6EI} + \frac{3W^2 l}{5abC}$$

End deflection

$$\delta = \frac{\partial U}{\partial W} = \frac{Wl^3}{3EI} + \frac{6Wl}{5abC}$$

The first expression is the bending deflection as given in Fig. 7.2, and the second expression is the shear deflection.

In applying strain energy theorems to the ring frameworks found in ship structures, it is common to ignore the effects of shear and direct load which are small in comparison with those due to bending. Confining attention to bending effects, the generalized expression becomes

$$\delta = \frac{\partial U}{\partial P} = \int \frac{M}{EI}\frac{\partial M}{\partial P}\,ds$$

Applying this, by example, to a simple ship-shape ring with a rigid centre line bulkhead which can be replaced by three unknown forces and moments, these can be found from the three expressions, since all displacements at B are zero:

$$0 = \frac{\partial U}{\partial H} = \int \frac{M}{EI}\frac{\partial M}{\partial H}\,ds$$

$$0 = \frac{\partial U}{\partial V} = \int \frac{M}{EI}\frac{\partial M}{\partial V}\,ds$$

$$0 = \frac{\partial U}{\partial M} = \int \frac{M}{EI}\frac{\partial M}{\partial M_B}\,ds,$$

summed for members BC and CDE.

Expressions for the bending moment M in terms of V, H, M_B and the applied load can be written down for members BC and CDE, added, and the above expressions determined. Hence V, H and M_B can be found and the bending

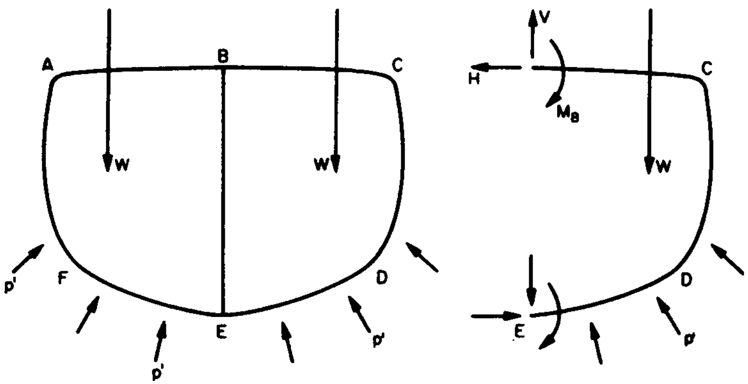


Fig. 7.21

moment diagram drawn. This method has the advantage that it deals readily with frames of varying inertia, the integrations being carried out by Simpson's rule. Like the slope deflection method, the arithmetic can become formidable.

(*d*) The *limit design method* is called also plastic design or collapse design. This method uses knowledge of the behaviour of a ductile material in bending beyond the yield point. A beam bent by end couples M within the elastic limit has a cross-section in which the stress is proportional to the distance from the neutral axis,

$$M = \sigma Z \text{ where } Z = \frac{I}{y}, \quad \text{the section modulus.}$$

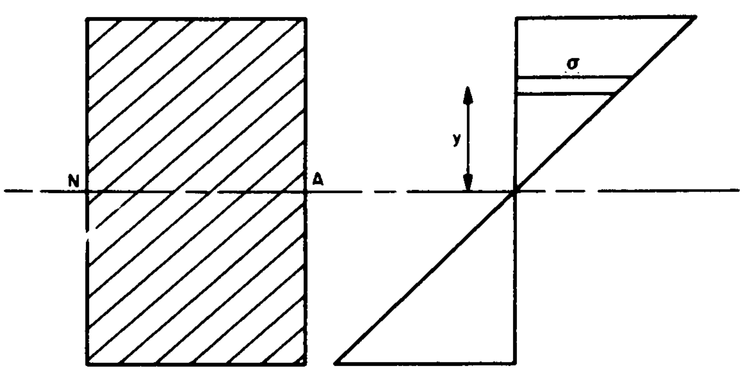


Fig. 7.22

Bent further, yield is reached first in the outer fibres and spreads until the whole cross-section has yielded, when the plastic moment, $M_p = \sigma_y S$ where S is the addition of the first moments of area of each side about the neutral axis and is called the plastic modulus. The ratio S/Z is called the *shape factor* which has a value 1.5 for a rectangle, about 1.2 for a rolled steel section and about 1.3 for a plating-stiffener combination. When a beam has yielded across its section, its

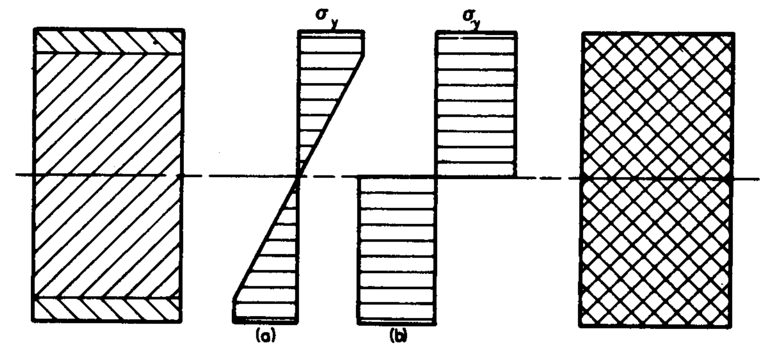


Fig. 7.23

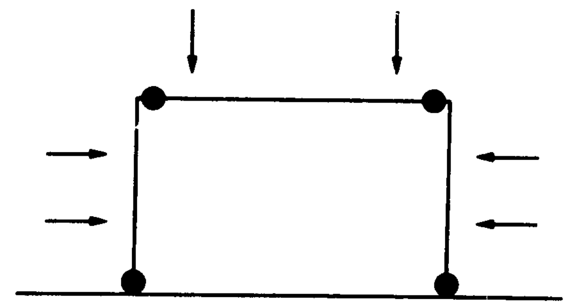


Fig. 7.24

moment of resistance is constant and the beam is said to have formed a plastic hinge.

The least load which forms sufficient plastic hinges in a framework to transform it into a mechanism is called the collapse load. A portal, for example, which has formed plastic hinges as shown in Fig. 7.24 has become a mechanism and has 'collapsed'. Let us apply this principle to an encastre beam (Fig. 7.25).

If yield is assumed to represent 'failure' by an elastic method of design, the maximum uniformly distributed load that the beam can withstand is (from Fig. 7.2)

$$p' = \frac{12Z\sigma_y}{l^2}$$

Using the definition of failure for the plastic method of design, however, collapse occurs when hinges occur at A, B and C, and they will all be equal to $\frac{1}{2}p'(l^2/8)$, i.e.

$$p' = \frac{16S\sigma_y}{l^2}$$

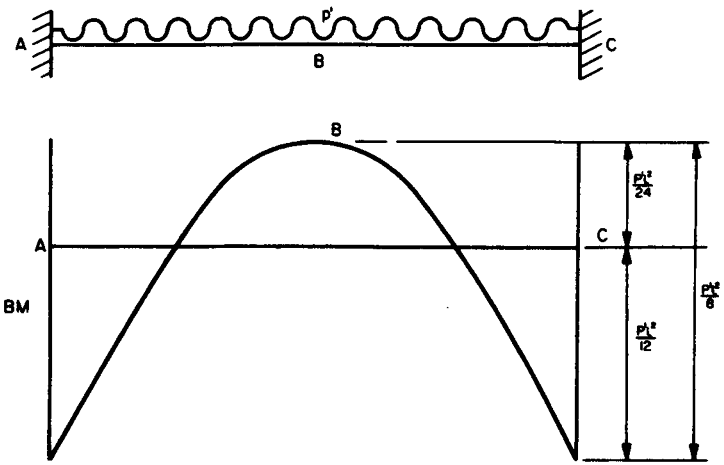


Fig. 7.25

If the shape factor for the beam section is taken as 1.2, the ratio of these two maximum carrying loads is 1.6, i.e. 60 per cent more load can be carried after the onset of yield before the beam collapses.

The plastic design method is often more conveniently applied through the principle of virtual work whence the distance through which an applied load moves is equated to the work done in rotating a plastic hinge. In the simple case illustrated in Fig. 7.26, for example,

$$W\frac{l\theta}{2} = M_p\theta + M_p\theta + M_p(2\theta) = 4M_p\theta$$

i.e. collapse load, $W = 8M_p/l$. This method can be used for finding the collapse loads of grillages under concentrated load; in this case, various patterns of plastic hinges may have to be tried in order to find the least load which would cause a mechanism (Fig. 7.27).

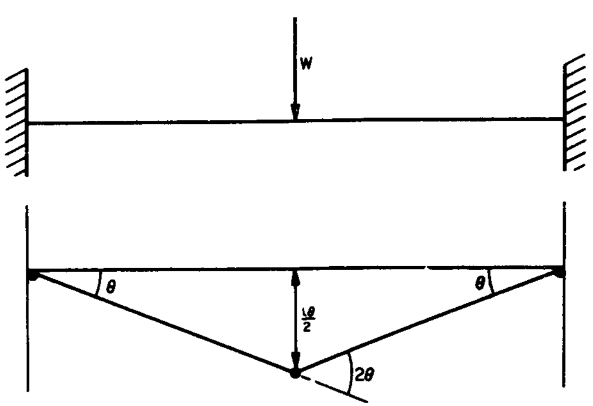


Fig. 7.26

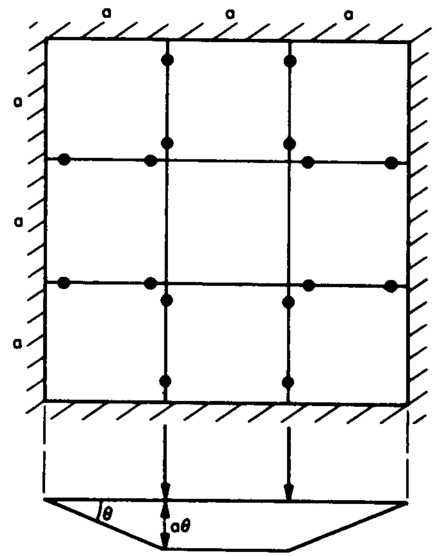


Fig. 7.27

Knowledge of the collapse loads of superstructures under nuclear blast or extreme winds is necessary, and the limit design method is the only way of calculating them. The method is suitable for the determination of behaviour of other parts of a ship structure subject to once-in-a-lifetime extreme loads such as bulkheads. Use of the method with a known factor of safety (or load factor), can ensure normal behaviour in the elastic range and exceptional behaviour in the plastic range-it is indeed, the only method which illustrates the real load factor over working load.

ELASTIC STABILITY OF A FRAME

The type of elastic instability of major concern to the designer of plated framed structures in a ship is that causing tripping, i.e. the torsional collapse of a stiffener sideways when the plating is under lateral load. Tripping is more likely

(*a*) with unsymmetrical stiffener sections,
(*b*) with increasing curvature,
(*c*) with the free flange of the stiffener in compression, rather than in tension,
(*d*) at positions of maximum bending moment, especially in way of concentrated loads.

Recommended spacing, l, for tripping brackets is summarized in Fig. 7.28 for straight tee stiffeners and for curved tee stiffeners for which R/W is greater than 70.

For curved stiffeners for which R/W is less than 70, Fig. 7.28 may be used by putting $l' = Rl/70W$. For straight unsymmetrical stiffeners l should not exceed $8W$. For curved unsymmetrical stiffeners for which R is more than $4f^2h/t^2$ (unsymmetrical stiffeners should not otherwise be used), $l = (RW/5)^{\frac{1}{2}}$ if the flange is in compression and $(2RW/5)^{\frac{1}{2}}$ if in tension.

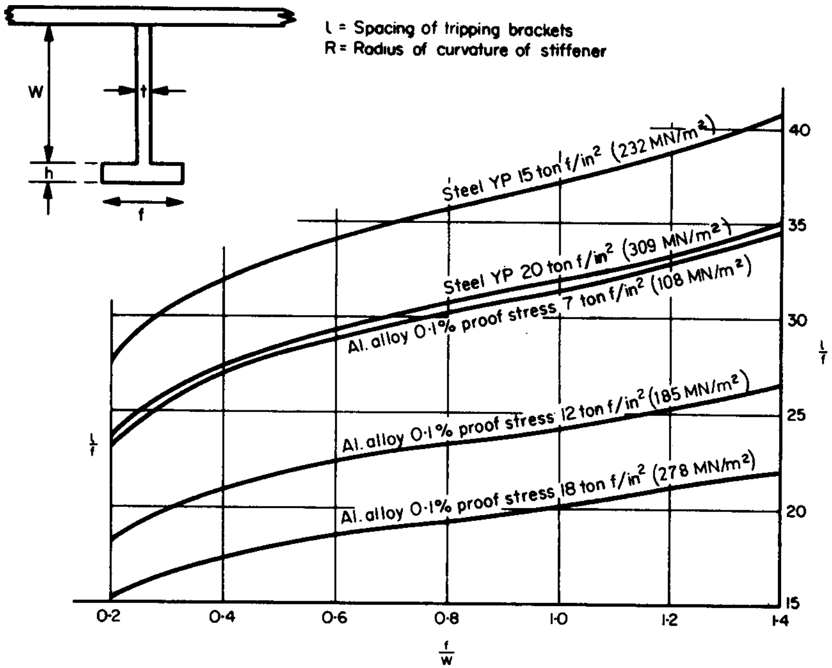


Fig. 7.28 Spacing of tripping brackets

The elastic stability of a circular ring frame under radial and end compression is the basis of important investigations in the design of submarines. Theoretically a perfectly circular ring under radial compression will collapse in a number of circumferential corrugations or lobes. Under additional end loading, a ring stiffened cylinder may collapse by longitudinal corrugation; this load, too, alters the number of circumferential lobes due to radial pressure. Elastic instability of the whole ring stiffened cylinder between bulkheads is also possible. Finally, built-in distortions in a practical structure have an important effect on the type of collapse and the magnitude of the collapse loads. These problems involve lengthy mathematics and will not be pursued in this book. The government of the diving depth of the submarine by consideration of such elastic stability should, nevertheless, be understood.

END CONSTRAINT

The degree of rotational end constraint has more effect on deflections than it has on stresses. End constraints in practical ship structures approach the clamped condition for flexural considerations, provided that the stiffeners are properly continuous at the joint. The degree of rotational end constraint of a member is due to

(a) the stiffness of the joint itself. It is relatively simple to produce a joint which can develop the full plastic moment of the strongest of the members entering the joint; no more is necessary;

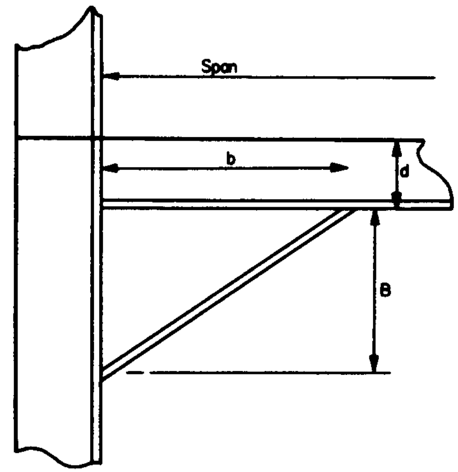


Fig. 7.29

(*b*) the effects of the other members entering the joint. These can be calculated to give the actual rotational stiffness pertaining to the member.

A square joint can provide entirely adequate stiffness. Brackets may be introduced to cheapen fabrication and they also reduce, slightly, the effective span of the member. The reduction in span (Fig. 7.29) is

$$b' = \frac{b}{1 + d/B}$$

EXAMPLE 2. The gantry for an overhead gravity davit comprises, essentially, a vertical deck edge stanchion 10 m high and an overhead member 15 m long sloping upwards at 30 degrees to the horizontal, inboard from the top of the stanchion. The two members are of the same section rigidly joined together and may be considered encastre where they join the ship.

A weight of 12 tonnef acts vertically downwards in the middle of the top member. Compare the analyses of this problem by (*a*) arched rib analysis, (*b*) slope deflection, and (*c*) moment distribution.

Solution (*a*): Arched rib analysis (Castigliano's theorem)

For a point P in AB distance x from A, $\mathrm{BM} = M_0 - Hx\sin 30 - Vx\cos 30 + 12\cos 30[x - 7.5]$, where the term [ ] is included only if positive, i.e.,

$$M = M_0 - \frac{Hx}{2} - V\frac{13x}{15} + 12 \times \frac{13}{15}\left[x - \frac{15}{2}\right]$$

$$\frac{\partial M}{\partial M_0} = 1, \quad \frac{\partial M}{\partial H} = -\frac{x}{2}, \quad \frac{\partial M}{\partial V} = -\frac{13x}{15}$$

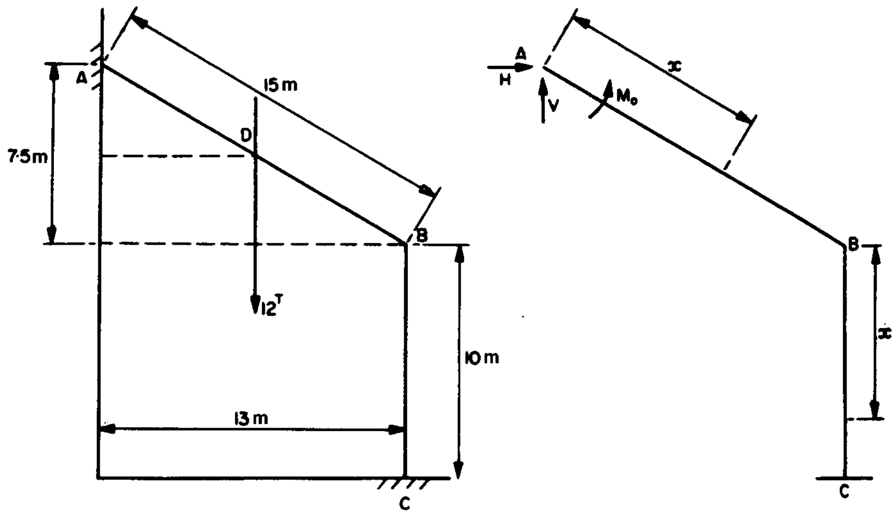


Fig. 7.30

Hence, taking care to integrate the Macaulay's brackets between the correct limits,

$$\int M\frac{\partial M}{\partial M_0}\,\mathrm{d}x = 15M_0 - 56.3H - 97.5V + 292.5$$

$$\int M\frac{\partial M}{\partial H}\,\mathrm{d}x = -56.3M_0 + 281.3H + 487.5V - 1828$$

$$\int M\frac{\partial M}{\partial V}\,\mathrm{d}x = -97.6M_0 + 487.5H + 845.0V - 3169$$

Similarly, for BC,

$$M = M_0 - H\frac{15}{2} - 13V + 78 - Hx$$

$$\frac{\partial M}{\partial M_0} = 1, \quad \frac{\partial M}{\partial H} = -\left(\frac{15}{2} + x\right), \quad \frac{\partial M}{\partial V} = -13$$

Hence, again, by integrating

$$\int M\frac{\partial M}{\partial M_0}\,\mathrm{d}x = 10M_0 - 125H - 130V + 780$$

$$\int M\frac{\partial M}{\partial H}\,\mathrm{d}x = -125M_0 + 1646H + 1625V - 9750$$

$$\int M\frac{\partial M}{\partial V}\,\mathrm{d}x = -130M_0 + 1625H + 1690V - 10{,}140$$

Adding the similar expressions for AB and BC and equating to zero gives three simultaneous equations. whence

$$V = 5.92, \quad H = 1.71, \quad M_0 = 23.4$$

Putting these values back into the expressions for bending moment gives

$$M_D = 21.5, \quad M_B = 11.6, \quad M_C = -5.5$$

The BM diagram can thus be plotted.

A good deal of algebra has, of course, been omitted above.

Solution (*b*): Slope deflection (Fig. 7.31)

$$\text{Moment of free BM diagram about A or B} = \frac{1}{2} \times 39 \times \frac{15^2}{2}$$

$$= 2195 = m_A = m_B$$

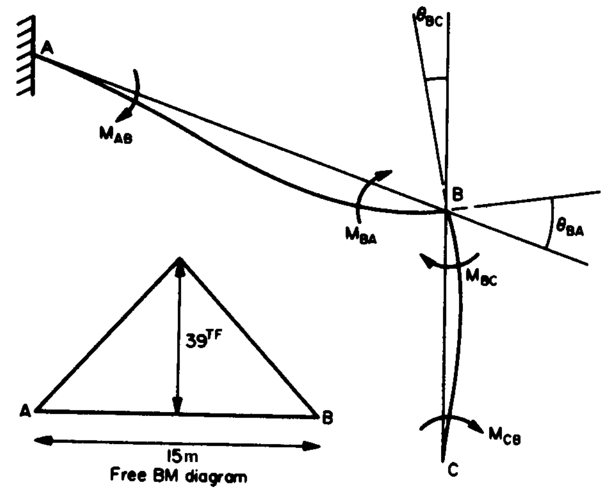


Fig. 7.31

From the slope deflection equations derived earlier:

$$\theta_{AB} = 0 = \frac{15}{6EI}\left(2M_{AB} - M_{BA} + \frac{6 \times 2195}{225}\right)$$

$$\theta_{BA} = -\frac{15}{6EI}\left(M_{AB} - 2M_{BA} + \frac{6 \times 2195}{225}\right)$$

$$\theta_{BC} = \frac{10}{6EI}(2M_{BC} - M_{CB})$$

$$\theta_{CB} = 0 = -\frac{10}{6EI}(M_{BC} - 2M_{CB})$$

Now $\theta_{BA} = \theta_{BC}$. Eliminating θ and solving the simultaneous equations gives

$$M_{AB} = -23.4$$
$$M_{BA} = 11.7 = -M_{BC}$$
$$M_{CB} = -5.9$$

Solution (*c*): Moment distribution, in the manner described in the text

	A M_{AB}	M_{BA}	B M_{BC}	C M_{CB}
Fixing moment $\frac{Wl}{8}$	19.5	−19.5	0	0
Distribution factors $\frac{I/l}{\Sigma I/l}$	—	$\frac{2}{5}$	$\frac{3}{5}$	—
Carry over factors	$\frac{1}{2}$			$\frac{1}{2}$
Fix	19.5	−19.5	0	0
Distribute		7.8	11.7	
Carry over	3.9			5.8
Totals	23.4	−11.7	11.7	5.8

From any of the above solutions the BM diagram can be sketched (Fig. 7.32).

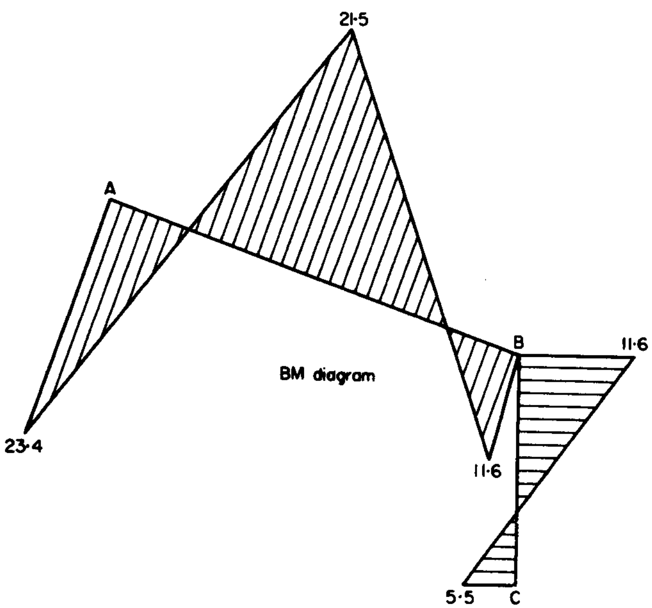


Fig. 7.32

EXAMPLE 3. An engine room bulkhead 10m high is propped on one side by three equally spaced intermediate decks. Stiffeners are uni-directional and, in order to be continuous with longitudinals, must be equally spaced 0.5 m apart.

Once-in-a-lifetime loading of flooding up to 5 m above the compartment crown is judged necessary. Top and bottom may be assumed encastre.

Compare the section moduli of stiffeners and plating thicknesses required by fully elastic and elasto-plastic analyses.

Solution: Assume, for simplicity that the hydrostatic loading is evenly distributed between each deck with a value equal to the value at mid-deck.

$$\text{Pressure A to B} = (5 + 1.25)\,\text{m} \times 1.025\,\frac{\text{tonnef}}{\text{m}^3} = 6.41\,\text{tonnef/m}^2$$

$$\text{Load per metre on each stiffener A to B} = 6.41 \times 0.5 = 3.20\,\text{tonnef/m}$$

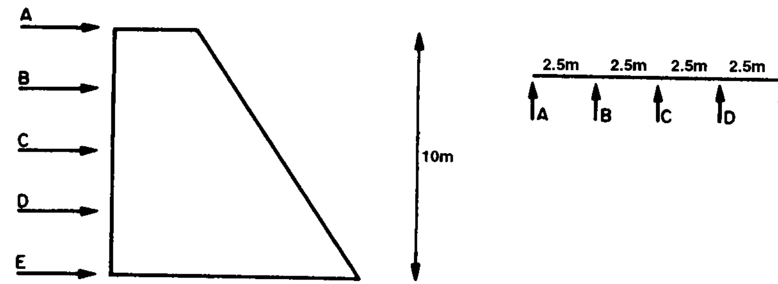


Fig. 7.33

Similarly, pressures and loads on other spans are

	AB	BC	CD	DE
p, press. (tonnef/m^2)	6.41	8.97	11.53	14.09
p', load (tonnef/m)	3.20	4.48	5.77	7.05

Plating thickness: Use large deflection elastic theory, the results of which are given in Fig. 7.13. The aspect ratio $a/b = 2.5/0.5 = 5$ so use curves for a/b = infinity. Maximum stress is at the plate edge given by the addition of σ_2 and σ_3 and it is necessary to use trial values of thickness. For AB,

$$\frac{p}{E}\left(\frac{b}{t}\right)^4 = \frac{6.41}{2.09 \times 10^7}\left(\frac{0.5 \times 100}{0.5}\right)^4 \quad \text{for trial value of } t = 0.5\,\text{cm},$$
$$E = 2.09 \times 10^7\,\text{tonnef/m}^2$$
$$= 31 \text{ for which, with } a/b = \text{infinity},$$

$$\sigma = \frac{2.09 \times 10^7}{(100)^2}(14 + 2) = 3.34\,\text{tonnef/cm}^2$$

For a second trial value of $t = 0.7$ cm, similarly with $(b/t)^4 p/E = 8.0$

$$\sigma = \frac{2.09 \times 10^7}{(100)^2} \times 5 = 1.05\,\text{tonnef/cm}^2$$

The value of σ_3 the membrane tension, is low and is having only a small effect. Adopting small deflection theory therefore, for a yield stress of 2.5 tonnef/cm^2,

$$2.5 = 1.0 \times \frac{6.41}{2 \times 10^4}\left(\frac{50}{t}\right)^2 \quad \therefore \; t = 50\left(\frac{6.41}{2 \times 10^4 \times 2.5}\right)^{\frac{1}{2}} = 0.57\,\text{cm}$$

On small elastic theory, plating thickness in centimetres required are

AB	BC	CD	DE
0.57	0.67	0.76	0.85

Implicit in these figures is an unknown factor of safety over 'failure'. Let us now adopt the elasto-plastic definition of failure and choose, deliberately, a load factor of 2.0. Referring to Fig. 7.14 for span DE,

$$\frac{pE}{\sigma_y^2} = \frac{2 \times 14.09 \times 2.09 \times 10^7}{(2.5 \times 10^4)^2} = 0.94$$

for which

$$\frac{b}{t}\left(\frac{\sigma_y}{E}\right)^{\frac{1}{2}} = 3.30$$

so that

$$t = \frac{1}{3.3}\left(\frac{2.5 \times 10^4}{2.09 \times 10^7}\right)^{\frac{1}{2}} = 0.53\,\text{cm}$$

Carrying out similar exercise for all bays,

Bay	AB	BC	CD	DE
$\dfrac{pE}{\sigma_y^2}$	0.42	0.60	0.77	0.94
$\dfrac{b}{t}\left(\dfrac{\sigma_y}{E}\right)^{\frac{1}{2}}$	5.40	4.70	3.95	3.30
t	0.32	0.37	0.44	0.53

A corrosion margin, say 40 per cent, now needs to be added to give thicknesses suitable for adoption. By this method, a saving has been achieved by adopting deliberate standards which are disguised by the elastic approach. Savings do not always occur, but the advantages of a rational method remain.

To determine the size of stiffeners, assumed constant for the full depth, elastic analysis is probably best carried out by moment distribution. Since the deck heights are equal at 2.5 m each,

$$\text{distribution factors for B, C and D} = \frac{I/l}{\Sigma I/l} = \frac{1}{2}$$

The fixing moments, i.e. the residuals at the ends of AB are due to a mean uniform pressure of 6.41 tonnef/m^2. Hence, with ends held encastre (Fig. 7.2)

$$M_{AB} = -M_{BA} = -\frac{pl^2b}{12} = -\frac{6.41(2.5)^2(0.5)}{12} = -1.67\,\text{tonnef/m}$$

Similarly, due to pressures of 9, 11.5 and 14.1 tonnef/m^2

$$M_{BC} = -M_{CB} = -2.34\,\text{tonnef/m}$$
$$M_{CD} = -M_{DC} = -3.00\,\text{tonnef/m}$$
$$M_{DE} = -M_{ED} = -3.67\,\text{tonnef/m}$$

The resultant moments at joints B, C and D are 0.67, 0.66 and 0.67 tonnef m. Since distribution factors are $\frac{1}{2}$, each member takes approximately 0.33 tonnef m at each joint. Carry-over factors are each $\frac{1}{2}$ so that relaxing the 0.33 tonnef/m moment at end B of AB induces 0.17 tonnef/m moment at A. This process is repeated for end B of BC, end C of BC, etc. The process is then repeated until the residuals at the ends of the members at each joint are substantially the same but of opposite sign, i.e. the joint as a whole is balanced. This process is worked through in the table below.

Distribution factors $\dfrac{I/l}{\Sigma I/l}$	A M_{AB}	B M_{BA} $\frac{1}{2}$	M_{BC} $\frac{1}{2}$	M_{CB} $\frac{1}{2}$	C M_{CD} $\frac{1}{2}$	D M_{DC} $\frac{1}{2}$	M_{DE} $\frac{1}{2}$	E M_{ED}
Fixing moments $\dfrac{p'l^2}{12}$	−1.67	+1.67	−2.34	+2.34	−3.00	+3.00	−3.67	+3.67
Distribute	0	+0.33	+0.33	+0.33	+0.33	+0.33	+0.33	0
Carry over (all factors $\frac{1}{2}$)	+0.17	0	+0.17	+0.17	+0.17	+0.17	0	+0.17
Distribute	0	−0.08	−0.08	−0.17	−0.17	−0.08	−0.08	0
Carry over	−0.04	0	−0.08	−0.04	−0.04	−0.08	0	−0.04
Totals	−1.54	+1.92	−2.00	+2.63	−2.71	+3.34	−3.42	+3.80
Distribute	0	+0.04	+0.04	+0.04	+0.04	+0.04	+0.04	0
Carry over	+0.02	0	+0.02	+0.02	+0.02	+0.02	0	+0.02
Totals	−1.52	+1.96	−1.94	+2.69	−2.65	+3.40	−3.38	+3.82

Maximum bending moment occurs in DE = 3.8 tonnef/m
For a maximum yield stress of 2.5 tonnef/cm^2

$$Z = \frac{I}{y} = \frac{3.8}{2.5} \times 100 = 150\,\text{cm}^3$$

Using a plastic method of analysis, the lowest span, which must be the critical span, will become a mechanism when plastic hinges occur at D, E and near the middle of this span (Fig. 7.33). Taking a load factor of 2.0, approx.

$$M_p = \frac{\frac{1}{2} \times p'l^2}{12} \times 2.0 = \frac{7.05 \times (2.5)^2}{24} \times 2 = 3.67\,\text{tonnef/m}$$

$$\therefore \; S = \frac{3.67}{2.5} \times 100 = 150\,\text{cm}^3$$

If the shape factor is taken as 1.3, Z = 115 cm^3. Thus, some savings occur by adopting plastic design methods for both plating and stiffeners. Because the loading is presumed to occur once in a lifetime, there is no reason why the smaller figures should not be adopted, provided that there are no other criteria of design.

While the theoretical considerations should follow, generally, the elasto-plastic and plastic methods illustrated in this example, there are other features which need to be brought out in the design of main transverse bulkheads. It will often be found economical to reduce scantlings of both stiffeners and plating towards the top, where it is warranted by the reduction in loading. Continuity of bulkhead stiffener, bottom longitudinal and deck girder is of extreme importance and will often dictate stiffener spacings. In ships designed to withstand the effects of underwater explosion, it is important to incorporate thicker plating around the edges and to permit no piercing of this area by pipes or other stress raisers; the passage of longitudinal girders must be carefully compensated; fillet connections which may fail by shear must be avoided.

Finite element techniques

The displacement D of a simple spring subject to a pull p at one end is given by $p = kD$ where k is the stiffness. Alternatively, $D = fp$ where f is the flexibility and $f = k^{-1}$.

If the forces and displacements are not in the line of the spring or structural member but are related to a set of Cartesian coordinates, the stiffness will differ in the three directions and, in general

$$P_1 = k_{11}D_1 + k_{12}D_2$$

and $$P_2 = k_{21}D_1 + k_{22}D_2$$

This pair of equations is written in the language of matrix algebra

$$\mathbf{P} = \mathbf{K}\mathbf{d}$$

P is the complete set of applied loads and d the resulting displacements. K is called the stiffness matrix and is formed of such factors k_{ll} which are called member stiffness matrices (or sub-matrices). For example, examine a simple member subject to loads p_X and p_Y and moments m at each end causing displacements δ_X, δ_Y and θ.

For equilibrium,

$$m_1 + m_2 + p_{Y_2}l = 0 = m_1 + m_2 - p_{Y_1}l$$

also $p_{X_1} + p_{X_2} = 0$

For elasticity,

$$p_{X_1} = -p_{X_2} = \frac{EA}{l}(\delta_{X_1} - \delta_{X_2})$$

From the slope deflection analysis discussed earlier can be obtained

$$m_1 = \frac{6EI}{l^2}\delta_{Y_1} + \frac{4EI}{l}\theta_1 - \frac{6EI}{l^2}\delta_{Y_2} + \frac{2EI}{l}\theta_2$$

These equations may be arranged

$$\begin{bmatrix} p_{X_1} \\ p_{Y_1} \\ m_1 \end{bmatrix} = \begin{bmatrix} \frac{EA}{l} & 0 & 0 \\ 0 & \frac{12EI}{l^3} & \frac{6EI}{l^2} \\ 0 & \frac{6EI}{l^2} & \frac{4EI}{l} \end{bmatrix} \begin{bmatrix} \delta_{X_1} \\ \delta_{Y_1} \\ \theta_1 \end{bmatrix} + \begin{bmatrix} -\frac{EA}{l} & 0 & 0 \\ 0 & -\frac{12EI}{l^3} & \frac{6EI}{l^2} \\ 0 & -\frac{6EI}{l^2} & \frac{2EI}{l} \end{bmatrix} \begin{bmatrix} \delta_{X_2} \\ \delta_{Y_2} \\ \theta_2 \end{bmatrix}$$

i.e. $\mathbf{p} = \mathbf{k}_{11}\boldsymbol{\delta}_1 + \mathbf{k}_{12}\boldsymbol{\delta}_2$

This very simple example is sufficient to show that a unit problem can be expressed in matrix form. It also suggests that we are able to adopt the very powerful mathematics of matrix algebra to solve structural problems which would otherwise become impossibly complex to handle. Furthermore, computers can be quite readily programmed to deal with matrices. A fundamental problem is concerned with the inversion of the matrix to discover the displacements arising from applied loads, viz.

$$\mathbf{d} = \mathbf{K}^{-1}\mathbf{P}$$

Now strains are related to displacements,

$$\boldsymbol{\epsilon} = \mathbf{B}\mathbf{d}$$

For plane strain for example,

$$\boldsymbol{\epsilon} = \begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{zy} \end{bmatrix} = \begin{bmatrix} \frac{\partial u}{\partial x} \\ \frac{\partial v}{\partial y} \\ \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \end{bmatrix} = \mathbf{B}\mathbf{d}$$

and stress is related to strain,

$$\boldsymbol{\sigma} = \mathbf{D}\boldsymbol{\epsilon}$$

D is a matrix of elastic constants which, for plane stress in an isotropic material is

$$\mathbf{D} = \frac{E}{1 - \nu^2}\begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1}{2}(1 - \nu) \end{bmatrix}$$

There are other relationships which are valuable such as the transformation matrix which changes reference axes. These together form the set of tools required for the solution of structural problems using finite element techniques. This brief description can do little more than explain the concept and the student must examine standard textbooks. Finite element analysis is approached broadly as follows:

(a) The structure is divided up by imaginary lines meeting at nodes, forming finite elements which are often triangular or rectangular and plane (but may be irregular and three-dimensional).

(b) For each element, a displacement function is derived which relates the displacements at any point within the element to the displacements at the

nodes. From the displacements strains are found and from the strains, stresses are derived.
(c) Forces at each node are determined equivalent to the forces along the boundaries of the element.
(d) Displacements of elements are rendered compatible with their neighbour's (this is not often totally possible).
(e) The whole array of applied loads and internal forces are arranged to be in equilibrium.

It is not within the scope of this book to describe how this analysis is carried out. It requires a good knowledge of the shorthand of matrix algebra and draws upon the work described in this chapter concerning various unit problems of beams, panels, grillages and frameworks and also the concepts of relaxation techniques and minimum strain energy.

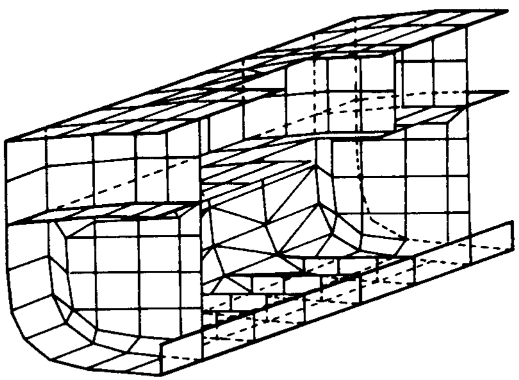


Fig. 7.34

It places in the hands of the structural analyst, a tool of enormous power and flexibility. There is no longer any need to make the assumption of simple beam theory for the longitudinal strength of the ship; the ship girder may now be built up from finite elements (Fig. 7.34) and the effects of the loads applied by sea and gravity determined. Indeed, this is now the basis for the massive suites of computer programs available for the analysis of total ship structure. The effects of the sea spectra are translated by strip theory into loads of varying probability and the effects of those loadings upon a defined structure are determined by finite element analysis. It is not yet a perfect tool. Moreover, it is a tool of analysis and not of design which is often best initiated by cruder and cheaper methods before embarking upon the expense of these programs.

Realistic assessment of structural elements

The division of the ship into small elements which are amenable to the types of analysis presented earlier in this chapter remains useful as a rough check upon more advanced methods. There is now becoming available a large stock of data

and analytical methods which do not have to adopt some of the simplifying assumptions that have been necessary up to now. This is due in large measure to the widespread use of finite element techniques and the computer programs written for them. Experimental work has carefully sifted the important parameters and relevant assumptions from the unimportant, so that the data sheets may present the solutions in realistic forms most useful to the structural designer. Once again the designer has to rely on information derived from computer analysis which cannot be checked readily, so that the wise will need to fall back from time to time upon simple analysis of sample elements to give confidence in them.

As explained in Chapter 6, the elemental behaviour of the whole ship cross section may be integrated to provide a knowledge of the total strength. Judgement remains necessary in deciding what elements should be isolated for individual analysis. This judgement has been assisted by extensive experimentation into box girders of various configurations under end load. Figure 7.35 shows the cross section of a typical specimen.

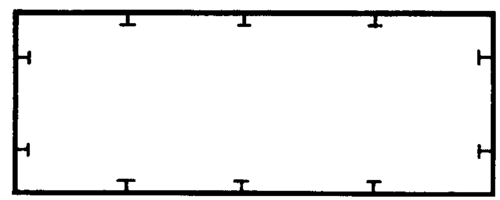


Fig. 7.35

The wisdom of generations of ship designers has steadily evolved a structure which:

(a) has more cross sectional area in the stiffener than in the plating;
(b) has longitudinals more closely spaced than transverses;
(c) favours quite deep longitudinals, preferably of symmetrical cross section.

Such structures tend to provide high collapse loads in compression and an efficient use of material. What happens as the load is steadily increased is firs,," buckling of the centres of panels midway between stiffeners. Shirking ofload by the panel centres throws additional load upon each longitudinal which, as load is further increased, will finally buckle in conjunction with the strip of plate to which it is attached. This throws all of the load upon the 'hard' corners which are usually so stiff in compression that they remain straight even after plasticity has set in. It has been found that these hard corners behave like that in conjunction with about a half the panel of plating in each direction. Thus the elements into which the box girder should be divided are plating panels, longitudinals with a strip of plating and hard corners.

Finite element analysis of these elements is able to take account of two factors which were previously the subject of simplifying assumptions. It can account for built-in manufacturing strains and for initial distortion. **Data** sheets or standard programs are available from software houses for a very wide range of geometry and manufacturing assumptions to give the stress-strain (or more correctly the load shortening curves) for many elements of structure. These can be integrated into total cross sectional behaviour in the manner described in Chapter 6.

By following the general principles of structural design above-i.e. close deep longitudinals, heavy transverse frames and panels longer than they are wide-two forms of buckling behaviour can usually be avoided:

(a) Overall grillage buckling of all plating and stiffeners together; this is likely only when plating enjoys most of the material and stiffeners are puny.
(b) Tripping of longitudinals by sideways buckling or twisting; this is likely with stiffeners, like flat bars which have low torsional stiffness.

The remaining single stiffener/plating behaviour between transverse frames may now be examined with varying material geometry and imperfections.

Data sheets on the behaviour of panels of plating shown in Figs 7.36 and 7.37 demonstrate firstly the wisdom of the general principles enunciated above and secondly how sensitive to imperfections they are. With moderate initial distortion and average built-in strains some 50-80 per cent of the theoretical yield of a square panel can be achieved (Fig. 7.36). Long panels as in transversely framed ships (Fig. 7.37) achieve only 10 per cent or so.

Data sheets for stiffened plating combinations are generally of the form shown in Fig. 7.38. On the tension side the stress/strain relationship is taken to be of the idealized form for ductile mild steel. In compression, the effects of progressive buckling of flanges is clearly seen. Sometimes the element is able to hold its load-bearing capacity as the strain is increased; in other cases the load-bearing capacity drops off from a peak in a form called catastrophic buckling.

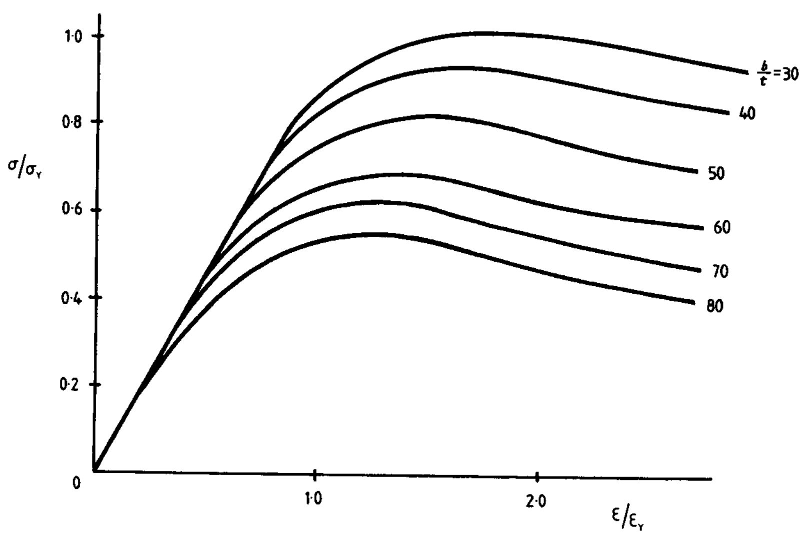


Fig. 7.36

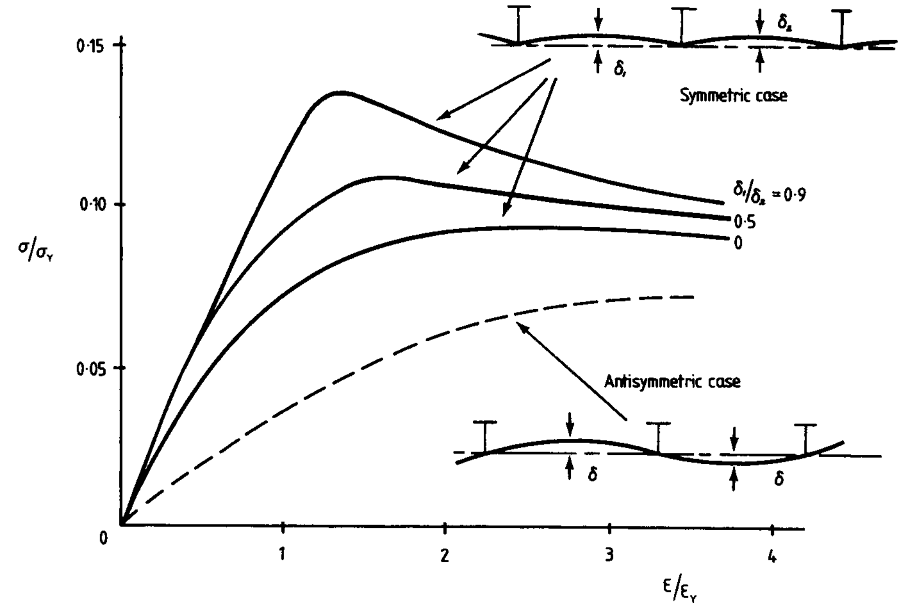


Fig. 7.37

Because the finite element analysis is able to account for the separate panel buckling the width of the associated plating may be taken to the mid panel so that the 30t assumption is no longer needed. Moreover, many of the elemental data sheets now available are for several longitudinals and plating acting together.

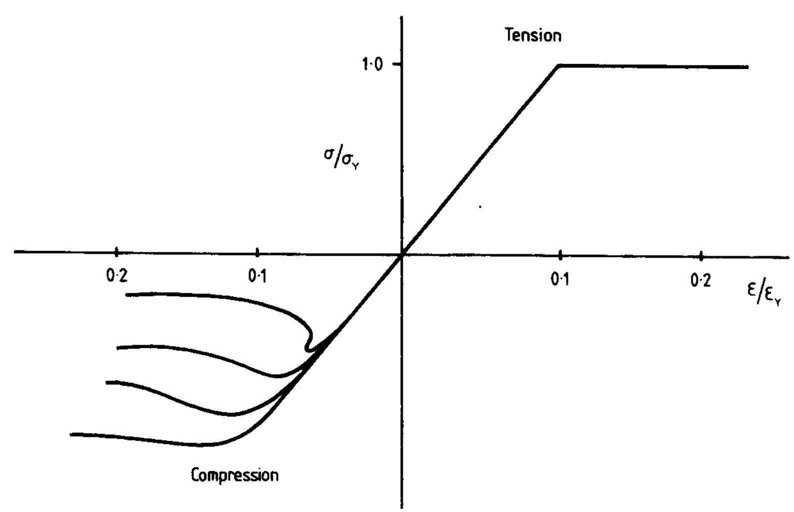


Fig. 7.38

Fittings

CONTROL SURFACES

Control surfaces in a ship are those which, through hydrodynamic lift and drag forces, control the motion of the ship. They include rudders, stabilizers, hydroplanes, hydrofoils and tail fins. The forces and their effects on the ship are discussed at length in other chapters. In this chapter, we are concerned with the strength of the surface itself, the spindle or stock by which it is connected to operating machinery inside the ship and the power required of that machinery.

Those control surfaces which are cantilevered from the ship's hull are the most easily calculable because they are not redundant structures. The lift and drag forces for ahead or astern motions (given in Chapter 13) produce a torque T and a bending moment M at section Z of the stock. If the polar moment of inertia of the stock is J, the maximum shear stress is given by

$$\tau_{\max} = \frac{D}{2J}(M^2 + T^2)^{\frac{1}{2}}$$

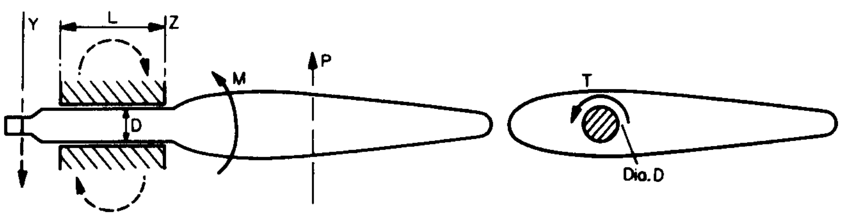


Fig. 7.39 Forces and moment on a control surface

Permissible maximum stresses are usually about 45 MN/m^2 or MPa for mild steel and 70 MPa for best quality forged steel. At section Y of the stock, torque required for the machinery to which the stock is subjected is augmented by the friction of the sleeve, whence

$$T^1 = T + P\mu\frac{D}{2}$$

The value of the coefficient of friction μ is about 0.2. The power required of the machinery is given by T^1 multiplied by the angular velocity of rotation of the surface: for a stabilizer fin this might be as high as 20 deg/s, while for a rudder it is, typically, 2–3 deg/s.

When there are pintle supports to the rudder, the reactions at the pintles and sleeve depend upon the wear of the bearings that is permitted before renewal is demanded. The effects of such wear may be assumed as given in Fig. 7.40.

Permitted bearing pressure at the sleeve which is usually about 12 mm thick is about 10 MPa

$$\frac{R}{L(D+2t)} \text{ should not exceed } 10\,\text{MPa}$$

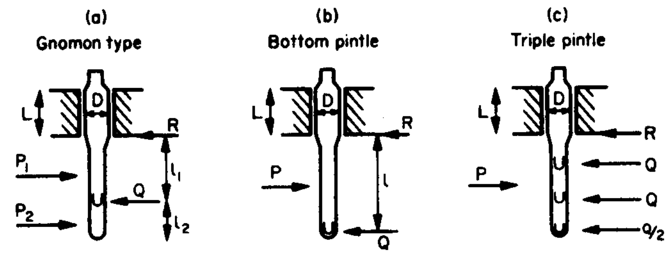


Fig. 7.40 Forces assumed acting on rudders

At the pintles, classification societies take about the same bearing pressure.

$$\frac{Q}{ld} \text{ should not exceed } 40\,\text{MPa}$$

The shearing stress in the steel pintle due to Q should not exceed the figures used for the stock itself. Structure of the rudder is analysed by conventional means, the stock continuation being regarded as a beam subject to the loads of Fig. 7.40. Because erosion can be severe the acceptable stresses are low.

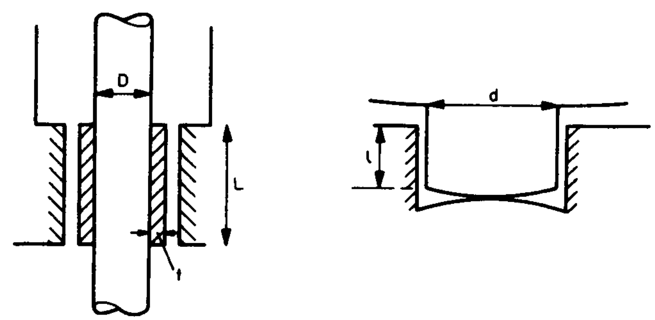


Fig. 7.41

Problems

1. A bottom longitudinal is in an area subject to an end compressive stress of 100 N/mm^2 and is at a still water draught of 4 m. With a width of steel plating of 28 cm assumed to be fully effective, estimate the maximum stress under the end load and a lateral load due to twice the draught. Spacing of longitudinals is 36 cm and 1 m (simply supported), plating thickness is 4 mm and the T bar stiffener has the following properties

flange (cm)	depth (cm)	c.s. area (cm^2)	MI (toe) (cm^4)	MI (N.A.) (cm^4)
4	8	5.0	196.0	30.0

2. A transverse floor in a double bottom is 53 cm deep and 1.6 cm thick. Adjacent floors are 3 m apart and the outer bottom is 1.9 cm thick and the

inner bottom is 0.8 cm thick. Each floor is supported by longitudinal bulkheads 4 m apart and is subjected to a uniformly distributed load from the sea.

Calculate the effective modulus of section.

3. A tubular strut of length 3 m external diameter 8 cm cross-sectional area 16 cm^2 and moment of inertia about a diameter 150 cm^4 when designed for an axial working load of 0.1 MN may be assumed to be pin-ended. It has an initial lack of straightness in the form of a half sine wave, the maximum (central) departure from the straight being 8 mm; find *(a)* the ratio of the maximum working stress to the yield stress of 270 MPa and *(b)* the ratio of the working load to the load which would just cause yield (E = 209 GN 1m^2).
4. A square grillage is composed of six equally spaced beams of length 3.08 m, three in each direction. The beams are fixed at their ends and are each acted on by a uniformly distributed load of 6.67 tonnef/m throughout the length. The central beam in each direction has a moment of inertia of 1498 cm^4. The others have the same moment of inertia which need not be that of the central beams; what should this moment of inertia be in order that no beam shall exert a vertical force on another? If, in fact, each of these four beams has a moment of inertia of 749 cm^4, what is the central deflection?
5. Four girders, each 12m long, are arranged two in each of two orthogonal directions to form a square grillage of nine equal squares. The ends of all girders are rigidly fixed and the joints where the girders intersect are capable of transmitting deflection and rotation. All four members are steel tubes with polar MI of 0.02 m^4.

   Find the deflection at an intersection when a uniformly distributed load of 2 MN/m is applied along each of two parallel girders and the two other girders are unloaded. (Modulus of rigidity is [illegible] of Young's modulus.)
6. Calculate by the plastic design method, the 'collapse' load of a grillage in which all members are tubes of mean diameter 203 mm, thickness 3.2 mm and of steel of yield point 247 MPa. The grillage is 9.75 m x 6.40 m consisting of three members 6.40 m long spaced 2.44 m apart in one direction, intersecting at right angles two members 9.75m long spaced 2.13m apart. The ends of all members are fixed. At each joint there is an equal concentrated load perpendicular to the plane of the grillage. Ignore torsion.
7. Three parallel members of a grillage are each 4 m long and equally spaced 2 m apart. They are intersected by a single orthogonal girder 8 m long. All ends are simply supported and each member has a second moment of area of 650 cm^4. Each member is subject to a uniformly distributed load of 15000 N/m. Calculate the maximum deflection. Neglect torsion. E = 209 GPa.
8. A long flat panel of plating is 1016mm wide and subject to a uniform pressure of 0.173 MPa. What thickness should it be

   (a) so that it nowhere yields,
   (b) to permit a membrane stress of 144 MPa, E = 209 GPa, υ = 0.3 and yield stress = 216 MPa
9. An alloy steel plate 1.50m x O.72m has an initial bow due to welding of 3 mm in the middle. It is 6 mm thick and made of steel with a Y.P. of 248 MPa.

What pressure would this plate withstand without causing further permanent set? If the plate had been considered initially flat, what pressure would small deflection theory have given to have caused yield? What would have been the deflection?

10. The yield stress of an alloy plate is 463 MPa and Young's modulus is 124 GPa. The plate is 1524 x 508 x 5mm thick and its edges are constrained against rotation. Calculate the approximate values of

    (a) the pressure first to cause yield;
    (b) the elasto-plastic design pressure assuming edges restrained from inward movement;
    (c) the design pressure assuming edges free to move in and a permanent set coefficient of 0.2;
    (d) the pressure which the plate could withstand without further permanent set if bowed initially by 2.5 mm, the edges assumed restrained.
11. A boat is supported from the cross bar of a portal frame at points 1.83 m from each end. The vertical load at each point of support is 2 tonnef. The portal is 3.05m high and 9.14m long and is made of thin steel tube 203mm in diameter and 3.2 mm thick; the steel has a yield point of 278 MPa.

    If both feet are considered encastre, find the load factor over collapse in the plane of the portal. Ignore the effects of axial load.

    Find also, by a moment distribution method, the ratio of the maximum working stress to the yield stress.
12. When a temporary stiffener is welded diametrically across a completed ring frame of a submarine, bad fitting and welding distortion cause a total diametral shrinkage of 8 mm.

    The mean diameter of the steel ring is 6.1 m. The MI of the frame section is 5000 cm^4 and the least section modulus of the ring is 213 cm^3. What is the greatest stress induced in the frame?

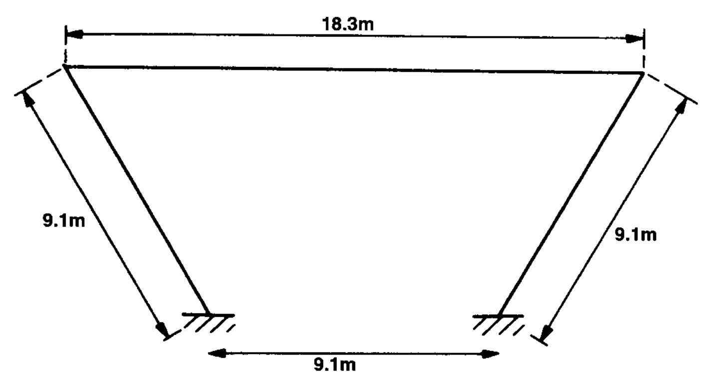


Fig. 7.42

13. Due to careless building, the ring frame shown in Fig. 7.42 is left unsupported while being built into a ship. The deck beam is a 203 mm. T bar of

mass *14.9kgjm* run with an MI of 1415cm^4, while the ribs are 152mm. T bars of 11.9*kgjm* run with an MI of 708 cm^4. Members are straight and the ribs rigidly attached to the double bottom. The neutral axes of section for the deck beams and ribs are respectively 152mm and 114mm from the toe.

Assuming that the frame does not buckle out of its plane, find the maximum stress due to its own weight.

14. Light alloy superstructure above the strength deck of a frigate may be treated as a symmetrical two-dimensional single storey portal as shown. The MI of AD, AE and DH is 5,000cm^4 and of BF and CG is 2330cm^4. All joints are rigid except F and G where the elasticity of the deck causes carry-over factors of ~ from B to C. Find the bending moment distribution in the structure when a load of *19,200Njm* is carried evenly along AD.

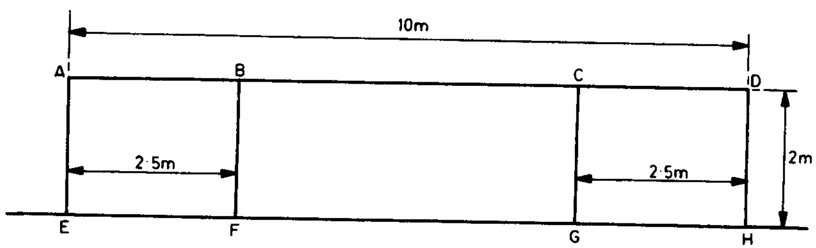


Fig. 7.43

15. The structure of a catapult trough is essentially of the shape shown. G and H are fixed rigidly but the rotary stiffnesses of A and F are such as to allow one-half of the slope which would occur were they to be pinned.

The mid-point of CD is subjected to a vertical force of 122 tonnef as shown.

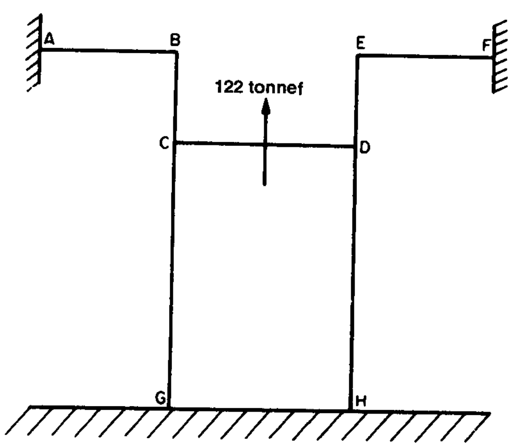


Fig. 7.44

Calculate the distribution of bending moment in the structure both by moment distribution and slope deflection analysis,

Members	AB, EF	BC,ED	CG,DH	CD
Length (m)	0.91	0.61	1.83	1.22
I(cm^4)	7491	5827	2497	6660

16. Calculate the collapse load of a rectangular portal having 3 m stanchions and a 8 m cross bar all made of tubes 20 em in diameter, 3 mm thick and having a yield stress of 250 MPa.

The lower feet are pinned and the upper corners rigid. Equal loads are applied horizontally inwards at the middle of each stanchion and vertically downwards at the middle of the crossbar. (Neglect the effect of end compression on the plastic moment of resistance.)

17. The balanced rudder shown (Fig. 7.45) has a maximum turning angle of 35 degrees and is fitted immediately behind a single propeller. What torque and bending moment are applied to the rudder stock at the lower end of the sleeve bearing when the rudder is put fully over at a ship's speed of26 knots?

In the force equation, take the constant $k = 0.100$. Also if the length of an elemental strip of the rudder surface, drawn at right angles to the centre line of the stock is *l*, then assume the centre of pressure of the strip to be *0.321* from the leading edge.

18. Limiting the shear stress to 69.5 MPa, what should be the diameter of the stock, if solid, of the previous question? If the coefficient of friction at the sleeve is 0.20 and the rate of turn at 35 degrees is 3 degrees per second, what power is required of the steering gear?

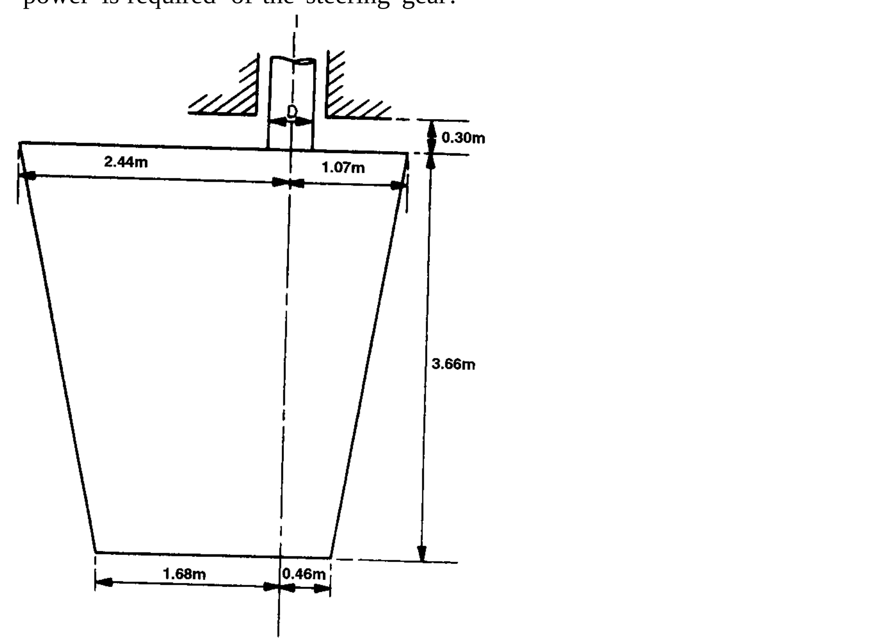


Fig. 7.45

8 Launching and docking

Few who have witnessed the launch of a vessel will have failed to experience the awesome nature of the event. The naval architect is responsible for the safe movement from land to water for the first time of vessels sometimes of thousands of tonnes. Predictions of the movement are vital to its safe control. The reverse procedure, of bringing a ship into dock also requires some sound practical knowledge assisted by theory.

Launching

A ship is best built under cover for protection from the elements leading to better quality and less disruption of schedules. Because of the capital costs it was not until the 1970s that covered slipways became fairly common. Partly to offset these costs a conveyor belt method of construction was pioneered in Scandinavia. The ship is built section by section and gradually pushed from the building hall into the open. A dry dock can be used and flooded up when the ship is sufficiently far advanced. A dry dock is too valuable a capital asset to be so long engaged and the method is generally confined to very large vessels where normal launching presents special difficulties. Occasionally, slipways are partially or wholly below high water level and the site is kept dry by doors which, on launch day are opened to let the water in and provide some buoyancy to the hull. More often, the hull is constructed wholly above water and slid into the water when ready. Very small hulls may be built on a cradle which is lowered on wheels down a ramp into the water, but ships over about 100 tonnef displacement must be slid on greased ways under the action of gravity.

Usually, the ship slides stern first into the water because this part of the ship is more buoyant, but bow first launches are not unknown. The theory presently described is applicable to both. Less often, the ship is slid or tipped sideways into the water and the considerations are then rather different.

In a well ordered, stern first launch, sliding ways are built around the ship, and shortly before the launch the gap separating them from the fixed groundways is filled by a layer of grease to which the weight of the ship is transferred from the building blocks.

There may be one, two or even four ways; generally, there are two although in Holland one way with propping ways at the sides are common. The ways are inclined and, often, cambered. Movement is prevented until the desired moment by triggers which are then knocked away to allow the ship to move under gravity down the inclined ways. After it begins to enter the water, buoyancy builds up at the stern until it reaches a value sufficient to pivot the entire ship about the fore poppet (i.e. the forward end of the launching cradle). The

ship continues down the ways until it is floating freely having slid away from. or dropped off the end of, the ways. If launched in a restricted waterway, its progress into the water is impeded by drags which are arranged so as gradually to bring the vessel to a stop before it strikes the far bank.

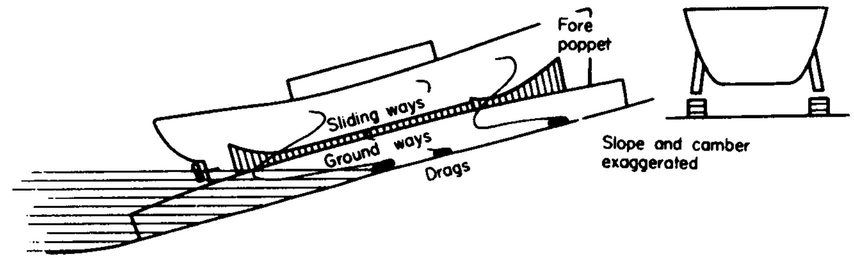


Fig. 8.1 A stern first launch

Consider what might go wrong with this procedure. The grease might be too slippery or not slippery enough. It might be squeezed out by the pressure. Instead of the stern lifting, the ship might tip the wrong way about the end of the groundways and plunge. The forefoot might be damaged by dropping off the end of the ways or it may dig into the slip when the stern has lifted. The ship might be insufficiently strong locally or longitudinally or the ways may collapse. The breaking effect of the drags might be too much or too little. The ship might, at some instant, be unstable.

Calculations are carried out before arranging the launch to investigate each one of these anxieties. The calculations predict the behaviour of the ship during launch, to enable a suitable high tide to be selected and to arrange the ship and slip to be in a proper and safe condition.

LAUNCHING CURVES

A set of six curves is prepared to predict the behaviour of the ship during launch. They are curves, plotted against distance of travel down the slip, of ,

(i) Weight, W
(ii) Buoyancy, ~
(iii) Moment of weight about fore poppet, Wa
(iv) Moment of buoyancy about fore poppet, d
(v) Moment of weight about after end of groundways, Wb
(vi) Moment of buoyancy about after end of ground ways, c

The geometry of the ship at any position in its travel down the ways is illustrated in Fig. 8.2 for the ways in contact. A distributed load $(W -)$ along the ways is not shown; after the stern lifts this becomes concentrated all at the fore poppet. Conventionally, the moment (v) is drawn positive when anti-clockwise and (vi) is drawn positive when clockwise. A typical set of launching curves is shown in Fig. 8.3.

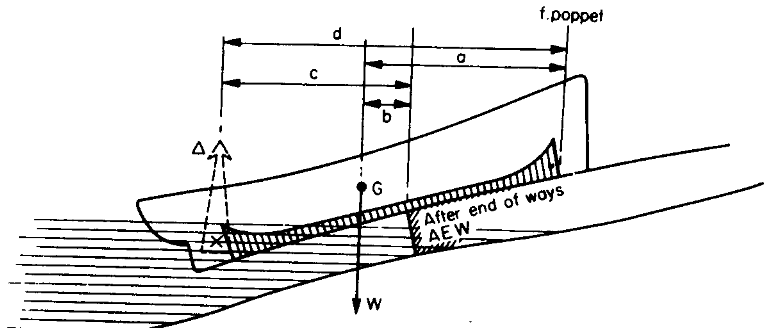


Fig. 8.2 *Ship and way geometry*

The important features of these curves are as follows:

(a) at the point at which the moment of buoyancy about the fore poppet equals the moment of weight about the fore poppet, the stern lifts;
(b) the difference between the weight and buoyancy curves at the position of stern lift, is the maximum force on the fore poppet;
(c) the curve of moment of buoyancy about the after end of the ways must lie wholly above the curve of moment of weight; the least distance between the two curves of moment about the after end of ways, gives the least moment against tipping about the end of ways;
(d) crossing of the weight and buoyancy curves before the after end of ways, indicates that the fore poppet will not drop off the end of the ways.

These curves answer some of the anxieties about the launch directly, but certain other investigations are necessary. From the difference between the weight and

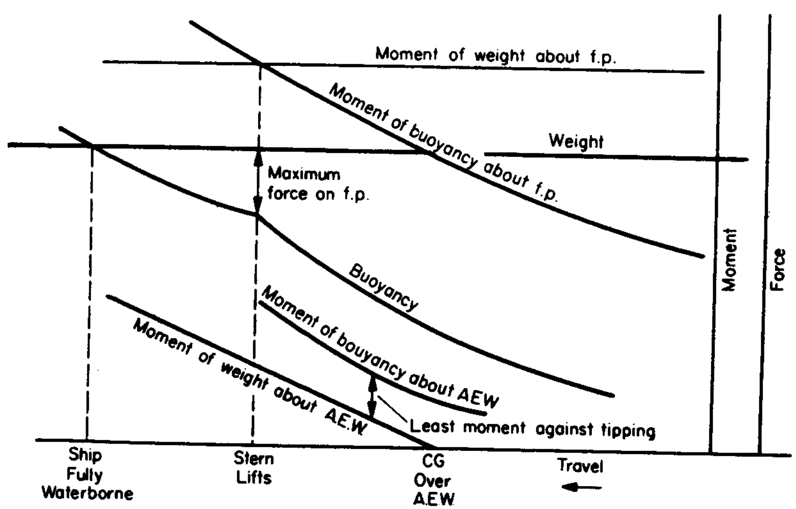


Fig. 8.3 Typical launching curves

buoyancy curves before stern lift, the grease pressures can be determined. **This** difference at stern lift (and for large vessels, it could well exceed 10,000 tonne!) giving the maximum force on the fore poppet, enables the poppet and internal strengthening of the ship to be devised; it also causes a loss of stability. From all of these features is judged the adequacy of the height of tide and the length of ways. Indeed, curves are usually constructed for more than one height of water to determine the minimum acceptable height. How are the curves constructed?

CONSTRUCTION OF LAUNCHING CURVES

The curve of weight results usually from the weights weighted into the ship plus an estimate of what remains to be built in during the period between calculation and launch. Centre of gravity position is similarly estimated and the two moment of weight curves produced.

Buoyancy and centre of buoyancy at any position of travel is determined by placing the relevant waterline over a profile of the ship with Bonjean curves drawn on and integrating in the usual fashion. For this calculation, dynamic considerations are ignored and the ship is assumed to travel very slowly. While the correct waterline can be determined by sliding a tracing of the ship over a drawing of the slip, it is more accurately found from the geometry of the situation. Let

a = the initial slope of the keel
L = length between perpendiculars
e = distance of the fore poppet abaft the FP
h = initial height of the fore poppet above water
j3 = declivity of groundways, i.e. slope of chord
f = camber of ways of length K
r = radius of camber
t = distance abaft the fore poppet

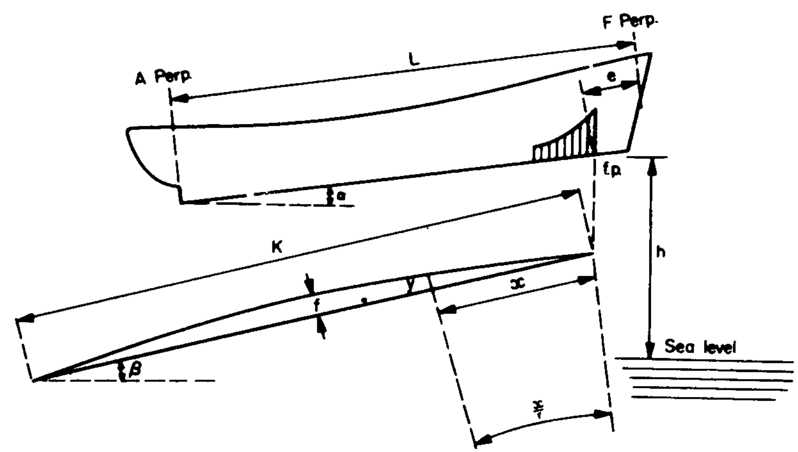


Fig. 8.4

From simple mensuration, the camber of an arc of a circle is given by

$$f=\frac{K^2}{8r}\quad\text{so that}\quad(f-y)=\frac{(K-2x)^2}{8r}$$

$$\therefore\ y=\frac{K^2}{8r}-\frac{(K-2x)^2}{8r}=x\frac{(K-x)}{2r}$$

After travelling a distance x, the fore poppet is raised y above the chord and the keel has moved through an angle x/r. The height of the fore poppet above the water is now therefore, approximately

$$h-\beta x+y=h-\beta x+\frac{x}{2r}(K-x),$$

while the height above water of a point t abaft the fore poppet is

$$h-\beta x+y-t\left(\alpha+\frac{x}{r}\right)=h-\beta x+\frac{x}{2r}(K-x)-t\left(\alpha+\frac{x}{r}\right)$$

If there is no camber, r is, of course, infinite.

If, in this expression t is put equal to $-e$, it will give the height of the keel at the FP above water or, when it is negative, below water, i.e. the draught at the FP. If t is put equal to $L-e$, the expression gives the negative draught at the AP. Thus, for a given travel down the ways x, a waterline can be drawn accurately on the Bonjean profile from which buoyancy and centre of buoyancy can be determined.

This is satisfactory when the ways are fully in contact, but after stern lift, while the draught at the fore poppet can be determined in this way, trim cannot. Now at all times after stern lift the moments of weight and buoyancy about the fore poppet must be equal. For the position of travel under consideration, the fore poppet draught is found as already described; buoyancy and moment of buoyancy are then calculated for several trims about the fore poppet and a curve plotted as shown in Fig. 8.5. Where the curves of moment about fore poppet cut, there is the correct trim; buoyancy can be read off and the position of centre of buoyancy calculated. Having determined the correct trim, the passage of the forefoot to assess the clearance from the slip can be drawn.

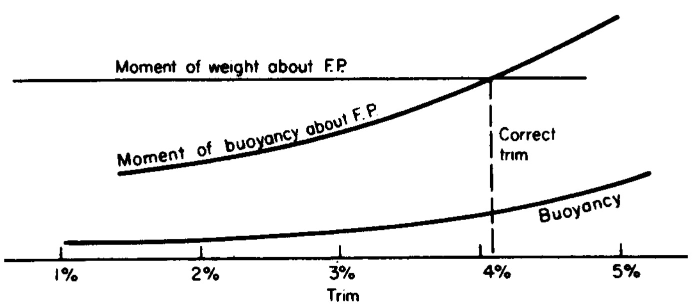


Fig. 8.5 Determination of correct trim after stern lift

If the launching curves indicate that the fore poppet will drop off the ways, adequate depth of water must be allowed under the forefoot for the momentum of the drop to prevent damage to the forefoot. What is adequate will depend on the amount of the excess of weight over buoyancy, the inertia of the ship and the damping effects of the water; it is best determined from an examination of previous launches.

G ROUNDW A YS

Typically, the declivity of the ground ways is I in 20 and the camber a half a metre in a groundway length of 300 m from fore poppet to after end. The radius corresponding to this camber is 22,500 m. Originally, camber was probably meant to offset the sinkage of the slip as the ship's weight grew. It has another important effect in rotating the ship to dip the stern deeply into the water; this increases the buoyancy force and causes an earlier stern lift than would be the case without camber. This increases the moment against tipping but also increases the load on the fore poppet.

The total load on the groundways is the difference between weight and buoyancy *W* - ~ Dividing by the length in contact gives a mean load per unit length and this, divided by the width of ways gives a mean pressure. The maximum total load on the ways is the initial one, *W,* and experience has shown that the mean pressure associated with this load should not, for many greases, exceed about 27 tonnef 1m² or the grease tends to get squeezed out. Both the ship, which has an uneven weight distribution, and the ways are elastic and they are separated by grease; what is the true distribution of pressure along the length has never been measured. Ignoring the moment causing the small angular acceleration of the ship due to camber, it is known that, before stern lift (Fig. 8.2),

(a) *W* - ~ = total load on ways,
(b) *Wa* - *d* = moment of way load about fore poppet.

Some appreciation of the load distribution at any instant can be obtained by assuming that the distribution is linear, i.e. that the curve ofload per unit length *p'* against length is a trapezoid. If the length of ways remaining in contact is *l* at any instant and the loads per unit length at the fore poppet and after end iJ.tt ways are, respectively, *Pr* and *P*;~ the conditions *(a)* and *(b)* become

$$(a)\ \ W-\Delta=\tfrac{1}{2}l(p'_{\mathrm{f}}+p'_{\mathrm{a}})$$

$$(b)\ \ Wa-\Delta d=p'_{\mathrm{f}}\frac{l^2}{2}+\frac{1}{3}l^2(p'_{\mathrm{a}}-p'_{\mathrm{f}})$$

Solving these equations for p'_{f} and p'_{a},

$$p'_{\mathrm{f}}=4\frac{W-\Delta}{l}-6\frac{Wa-\Delta d}{l^2}$$

$$p'_{\mathrm{a}}=6\frac{Wa-\Delta d}{l^2}-2\frac{W-\Delta}{l}$$

This is a satisfactory solution while p_f' and p_a' are positive, and the load per unit length can be represented by a trapezium. When $(Wa - \Delta d)/(W - \Delta)$ is greater than $\frac{2}{3}l$, p_f' becomes negative and when $(Wa - \Delta d)/(W - \Delta)$ is less than $\frac{1}{3}l$, p_a' becomes negative. There cannot be a negative load so for these conditions, the trapezoidal presumption is not permissible. It is assumed, instead, that the distribution is triangular whence, for

$$\frac{Wa - \Delta d}{W - \Delta} > \frac{2}{3}l \quad \text{then} \quad p_a' = \frac{2(W - \Delta)^2}{3l(W - \Delta) - 3(Wa - \Delta d)}$$

and for

$$\frac{Wa - \Delta d}{W - \Delta} < \frac{1}{3}l \quad \text{then} \quad p_f' = \frac{2(W - \Delta)^2}{3(Wa - \Delta d)}$$

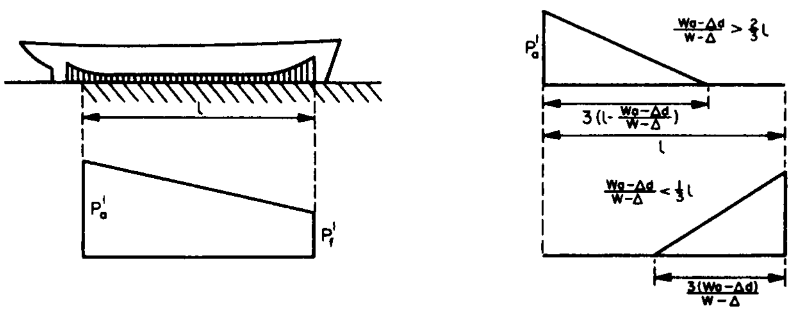


Fig. 8.6 Assumed load distribution

It is now possible to plot the maximum loads or pressures against travel down the slip (Fig. 8.7). Permissible grease pressures vary with grease, temperature and past experience. 55 tonnef/m^2 is typical but figures of 100 or 150 are not unknown. The calculation helps determine poppet strength and where internal shoring is needed.

Apart from the assumption of linear variation in pressure the above method of calculation assumes the ship and slipway to be rigid. It has been suggested

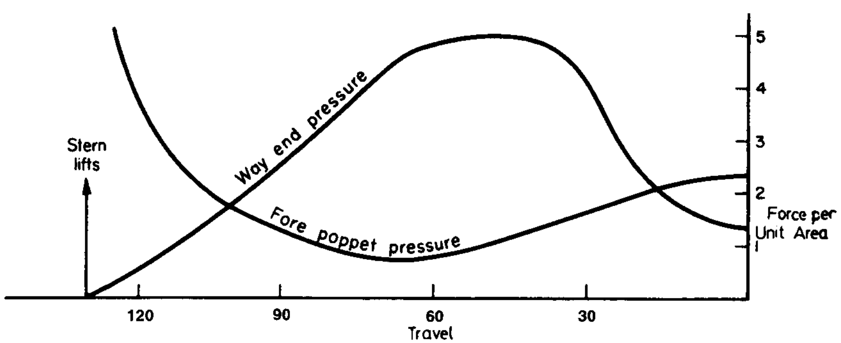


Fig. 8.7 End pressures

that the simple method predicts way end pressures which are unrealistically low and a more rigorous treatment allowing for ship elasticity, flexibility of the ways and the effects of bottom panel deflection may be wise. Pressures may be reduced by champfering the ends of the ways and may be physically limited by using deformable packing or collapsible cushions.

THE DYNAMICS OF LAUNCHING

The force accelerating the ship down the groundways is, at any instant, approximately

$$(W - \Delta)\theta - \text{way friction} - \text{water resistance} - \text{drag forces}$$

θ is the slope of the ways at the centre of gravity of the ship. Way friction is $\mu(W - \Delta)$; the coefficient of friction μ is usually less than 0.02, although at the commencement it can be slightly higher and tests should be conducted to establish the figures for the particular lubricant over a range of temperatures. Water resistance is due to the hull friction, the creation of the stern wave and to the resistance of locked propellers, water brakes or masks, etc., where fitted; this resistance is expressed by $K\Delta^{2/3}V^2$ where V is the velocity of travel and K is a constant determined from similar ships with similar water braking devices. Retarding forces due to chain drags are found to follow a frictional law, $\mu' w$ where w is the weight of the chains and μ' is 0.40–0.80 depending upon the state of the slipway; this figure must be determined from trials or from previous launches.

For a particular ship the effects of entrained water can be expressed as a fraction z of the buoyancy. The equation of motion of the ship before it is waterborne is then

$$(W - \Delta)\theta - \mu(W - \Delta) - K\Delta^{\frac{2}{3}} V^2 - \mu' w = \text{net force} = \frac{(W + z\Delta)}{g}\frac{dV}{dt}$$

This differential equation cannot, of course, be solved mathematically because of the presence of ~

Consideration of each of the factors at intervals of travel down the slip, however, enables a component force diagram to be built up as shown in Fig. 8,1:- and a distance-time relationship estimated from it, by equating the nett force to the mass x acceleration. After the ship has become waterborne, the first two components of the expression become zero. Integration of each component force-distance curve gives the work done in overcoming that resistance. Velocity at any point of travel may therefore be checked by relating the kinetic energy at that point to the loss of potential energy minus work done in overcoming friction and resistance.

STRENGTH AND STABILITY

Problems of local strength occur at the keel over the after end of the ways and at the fore poppet both for the ship and for the poppet structure. The difficulties of containing the large forces at the fore poppet account for the increasing

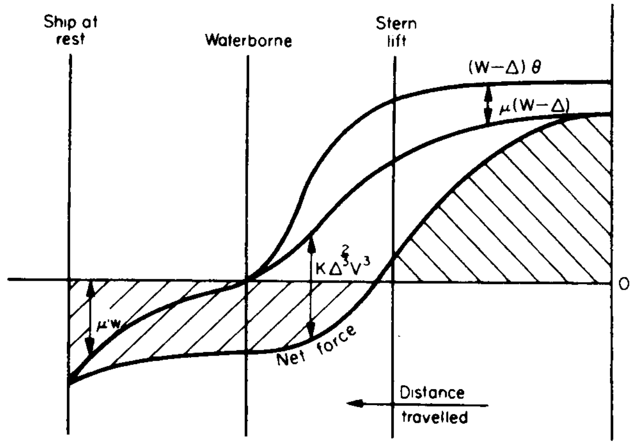


Fig. 8.8 Forces acting during launch

number of very large oil tankers and other bulk carriers being built in dry docks (productivity is of course high).

At the instant of stern lift, the hull girder undergoes a maximum sagging bending moment, the longitudinal strain from which in some large ships has been measured to give stresses of 90 MN/m^2. The bending strain is easily found from the data already compiled to ensure a safe launch but strain measured in practice is usually somewhat below this, due probably to the effects of water resistance. Breakage, that is the longitudinal bending of the ship, is usually measured.

Stability is also at a minimum at the instant of stern lift due to the large fore poppet load at the keel. The $\overline{GM}$ is calculated by the methods described in Chapter 5 both for the ship with the concentrated load at the keel and fully afloat. While instability with the ship supported each side by the ways is unlikely, no naval architect would permit a ship to be launched with a negative virtual $\overline{GM}$ at the instant of stern lift. Any sudden sinking of the ways, for example, could be increased by an unstable ship. Ballasting is, in any case, often carried out to increase the longitudinal moment against tipping and presents no difficulty if needed for stability purposes.

SIDEWAYS LAUNCHING

When the ship is small or waterfront space is not at a great premium, ships may be built on an even keel broadside on to the water and consigned to the water sideways. There are three common methods of sideways launching:

(a) the ship slides down ways which are built well down under the water;
(b) the ship tips or drops off the end of the ways into the water, sometimes tipping a part of the ways too;
(c) the ship is built on piles which are made to collapse by a sideways push to allow the ship to fall into the water.

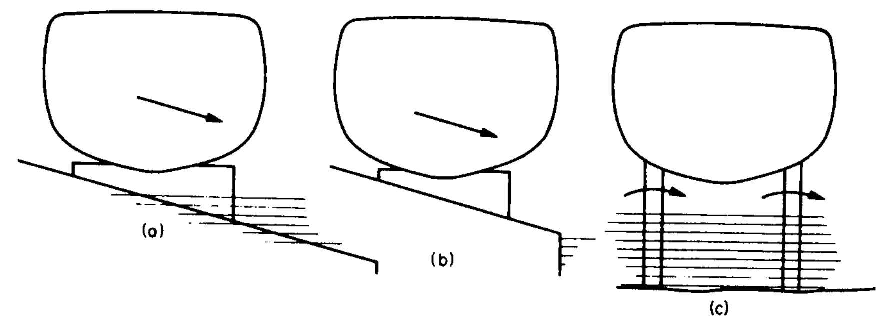


Fig. 8.9 Sideways launching

In all of these methods the ship takes to the water violently and may roll heavily-on entry, the ship may roll thirty degrees or more. Stability at large angles and watertightness are therefore important considerations. Waves may cause damage on adjacent shores.

Conventional calculations are not performed. Declivity of ways is usually of the order of 1 in 8 in order to give a high speed of launch to clear the end of the ways. Grease pressures permitted, 0.25 to 0.30 MN/m^2, are somewhat higher than with end launch.

Docking

From time to time, attention is needed to the outer bottom of a ship either for regular maintenance or to effect repairs. Access to the bottom may be obtained by careening, by hauling up a slipway or by placing in a dock from which the water is removed. There are limitations to what work can be done by careening, while the magnitude of the machinery needed to haul a ship up a slipway is such that this procedure is confined to ships up to a few hundred tonnef in displacement.

It has become common practice in some countries to build large vessels totally in a drydock and to allow the water to enter when the hull is in a satisfactory state. This undocking procedure must be investigated in a manner similar to that for normal docking or undocking to ensure that forces at the keel do not cause instability to the ship or that they do not trip the blocks. These matters are investigated in advance while evidence that affairs are what they should be is rigorously observed during the procedure by the dockmaster. This chapter is concerned with theory which assists such successful operations and some design features of floating docks. Features of principal concern are

(a) load distribution between dock and ship;
(b) behaviour of blocks;
(c) strength of floating docks;
(d) stability.

This list by no means exhausts the problems associated with docking and constructing docks, which include design of caissons and pumping systems and the whole field of difficulties facing civil engineers in the construction of graving docks. It does, however, embrace the major concerns of the practising naval architect. Basically, docking is the placing of an elastic ship with an uneven weight distribution on to an elastic set of blocks supported in turn by an elastic floating dock or a relatively rigid graving dock. Blocks will not be of an even height and are subject to crush, creep and instability. In many aspects of ship design, the criteria of design are based on misuse or accident; with floating docks, there is little scope for permitting maloperation because if this were a major criterion of design, the lifting capacity of the dock would be unduly penalized.

LOAD DISTRIBUTION

Theoretical analyses are available to study the behaviour of a ship in a floating dock to produce with some rigour the longitudinal distribution ofloading on the blocks. A simplified procedure has often been found adequate as follows.

It is assumed that the load distribution along the blocks follows the weight distribution of the ship except at the after cut up and at any other gap in the blocks. Forward of the after cut up, the weight distribution is assumed augmented by the weight abaft the cut up spread over a length equal to twice the length of the overhang, and distributed according to a parabolic law such that the moment of weight is the same as the moment of overhang about the cut up. If the weight and moment of the overhang about the cut up are respectively W and M, the augment in load per unit length at the cut up, a, is

$$a = \frac{9W^2}{16M}$$

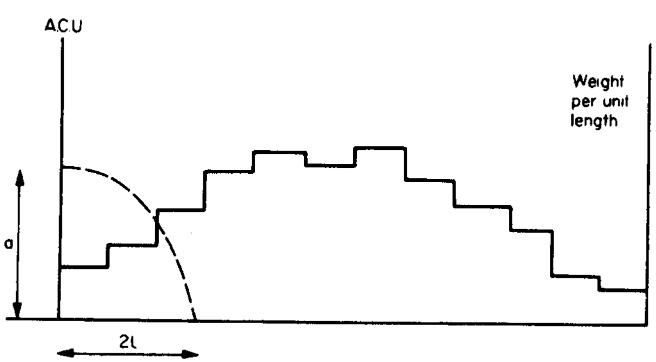


Fig. 8.10 Load distribution in dock

In a floating dock, the load distribution is further affected by the buoyancy distribution along the dock which is under the control of the dockmaster. For a given buoyancy distribution, the loading in the past has been determined by a method similar to the longitudinal strength calculation for a ship. The difference between the total buoyancy and ship's weight plus weight of dock and contained water gives the net loading. As explained in Chapter 6, shearing force

on a girder is obtained by integrating the net loading, and bending moment is obtained by integrating the shearing force along the length. In the case of the docked ship, the loading and the subsequent integrations apply to the combined ship and dock. Difficulty arises in trying to separate the two because their interaction is not known. In the past, bending moment has been divided up in the ratio of their respective second moments of area of section. This is not satisfactory and comprehensive analysis is needed.

Floating docks are classified by their lifting capacity. Lifting capacity is the buoyancy available when the dock is floating at its working freeboard. Working freeboard is often 0.15 m at the pontoon deck. Buoyancy available arises from the tanks which can be emptied to provide lift to the dock without undue strain.

BLOCK BEHAVIOUR

Depending as it does on defects, age, grain, moisture content and surface condition, it is not surprising to find wide variation in the behaviour of a stack of wooden dock blocks, as experiments have shown. Provided that the blocks are cribbed to prevent instability, it is common practice not to permit block pressures in excess of 440 tonnef $1m^2$, although the ultimate load intensity of a stack of English oak blocks may be as high as 1090 tonnef $1m^2$.

Experiments have shown that the crippling pressure for a stack of blocks h high and a in width is given approximately by

$$p = 11\left(40 - \frac{h^2}{a^2}\right)\text{tonnef/m}^2 = 1.07\left(40 - \frac{h^2}{a^2}\right) \times 10^5\ \text{N/m}^2$$

where p is the load divided by the total plan area of a stack of blocks, irrespective of the half siding of keel. The formula is suitable for English oak and is obviously an approximation. A 2 m stack of 40 cm blocks has a crippling load of 165 tonnef/m^2. In practice, 2 m stacks are rare and pressures in excess of 220 tonnef/m^2 for a single uncribbed stack are not permitted. It is normal good practice to crib a certain number of blocks, which increases the crippling load as well as preventing tripping due to the trim of the docking ships.

The importance of exact alignment of blocks cannot be over-emphasized. Measurements of the loads in dock blocks have shown that a stack will take many times the load of its neighbour by being only a little higher. A lack of fit of 12 mm between adjacent blocks is not unusual at present and is responsible for wild variation in dock block loading. It is responsible, too, for the departures of the block loading curve from the ship's weight distribution in graving docks.

Great care is needed to ensure that blocks are all of even stiffness. The number of blocks to ensure a mean deflection of stacks of x is

$$\text{Number of blocks} = \frac{2 \times \text{ship's weight}}{x \times \text{block stiffness}}$$

For a mean deflection of 1cm on blocks with a stiffness of 20 tonnef *jcm,* the minimum number of blocks required is *WjlOtonnef.* Clearly, this limitation is impractical for ships over about 2000 tonnef in displacement and, in general, the larger the ship, the closer the blocks are together and side blocks introduced.

STRENGTH OF FLOATING DOCKS

It is clear even from a cursory examination, that a dockmaster can place severe strains on a dock by a thoughtless selection of buoyancy tanks. In the transverse direction, if the dockmaster first selects the buoyancy tanks in the dock walls to begin raising the dock, a couple will be applied to the dock as shown in Fig. 8.11 which could cause damage. Similarly, in the longitudinal direction, if a ship has much weight amidships, the use of the buoyancy tanks at the ends of the dock could break it in two. It is good practice to select those buoyancy tanks immediately beneath the heaviest weights. Dock behaviour longitudinally, if the dock is made of steel, is checked continuously during lift by measuring the breakage. A strict limit to breakage is placed by the naval architect in the instructions for use of the dock. Reinforced concrete docks do not suffer much elastic distortion and breakage is not used as a measure of behaviour; instead, strict pumping patterns are imposed.

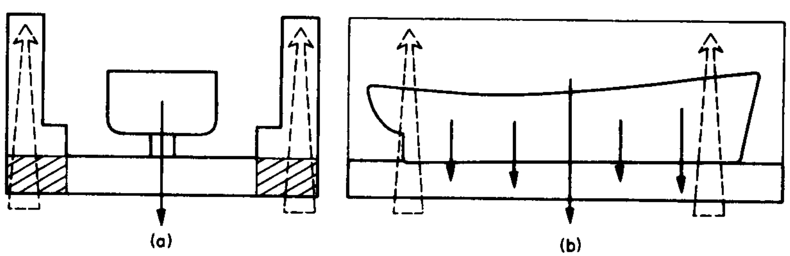


Fig. 8.11 *Strains imposed by maloperation*

Analysis of the strength of a floating dock was too difficult before computers had been satisfactorily programmed. Analyses now treat the dock bottom as a grillage simply supported at its edges by the two dock walls. The lumped weights of the docked vehicle are assumed supported by three rows of blocks constituted of stacks with a declared lack of fit. The problem is then analysed as an unevenly loaded beam on an elastic foundation for which programs are available.

The solution is equally applicable to graving docks by applying very high figures to the dock rigidity. Standard comparative longitudinal strength calculations to cater for transit conditions are also performed on the floating dock structure itself as described for the ship in Chapter 6.

The conventional floating dock is capable of taking very large vessels. A concrete floating dock may be capable of lifting a 350,000 dwt ship and a steel one of 400,000dwt capacity.

STABILITY DURING DOCKING

Loss of stability due to grounding or docking has been discussed fully in Chapter 5. Calculation of this loss is a standard procedure before every docking, since a list developing before the ship is in contact with the blocks completely along the keel can be extremely dangerous and can dislodge the blocks.

A floating dock is trimmed approximately to the trim of the ship and large suing forces before the keel takes the blocks are thereby avoided. There is a critical stability condition when the ship is just clear of the water and the restoring waterplane for both ship and dock is provided only by the dock walls. In this condition, the dock designers normally demand a minimum -G-Mof 1.6m. When the pontoon is lifted clear, the waterplane is, of course, much larger. Free surface effects in a floating dock are also large.

Large angle stability is important not only for ocean transit conditions but for the careening which is necessary during maintenance and repair operations.

SHIPLIFTS

Shiplifts can be used for launching or docking or building but the usual application is for docking. In a typical arrangement a vessel is floated above a cradle on a platform which can be raised to bring the cradle and ship to normal ground level. There is a system of rails which can then traverse the vessel sideways until it is in line with a shed into which it can be moved. In this way one lift can serve a number of docking sheds. Clearly care must be taken to avoid straining the ship or the platform during the lifting operation. There are a number of hydraulic hoisting points along the length of the platform and these can be fitted with load measuring devices to assist the operator, the data being fed into a computer to calculate load distribution and so assess trim, heel and draught before lowering into the water.

Platform elevator systems which lift vessels from the water and transfer them to the shore can take vessels of up to 25,000 tonnef. In other devices the vessel is floated over a cradle and both are winched up an inclined slipway. Often a transfer system is provided so that a single lift system can serve a number of refitting bays. Shiplifts are claimed to be significantly more economical than the corresponding floating dock system. .,.

Problems

1. A vessel whose launching weight is 25 MN with CG 5 m abaft amidships has the fore poppet positioned 50 m forward of amidships. Find the force on the fore poppet and the travel when the stern lifts from the following data:

Travel down slip, m	50	60	70	80	90
Buoyancy, MN	10.1	12.3	15.3	19.4	24.6
CB abaft amidships, m	47	40	33	26	19

Sliding ways 70 cm wide each side are proposed, with a length of 95 m. What is the mean pressure on the lubricant before motion takes place?

2. In a certain ship, the length of sliding ways was 163m and the breadth 1.63m. The launching weight was 9755 tonnef and c.g. estimated at 75.50 m forward of the after end of the sliding ways. Calculate the mean load per square metre on the ways. Assuming the pressure to vary in a linear manner from forward to the after end of ways, what is the pressure at each end?
3. A ship of length 152m between perpendiculars is built on a slip with its keel at a slope of 2.98 degrees. It is launched on groundways which extend to the fore poppet only and have a camber of 0.61 m over their length of 183m and a slope of 3.58 degrees. If the underside of keel produced at the fore perpendicular is initially 5.5 m above water level and the fore poppet is 18.3 m aft of the fore perpendicular, what are the draughts at the fore perpendicular and at the after perpendicular when the ship has travelled 61 m down the slip?
4. A ship is launched from ways 219 m long, with a camber of 457 mm and a declivity of 2.69 degrees. The slope of the keel is 2.46 degrees. If the minimum height of stopping up is 762 mm, find the height of the fore poppet above the after end of the ways when the ship has travelled 61 m down the ways. Assume the fore poppet initially at the fore end of ways.
5. Construct a set of launching curves from the following information:

launching weight	5230 tonnef
CG abaft midships	7m
fore poppet before midships	69m
after end of groundways from c.g. at rest	100m

Travel down slip (m)	30	60	75	90	105	120	135
Buoyancy (tonnef)							
before stern lift	830	1960	2600	3350	4280	5600	
after stern lift					4050	4700	5550
Moment of buoyancy about fp before stern lift (IOOOtonnefm)	132	219	282	356	450	585	
Moment of buoyancy about AEGW (1000 tonnefm)				42	76	160	316

(a) When does the stern lift?
(b) What is the maximum force at the fore poppet?
(c) What is the minimum moment against tipping?
(d) How does the ship leave the ways?

6. A ship weighs 28,500 tonnef at launch. The fore poppet is 72 m before the CG of the ship. The after end of the ground ways is 155m from the CG before the ship moves. The following table gives the data derived for the passage of the ship in full contact with the ways:

Travel, m	120	135	150	157.5	165
Buoyancy, tonnef	9000	15,600	24,400	29,200	35,000
CB abaft AEGW, m		7.5	21.5	28.0	33.0

Deduce a set of launching curves and pick off the important data.

7. A floating dock of rectangular bottom shape, 122m long and 27.4 m wide floats, when empty, at a draught of 1.07m. It is used to dock a ship 0Γ 3658 tonnef displacement, 110m long, which is symmetrically placed in the dock. The weight distribution of the ship is symmetrical about amidships and has the values shown in the table below:

Distance from midships, m	0	6	12	18	24	30	36	42	48	54
Section from midships		1	2	3	4	5	6	7	8	9
Weight per section, tonnef		294	289	276	256	228	194	152	103	38

Draw the curves of load, shearing force and bending moment for the dock and ship combination, in still water of 0.975 m^3/tonnef.

The weight of the dock may be assumed equally distributed along its length.

8. A vessel has a launching weight of 5893 tonnef, the CG being 7.9 m abaft the mid-length and the fore poppet 70 m before the mid-length. Construct a launching diagram from the following data:

Mid-length abaft AEGW, m	0	6	12	18	24
Buoyancy, tonnef	2601	3241	3901	4602	5415
CB abaft AEGW, m	39.9	43.6	48.2	52.7	56.4

State:

(a) distance of mid-length abaft the after end of groundways when the stern lifts;
(b) force on fore poppets when the stern lifts;
(c) reserve moment against tipping.

9 The ship environment and human factors

The naval architect, like the designer of any successful engineering product, must know the conditions under which the equipment is to exist and operate for its full life cycle, the attributes of those who will operate and maintain it and how those attributes vary with changing environment.

For convenience the environment can be divided into:

(a) the environment external to the ship which affects the ship as a whole and all exposed equipment. These conditions are caused by the sea and climate. Also the ship performance can be influenced by the depth or width of water present;
(b) the internal environment which affects the personnel and the internal equipment. To some extent, this environment is controllable, e.g. the temperature and humidity can be controlled by means of an air conditioning system.
(c) the interface between the human and machine and the effect of the environment on the human being.

The human element is covered by what is termed Human Factors (HF); a full study of which involves multi-discipline teams of physiologists, psychologists, engineers and scientists. The HF team can advise the naval architect on:

(i) how to design a system or equipment so that the operator can most effectively play a proper part, so giving the greatest overall system efficiency. It is not necessarily, and in general is not, true that the maximum degree of automation is desirable. The blend of human and machine should build up the strengths of each and minimize the weaknesses;
(ii) in what way, and to what degree, the system efficiency will be reduced due to degradations in the person's performance due to the environment;
(iii) the levels of environmental parameters (e.g. noise, vibration) which should not be exceeded if a person's physical state is not to be temporarily or permanently harmed.

Clearly there is an interaction between (i) and (ii) above in that the initial design must allow for the likely in-service environment.

The external environment. The sea

WATER PROPER TIES

Certain physical properties of the sea are of considerable importance to the designer. They are:

(a) *Density.* The density of the water in which the ship floats affects her draught and trim (Chapter 3) and depends mainly upon the temperature and salinity. The standard values of density of fresh and salt water at various temperatures are given in Tables 9.1 and 9.2.
(b) *Kinematic viscosity.* This is particularly relevant to the frictional resistance experienced by a ship as it defines the Reynolds' number (Chapter 10). The standard values of the kinematic viscosity of fresh and salt water at various temperatures are given in Tables 9.3 and 9.4.

Table 9.1
Mass densities for fresh water (last decimal figure is doubtful)

°C	ρ	°C	ρ	°C	ρ
0	999.79	10	999.59	20	998.12
1	999.79	11	999.49	21	997.92
2	999.89	12	999.40	22	997.72
3	999.89	13	999.30	23	997.43
4	999.89	14	999.1 0	24	997.24
5	999.89	15	999.00	25	996.94
6	999.89	16	998.91	26	996.75
7	999.79	17	998.71	27	996.45
8	999.79	18	998.51	28	996.16
9	999.69	19	998.32	29	995.87
				30	995.57

Metric units; ρ in kg/m^3

Table 9.2
Mass densities for salt water (salinity 3.5 per cent) (last decimal figure is doubtful)

°C	ρ	°C	ρ	°C	ρ
0	1028.03	10	1026.85	20	1024.70
1	1027.93	11	1026.66	21	1024.40
2	1027.83	12	1026.56	22	1024.11
3	1027.83	13	1026.27	23	1023.81
4	1027.74	14	1026.07	24	1023.52
5	1027.64	15	1025.87	25	1023.23
6	1027.44	16	1025.68	26	1022.93
7	1027.34	17	1025.38	27	1022.64
8	1027.15	18	1025.19	28	1022.25
9	1027.05	19	1024.99	29	1021.95
				30	1021.66

Metric units; ρ in kg/m^3

Table 9.3
Values of kinematic viscosity for fresh water, v, in metric units of (m^2s^{-1}) $\times$ 10^6.
Temp. in degrees Celsius

Deg. C	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	1.78667	1.78056	1.77450	1.76846	1.76246	1.75648	1.75054	1.74461	1.73871	1.73285
1	1.72701	1.72121	1.71545	1.70972	1.70403	1.69836	1.69272	1.68710	1.68151	1.67594
2	1.67040	1.66489	1.65940	1.65396	1.64855	1.64316	1.63780	1.63247	1.62717	1.62190
3	1.61665	1.61142	1.60622	1.60105	1.59591	1.59079	1.58570	1.58063	1.57558	1.57057
4	1.56557	1.56060	1.55566	1.55074	1.54585	1.54098	1.53613	1.53131	1.52651	1.52173
5	1.51698	1.51225	1.50754	1.50286	1.49820	1.49356	1.48894	1.48435	1.47978	1.47523
6	1.47070	1.46619	1.46172	1.45727	1.45285	1,44844	1.44405	1.43968	1.43533	1.43099
7	1.42667	1.42238	1.41810	1.41386	1.40964	1.40543	1.40125	1.39709	1.39294	1.38882
8	1.38471	1.38063	1.37656	1.37251	1.36848	1.36445	1.36045	1.35646	1.35249	1.34855
9	1.34463	1.34073	1.33684	1.33298	1.32913	1.32530	1.32149	1.31769	1.31391	1.31015
10	1.30641	1.30268	1.29897	1.29528	1.29160	1.28794	1.28430	1.28067	1.27706	1.27346
11	1.26988	1.26632	1.26277	1.25924	1.25573	1.25223	1.24874	1.24527	1.24182	1.23838
12	1.23495	1.23154	1.22815	1.22478	1.22143	1.21809	1.21477	1.21146	1.20816	1.20487
13	1.20159	1.19832	1.19508	1.19184	1.18863	1.18543	1.18225	1.17908	1.17592	1.17278
14	1.16964	1.16651	1.16340	1.16030	1.15721	1.15414	1.15109	1.14806	1.14503	1.14202
15	1.13902	1.13603	1.13304	1.13007	1.12711	1.12417	1.12124	1.11832	1.11542	1.11254
16	1.10966	1.10680	1.10395	1.10110	1.09828	1.09546	1.09265	1.08986	1.08708	1.08431
17	1.08155	1.07880	1.07606	1.07334	1.07062	1.06792	1.06523	1.06254	1.05987	1.05721
18	1.05456	1.05193	1.04930	1.04668	1.04407	1.04148	1.03889	1.03631	1.03375	1.03119
19	1.02865	1.02611	1.02359	1.02107	1.01857	1.01607	1.01359	1.01111	1.00865	1.00619
20	1.00374	1.00131	0.99888	0.99646	0.99405	0.99165	0.98927	0.98690	0.98454	0.98218
21	0.97984	0.97750	0.97517	0.97285	0.97053	0.96822	0.96592	0.96363	0.96135	0.95908
22	0.95682	0.95456	0.95231	0.95008	0.94786	0.94565	0.94345	0.94125	0.93906	0.93688
23	0.93471	0.93255	0.93040	0.92825	0.92611	0.92397	0.92184	0.91971	0.91760	0.91549
24	0.91340	0.91132	0.90924	0.90718	0.90512	0.90306	0.90102	0.89898	0.89695	0.89493
25	0.89292	0.89090	0.88889	0.88689	0.88490	0.88291	0.88094	0.87897	0.87702	0.87507
26	0.87313	0.87119	0.86926	0.86734	0.86543	0.86352	0.86162	0.85973	0.85784	0.85596
27	0.85409	0.85222	0.85036	0.84851	0.84666	0.84482	0.84298	0.84116	0.83934	0.83752
28	0.83572	0.83391	0.83212	0.83033	0.82855	0.82677	0.82500	0.82324	0.82148	0.81973
29	0.81798	0.81625	0.81451	0.81279	0.81106	0.80935	0.80765	0.80596	0.80427	0.80258
30	0.80091	0.79923	0.79755	0.79588	0.79422	0.79256	0.79090	0.78924	0.78757	0.78592

(c) *Salinity.* 'Standard' density and kinematic viscosity have been mentioned above. Values for actual samples of sea water will vary from area to area and will depend, amongst other things, upon the salinity. For instance many objects will float in the Dead Sea, which has a very high salt content, which would sink in fresh water. For most standard calculations and tests a salinity of 3.5 per cent is assumed.

THE SEA SURFACE

The sea presents an ever changing face to the observer. In the long term, the surface may be in any condition from a flat calm to extreme roughness. In the short term, the surface may present the appearance of a fairly steady level of roughness but the actual surface shape will be continuously varying.

Any observer is aware that a stream of air passing over a water surface causes ripples or waves to form, e.g. by blowing across the top of a cup of tea ripples

Table 9.4
Values of kinematic viscosity for salt water, v, in metric units of (m^2s^{-1}) $\times$ 10^6.
(Salinity 3.5 per cent.) Temp. in degrees Celsius

Deg. C	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	1.82844	1.82237	1.81633	1.81033	1.80436	1.79842	1.79251	1.78662	1.78077	1.77494
1	1.76915	1.76339	1.75767	1.75199	1.74634	1.74072	1.73513	1.72956	1.72403	1.71853
2	1.71306	1.70761	1.70220	1.69681	1.69145	1.68612	1.68082	1.67554	1.67030	1.66508
3	1.65988	1.65472	1.64958	1.64446	1.63938	1.63432	1.62928	1.62427	1.61929	1.61433
4	1.60940	1.60449	1.59961	1.59475	1.58992	1.58511	1.58032	1.57556	1.57082	1.56611
5	1.56142	1.55676	1.55213	1.54752	1.54294	1.53838	1.53383	1.52930	1.52479	1.52030
6	1.51584	1.51139	1.50698	1.50259	1.49823	1.49388	1.48956	1.48525	1.48095	1.47667
7	1.47242	1.46818	1.46397	1.45978	1.45562	1.45147	1.44735	1.44325	1.43916	1.43508
8	1.43102	1.42698	1.42296	1.41895	1.41498	1.41102	1.40709	1.40317	1.39927	1.39539
9	1.39152	1.38767	1.38385	1.38003	1.37624	1.37246	1.36870	1.36496	1.36123	1.35752
10	1.35383	1.35014	1.34647	1.34281	1.33917	1.33555	1.33195	1.32837	1.32481	1.32126
11	1.31773	1.31421	1.31071	1.30722	1.30375	1.30030	1.29685	1.29343	1.29002	1.28662
12	1.28324	1.27987	1.27652	1.27319	1.26988	1.26658	1.26330	1.26003	1.25677	1.25352
13	1.25028	1.24705	1.24384	1.24064	1.23745	1.23428	1.23112	1.22798	1.22484	1.22172
14	1.21862	1.21552	1.21244	1.20938	1.20632	1.20328	1.20027	1.19726	1.19426	1.19128
15	1.18831	1.18534	1.18239	1.17944	1.17651	1.17359	1.17068	1.16778	1.16490	1.16202
16	1.15916	1.15631	1.15348	1.15066	1.14786	1.14506	1.14228	1.13951	1.13674	1.13399
17	1.13125	1.12852	1.12581	1.12309	1.12038	1.11769	1.11500	1.11232	1.10966	1.10702
18	1.10438	1.10176	1.09914	1.09654	1.09394	1.09135	1.08876	1.08619	1.08363	1.08107
19	1.07854	1.07601	1.07350	1.07099	1.06850	1.06601	1.06353	1.06106	1.05861	1.05616
20	1.05372	1.05129	1.04886	1.04645	1.04405	1.04165	1.03927	1.03689	1.03452	1.03216
21	1.02981	1.02747	1.02514	1.02281	1.02050	1.01819	1.01589	1.01360	1.01132	1.00904
22	1.00678	1.00452	1.00227	1.00003	0.99780	0.99557	0.99336	0.99115	0.98895	0.98676
23	0.98457	0.98239	0.98023	0.97806	0.97591	0.97376	0.97163	0.96950	0.96737	0.96526
24	0.96315	0.96105	0.95896	0.95687	0.95479	0.95272	0.95067	0.94862	0.94658	0.94455
25	0.94252	0.94049	0.93847	0.93646	0.93445	0.93245	0.93046	0.92847	0.92649	0.92452
26	0.92255	0.92059	0.91865	0.91671	0.91478	0.91286	0.91094	0.90903	0.90711	0.90521
27	0.90331	0.90141	0.89953	0.89765	0.89579	0.89393	0.89207	0.89023	0.88838	0.88654
28	0.88470	0.88287	0.88105	0.87923	0.87742	0.87562	0.87383	0.87205	0.87027	0.86849
29	0.86671	0.86494	0.86318	0.86142	0.85966	0.85792	0.85619	0.85446	0.85274	0.85102
30	0.84931	0.84759	0.84588	0.84418	0.84248	0.84079	0.83910	0.83739	0.83570	0.83400

are formed. The precise mechanism by which the transfer of energy takes place is not known. Nor is the minimum velocity of air necessary to generate waves known accurately and figures between 0.6 and 6 *m/s* have been suggested by various workers in this field.

The surface, then, is disturbed by the wind, the extent of the disturbance depending upon the strength of the wind, the time for which it acts and the length of the water surface over which it acts. These three qualities are referred to as the *strength, duration* and *fetch* of the wind, respectively. The disturbance also depends on tide, depth of water and local land contours. Wave characteristics become practically independent of fetch when the fetch is greater than about 500 km.

Once a wave has been generated it will move away from the position at which it was generated until all its energy is spent. Waves generated by local winds are termed *sea* and those which have travelled out of their area of generation are termed *swell.* Sea waves are characterized by relatively peaky crests and the crest length seldom exceeds some two or three times the wavelength. Swell waves are generally lower with more rounded tops. The crest length is typically six or seven times the wave-length. In a swell, the variation in height between successive waves is less than is the case for sea waves.

The sea surface presents a very confused picture which defied for many years any attempts at mathematical definition. Developments in the theory of random variables now make it possible to construct a mathematical model of the sea surface.

Before proceeding to consider the irregular sea surface, it is first of all necessary to consider a regular wave system as this is the basic building brick from which the picture of any irregular system is built up. Such a system with crests extending to infinity in a direction normal to the direction of propagation of the wave has the appearance of a large sheet of corrugated iron. The section of the wave is regular and two cases are of particular significance. They are: (i) the trochoidal wave, and (ii) the sinusoidal wave.

The theory of these two wave forms is developed in the following paragraphs which can be omitted by the student wishing to have only a general appreciation of wave systems.

Waves

TROCHOIDAL WAVES

The trochoidal wave theory has been adopted for certain standard calculations. e.g. that for longitudinal strength. By observation, the theory appears to reflect many actual ocean wave phenomena although it can be regarded as only an approximation to the complex wave form actually existing.

The section of the trochoidal wave surface is defined mathematically as the path traced by a point fixed within a circle when that circle is rolled along and below a straight line as shown in Fig. 9.1.

Suppose the height and length of the wave to be generated are h_w and λ respectively: then the radius of the generating circle is R, say, where

$$\lambda = 2\pi R$$

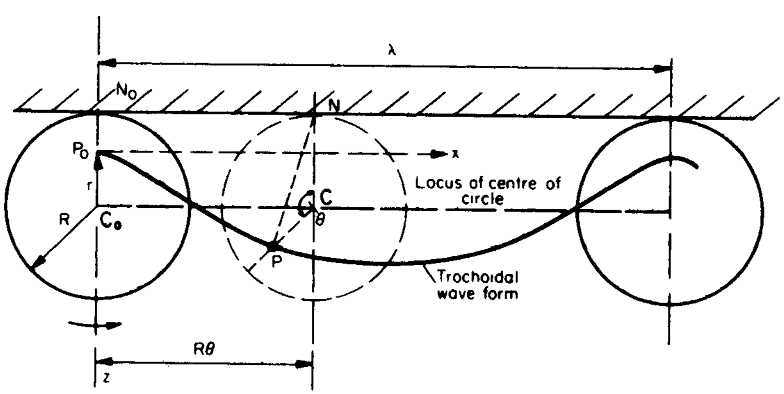


Fig. 9.1 Generation of trochoidal wave form

and the distance of the generating point from the centre of the circle is r, say, where

$$h_w = 2r$$

Let the centre of the generating circle start at position C_0 and let it have turned through an angle θ by the time it reaches C. Then

$$C_0C = N_0N = R\theta$$

Relative to the x-, z-axes shown in Fig. 9.1, the co-ordinates of P are

$$x = R\theta - r\sin\theta$$
$$= \frac{\lambda}{2\pi}\theta - \frac{h_w}{2}\sin\theta$$
$$z = r - r\cos\theta = \frac{h_w}{2}(1-\cos\theta)$$

As the circle rolls it turns, instantaneously, about its point of contact, N, with the straight line. Hence $\overline{NP}$ is the normal to the trochoidal surface and the instantaneous velocity of P is

$$\overline{NP}\frac{d\theta}{dt} \text{ normal to } \overline{NP}$$

Surfaces of equal pressure below the water surface will also be trochoidal. Crests and troughs will lie vertically below those of the surface trochoid, i.e. the length of the trochoidal wave is the same in all cases and they are all generated by a rolling circle of the same diameter. At great depths, any surface disturbance is not felt so that the radius of the point generating the trochoidal surface must reduce with increasing depth.

Consider two trochoidal sub-surfaces close to each other as shown in Fig. 9.2.

By definition, no particles of water pass through a trochoidal surface. Hence, for continuity the space shown shaded in Fig. 9.2 must remain filled by the same volume of water.

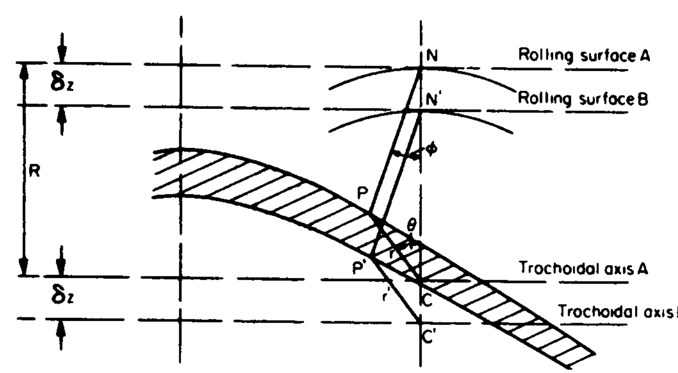


Fig. 9.2 Sub-surface trochoids

Let t = thickness of layer at P (measured normal to the trochoidal surface).

$$\text{Velocity at P} = \overline{\text{PN}}\frac{d\theta}{dt}$$

Hence the condition of continuity implies

$$t\overline{\text{PN}}\frac{d\theta}{dt} = \text{constant for all values of } \theta$$

i.e.

$$t\overline{\text{PN}} = \text{constant for all values of } \theta$$

Now $\overline{\text{CP}}$ and $\overline{\text{C'P}}'$ are parallel and, provided δz is small, $\overline{\text{PN}}$ and $\overline{\text{P'N}}'$ are very nearly parallel. Resolving along PN

$$\overline{\text{P'N}}' + \delta z\cos\phi = \overline{\text{PN}} + t$$

$$\therefore\ t = \delta z\cos\phi + \delta\overline{\text{PN}},\ \text{where } \delta\overline{\text{PN}} = \overline{\text{P'N}'} - \overline{\text{PN}}$$

$$= \delta z\left(\frac{R - r\cos\theta}{\overline{\text{PN}}}\right) + \delta\overline{\text{PN}}$$

$$\therefore\ t\overline{\text{PN}} = \delta z(R - r\cos\theta) + \overline{\text{PN}}\delta\overline{\text{PN}}$$

Also

$$\overline{\text{PN}}^2 = R^2 + r^2 - 2Rr\cos\theta$$

Differentiating and remembering that R and θ are constant

$$2\overline{\text{PN}}\delta\overline{\text{PN}} = 2r\delta r - 2R\cos\theta\ \delta r$$

$$\therefore\ t\overline{\text{PN}} = \delta z(R - r\cos\theta) + r\ \delta r - R\cos\theta\ \delta r$$

$$= R\ \delta z + r\ \delta r - \cos\theta(r\ \delta z + R\ \delta r)$$

Since $t\overline{\text{PN}}$ is constant for all values of θ

$$r\ \delta z + R\ \delta r = 0$$

$\therefore$ in the limit

$$\frac{1}{r}dr = -\frac{1}{R}dz$$

Integrating

$$\log_e r = c - \frac{z}{R} \text{ where } c = \text{constant}$$

If $\quad r = r_0$ at $z = 0,\ c = \log_e r_0$

$$\therefore\quad \log_e r = \log_e r_0 - \tfrac{z}{R}$$

i.e.

$$r = r_0\exp\left(-\frac{z}{R}\right)$$

That is to say, the radius of the point tracing out the sub-surface trochoid decreases exponentially with increasing depth. This exponential decay is often met within natural phenomena, and is very rapid as is illustrated in Table 9.5 for a surface wave 150 m long and 15 m high, i.e. a wave for which

$$R = \frac{\lambda}{2\pi} = \frac{75}{\pi} = 23.87\text{ m}$$

$$r_0 = \frac{15}{2} = 7.5\text{ m}$$

Table 9.5
Decay of orbital radius with depth

z (m)	r (m)	z (m)	r (m)
0	7.50	25	2.57
5	6.06	50	0.87
10	4.88	100	0.10
23.87	2.76		

Although it has been convenient to study the shape of the trochoidal wave using the artifice of a point within a rolling circle, a particle in the water surface itself does not trace out a trochoid. The absolute motion is circular and neglecting bodily movement of masses of water such as those due to tides, is obtained by superimposing on Fig. 9.1 an overall velocity of C from right to left such that

$$C = \frac{\lambda}{T}$$

This superposition of a constant velocity in no way invalidates the results obtained above or those which follow. The use of the 'stationary' trochoid as in Fig. 9.1 is merely one of mathematical convenience.

The magnitude of C can be expressed in terms of λ by considering the forces acting on a particle at P in Fig. 9.1. Such a particle will suffer a downward force due to gravity and a 'centrifugal force' in the line $\overline{\text{CP}}$.

If the mass of the particle is m, the gravitational force is mg and the centrifugal force $mr\omega^2$ where $\omega = d\theta/dt$.

Since the surface is one of equal pressure, the resultant of these two forces must be normal to the surface, i.e. it must lie in the direction of $\overline{\text{NP}}$.

Referring to Fig. 9.3,

$$\frac{mg}{\overline{\text{NC}}} = \frac{mr\omega^2}{\overline{\text{CP}}}$$

i.e.

$$\frac{mg}{\text{R}} = \frac{mr\omega^2}{r}$$

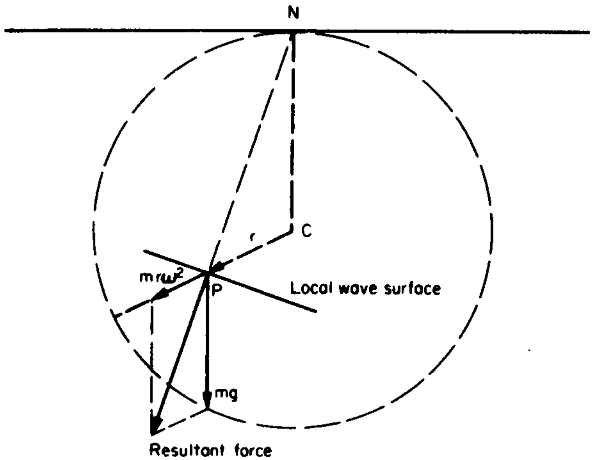


Fig. 9.3 Forces on particle in wave surface

Whence

$$\omega^2 = \frac{g}{R}$$

Now $C = \lambda/T$ where T, the wave period $= 2\pi/\omega$.

$$\therefore\ C = \lambda\omega/2\pi = \omega R$$

Hence

$$C^2 = \omega^2 R^2 = gR = \frac{g\lambda}{2\pi}$$

Relationship between line of orbit centres and the undisturbed surface

In Fig. 9.4, the trochoidal wave form is superimposed on the still water surface. It is clear that the volume of water in a crest, as measured above the still water level, must be equal to the volume of a trough measured below this level. Put another way, the area $P_0\ P_0'\ P_0''$ must equal the area of the rectangle $P_0\ P_0'\ L'L$ or, for simplification of the integration

$$\int_0^{x=\pi R} z\ \mathrm{d}x = \pi R(r + \delta z)$$

where δz = distance of still water surface below the line of orbit centres. Since $x = R\theta - r\sin\theta$ and $z = r - r\cos\theta$, we obtain

$$\int_{\theta=0}^{\theta=\pi} (r - r\cos\theta)(R - r\cos\theta)\mathrm{d}\theta = \pi R(r + \delta z)$$

$$\int_0^{\pi} (rR - rR\cos\theta - r^2\cos\theta + r^2\cos^2\theta)\mathrm{d}\theta = \pi R(r + \delta z)$$

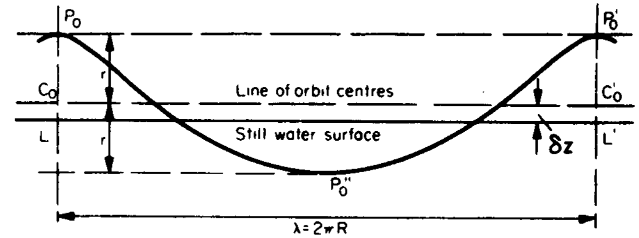


Fig. 9.4 Line of orbit centres in relation to the still water surface

i.e.

$$\pi Rr + \frac{\pi}{2}r^2 = \pi R(r + \delta z)$$

$$\therefore\ \delta z = \frac{r^2}{2R}$$

This property has been used in the past as a wave pressure correction in the longitudinal strength calculations.

SINUSOIDAL WAVES

Mathematically, the trochoidal wave is not easy to manipulate and the basic units from which an irregular sea is assumed to be built up are sinusoidal in profile.

For a wave travelling with velocity C in the direction of decreasing x, Fig. 9.5, the profile of the wave can be represented by the equation

$$z = \frac{H}{2}\sin(qx + \omega t)$$

In this expression, $q = 2\pi/\lambda$ is known as the *wave number* and $\omega = 2\pi/T$ is known as the *wave frequency*. The wave velocity as in the case of the trochoidal wave, is given by

$$C = \frac{\lambda}{T} = \frac{\omega}{q}$$

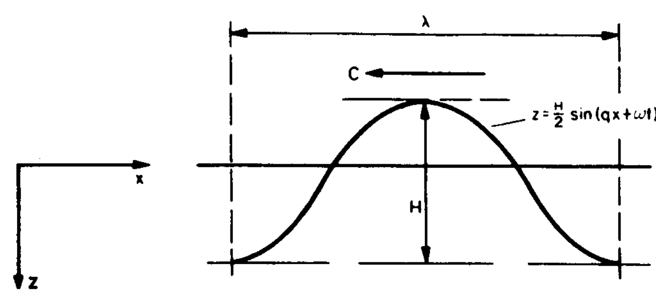


Fig. 9.5 Profile of sinusoidal wave

Other significant features of the wave are

$$T^2 = \frac{2\pi\lambda}{g}$$

$$\omega^2 = \frac{2\pi g}{\lambda}$$

$$C^2 = \frac{g\lambda}{2\pi}$$

Water particles within the waves move in orbits which are circular, the radii of which decrease with depth in accordance with the expression

$$r = \frac{H}{2}\exp(-qz) \text{ for depth } z$$

From this it can be deduced that at a depth $z = +\lambda/2$, the orbit radius is only $0.02H$ so that for all intents and purposes motion is negligibly small at depths equal to or greater than half the wavelength (see Table 9.5). The proof of this relationship is similar to that adopted for the trochoidal wave.

The hydrodynamic pressure at any point in the wave system is given by

$$p = \rho g\frac{H}{2}\exp(-qz)\sin(qx + \omega t)$$

The average potential and kinetic energies per unit area of the wave system are equal, and the average total energy per unit area is

$$\rho g\frac{H^2}{8}$$

The energy of the wave system is transmitted at half the velocity of advance of the waves. Thus, when a train of regular waves enters calm water, the front of the waves advances at the velocity of energy transmission, and the individual waves travel at twice this velocity and 'disappear' through the front.

Two of the above relationships are presented graphically in Fig. 9.6.

A fuller treatment of sinusoidal waves will be found in standard textbooks on hydrodynamics. As mentioned above, this wave form is assumed in building up irregular wave systems. It is also used in studying the response of a ship to regular waves. Differences in response to a trochoidal wave would be small if the waves are of the same height and length.

IRREGULAR WAVE PATTERNS

Behind the apparent confusion of the sea there is statistical order and the sea surface may be regarded as the result of superimposing a large number of regular sinusoidal waves of different, but small, heights and various lengths in random phase. An important parameter is the range of lengths, and hence frequencies, present in the system.

If all these wave components travel in the same direction, the irregular pattern will exhibit a series of straight crests extending to infinity in a direction normal to the direction of wave travel. Such a system is termed a *long-crested*

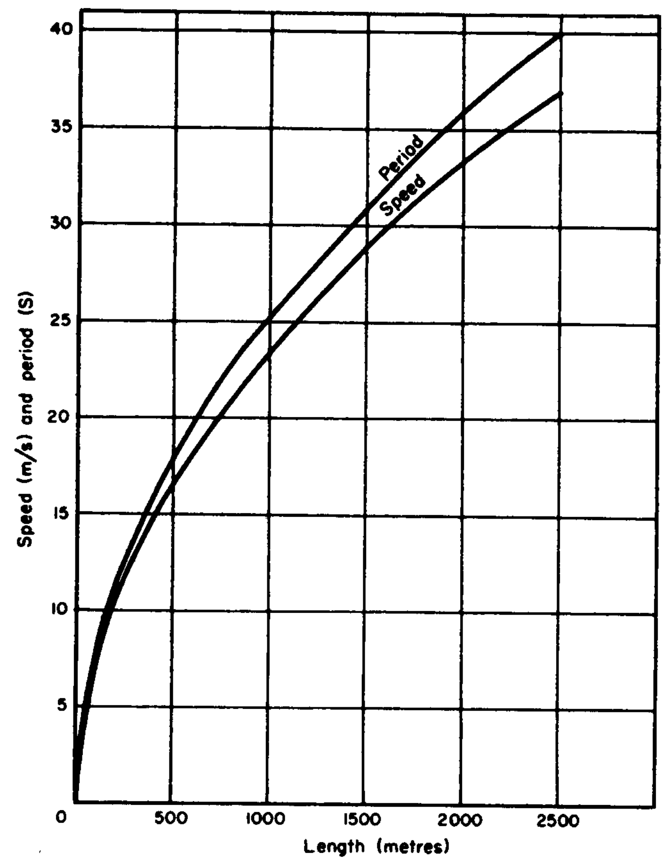


Fig. 9.6

irregular wave system and is referred to as *one-dimensional* (frequency). In the more general case, the individual wave components travel in different directions and the resultant wave pattern does not exhibit long crests but rather a series of humps and hollows. Such a system is termed an *irregular wave system* or *two-dimensional system* (frequency and direction). The remainder of this section is devoted mainly to the long-crested system.

How is any irregular system to be defined? It is possible to measure the time interval between successive crests passing a fixed point and the heights between successive troughs and crests. These vary continuously and can be misleading in representing a particular sea. For instance, a part of the system in which many component waves cancelled each other out because of their particular phase relationships would appear to be less severe than was in fact the case.

Provided the record of surface elevation against time is treated carefully, however, it can yield some very useful information. If ..\is the average distance between crests and t the average time interval in seconds, it can be shown that, approximately

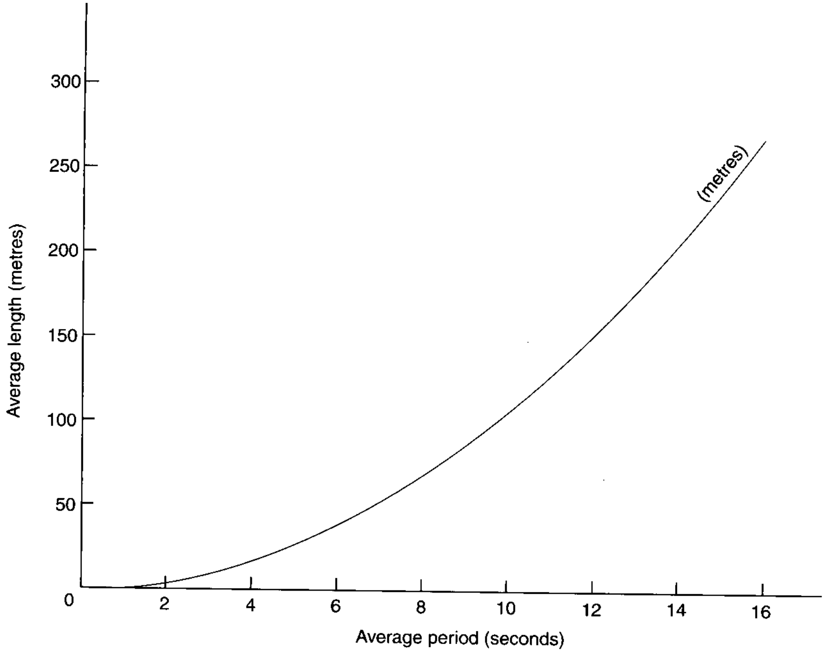


Fig. 9.7 Average period and wave-length

$$\bar{\lambda} = \frac{2g\bar{T}^2}{3 \times 2\pi} = 1.04\bar{T}^2, \text{ m}$$

This relationship between $\bar{\lambda}$ and $\bar{T}$ is represented in Fig. 9.7. Both can be related to wind speed as defined by the expression:

Average period $= 0.285\ V_w$, S

where V_w = wind speed in knots.

Since an observer is more likely to record $\bar{T}$ using a stop watch, applying the classical relationship $\lambda = gT^2/2\pi$ would lead to an error.

In general, in this chapter record is used to refer to a record of the variation of sea surface elevation, relative to its mean level, with time. A complete record would be too difficult to analyse and a sample length is usually taken from time to time, the duration of each being such as to ensure a reasonable statistical representation of the sea surface.

The values of wave height in a sample can be arranged in descending order of magnitude and the mean height of the first third of the values obtained. This mean height of the highest one-third of the waves is termed the *significant wave height*. The height of the highest wave occurring in a given record can also be found. If a large number of wave records is so analysed for a given ocean area, it is possible to plot the probability of exceeding a given significant wave height. Results are presented in Fig. 9.8 for the North Atlantic, Northern North

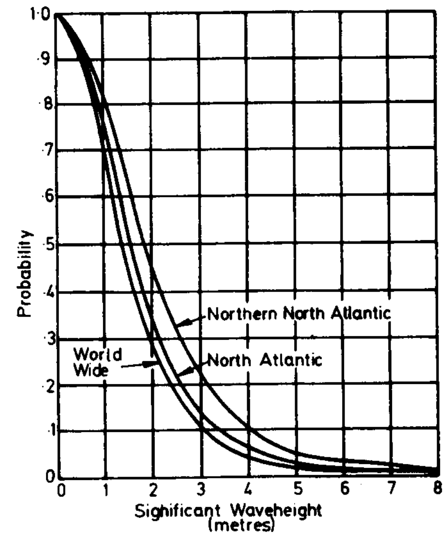


Fig. 9.8 Waves at OWS station India

Atlantic and world wide. It will be noted that the Northern North Atlantic is the most severe area as would be expected from general experience.

SEA STATE CODE

A generally accepted description of the sea state appropriate to various significant wave heights is afforded by the sea state code.

Sea state code

Code	Description of sea	Significant wave height (m)
0	Calm (glassy)	0
1	Calm (rippled)	0–0.1
2	Smooth (wavelets)	0.1–0.5
3	Slight	0.5–1.25
4	Moderate	1.25–2.50
5	Rough	2.50–4.00
6	Very rough	4–6
7	High	6–9
8	Very high	9–14
9	Phenomenal	Over 14

HISTOGRAMS AND PROBABILITY DISTRIBUTIONS

The wave heights can also be arranged in groups so that the number of waves with heights falling into various intervals can be counted. These can be plotted as a histogram as in Fig. 9.9 in which the area of each rectangle represents the number of waves in that interval of wave height. In practice, it is usual to arrange scales such that the total area under the histogram is unity. The histogram shows the distribution of heights in the wave sample, but in many applications subsequent analysis is easier if a mathematical curve can be fitted to the results. Three theoretical distributions are of particular importance.

The Normal or Gaussian distribution which is defined by the equation

$$p(x) = \frac{1}{\sigma\sqrt{(2\pi)}}\exp\left\{-\frac{1}{2}\frac{(x-\mu)^2}{\sigma^2}\right\}$$

where μ = mean of value of x, σ_1^2 = variance, and σ_1 = standard deviation.

The Logarithmically Normal or Log-Normal Distribution which is defined by the equation

$$P(\log x) = \frac{1}{\sigma_1\sqrt{(2\pi)}}\exp\left\{-\frac{1}{2}\frac{(\log x - u)^2}{\sigma_1^2}\right\}$$

where u = mean value of $\log x$, σ_1^2 = variance, and σ_1 = standard deviation.

The Rayleigh distribution which is defined by the equation

$$p(x) = \frac{x}{a}\exp\left\{-\frac{x^2}{2a}\right\}, \quad x > 0$$

where $2a$ is the mean value of x^2.

In these expressions, $p(x)$ is a probability density. Plotted to a base of x the total area under the curve must be unity, expressing the fact that it is certain that the variable under examination will take some value of x. The area under the curve in a small interval of x, dx, is $p(x)\,dx$ and this represents the probability that the variable will take a value in the interval dx.

Integrating the curve leads to a *cumulative probability distribution* in which the ordinate at a given value of x represents the area under the probability distribution between 0 and x. The ordinate thus represents the probability that the variable will have a value less than, or equal to, x.

It will be noted that the normal and log-normal distributions as functions of x are defined by two variables, μ and σ or u and σ_1, and the Rayleigh distribution by one variable a. Hence, it is necessary to determine the values of these quantities which most closely fit the observed wave data.

EXAMPLE 1. A record contains 1000 waves with heights up to 10 m, the numbers of readings falling into various 1-m groups are

Height (m)	0–1	1–2	2–3	3–4	4–5	5–6	6–7	7–8	8–9	9–10
Number of waves	6	29	88	180	247	260	133	42	10	5

Plot a histogram for these data and derive the corresponding normal distribution.

Solution: The histogram is obtained by erecting a series of rectangles on each height band such that the area of each rectangle is proportional to the number of waves in that height band. For ease of comparison with the probability curve, the total area of all rectangles is made unity by dividing by 1000 and the histogram is plotted as in Fig. 9.9.

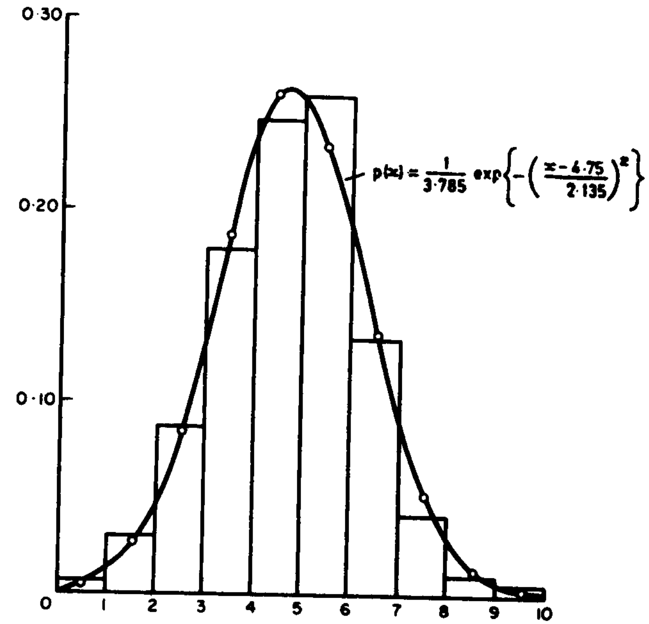


Fig. 9.9 Histogram and normal probability curve superimposed

The mean, μ, of the wave heights is found by summating the products of the average height in each band (x, say) by its frequency of occurrence (F_r) and the variance follows as in the table. Using Table 9.6,

$$\mu = \frac{4750}{1000} = 4.75$$

$$\sigma^2 = \frac{2282}{1000} = 2.282$$

$$\sigma = 1.510$$

Hence the equation to the normal curve is

$$p(x) = \frac{1}{1.510\sqrt{(2\pi)}}\exp\left\{-\frac{1}{2}\left(\frac{x-4.75}{1.51}\right)^2\right\}$$

$$= \frac{1}{3.785}\exp\left\{-\left(\frac{x-4.75}{2.135}\right)^2\right\}$$

Corresponding values of x and $p(x)$ are given in Table 9.6 and these have been plotted in Fig. 9.9 superimposed on the histogram.

Table 9.6

Xr	Fr	XrFr	Xr -/-t	(xr - /-ti	Fr(xr - /-t)2
0.5	6	3	-4.25	18.06	108.5
1.5	29	43.5	-3.25	10.56	306.2
2.5	88	220	-2.25	5.06	445.5
3.5	180	630	-1.25	1.56	281.3
4.5	247	1111.5	-0.25	0.06	15.5
5.5	260	1430	+0.75	0.56	146.2
6.5	133	864.5	+1.75	3.06	407.3
7.5	42	315	+2.75	7.56	317.7
8.5	10	85	+3.75	14.06	140.8
9.5	5	47.5	+4.75	22.56	112.7
	1000	4750			2281.7

x	0.5	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5
p(x)	0.0050	0.0260	0.0870	0.1874	0.2606	0.2340	0.1350	0.0504	0.0120	0.0018

In applying the above techniques, it must be realized that particular sets of records are only *samples* of all possible records, known in statistics as the *population,* e.g. the wave data reproduced above for weather station India were made in one area only and for limited periods of time. When an analytical expression is fitted to the observed data, it is desirable to demonstrate that the expression is reasonably valid. The reliability of the hypothesis can be evaluated by applying statistical tests of significance which enable estimates to be made of the expected variation of the measured data from the analytically defined data if the hypothesis is true.

WAVE SPECTRA

A very useful method of presenting data on the wave system is in the form of a wave spectrum. When there are n regular sine wave components present and n is not too large, a *line spectrum* can be drawn as Fig. 9.10 where the height of each ordinate represents the component wave amplitude or half the square of the component wave amplitude. In this latter case the ordinate is proportional to the component wave energy.

In practice, n is usually so large and the amplitude of each component so small, that the concept of a *continuous spectrum* is more convenient. Taking the energy spectrum form the area between frequencies *I* and *I* + *81* represents

$$\sum_{f}^{f+\delta f} \frac{1}{2}\zeta_A^2$$

That is , the area under the curve between frequencies **/** and **/** + *81* represents the sum of the energy of all component waves having frequencies in the band. In the limit, as *81* tends to zero, the curve becomes smooth. This more general curve can be represented by a plot of Sew) against w as in Fig. 9.11.

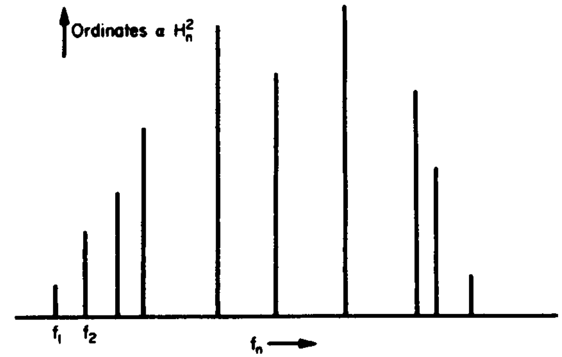


Fig. 9.10 Line spectrum

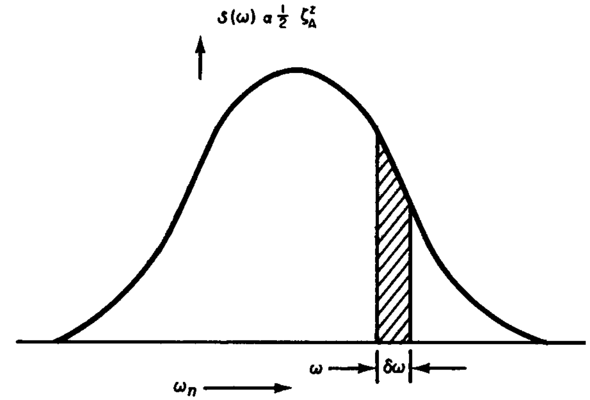


Fig. 9.11 Continuous spectrum

Area under spectrum $= m_0 =$ mean square of surface elevation

The area under the energy spectrum is given by

$$\int_0^\infty S(\omega)d\omega$$

In more general terms the nth moment of the curve about the axis $\omega = 0$ is

$$m_n = \int_0^\infty \omega^n S(\omega)d\omega$$

It can be shown that

$$m_0 = \text{area under spectrum} = \text{mean square of wave surface elevation}$$

Referring to Fig. 9.12, m_0 can be obtained by measuring the surface elevation ζ, relative to the mean level, at equal time intervals. If n such measurements are taken:

$$m_0 = \frac{1}{2n}\Sigma\zeta^2 = \frac{1}{2n}\left(\zeta_1^2 + \zeta_2^2 + \cdots + \zeta_n^2\right)$$

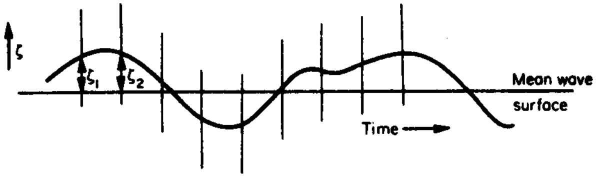


Fig. 9.12

For reasonable accuracy, the time interval chosen should not be greater than half the minimum time interval between successive crests and the number of readings must be large.

WAVE CHARACTERISTICS

If the distribution of wave amplitude is Gaussian then the probability that at a random instant of time the magnitude of the wave amplitude exceeds some value ζ is given (for a normal distribution) by

$$P(\zeta) = 1 - \mathrm{erf}[\zeta/\sqrt{(2m_0)}]$$

The error function (erf) is tabulated in standard mathematical tables.

The percentages of wave amplitudes exceeding certain values are shown in Fig. 9.13. It will be seen that more than 60 per cent of waves have amplitudes of $\sqrt{m_0}$ or less.

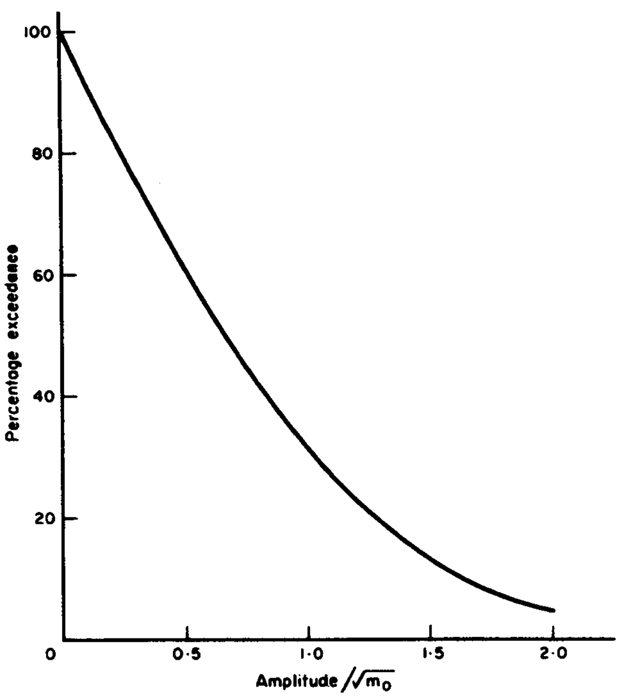


Fig. 9.13

However, observations at sea show that many statistical properties of the wave surface can be closely represented by the Rayleigh distribution. It can be shown that for waves represented by a Rayleigh distribution:

Most frequent wave amplitude $= 0.707\sqrt{(2m_0)} = \sqrt{m_0}$
Average wave amplitude $= 0.886\sqrt{(2m_0)} = 1.25\sqrt{m_0}$
Average amplitude of $\frac{1}{3}$ highest waves $= 1.416\sqrt{(2m_0)} = 2\sqrt{m_0}$
Average amplitude of $\frac{1}{10}$ highest waves $= 1.800\sqrt{(2m_0)} = 2.55\sqrt{m_0}$

Note: Some authorities plot ζ^2 against f as the spectrum. In this case, the area under the spectrum is $2m_0$ where m_0 is defined as above.

The average amplitude of $\frac{1}{3}$ highest waves is denoted by $\zeta_{\frac{1}{3}}$, and is termed the *significant wave amplitude*.

More generally if the values of ζ were to be arranged in descending order of magnitude, the mean value of the first *cn* of these is given by $k_1\zeta_S$ where corresponding values of c and k_1 are given below:

Table 9.7
Corresponding values of c and k_1

c	k_1	c	k_1
0.01	2.359	0.4	1.347
0.05	1.986	0.5	1.256
0.1	1.800	0.6	1.176
0.2	1.591	0.7	1.102
0.25	1.517	0.8	1.031
0.3	1.454	0.9	0.961
0.3333	1.416	1.0	0.886

It also follows that:

(*a*) the probability that ζ should exceed a certain value r is given by

$$\int_r^\infty p(r)dr = \exp(-r^2/\bar{\zeta}^2) \text{ where } \bar{\zeta}^2 = \frac{1}{N}(\zeta_1^2 + \zeta_2^2 + \cdots + \zeta_N^2)$$

(*b*) the expected value of the maximum amplitude in a sample of n values is $k_2\zeta_S$ where corresponding values of n and k_2 are given in Table 9.8.

Table 9.8
Corresponding values of n and k_2

n	k_2	n	k_2
1	0.707	500	2.509
2	1.030	1000	2.642
5	1.366	2000	2.769
10	1.583	5000	2.929
20	1.778	10,000	3.044
50	2.010	20,000	3.155
100	2.172	50,000	3.296
200	2.323	100,000	3.400

(*c*) in such a sample the probability that ζ will be greater than r is

$$1 - \exp[-\exp\{-(r^2 - r_0^2)/\bar{\zeta}^2\}]$$

where $r_0^2/\bar{\zeta}^2 = \log n$

(*d*) The number of zero up-crossings (Fig. 9.14) per unit time is:

$$\frac{1}{2\pi}\sqrt{\frac{m_2}{m_0}}\text{; Average period } \bar{T}_0 = 2\pi\frac{m_0}{\sqrt{m_2}}$$

(*e*) The number of maxima per unit time is:

$$\frac{1}{2\pi}\sqrt{\frac{m_4}{m_2}}\text{; Average period } \bar{T}_m = 2\pi\sqrt{\frac{m_2}{m_4}}$$

(*f*) Average wavelengths are:

$$\bar{\lambda}_0 = 2\pi g\sqrt{\frac{m_0}{m_4}} \text{ based on zero up-crossings}$$

$$\bar{\lambda}_m = 2\pi g\sqrt{\frac{m_4}{m_8}} \text{ based on maxima}$$

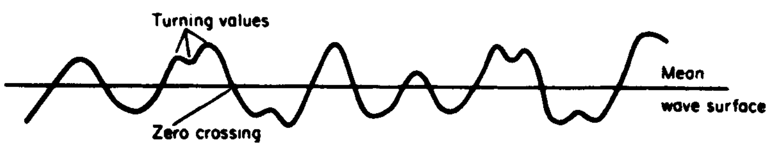


Fig. 9.14 Wave elevation recorded at a fixed point

In the more general case when the spectrum is not narrow its width can be represented by a parameter ϵ. The extreme values of ϵ are 0 and 1 representing a Rayleigh and normal distribution of wave amplitude respectively.

Then, if m_0 is the mean square value of the wave surface relative to the mean level, the variance as previously defined, it can be shown that the

mean square crest height $= (2 - \epsilon^2)m_0$

By putting $\epsilon = 0$, it follows that

$$\bar{\zeta}^2 = 2m_0$$

On the other hand in a very broad spectrum with $\epsilon = 1$,

mean square crest height $= m_0$

Also, mean crest to trough height is:

$$2\left\{\frac{\pi}{2}(1-\epsilon^2)m_0\right\}^{1/2}$$

and

$$\epsilon^2 = 1 - \frac{m_2^2}{m_0 m_4} = 1 - \frac{\bar{T}_\lambda^2}{\bar{T}_0^2}$$

$$\text{hence}(1-\epsilon^2)^{1/2} = \frac{\text{number of zero crossings}}{\text{number of turning points}}$$

Thus, referring to Fig. 9.14 and remembering that, in practice, a much longer record would be used

Number of zero crossings = 11; Number of turning values = 19

Hence

$$(1-\epsilon^2)^{1/2} = \frac{11}{19} = 0.58$$

i.e.

$$\epsilon = 0.815$$

FORM OF WAVE SPECTRA

There has been much debate concerning the form of representative wave spectra. The dependence of the wave system upon the strength, duration and fetch of the wind has already been discussed. Some of the differences between the spectra advanced by various authorities are probably due, therefore, to the difficulty of deciding when a sea is fully developed because a uniform wind

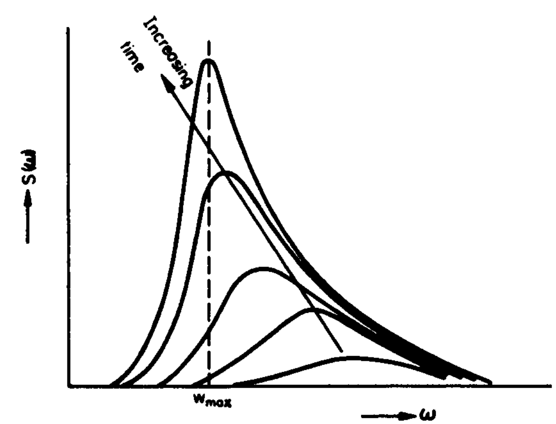


Fig. 9.15 Growth of spectra

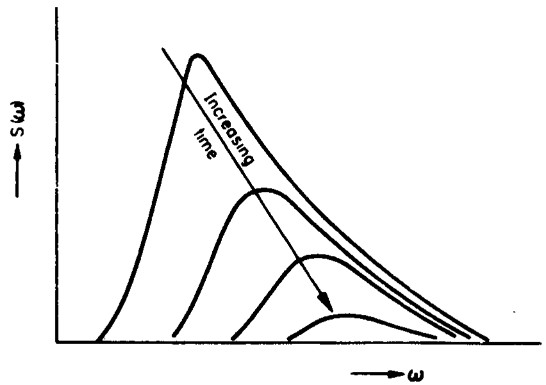


Fig. 9.16 Decay of spectra

seldom blows long enough for this to be determined. When a wind begins to blow, short, low amplitude waves are formed at first. As the wind continues to blow, larger and longer waves are formed. As the wind dies down the longer waves move out of the area by virtue of their greater velocity, leaving the shorter waves. Thus the growth and decay of wave spectra is as shown in Figs. 9.15 and 9.16.

Because of the dependence of a fully developed sea on wind strength earlier attempts to define wave spectra used wind speed as the main parameter. One such was that due to Pierson and Moskowitz, viz.:

$$S(\omega) = \frac{8.1 \times 10^{-3} g^2}{\omega^5} \exp\left\{-0.74\left(\frac{g}{V\omega}\right)^4\right\} m^2 S$$

g in ms^{-2}
V in ms^{-1}

Typical plots are given in Fig. 9.17. These show how rapidly the area under the curves increases with increasing wind speed, while the peak frequency decreases.

The spectrum now most widely adopted is that recommended by the International Towing Tank Conference:

$$S(\omega) = \frac{A}{\omega^5} \exp(-B/\omega^4) \text{ (the Bretschneider spectrum)}$$

ω is the circular frequency of the waves (s^{-1}) and A and B are constants.

It will be noted that this is of the same general form as that due to Pierson and Moskowitz. For this form

$$\text{Maximum } S(\omega) = A(0.8B)^{-5/4} e^{-5/4} \text{at } \omega = (0.8B)^{1/4}$$

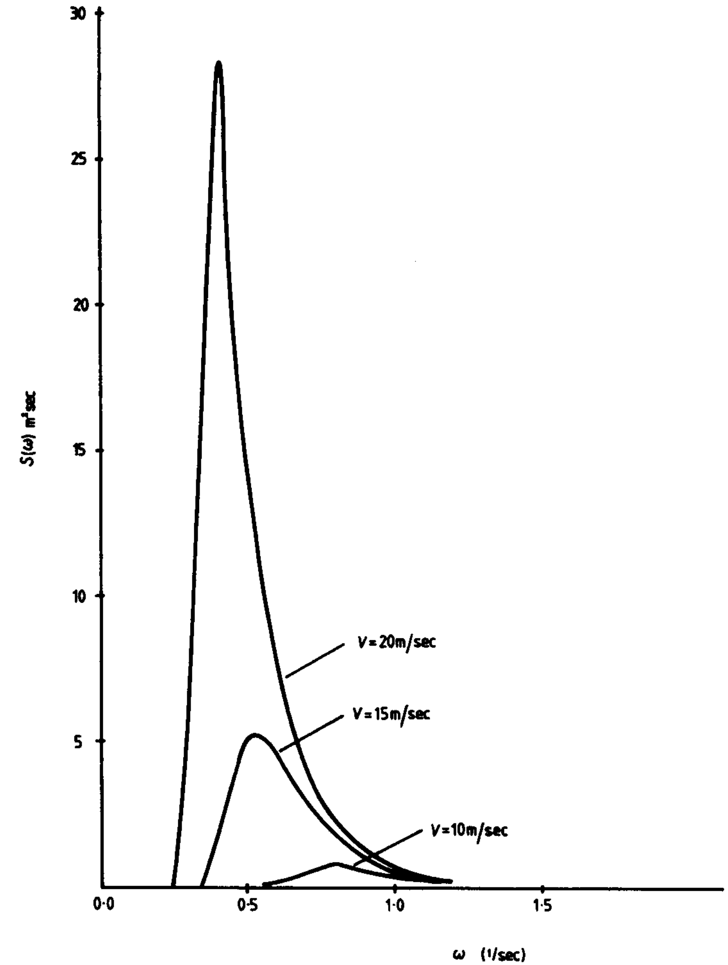


Fig. 9.17

When both significant wave height and the characteristic wave period, T_1, are known,

$$A = \frac{173\,\zeta_{\frac{1}{3}}^2}{T_1^4}$$

$$B = 691/T_1^4 \qquad \zeta_{\frac{1}{3}} = \text{significant wave height in m}$$

$$T_1 = 2\pi\frac{m_0}{m_1} \qquad = 4\sqrt{m_0}$$

When only the significant wave height is known, an approximation to $S(\omega)$ is given using

$$A = 8.10 \times 10^{-3} g^2$$

$$B = 3.11/\zeta_{\frac{1}{3}}^2$$

A suggested relationship for significant wave height in terms of wind speed for a fully developed sea is shown in Fig. 9.23. A typical family of spectra for long crested seas using the single parameter formula is shown in Fig. 9.18. With this simple version the ratio between the frequency or wave length at peak energy and the significant wave height is constant for all spectra the value of the ratio being 39.2. This is not true for all ocean spectra although many have values in the range 30 to 50. Hence for general purposes, such as prediction of ship motion likely to be experienced by a new design, the use of the simple formula is acceptable. Where it is necessary to vary the ratio the more elaborate formula can be used. Figure 9.19 presents a family of spectra with significant wave height of 7.5 metres.

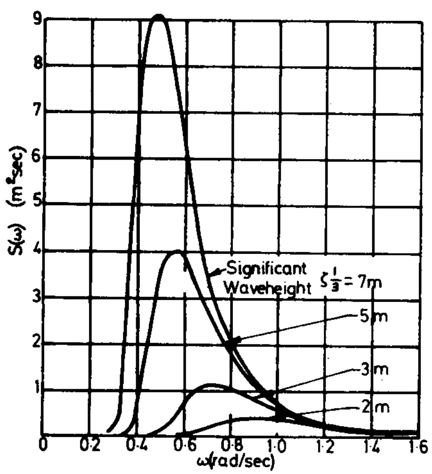


Fig. 9.18

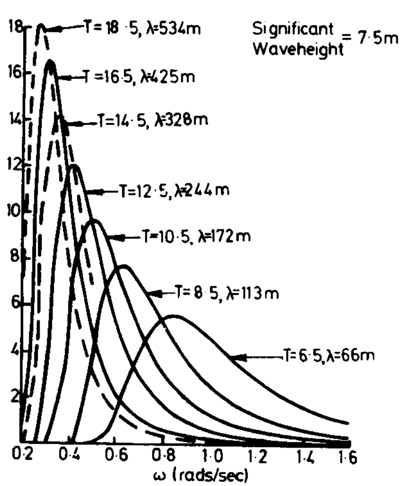


Fig. 9.19

For limited fetch conditions, the ITTC recommend

$$S_J(\omega) = 0.658\ S(\omega)(3.3)\exp\left(-\frac{0.206\omega T_1 - 1}{\sqrt{2}\sigma}\right)^2 \mathrm{m}^2 S$$

where $\sigma = 0.07$ for $\omega \leq 4.85T_1$
$\sigma = 0.09$ for $\omega > 4.85T_1$

For a multi-directional sea the ITTC recommend applying a spreading function as follows

$$S(\omega, \mu) = K \cos^n \mu S(\omega) \qquad -\frac{\pi}{2} < \mu < \frac{\pi}{2}$$

where μ = direction of wave components relative to the predominant wave direction.

The 1978 ITTC recommended use of $K = 2/\pi$ and $n = 2$ in the absence of better data.

EXTREME WAVE AMPLITUDES

The naval architect is often concerned with the extreme waves that can occur. Suppose a number of records, say M, are taken with the same significant wave amplitude and N values of wave amplitude are recorded from each record. If the greatest wave amplitude in each record is noted there will be M such maxima. Then Table 9.9 expresses the maximum wave amplitudes in terms of the significant amplitude $\zeta_{\frac{1}{3}}$ for various values of N.

Table 9.9
Exceptionally high waves

N	5% lower	Most frequent	Average	5% greater
20	0.99	1.22	1.32	1.73
50	1.19	1.40	1.50	1.86
100	1.33	1.52	1.61	1.94
200	1.45	1.63	1.72	2.03
500	1.60	1.76	1.84	2.14
1000	1.70	1.86	1.93	2.22

Thus, if each record has 200 wave amplitudes then, of the M maximum readings.

5 per cent can be expected to be lower than $1.45\zeta_{\frac{1}{3}}$
The most frequent value = $1.63\zeta_{\frac{1}{3}}$
The average value = $1.72\zeta_{\frac{1}{3}}$
5 per cent can be expected to be greater than $2.03\zeta_{\frac{1}{3}}$

Extreme value statistics were studied by Gumbel and are based on a probability distribution function with a pronounced skew towards the higher values of the variate. Type I extreme value distributions represent the numerical value of probability for the largest values of a reduced variate y as

$$\Phi(x) = \exp[-\exp(-y)]$$

where $y = \alpha(x - u)$

x = extreme variate

α = a scale parameter

u = the mode of the distribution

The probability distribution for the smallest values is

$$\Phi(x) = 1 - \exp[-\exp(y)]$$

In type II distributions the logarithm of the variate is used.

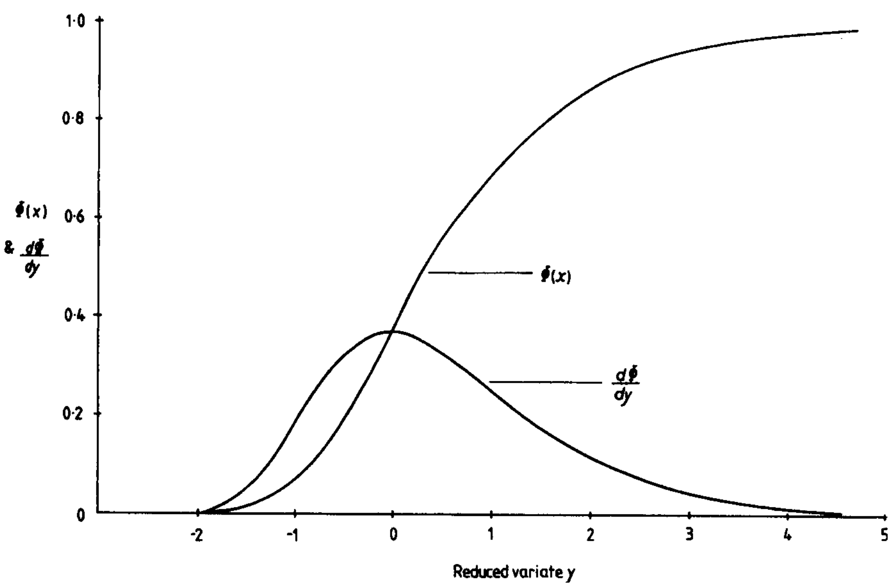


Fig. 9.20 Gumbel Type 1 Extreme value probability and cumulative probability distribution

Plots of $\Phi(x)$ and $d\Phi/dy$ are presented in Fig. 9.20. In this distribution the mode occurs at $y = 0$, the median at $y = 0.367$ and the mean at $y = 0.577$. A special paper is available for plotting data for extreme value statistics (see Fig. 9.21). The reduced variate and ordinate scales are linear. The frequency scale follows from the relationship of Fig. 9.20. The 'return period' scale at the top derives its name from an early application of extreme value statistics to flooding problems.

If observed data are in accord with the Gumbel distribution they will lie close to a straight line when plotted on the special paper, the line being

$$x = u + y/\alpha$$

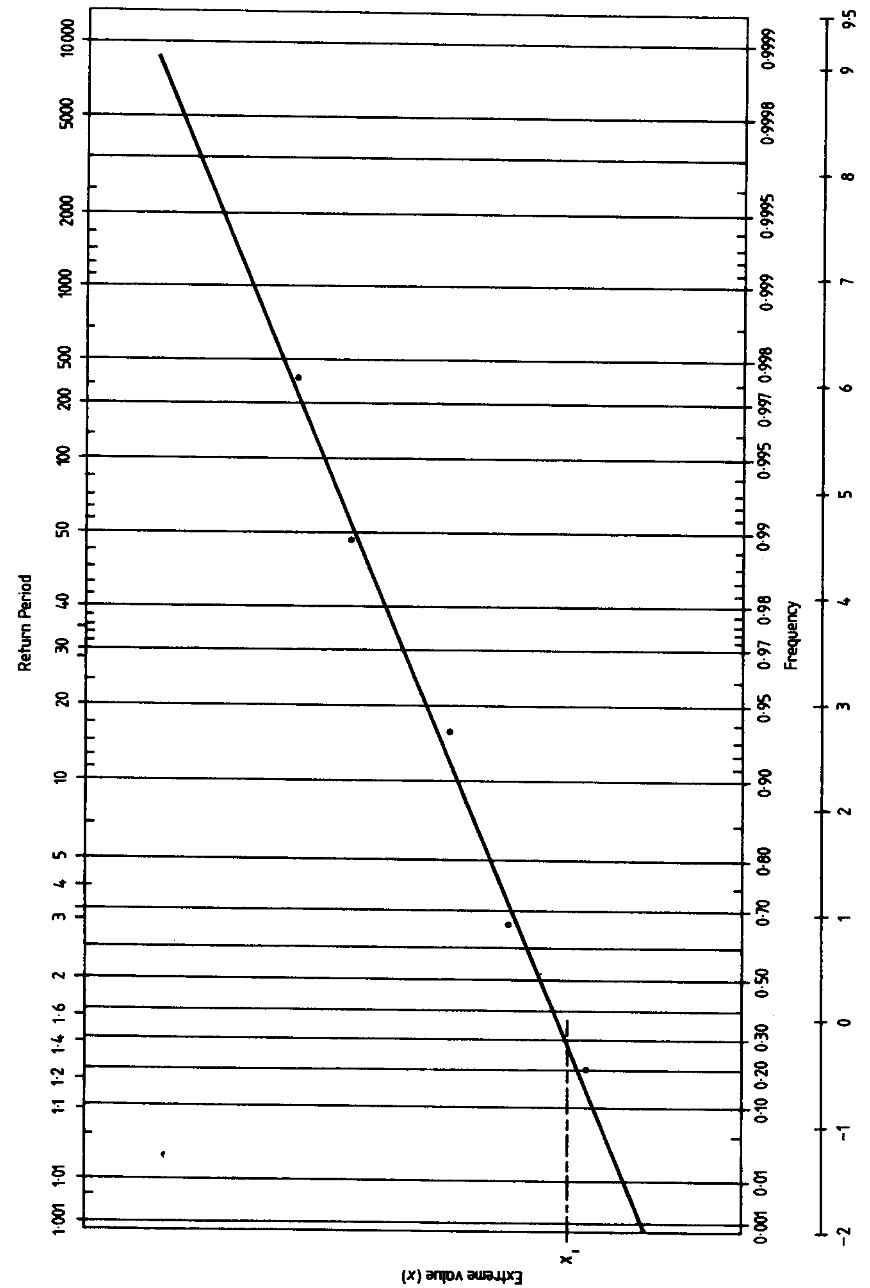


Fig. 9.21

Such a plot can provide a lot of useful information, e.g.

(a) If x_1 is the ordinate of the straight line corresponding to a probability of 0.30 then only 30 per cent of the results will be x_1 or less in value. Conversely, the probability of a result being less than any particular value of x can be found.

(b) The most probable extreme value is obtained by noting the ordinate corresponding to zero on the reduced variate scale. The mean and median values follows from the corresponding reduced variate values quoted above.

(c) To discover how many results are needed to give a 50 per cent probability of observing an extreme value of x_1, the return period corresponding to this ordinate on the straight line is read off.

OCEAN WAVE STATISTICS

The ITTC formulation defines the wave spectrum in terms of the characteristic wave period and the significant wave height. If a designer wishes to ascertain the motions of a design in waves with specific values of these parameters, this can be done provided that the response in regular waves is known. This aspect of ship motions is dealt with later.

More generally the designer is concerned with the motions likely to be experienced during a given voyage or during the life of the ship. For this statistics are needed on the probability of occurrence of various sea conditions at different times of the year on the likely trade routes together with the predominant wave directions. Such statistics are available in a publication by The Stationery Office called *Ocean Wave Statistics,* which has some 3,000 tables based on a million sets of observations of wave heights, periods and directions. Although individually the observations may be suspect, taken in aggregate the overall statistics can provide an accurate picture. There is evidence to show that the characteristic period and significant wave height of the ITTC formula correspond closely to the observed period and height.

The data are presented for 50 sea areas (Fig. 9.22) for which a reasonable number of observations are available and in which conditions are fairly homogeneous. Thus they cover the well-frequented sea routes. Where observers reported both sea and swell the tables present the group of waves with the greater height or, if heights are equal, that with the longer period. The heights and periods are those obtained from the larger well-formed waves of the system observed.

The detailed tables of wave period against wave height are arranged by area, season and wave direction. Summary tables covering all seasons and directions are provided. They are illustrated by Tables 9.10 to 9.12.

The data can be combined and presented in a number of ways. Table 9.13 presents the data for areas 1, 2, 6, 7 and 8 for all seasons and all directions in terms of the expected number of days occurrence in the year. Table 9.14 presents the same data in terms of percentage frequency of occurrence with the data reported under 'calm or period undetermined' spread over the range

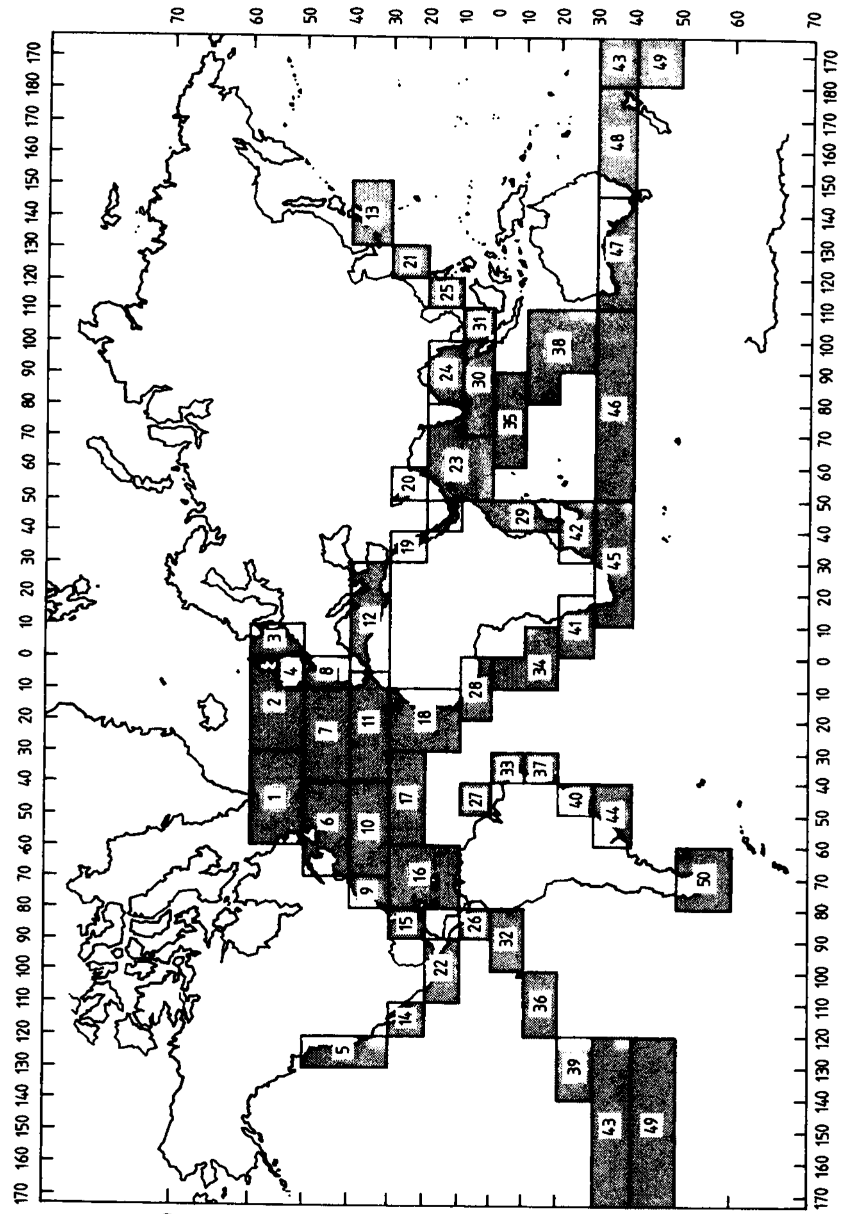


Fig. 9.22

of periods reported in proportion to the observations recorded. Tables 9.15 and 9.16 give similar percentage frequencies for areas 1, 2, 6-11, 16-18 and world wide.

Table 9.10
Area 2. All seasons. All directions (from Ocean Wave Statistics)

	Wave period code											
	x	2	3	4	5	6	7	8	9	0	1	Totals
00	524	514	24	18	11	4		2	1	7	4	1109
01	73	1558	179	46	20	8	8	4	6	5	63	1970
02	186	2800	1448	321	123	42	6	3	6	5	48	4988
03	289	1565	2860	1083	275	85	35	13	6	3	4	6218
04	232	476	2049	1460	388	119	30	10	3			4767
05	190	229	1325	1455	620	185	65	13	6	1	2	4091
06	118	96	681	1015	584	224	48	13	10	2		2791
07	97	48	435	769	555	253	75	13	5			2250
08	52	25	204	480	411	195	58	28	7	2	1	1463
09	69	25	193	421	397	241	69	28	17	1		1461
10	14	3	34	71	80	25	14	2		1	1	245
11	15		21	79	65	56	24	5	1		1	267
12	12	3	34	97	120	78	33	15	3		2	397
13	10	6	40	68	103	71	24	13	2		3	340
14	4	5	21	35	34	31	18	8	1			157
15	4	1	16	45	41	30	14	7	3			161
16	10	2	10	31	35	40	26	9	2		3	168
17	4		1	15	13	21	13	4				71
18	3		4	7	25	15	7	3				64
19	8	2	8	26	32	47	23	12	3			161
91					1				1			2
92					1	2						3
95					1	1						2
Totals	1914	7358	9587	7542	3935	1773	590	205	83	27	132	33146

Wave period code	Wave period seconds
x	Calm or period undetermined
2	5 Or Less
3	6 Or 7
4	8 Or 9
5	10 Or 11
6	12 Or 13
7	14 Or 15
8	16 Or 17
9	18 Or 19
0	20 Or 21
1	Over 21

Wave height code number in left hand column is twice wave height in metres except:

00 = 0.25 m
91 = 11 m
92 = 12 m
95 = 15 m

Table 9.11
Area 2. All seasons

			260°		270°			280°			
	2	3	4	5	6	7	8	9	0	1	
00	50		2	3	1				1		57
01	199	16	7	3	1	2	1			6	235
02	381	252	46	16	5	1	1	1		8	711
03	248	498	191	46	20	4	3			1	1011
04	72	382	276	76	22	7	5	1			841
05	37	285	317	159	46	12	2	3			861
06	19	149	220	131	54	10	1	4	1		589
07	8	84	197	150	67	22	4	2			534
08	6	49	136	105	54	17	14	2	1		384
09	7	48	120	115	82	19	15	5	1		412
10	1	11	29	32	10	3	1			1	88
11		3	27	19	12	5	1				67
12		11	21	42	28	8	3	1			114
13	1	14	18	35	29	5	4	1			107
14		8	16	10	7	7	1				49
15		5	16	18	11	7	3	3			63
16	2	2	16	9	11	13	5	1		3	62
17			5	4	11	7	2				29
18		2	5	13	6	1	1				28
19		1	6	8	17	3	1	1			37
91				1				1			2
92				1	2						3
95					1						1
	1031	1820	1671	996	497	153	68	26	4	19	6285

Table 9.12
Area 2. December to February

			260°		270°		280°				
	2	3	4	5	6	7	8	9	0	1	
00	5		1								6
01	28			1							29
02	39	23	4	2	1					1	70
03	43	53	21	4	2						123
04	12	56	42	9	6	2	3	1			131
05	5	52	39	19	5	2		1			123
06	2	23	45	22	10	1		1			104
07	2	19	42	44	13	4	2				126
08	1	10	24	23	17	5	2				82
09	4	15	38	37	21	7	4				126
10		2	2	13	1		1			1	20
11			12	6	6	2					26
12		2	7	17	6	1	2				35
13	1	4	3	9	8	1	2				28
14			4		5	4	1				14
15			2	6	5	1	2	2			18
16	1		6	8	4	6	2			2	29
17			3	3	7	4	1				18
18			3	7	5						15
19			1	3	10	3	1				18
91								1			1
92					1						1
95					1						1
	143	259	299	233	134	43	23	6		4	1144

Table 9.13
Wave statistics. All seasons. All directions. Days per year northern North Atlantic

Wave height (metres)	Wave Period T_1 (seconds)										Total
	*	≤ 5	6–7	8–9	10–11	12–13	14–15	16–17	18–19	20–21	> 21	
0.25	6.127	7.163	0.264	0.151	0.106	0.036	0.014	0.022	0.012	0.156	0.102	14.153
0.5	0.706	18.977	2.224	0.612	0.295	0.088	0.055	0.028	0.024	0.043	0.863	23.915
1.0	1.747	36.768	17.697	4.029	1.342	0.485	0.146	0.061	0.040	0.071	0.395	62.781
1.5	2.136	19.507	34.577	12.423	3.249	1.071	0.380	0.140	0.045	0.029	0.066	73.623
2.0	1.721	5.638	23.020	17.059	5.355	1.487	0.430	0.133	0.050	0.014	0.014	54.921
2.5	1.272	2.349	13.101	14.968	7.236	2.207	0.725	0.191	0.061	0.021	0.017	42.148
3.0	0.966	0.940	6.340	10.187	6.529	2.437	0.747	0.194	0.055	0.009	0.009	28.413
3.5	0.744	0.555	3.771	7.080	5.353	2.543	0.860	0.225	0.080	0.007	0.007	21.225
4.0	0.437	0.238	1.973	4.128	3.748	2.018	0.690	0.277	0.087	0.012	0.005	13.613
4.5	0.541	0.222	1.642	3.580	3.367	1.992	0.829	0.347	0.106	0.016	0.012	12.654
5.0	0.128	0.038	0.236	0.624	0.648	0.362	0.158	0.040	0.005	0.007	0.005	2.251
5.5	0.097	0.045	0.220	0.576	0.603	0.397	0.212	0.062	0.007	0.005	0.003	2.227
6.0	0.102	0.055	0.342	0.844	0.967	0.586	0.258	0.102	0.028	0.002	0.007	3.293
6.5	0.076	0.057	0.293	0.687	0.927	0.584	0.276	0.107	0.023	0.003	0.007	3.040
7.0	0.047	0.026	0.120	0.347	0.350	0.234	0.113	0.040	0.007	0.000	0.002	1.286
7.5	0.059	0.009	0.149	0.350	0.437	0.298	0.158	0.068	0.016	0.002	0.005	1.551
8.0	0.061	0.016	0.081	0.272	0.317	0.316	0.123	0.073	0.010	0.009	0.012	1.290
8.5	0.029	0.003	0.057	0.120	0.189	0.163	0.102	0.049	0.007	0.005	0.003	0.727
9.0	0.023	0.003	0.031	0.087	0.179	0.140	0.062	0.042	0.007	0.000	0.005	0.579
9.5	0.054	0.010	0.055	0.192	0.321	0.274	0.161	0.085	0.057	0.028	0.014	1.251
> 10	0.000	0.000	0.004	0.007	0.015	0.024	0.005	0.000	0.002	0.002	0.000	0.059
Totals	17.073	92.619	106.197	78.323	41.533	17.742	6.504	2.286	0.729	0.441	1.553	365.000

*Calm or period undetermined

Table 9.14
Percentage frequency of occurrence of sea states, northern North Atlantic

Wave height (metres)	Wave Period T_1 (seconds)									Total
	≤ 5	6–7	8–9	10–11	12–13	14–15	16–17	18–19	20–21	> 21	
Up to 0.50	5.1870	0.3864	0.1364	0.0785	0.0254	0.0125	0.0122	0.0077	0.0655	0.1500	6.0616
0.5–1.25	13.9308	5.4071	1.2508	0.4336	0.1534	0.0515	0.0223	0.0158	0.0282	0.2738	21.5673
1.25–2.5	7.4338	18.1186	10.4412	3.4524	1.0344	0.3308	0.1043	0.0356	0.0153	0.0251	40.9915
2.5–4	0.7898	5.0003	7.5983	4.9232	2.0100	0.6557	0.1850	0.0592	0.0091	0.0077	21.2383
4–6	0.1290	0.9294	2.0748	1.9919	1.1577	0.4774	0.1824	0.0500	0.0099	0.0076	7.0101
6–9	0.0398	0.2517	0.6354	0.7925	0.5554	0.2644	0.1158	0.0227	0.0055	0.0099	2.6931
9–14	0.0033	0.0209	0.0691	0.1201	0.1039	0.0564	0.0302	0.0178	0.0082	0.0047	0.4346
14+	—	0.0005	0.0002	0.0014	0.0012	—	—	—	0.0002	—	0.0035
Totals	27.5135	30.1149	22.2062	11.7936	5.0414	1.8487	0.6522	0.2088	0.1419	0.4788	100.0000

Table 9.15
Percentage frequency of occurrence of sea states, North Atlantic

Wave height (metres)	Wave Period T_1 (seconds) ≤ 5	6–7	8–9	10–11	12–13	14–15	16–17	18–19	20–21	> 21	Total
Up to 0.50	7.1465	0.5284	0.1608	0.0792	0.0318	0.0147	0.0140	0.0083	0.0876	0.2390	8.3103
0.5–1.25	18.6916	6.8744	1.4616	0.4616	0.1798	0.0616	0.0355	0.0158	0.0392	0.3785	28.1996
1.25–2.5	8.6495	19.1881	9.7609	3.0633	0.9056	0.2969	0.0967	0.0269	0.0104	0.0290	42.0273
2.5–4	0.6379	3.7961	5.4827	3.4465	1.4291	0.4597	0.1368	0.0420	0.0067	0.0060	15.4435
4–6	0.0863	0.5646	1.2574	1.2199	0.7040	0.3005	0.1143	0.0338	0.0076	0.0054	4.2938
6–9	0.0258	0.1468	0.3486	0.4417	0.3052	0.1451	0.0614	0.0126	0.0039	0.0057	1.4968
9–14	0.0018	0.0114	0.0362	0.0614	0.0520	0.0320	0.0163	0.0082	0.0046	0.0024	0.2263
14+	—	0.0002	0.0001	0.0006	0.0006	—	—	—	0.0001	—	0.0016
Totals	35.2394	31.1100	18.5083	8.7742	3.6081	1.3105	0.4750	0.1476	0.1601	0.6660	99.9992

Table 9.16
Percentage frequency of occurrence of sea states, world wide

Wave height (metres)	Wave Period T_1 (seconds) ≤ 5	6–7	8–9	10–11	12–13	14–15	16–17	18–19	20–21	> 21	Total
Up to 0.50	10.0687	0.5583	0.1544	0.0717	0.0272	0.0117	0.0087	0.0076	0.0959	0.2444	11.2486
0.5–1.25	22.2180	6.9233	1.4223	0.4355	0.1650	0.0546	0.0268	0.0123	0.0394	0.3878	31.6850
1.25–2.5	8.6839	18.1608	9.0939	2.9389	0.8933	0.2816	0.0889	0.0222	0.0079	0.0230	40.1944
2.5–4	0.5444	3.0436	4.4913	2.9163	1.2398	0.4010	0.1220	0.0322	0.0050	0.0049	12.8005
4–6	0.0691	0.3883	0.8437	0.8611	0.5214	0.2244	0.0866	0.0229	0.0045	0.0033	3.0253
6–9	0.0223	0.0928	0.2164	0.2632	0.1842	0.0908	0.0419	0.0081	0.0032	0.0034	0.9263
9–14	0.0014	0.0067	0.0186	0.0316	0.0258	0.0177	0.0094	0.0044	0.0023	0.0011	0.1190
14+	—	0.0001	0.0001	0.0003	0.0003	—	—	—	0.0001	—	0.0009
Totals	41.6078	29.1739	16.2408	7.5187	3.0571	1.0814	0.3844	0.1098	0.1583	0.6679	100.0001

Climate

THE WIND

The main influence of the wind is felt indirectly through the waves it generates on the surface of the sea. As stated above, the severity of these waves will depend upon the strength (i.e. velocity) of the wind, the time it acts (i.e. its duration) and the distance over which it acts (i.e. the fetch).

The strength of the wind is broadly classified by the *Beaufort Scale*. The numbers 0 to 12 were introduced by Admiral Sir Francis Beaufort in 1806, 0 referring to a calm and force 12 to a wind of hurricane force. Originally, there were no specific wind speeds associated with these numbers but the values shown in Table 9.17 have now been adopted internationally. The figures relate to an anemometer at a height of 6 m above the sea surface.

No fixed relation exists between the spectra of a sea and the speed of wind generating it. Figure 9.23 shows a relationship agreed by the **ITTC** but it applies only to fully developed seas where duration and fetch are large. Figure 9.24 shows the probability of exceeding a given wind speed in the North Atlantic.

Table 9.17
The Beaufort Scale

	(Kts)	*Limits of speed (m/s)*
0 Calm	1	0.3
1 Light air	1–3	0.3–1.5
2 Light breeze	4–6	1.6–3.3
3 Gentle breeze	7–10	3.4–5.4
4 Moderate breeze	11–16	5.5–7.9
5 Fresh breeze	17–21	8.0–10.7
6 Strong breeze	22–27	10.8–13.8
7 Near gale	28–33	13.9–17.1
8 Gale	34–40	17.2–20.7
9 Strong gale	41–47	20.8–24.4
10 Storm	48–55	24.5–28.4
11 Violent Storm	56–63	28.5–32.6
12 Hurricane	64 & over	32.7 & over

The wind velocity varies with height. At the surface of the water, the relative velocity is zero due to a boundary layer effect. Stability calculations often adopt a nominal wind at 10m above the waterline, and the variation in wind velocity with height is assumed to be in accord with Fig. 9.25. It will be noted, that these nominal velocities will be about 6 per cent less than those defined by the Beaufort Scale at a height of 6 m above the surface. Also given in Fig. 9.25 is a curve based on this nominal height. Because of this variation of wind speed with height, it is important to define both in any calculati on.

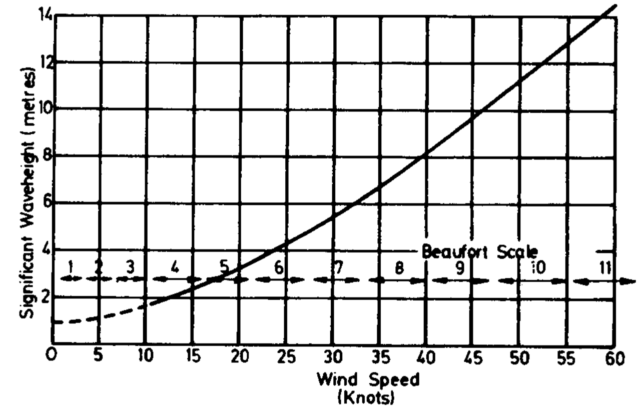


Fig. 9.23

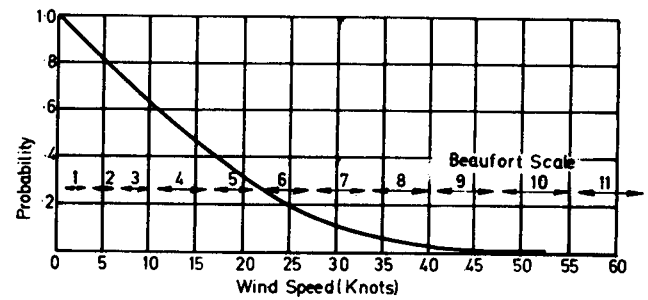
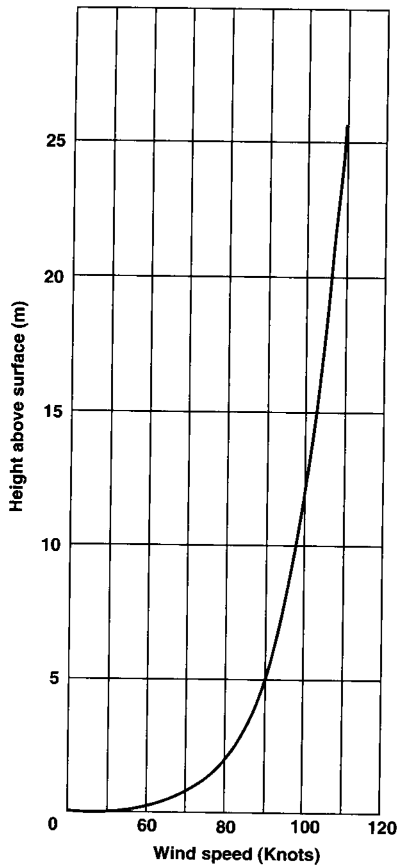


Fig. 9.24

For nominal velocities other than 100 knots, the true velocities can be scaled in direct proportion to the nominal velocities.

More direct influences of the wind are felt by the ship as forces acting on the above water portions of the hull and the superstructure. The fore and ~ components of such forces will act either as a resistive or propulsive force. The lateral force will act as a heeling moment. Thus, standards of stability adopted in a given design must reflect the possible magnitude of this heeling moment, e.g. large windage area ships will require a greater stability standard as discussed in Chapter 4.

Another direct influence of the wind, including the effect of the ship's own speed, is to cause local wind velocities past the superstructure which may entrain the funnel gases and bring them down on to the after decks, or which may produce high winds across games decks to the discomfort of passengers. These effects are often studied by means of model experiments conducted in a wind tunnel, in order to obtain a suitable design of funnel and superstructure.



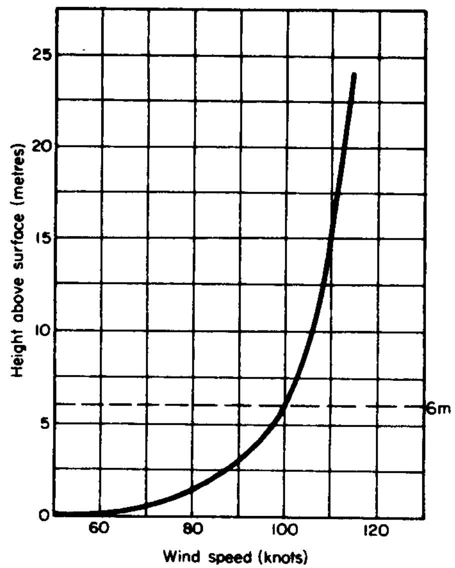


Fig. 9.25 Variation of wind speed with height above sea surface (nominal wind speed 100 knots)

AMBIENT AIR

A ship is completely immersed in two fluids-air for the upper portion and water for the lower portion. Because the specific heat of water is much greater than that for air, the former suffers a much slower variation of temperature due to climatic changes. In defining the temperature environment of the ship, it is necessary to quote temperatures for both the air and water. Further, it is common experience that the humidity of air is important in determining the degree of comfort in hot climates. This idea is elaborated later in this chapter, but for the present, it is enough to know that for air it is necessary to specify the temperature that would be recorded on both a 'wet bulb' and a 'dry bulb' thermometer.

Standard temperatures considered by the UK Ministry of Defence as being appropriate to different regions of the world at various times of the year are given in Table 9.18.

Table 9.18
Standard climatic figures, UK Ministry of Defence

Region	Average max. Summer temperature (°C)			Average min. Winter temperature (°C)		
	Air		Deep sea	Air		Deep sea
	Dry bulb	Wet bulb	Water temp. near surface	Dry bulb	Wet bulb	Water temp. near surface
Extreme Tropics	34.5	30	33			
Tropics	31	27	30			
Temperature	30	24	29			
Temperate Winter				−4	—	2
Sub-Arctic Winter				−10	—	1
Arctic and Antarctic Winter				−29	—	−2

For air conditioning calculations, conditions given in Table 19.19 are recommended; figures in parentheses are the approximate values in degrees Celsius.

Table 9.19

	Dry bulb (°C)	Wet bulb (°C)	Relative humidity (%)
Red Sea passenger vessels	32	30	85
Other passenger vessels	31	27	70
Extreme Tropical cargo vessels and tankers	32	29	78

Table 9.20
Extreme recorded air temperatures (°C)

	Harbour	At sea
Tropics	52	38
Arctic	−40	−30

These temperatures are important to the ship design in a number of ways. The air temperature, for instance, influences: ,

(a) the amount of heating or cooling required for air entering the ship to maintain a given internal temperature;
(b) the amount of thermal insulation which can be economically justified;
(c) efficiency of some machinery;
(d) the amount of icing likely to occur in cold climates and thus the standard of stability required to combat this (Chapter 4), and the need to provide special de-icing facilities.

The water temperature influences:

(a) density of water and thus the draught;
(b) the efficiency of, and hence the power output from, machinery, e.g. steam machinery may suffer a loss of about 3 per cent of power in the tropics

compared with temperate conditions. The effect on other machinery may be greater, but all types are affected to some degree including auxiliaries such as electrical generators, pumps, and air conditioning plant;

(c) whether ice is likely to be present and if so how thick and, thus, whether special strengthening of the ship is necessary.

Both air and water temperatures can cause expansion or contraction of materials leading to stresses where two dissimilar materials are joined together. Temperature differences between the air and water can lead to relative expansion or contraction of the upper deck relative to the keel causing the ship to hog or sag. This leads to stresses in the main hull girder and can upset the relative alignment of items some distance apart longitudinally. Similarly, if one side of the ship is in strong sunlight the main hull will tend to bend in a horizontal plane.

The above temperatures are used in calculations for assessing the capacity required of ventilation or air conditioning systems or for the degradation in machinery power generation.

Equipment should be tested in accordance with standard specifications:

(a) in locations where the ambient temperature rises and remains at a relatively constant level, e.g. in machinery spaces where the temperature is mainly influenced by the heat generated by machinery;
(b) where the adjacent air temperatures are indirectly affected by solar heating, e.g. equipment in the open but not fully exposed;
(c) fully exposed to solar radiation. In this case not only surface heating is important but also the degradation of materials caused by the ultra-violet radiation;
(d) in low temperature environments.

The tests may have to cater for high or low humidity conditions and may need to allow for diurnal variations.

CLIMATIC EXTREMES

The designer must allow for extreme climatic conditions although these may only be met on a few occasions. These are usually specified by an owner. British Standards Specifications and the MIL Specifications of the USA are usual. Care is necessary to ensure that the latest specifications are used.

Dust and sand. These cause abrasion of surfaces and penetrate into joints where movements due to vibration, and to changes in temperature, may bring about serious wear. Movements of particles can also set up static electrical charges. Although arising from land masses, dust and sand can be carried considerable distances by high winds. Specifications define test conditions, e.g. grade of dust or sand, concentration, air velocity and duration.

Rain. Equipments should be able to operate without degradation of performance in a maximum rainfall intensity of 0.8 mm/ min at 24°C and a wind speed of 8 m/s for periods of 10 min with peaks of 3 mm/ min for periods of 2 min. The test condition is 180mm/h of rain for 1 hour. Driving sea spray can be more corrosive than rain.

Icing. Equipments fitted in the mast head region must remain operational unll safe with an ice accretion rate of 6.4 mm/h with a total loading of 24 kg/m^2 Corresponding figures for equipment on exposed upper decks are 25 mm/h and 70 kg/m^2. Such equipments must be able to survive a total loading or 120kg/m^2. Equipments fitted with de-icing facilities should be able to shed 25 mm ice per hour.

Hail. Equipment should be able to withstand without damage or degradation of performance a hailstorm with stones of 6 to 25 mm, striking velocity 14 to 25 mis, duration 7 min.

High winds. Exposed equipments should remain secure and capable of their design performance in a relative steady wind to ship speed of 30 m/ s with gusts of 1 min duration of 40m/s. They must remain secure, albeit with some degradation of performance in wind to ship speed of 36 m/s with gusts of 1 min duration of 54 m/ s.

Green sea loading. With heavy seas breaking over a ship, exposed equipment and structure are designed to withstand a green sea frontal pulse loading of 70 kPa acting for 350 ms with transients to 140kPa for 15 ms.

Solar radiation. Direct sunlight in a tropical summer will result in average surface temperatures of up to 50°C for a wooden deck or vertical metal surfaces and 60°C for metal decks. Equipment should be designed to withstand the maximum thermal emission from solar radiation which is equivalent to a heat flux of 1120 W/m^2 acting for 4 hours causing a temperature rise of about 20°C on exposed surfaces.

Mould growth. Mould fungi feed by breaking down and absorbing certain organic compounds. Growth generally occurs where the relative humidity (rh) is greater than 65 per cent and the temperature is in the range 0 to 50°C. Most rapid growth occurs with rh greater than 95 per cent and temperature 20 to 35°C and the atmosphere is stagnant. Equipments should generally be able to withstand exposure to a mould growth environment for 28 days (84 for some critical items) without degradation of performance. Moulds can damage or degrade performance by:

(a) direct attack on the material, breaking it down. Natural products such as wood, cordage, rubber, greases, etc., are most vulnerable;
(b) indirectly by association with other surface deposits which can lead to acid formation;
(c) physical effects, e.g. wet growth may form conducting paths across insulating materials or change impedance characteristics of electronic circuits.

Physical limitations

There are various physical limitations, either natural or man-made, which must be considered by the designer. They are:

(a) Depth of water available

The draught of the ship including any projections below keel level must not exceed the minimum depth of water available at any time during service. This depth may be dictated by the need to negotiate rivers, canals, harbour or dock entrances. If the vessel has to operate for any distance in shallow water, the increased resistance which occurs in these conditions must be taken into account. Various empirical formulae exist for estimating the depth of water needed to avoid loss of speed. One is:

$$\text{Depth in fathoms} > 3.02\,\frac{V\sim}{(L)2}$$

where T and L are in metres and V is in knots.

Allowance must be made for the sinkage and trim that will occur. For example, a large ship drawing say 7.5 m of water may strike bottom in 8.5 m of water if it attempts to negotiate the shallows too quickly.

(b) Width of water available

This is usually limited by the need to negotiate canals, harbour or dock entrances. As with shallow water the ship is acted upon by additional interference forces when negotiating narrow stretches at speed. It may be necessary to adopt special manoeuvring devices for ships habitually operating in confined waters.

(c) *Height available above the water line*

Most ships have, at one time or another, to pass below bridges in entering ports. Unless the bridge itself can be opened, the ship must either keep masts and funnels below the level of the bridge or the uppermost sections must be capable of being lowered. Tidal variations must be catered for.

(d) Length

The length of a ship may be limited by the building slips, fitting-out berths and docks available or by the length of canal lock or harbour dimensions.

The internal environment

A full study must embrace all those aspects of the environment which affect the efficiency of equipment and/or the crew. Thus it must include

(a) Movement. The ship is an elastic body acted on by a series of external forces. A thorough study should deal with the vital ship response in an integrated way. In practice it is usual to deal separately with the response of the ship as a rigid body (ship motions), as an elastic body (vibrations) and under impulsive loading (wave impact, collision or enemy action). The last item is dealt with in Chapter 5.

(b) Ambient conditions including quality of air (temperature, humidity, freshness), noise levels (from machinery, sea, wind) and lighting levels. In some spaces the levels of the ambient conditions will be more critical than in others. This may dictate layout or special environmental control systems.

Motions

The study of ship motions *per se* is considered in Chapter 12. As far as the ship itself is concerned, the designer is concerned with the effects on the structure, equipment and crew.

For structure and equipment it is often sufficient to consider certain limiting conditions of amplitude and frequency for design purposes when associated with a factor of safety based upon previous successful design. More strictly, any amplitudes of motion should be associated with the probability of their occurrence during the life of a ship.

To avoid overdesign, it is usual to consider two sets of motion figures; those under which equipment should be capable of meeting its specification fully, and a second set in which the equipment must be able to function albeit with reduced performance. Again, certain equipment needs only to function in certain limited sea states, e.g. that associated with transfer of stores between two ships at sea.

Typical figures for which equipment must remain secure and able to operate without degradation are given in Table 9.21 for a medium size warship. Figures for acceleration are taken to be 1.5 times those calculated assuming simple harmonic motion. They are based on Sea State 7.

Table 9.21

	Roll	Pitch	Yaw	Heave
Period (s)	10	5-6.5		7
Amplitude (±)	18°	8°		3.5m
Acceleration *(o/s2)*			1.75	

As regards effects on the crew, most people have felt nausea when subject t~ large motion amplitudes, having earlier experienced a reduction in mentaJ alertness and concentration. Whilst the underlying cause is reasonably clear surprisingly little is known about the degree of degradation suffered and the motions which contribute most to this degradation. The subject assumes greater significance for modern ships with smaller complements and more complex systems requiring greater mental capability. For warships the problems are compounded by the desire to go to smaller ships which, for conventional hull forms at least, implies larger motion amplitudes.

It is the effect that motions have on the vestibula and labyrinthine systems of the human body that causes the feelings of nausea. In one interesting experiment carried out by the US Coastguard, a number of 'labyrinthine defectives' were subject to very severe motion conditions. None of these subjects vomited although normal subjects in the same situation all did. All the stimulants believed to be relevant to inducing nausea in subjects were present including fear for own safety and witnessing others being sick.

What is not clear is which motions (roll, pitch) and which aspects of those motions (amplitude, velocity, acceleration) have most effect. It appears that linear or angular accelerations provide the best correlation between motion and sickness, the linear acceleration being caused mainly by a combination of pitch and heave with a smaller contribution from roll and the most significant angular acceleration being that due to roll. Frequency of motion is a vital factor (see Fig. 9.26).

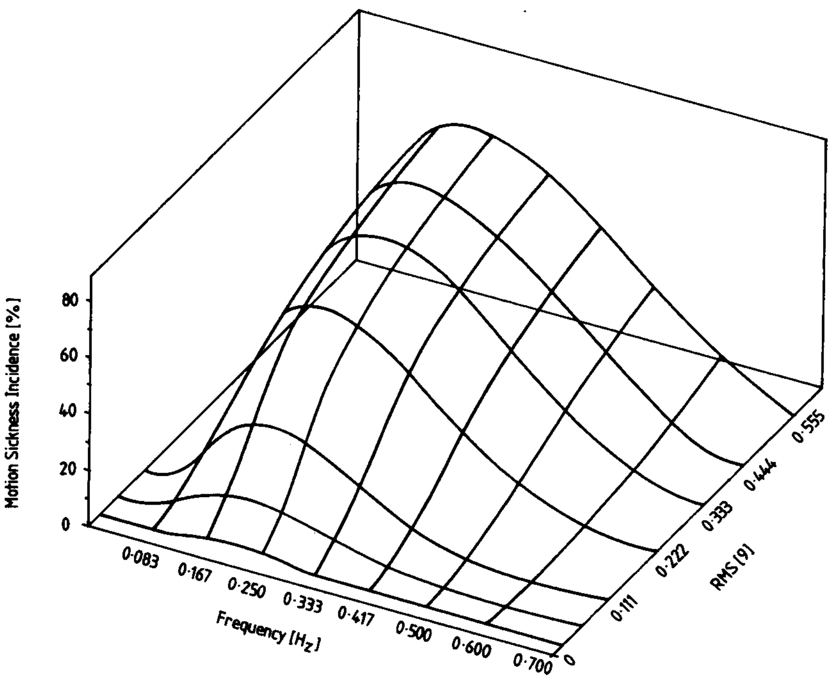


Fig. 9.26 *Motion sickness incidence*

A number of steps can be taken to mitigate the effects of motion. Drugs can suppress feelings of nausea although it is not clear whether the side effects, such as drowsiness, mean the person is any better on balance in carrying out some complex mental task. The ship designer can arrange for vital human operations to be arranged in an area of minimum motion or for the operators to be in a prone position. The designer could even provide a stabilized platform but all these imply some offsetting disadvantage in the overall design and therefore some objective means of judging the 'trade-offs' is needed. These are not yet available but the problem is receiving increasing attention.

Motions may cause not only a fall off in human abilities but also make a task itself physically more difficult. Thus moving heavy weights about the ship, during a replenishment at sea operation, say, is made more difficult and

dangerous. The designer is normally expected to design for such operations up to some limiting sea state. See also Chapter 12.

The air

If no action were taken to modify the air in a closed compartment within the ship, what would happen? Consider the case where people and heat producing equipment are both present. Then:

(a) due to the people and equipment the temperature within the space would rise steadily until the heat passing through the compartment boundaries matched the heat input from within;
(b) due to the moisture breathed out by the people the moisture content of the air would increase;
(c) due to the people breathing, the oxygen content would decrease and the carbon dioxide content increase;
(d) due to the movement of the people dust would be created;
(e) smells would be produced.

Put briefly, the temperature, humidity, chemical purity and physical purity are changed. Complete air conditioning therefore involves control of all these factors. Only in special cases, such as the submarine, is the chemical purity of the air controlled by using special means to produce new oxygen and absorb the carbon dioxide, although in all surface ships a certain minimum quantity of fresh air is introduced to partly control the chemical balance. This is typically about 0.3 m^3 of fresh air per person per minute. In general, the physical purity is only controlled within broad limits by filters in air distribution systems. Exceptions to these may be special 'clean' workshops or operating theatres. With only one device, namely cooling, to effect control of both temperature and humidity some compromise between the two is necessary. More precise control of both can be effected by cooling in conjunction with after-warming. This is discussed more fully in Chapter 14.

Temperature is measured by means of a thermometer in *degrees Celsi1lll (Centigrade)* (0C). Pure water freezes at O°C and boils at 100°C at sea level.

It is a common experience, that the sunny side of a house feels warmer than the side in shade. This is because the body feels not only the warmth of the surrounding air but also the warmth of the sun's radiation. To define temperature precisely it is necessary to distinguish between these two. The *dry bulb temperature* is that measured in still air by a thermometer which is unaffected by radiated heat. To measure the effect of radiated heat a special device known as a globe thermometer is used. In this, the conventional thermometer is enclosed in a black sphere. This device records the *globe temperature.*

Another common experience is that one feels chilly in wet clothing. This is because the water, as it evaporates, absorbs heat from the body. This effect is measured by the *wet bulb temperature* which is that recorded by a thermometer with its bulb covered by wet muslin and subject to a moving air stream. It follows, that the wet bulb temperature can never be higher than the dry bulb and will, in general, be lower. Air can only hold a certain quantity of water vapour at a given temperature. The higher its moisture content the slower the rate of water evaporation from a wet body and the closer the wet bulb temperature will approach the dry bulb measurement.

Thus, the amount of water vapour present is important in relation to the maximum amount the air can contain at that temperature. This ratio is known as the *relative humidity.* The ability of air to hold water vapour increases with increasing temperature. If the temperature of a given sample of air is lowered, there comes a time when the air becomes *saturated* and further reduction leads to condensation. The temperature at which this happens is known as the *dew point* for that sample of air. If a cold water pipe has a temperature below the dew point of the air in the compartment, water will condense out on the pipe. This is why chilled waterpipes are lagged if dripping cannot be tolerated.

Since a human being's comfort depends upon temperature, humidity and air movement, it is difficult to define comfort in terms of a single parameter. For a given air state, an *effective temperature* is defined as the temperature of still, saturated air which produces the same state of comfort.

When a kettle of water is heated, the temperature rises at first and then remains constant (at 100°C) while the water is turned into steam. To distinguish between these two conditions, heat which causes only a temperature change is called *sensible heat* (i.e. it is clearly detectable) and heat which causes only a change of state is called *latent heat* (i.e. hidden heat). The sum of these two heats is known as the *total heat.* The unit used to measure heat is the joule.

This section covers briefly the basic definitions associated with air conditioning comfort. These are put to use in the design of an air conditioning system in Chapter 14.

Lighting

There is a clear need for certain compartments to have a minimum level of illumination in order that work in them can be carried out efficiently.

Light is an electromagnetic radiation and the eye responds to radiation in that part of the spectrum between 7600 Å and 3800 Å. In this, Å stands for angstrom which is a wave-length of 10^{-10} m. The energy of the radiation can be expressed in terms of watts per unit area but the eye does not respond equally to all frequencies. Response is greater to greens and yellows than to reds and blues. Initially, wax candles were used as standards for illumination. A *lumen* (1m) is the amount of luminous energy falling per unit area per second on a sphere of unit radius from a point source of one candle-power situated at the centre of the sphere. The intensity of illumination is defined by:

$$1\,\mathrm{lm/m^2} = 1\,\mathrm{lux} = 1\ \text{metre-candle}$$

Suggested standards of illumination for merchant ships are given in Table 9.22.

Table 9.22
Suggested standards of illumination

Space	Im/m2	Space	Im/m2
Passenger cabins	75	Nursery	108
Dining rooms	108-161	Engine rooms	161-215
Lounges	75-108	Boiler rooms	108
Passageways	22-54	Galleys	161
Toilets	75	Laundries	161
Shops	215	Store-rooms	75

Standards used within the UK Ministry of Defence are given in Table 9.23- local figures relate to desk tops, instrument dials, etc. Special fittings are used in dangerous areas such as magazines, paint shops, etc., and an emergency system of red lighting is also provided. In darkened ship conditions, only this red lighting is used over much of the ship so as not to impair night vision.

In some ships, lighting may be regarded as merely a necessary service but in large ships and particularly in passenger liners, lighting can have a considerable influence on atmosphere. This can be especially important in the public rooms, and shipowners often enlist the services of lighting engineers and designers. Light intensity and colour is important and also the design of the light fittings themselves and their arrangement, e.g. it is essential to avoid glare.

Table 9.23
Lighting standards, RN ships

Space	Level of illumination (lux)	
	General	Local
Accommodation spaces	150	300
Machinery spaces	100	300
Galley, bakery	100	200
Workshops	150	400
Magazines	150	300
Stores, general	100	300
Switchboards	150	300
Bathrooms	100	
Dental clinic	200	400

Economic considerations lead to a desire to rationalize light fittings throughout the major part of the ship and suggests that a 230 volt, 60 hertz unearthed system will become standard for merchant ships. The Ministry of Defence favours a three wire 115/0/115 volt, 60 hertz, unearthed system which provides a very flexible system with the choice of 115 or 230 volts to suit electrical demands of all sorts, including lighting.

In deciding upon the type of light fitting, it is necessary to consider the efficiency of the appliance, its probable life and the effect it will have on the general lighting system. Efficiency is important in air-conditioned spaces as less heat is generated in the production of a given light intensity. Fluorescent lighting is fitted in all important, regularly used, compartments in RN ships. The levels of illumination achieved can be expected to drop by 30 per cent over time due to the slow deterioration of the reflectance of bulkheads etc. and within the light fitting itself.

Table 9.24
Efficiency and life of light fittings

Type of fitting	Efficiency (lm/W)	Life (hr)
Single coil tungsten	8-20	1000
Tungsten-quartz iodine	22	2000
Fluorescent	24-66	5000
Cold cathode	10-40	30,000

Vibration and noise

Vibration theory is dealt with in many textbooks. In the present context, it is sufficient to point out that the ship is an elastic structure containing a number of discreet masses and, as such, it will vibrate when subject to a periodic force.

VIBRA TION

Ships must be designed to provide a suitable environment for continuous and efficient working of equipment and in which the crew can perform comfortably, efficiently and safely. Criteria relating to the evaluation of vibration in merchant ships is laid down in British Standards and International Standards as are tolerance to whole-body vibration.

EXCITATION

Periodic forces causing excitation can arise from:

The propulsion system. Misalignment of shafts and propeller imbalance can cause forces at a frequency equal to the shaft revolutions. Forces should be small with modern production methods. Because it operates in a non-uniform flow the propeller is subject to forces at blade rate frequency - shaft revs x number of blades. These are unlikely to be of concern unless there is resonance with the shafting system or ship structure. Even in a uniform flow a propulsor induces pressure variations in the surrounding water and on the ship's hull in the vicinity. The variations are more pronounced in non-uniform flow particularly if cavitation occurs. If cavitation is fairly stable over a relatively large arc it represents in effect an increase in blade thickness and the blade rate pressures increase accordingly. If cavitation is unstable pressure amplitudes may be many times greater. Whilst the number of blades is important to the frequency it has little effect on pressure amplitude. The probability of vibration problems in single screw ships can be reduced by using bulbous or *V-rather* than V-sections

in the after body, avoiding near horizontal buttock lines above the propeller, providing good tip clearance between propeller and hull, avoiding shallow immersion of the propeller tips to reduce the possibility of air drawing and avoiding low cavitation numbers. Generally the wake distribution in twin screw ships is less likely to cause vibration problems. If A-brackets are used, the angle between their arms must not be the same as that between the propeller blades.

Wavejorces. A ship in waves is subject to varying pressures around its hull. The ship's rigid body responses are dealt with in Chapter 12. Because the hull is elastic some of the wave energy is transferred to the hull causing main hull and local vibrations. They are usually classified as springing or whipping vibrations. The former is a fairly continuous and steady vibration in the fundamental hull mode due to the general pressure field acting on the hull. The latter is a transient vibration caused by slamming or shipping green seas. Generally vertical vibrations are most important because the vertical components of wave forces are dominant. However, horizontal and torsional vibrations can be important in ships with large openings or of relatively light scantlings, e.g. container ships or light carriers. The additional bending stresses due to vibration may be significant in fatigue because of the frequency, and the stresses caused by whipping can be of the same order of magnitude as the wave bending stresses.

Machinery. Rotating machinery such as turbines and electric motors generally produce forces which are of low magnitude and relatively high frequency. Reciprocating machinery on the other hand produces larger magnitude forces of lower frequency. Large diesels are likely to pose the most serious problems particularly where, probably for economic reasons, 4 or 5 cylinder engines are chosen with their large unbalance forces at frequencies equal to the product of the running speed and number of cylinders. Auxiliary diesels are a source of local vibrations. Vibration forces transmitted to the ship's structure can be much reduced by flexible mounting systems. In more critical cases vibration neutralizers can be fitted in the form of sprung and damped weights which absorb energy, or active systems can be used which generate forces equal but anti-phase to the disturbing forces.

RESPONSES

As with any vibratory phenomenon, the response of the ship, or part of the ship, to an exciting force depends upon the frequency of the excitation compared with the natural frequency of the structure and the damping present as indicated in Fig. 9.27.

In this figure, the magnification Q is defined as

$$Q = \frac{\text{dynamic response amplitude}}{\text{static response amplitude}}$$

and w/Wo is the ratio of the frequency of the applied disturbance to the natural frequency of the structure. It should be noted, that the most serious vibrations occur when the natural frequency of the structure is close to that of the applied force, i.e. at resonance.

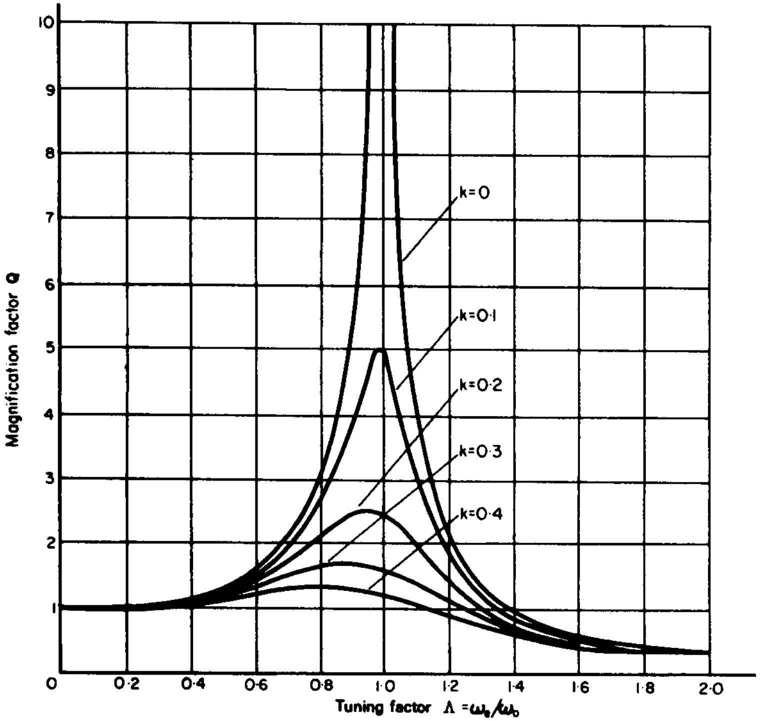


Fig. 9.27 Magnification factor

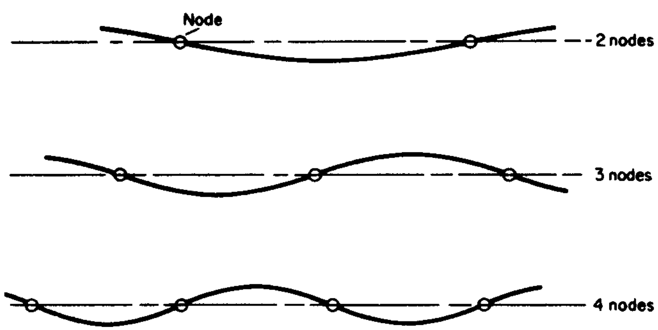


Fig. 9.28 Modes of vertical vibration

The response by the ship may be as a whole or in a local area or piece of structure. In the former case the ship responds to the exciting forces by vibrating as a free-free beam. In this type of vibration, certain points along the length suffer no displacement and these points are called *nodes*. The term *anti-nodes* is used for the points of maximum displacement between nodes. The hull can vibrate in different ways, or *modes*, involving 1, 2, 3,4 or more nodes (Fig. 9.28), although the single node mode applies only to torsional vibration. The natural frequency of the vibration increases as the number of nodes increases as is discussed later.

There is little that a designer can do to prevent this free-free vibration and there is little that can be done to alter the frequencies at which the resonances occur. Their existence must be recognized, however, and the critical frequencies calculated so that an endeavour can be made to avoid any exciting forces at these values. Often, a Master gets to know when to drive fast through such a problem.

The figures given in Fig. 9.29 and Table 9.25 relate specifically to the vertical vibration. Figure 9.29 is intended to be used for evaluating hull and superstructure vibration indicating where adverse comment is to expected. It is applicable to turbine and diesel driven merchant ships 100 m long and longer. It is not intended to establish vibration criteria for acceptance or testing of machinery or equipment. The figures in the third column of Table 9.25 are used to evaluate the responses of equipments and detect resonances which the designer will endeavour to design out. Resonances are considered significant when the dynamic magnification factor, Q, exceeds 3. The endurance tests are then conducted at the fixed frequencies shown in the fourth column plus any frequencies, determined by the response tests, giving rise to significant resonances which the designer was unable to eliminate. Transverse vibration will generally be rather less, but for design purposes is usually assumed equal to it. Fore and aft vibration amplitudes are generally insignificant except at the masthead position, but even here they are low compared with the vertical and transverse levels. The remaining vibratory mode—the torsional—is not common and there is not a lot of available evidence on its magnitude.

Table 9.25
Vibration response and endurance test levels for surface warships

Ship type	Region	Standard test level Peak values and frequency range	Endurance tests
Minesweeper size and above	Masthead	1.25 mm, 5 to 14 Hz 0.3 mm, 14 to 23 Hz 0.125 mm, 23 to 33 Hz	1.25 mm, 14 Hz 0.3 mm, 23 Hz 0.125 mm, 33 Hz Each 1 hour
	Main	0.125 mm, 5 to 33 Hz	0.125 mm, 33 Hz For 3 hours
Smaller than minesweeper	Masthead and main	0.2 mm or a velocity of 63 mm/s whichever is less 7 to 300 Hz	0.2 mm, 50 Hz For 3 hours
	Aftermost 1/8 of ship length	0.4 mm or a velocity of 60 mm/s whichever is less 7 to 300 Hz	0.4 mm, 24 Hz For 3 hours

Notes:
1. The masthead region is that part of the ship above the main hull and superstructure.
2. The main hull includes the upper deck, internal compartments and the hull.

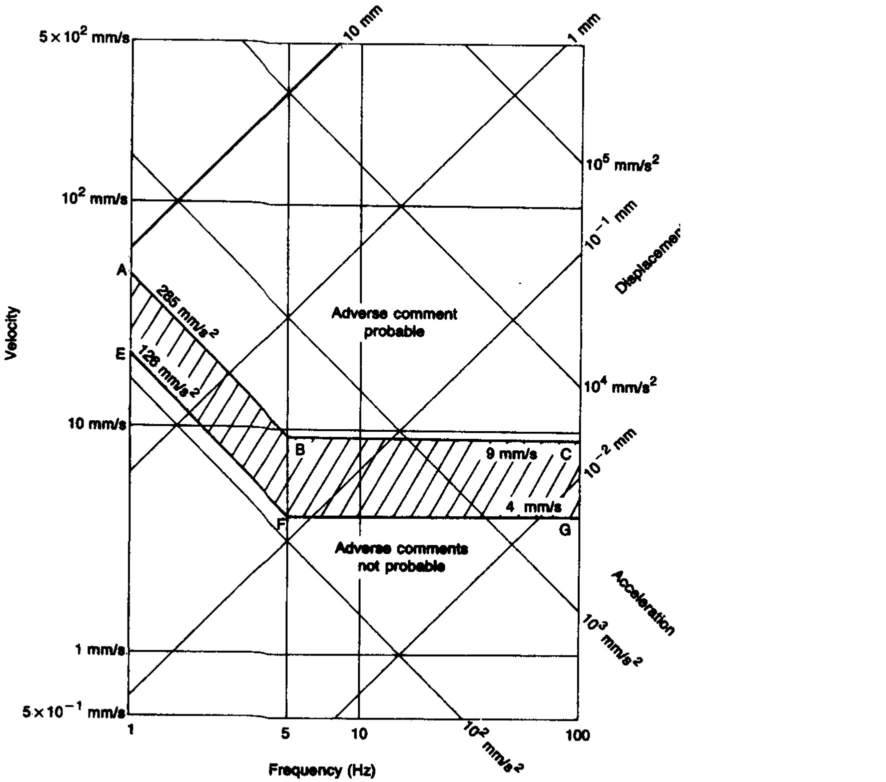


Fig. 9.29 Guidelines for the evaluation of vertical and horizontal vibration in merchant ships (peak values)

The local response of the ship, due either to ship girder vibration or direct resonance with the disturbing forces, is curable and is therefore of less vital interest. A deck, bulkhead, fitting or panel of plating may vibrate. Considering all the possible vibratory systems in local structures, it is impossible to avoid some resonances with the exciting forces. Equally, it is impossible to calculate all the frequencies likely to be present. All the designer can do is select for calculation those areas where vibration would be particularly obnoxious; for the rest, troubles will be shown up on trial and can be cured by local stiffening, although this is an inconveniently late stage.

BODY RESPONSE

The human body responds to the acceleration rather than the amplitude and frequency of vibration imposed upon it. Humans can also be upset by vibrations of objects in their field of view, e.g. by a VDU which they need to monitor. Fig. 9.30 is concerned with vibrations transmitted to the body through a supporting surface such as feet or buttocks. The limits shown in Fig. 9.30 relate to vibration at the point of contact with the subject. Thus for a seated person

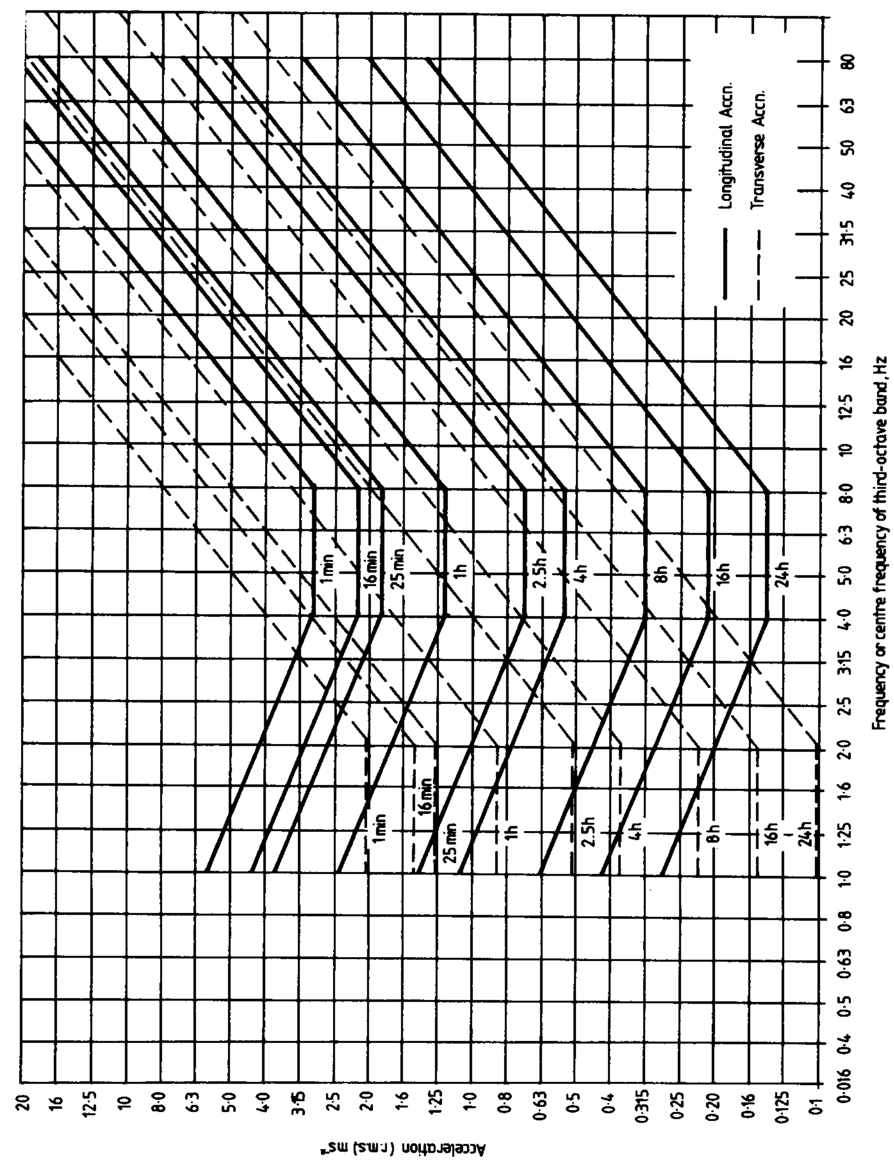


Fig. 9.30 Acceleration limits as a function of frequency and exposure time; 'fatigue—decreased proficiency boundary'

they refer to the vibration transmitted through the cushion. In working from the basic ship structure vibration levels allowance must be made for the transmission properties of anything, e.g. the chair, between the structure and the subject. The curves are presented as r.m.s. acceleration against frequency and refer to exposure times beyond which there is likely to be a decrease in proficiency particularly in carrying out tasks where fatigue is likely to be a significant factor. The limiting accelerations at which humans are likely to experience some discomfort can be obtained by dividing the accelerations from Fig. 9.30 by 3.15 (i.e. they are 10dB lower) and those for maximum safe exposure are double those in Fig. 9.30 (i.e. 6dB higher). The figures apply to each 24 hour period.

The accelerations are related to orthogonal axes with origin at the heart. The 'longitudinal' acceleration is in the line from foot (or buttock) to head. 'Transverse' accelerations are those from the chest to the back and from right to left side. In using the limiting curve, it is recommended that

(a) if vibration occurs simultaneously at more than one discrete frequency or in more than one direction the r.m.s. acceleration of each component shall be evaluated separately with reference to the appropriate limits;
(b) for vibration concentrated in a third-octave band or less the r.m.s. acceleration within the band shall be evaluated with reference to the appropriate limit at the centre frequency of the band;
(c) for broad band distributed vibration occurring in more than one third-octave band, the r.m.s. acceleration in each such band shall be evaluated separately with respect to the appropriate limit;
(d) the effective total daily exposure to an interrupted constant intensity vibration is obtained by summating the individual exposure times, i.e. no allowance is made for human recovery which is likely to occur during pauses;
(e) where intensity varies significantly with time the total exposure is divided into a number of exposures t_i at level A_i. A convenient notional A_1 is chosen in the range of values A_i. If T_i is the permissible time at A_i then the equivalent time for exposure at A_1 is

$$t_i \frac{T^1}{T_i} \text{ where } T^1 = \text{permissible exposure at } A^1$$

The equivalent total exposure time at the notional level A^1 is then given by

$$T^1 \sum \frac{t_i}{T_i}$$

CALCULATIONS

Textbooks discuss design procedures capable of dealing with vibration problems other than those arising from wave excitation. Because of the complicated mathematics, cost and time may limit what can be achieved in practice. In the early design stages of a ship, it is more likely that use will be made of one of the empirical formulae available.

One of the earliest empirical formulae was proposed by *Schlick*, viz.

$$\text{Frequency} = \phi\left(\frac{I}{\Delta L^3}\right)^{\frac{1}{2}} \text{c.p.m.}$$

where Δ = ship displacement in MN, L = length of ship in m, and I = moment of inertia of the midship section including all continuous members in m^4. ϕ is a coefficient which is best calculated from data for a ship similar to that being designed. Typical values of ϕ for the 2-node vertical vibration are given:

Large tankers	282,000
Small trunk deck tankers	217,000
Cargo ships at about 60 per cent load displacement	243,000

For warships, with their finer lines, values of about 347,000 are more appropriate.

For merchant ships the fundamental frequencies of the hull are generally larger than the encounter frequency with the waves. However, increasing size accompanied by increased flexibility results in lowering the hull frequencies. Taken with higher ship speeds resonances become more likely. To illustrate the point the value of $(I/\Delta L^3)^{1/2}$ in Schlick's formula for a 300,000 DWT tanker may be only a third of that of a 100,000 DWT ship.

The Schlick formula is useful for preliminary calculations but it does ignore the effects of entrained water and is therefore only likely to give good results where a similar ship is available. It also involves a knowledge of I which may not be available during the early design stages. This is overcome in the *Todd* formula which can be expressed as:

$$\text{Frequency} = \beta\left(\frac{BH^3}{\Delta L^3}\right)^{\frac{1}{2}} \text{c.p.m.}$$

where Δ is measured in MN and B, H and L in m.

Values of β were found to be

Large tankers	11,000
Small trunk deck tankers	8,150
Cargo ships at about 60 per cent load displacement	9200

If Δ is expressed in tonnef and linear dimensions in metres the constants, within the accuracy of the formula, can be taken as ten times these values.

To account for the entrained water, a virtual weight Δ_1 can be used instead of the displacement Δ, where

$$\Delta_1 = \Delta\left(1 \cdot 2 + \frac{1}{3}\frac{B}{T}\right)$$

Having estimated the frequency of the 2-node vibration, it is necessary to estimate the 3-node period and so on. The theory of a vibrating uniform beam predicts that the 3-node and 4-node vibrational frequencies should be 2.76 and 5.40 times those for the 2-node mode. It is found that in ships the ratios are lower and for very fine ships may be 2, 3, 4, etc.

Some typical natural frequencies of hull for various ship types are given in Table 9.26.

Table 9.26
Natural frequencies of hull for several types of ship

Type of ship	Length (m)	Condition of loading	Frequency of vibration: Vertical 2-node	Vertical 3-node	Vertical 4-node	Vertical 5-node	Horizontal 2-node	Horizontal 3-node	Horizontal 4-node
Cargo ship	85	Light	150	290			230		
		Loaded	135	283			200		
Cargo ship	130	Light	106	210			180	353	
		Loaded	85	168			135	262	
Passenger ship	136		104	177			155	341	
Tanker	227	Light	59	121	188	248	103	198	297
		Loaded	52	108	166	220	83	159	238
Destroyer	160	Average action	85	180	240		120	200	
G.M. Cruiser	220	Average action	70	130	200		100	180	270

NOISE

Noise is of growing importance both for merchant ships and warships. This book can only introduce the subject briefly. The internationally agreed unit for sound intensity is 10^{-16} watts/cm^2 and at 1000 hertz this is close to the threshold of hearing. Due to the large range of intensity to which a human ear is sensitive a logarithmic scale is used to denote the intensity of sound and the usual unit in which noise levels are expressed is the *decibel*. If two noise intensities are w_1 and w_2 then the number of decibels, n, denoting their ratio is

$$n = 10\log_{10}\frac{w_1}{w_2}\,\mathrm{dB}$$

As a reference level, $w_2 = 10^{-16}$ watts/cm^2 is used.
Instruments measuring noise levels in air record sound pressure so that

$$n = 10\log_{10}\frac{w_1}{w_2} = 10\log_{10}\left(\frac{p_1}{p_2}\right)^2 = 20\log_{10}\frac{p_1}{p_2}$$

where p_2 is the pressure in dynes/cm^2 corresponding to the threshold of hearing. This becomes $p_2 = 2 \times 10^{-5}\,\mathrm{N/m^2}$.

Thus a sound pressure level of 0.1 N/m^2 (1 dyne/cm^2) is represented by

$$\text{Level in dB} = 20\log 1/0\cdot 0002 = 74\,\mathrm{dB}$$

In the open, the sound intensity varies inversely as the square of the distance from the source. Hence halving the distance from the noise source increases the level in dB by $10\log 4 = 6\,\mathrm{dB}$ and doubling the distance reduces the level by 6 dB. The addition of two equal sounds results in an increase of 3 dB and each 10 dB increase is roughly equivalent to a doubling of the subjective loudness.

A noise typically contains many components of different frequency and these varying frequencies can cause different reactions in the human being. In order to define fully a given noise it is necessary, therefore, to define the level at all frequencies and this is usually achieved by plotting results as a noise spectrum. The alternative is to express noise levels in dB(A). The A weighted decibel is a measure of the total sound pressure modified by weighting factors, varying with frequency, reflecting a human's subjective response to noise. People are more sensitive to high (1000 Hz+) than low (250 Hz and less) frequencies.

Primary sources of noise are machinery, propulsors, pumps and fans. Secondary sources are fluids in systems, electrical transformers and ship motions. Noise from a source may be transmitted through the air surrounding the source or through the structure to which it is attached. The actions are complex since airborne noise may excite structure on which it impacts and directly excited structure will radiate noise to the air. Taking a propulsor as an example, much of the noise will be transmitted into the water. That represented by pressure fluctuations on the adjacent hull will cause the structure to vibrate transmitting noise both into the ship and back into the water. Other transmissions will be through the shaft and its support system.

A designer will be concerned to limit noise for two reasons:

(a) noise transmitted into the water can betray the presence of the ship, provide a signal on which weapons can home and reduce the effectiveness of the ship's own sensors;
(b) internal noise levels may have effects on crew and passengers.

It is the latter effects with which we are primarily concerned here. Noise may:

(a) annoy;
(b) disturb sleep;
(c) interfere with conversation. Conversational speech at a distance of 1,, represents a noise level of about 60 dB. The greatest interference is created by noises in the frequency range 300-5000 hertz;
(d) damage hearing. Noise levels of 130-140 dB can cause pain in the ear and higher levels can cause physical damage;
(e) upset the normal senses, e.g. it can lead to a temporary disturbance of vision.

Thus noise effects can vary from mere annoyance, itself very important in passenger vessels, to interference with the human being's ability to function efficiently and eventually can cause physical damage to the body.

A designer can apply basic acoustical calculations to ship noise estimation and the design of noise control systems.

Various methods are available for reducing the effect of noise, the particular method depending upon the noise source and why it is undesirable. For instance, the noise level within a communications office with many teleprinters can be reduced by treating the boundary with noise absorption material. A heavy, vibrating, machine transmitting noise through the structure can be mounted on special noise isolating mounts, care being taken to ensure that the mounts are not 'short-circuited' by pipes connected to the machine. Such pipes must include a flexible section. Where a piece of equipment produces a high level of air-borne noise in a compartment adjacent to an accommodation space the common boundary can be treated with noise insulation material. For the best results, the whole boundary of the offending compartment should be treated to reduce the level of structure borne .noise which can arise from the impact of the pressure waves on, say, deck and deckhead.

In recent years active noise cancellation techniques have been developing. The principle used is to generate a noise of equivalent frequency content and volume but in anti-phase to the noise to be cancelled. Thus to cancel the noise of a funnel exhaust a loudspeaker could be placed at the exhaust outlet. For structure borne noise from a machine force generators could be used at the mounting.

ICE

When the temperature falls sufficiently, ice forms on the rigging and upperworks of ships constituting a hazard, as discussed in Chapter 4. Remedial action may also be difficult because the deck machinery itself may have been rendered unusable by ice. As well as reducing stability, ice may hazard the whole ship by structural damage caused by sea ice or by the icebergs which originate in glaciers. While fresh water has its maximum density at 4°C sea water of salinity greater than 24.7 per cent increases in density right down to its freezing point (which is- 1.33°C at 24.7 per cent salinity). Thus freezing does not occur at the surface until the water below is also at the freezing point. Initially individual ice crystals are formed, creating a slush. Movement of the water surface leads to the formation of flat circular discs 30 to 100 cm in diameter with upturned rims, called pancake ice. As the temperature drops further the discs become cemented together to form a continuous sheet which in a typical Arctic winter reaches an average thickness of about 3 metres. In way of pressure ridges the thickness may be much greater.

The majority of icebergs in the North Atlantic originate at the Greenland glaciers. They vary a great deal in size but whatever the size represent a formidable threat to shipping, as witnessed by the *Titanic.* Due to their porosity, their effective density may be appreciably less than that of pure ice and the depth below water can vary between three and eight times their height above water.

Human Factors

The human element in design is covered by what is termed Human Factors (HF), a full study of which involves multi-discipline teams of physiologists, psychologists, engineers and scientists. The HF team can advise the naval architect on:

(i) how to design a system or equipment so that the human can perform most effectively, so giving the greatest overall system efficiency. It is not necessarily, and in general is not, true that the maximum degree of automation is desirable. The blend of human and machine should build upon the strengths of each and minimize the influence of any weaknesses;

(ii) in what way, and to what degree, the system efficiency will be reduced due to degradation in the human's performance due to the environment;

(iii) the levels of environmental parameters (e.g. noise, motions and vibrations) which should not be exceeded if a human's physical state is not to be temporarily or permanently harmed. This has already been discussed in sections above.

A designer has, or should have, always taken account of ergonomic factors. Chairs must be comfortable, writing surfaces at the right height and so on. What is not always appreciated is that the average statistics for the human body are changing with time; for instance people are generally getting taller, so the designer must use the latest available anthropological data for both men and women.

Operators often spend hours in front of displays. HF research can do much to make the operator's task easier. Dials and buttons which are most frequently consulted or activated should be most readily to hand. A digital display helps with taking accurate readings but if the need is to monitor that the value of a variable is below a particular figure, or lies within a given band, an analogue display gives a quicker feel for the situation. Analogue displays also give an appreciation of the value of a variable when it must be read at a distance. Colour can be important. A human being is conditioned to regard red as a danger signal and green as indicating that all is well. Some colours are restful and can convey feelings of warmth or cold. Colour is not always beneficial in differentiating between different levels of a parameter being displayed. It is not immediately obvious whether blue or red represents a higher or lower value. A human will often find a grey scale, or monochrome, presentation easier to interpret for a single variable.

Dials and switches can be grouped into a diagrammatic display of a system, say the fire fighting ring main, the diagram showing the distribution of the main, location of pumps and so on. The state of valves can be readily seen~ represented by a bar-in line with the pipe if open and at right angles to it'Tr closed. The pump symbol can be illuminated when it is running.

In some cases an operator has a single VDU and calls up a variety of displays, at different levels of detail, as required. The system can even decide, in the light of data it is receiving from sensors, which display to present. For instance if the computer senses an alarm from a system not currently being monitored it can draw it to the attention of the appropriate operator. If it does this it is important that the reason it has done so should be immediately apparent to the operator. For instance, if a pump is running hot its symbol can be made to flash and the word 'overheating' appear alongside.

The aim of the HF practitioner is to aid the user or maintainer of a system or equipment. In research, a study is made of the user's actions to deduce the thought processes being followed. Then the displays can be arranged to present the data needed at any instant in a form the user will most readily comprehend. Any system must recognize that the computer is very good at 'number crunching' but poor at pattern recognition. The reverse is true of human beings. Modern technology means that sensors can measure and present more and more data, with greater accuracy. The user may become overwhelmed by the sheer volume of data and may not be able to utilize the level of accuracy. The system should pre-analyse the data, in a predetermined way, to help the user. Presentation of statistical data in a meaningful way can be difficult. Many methods are feasible but the problem lies with the user's ability to interpret it easily. If presented, for instance, with an ellipse on a chart-like display, the ellipse representing the boundary within which a target has a 90% probability of lying, the human may ignore the 10% chance that the target lies outside the ellipse. **HF** research into such matters can point the way to suitable training programmes.

It is not enough simply to study a system with a human 'in the circuit'. This will give an appreciation of overall performance but will not necessarily show whether the human is performing as efficiently as could be the case. A change in what the user is asked to do could lead to a significant increase in overall performance by changing the balance of what is asked of the machine and the human. Such studies must be carried out early enough for the hardware and software to be amended. In modern systems it is often the interaction between the user and the software that is important, rather than the interface with the hardware.

Problems

1. Discuss the process by which sea waves are thought to be formed and the factors on which the wave characteristics depend. What do you understand by the term *swell?*

   Describe the type of sea you would expect to be generated by a wind blowing for some time in one direction. How would you describe, quantitatively, the resulting surface characteristics?
2. For a regular sinusoidal wave deduce expressions for speed of propagation and period.

   Calculate the speed and period of a wave 100m long. If the surface wave height is 3m calculate the depth at which the height of the sub-surface profile is 0.5 m.
3. A wave spectrum is defined by the following table:

w(l/s)	0.20	0.40	0.60	0.80	1.00	1.20	1.40
S«(w)(m² s)	0.00	7.94	11.68	5.56	2.30	0.99	0.45
w(l/s)	1.60	1.80	2.00	2.20			
S«(w)(m² s)	0.23	0.12	0.07	0.00			

   Calculate the significant wave height, average wave height and mean of the tenth highest waves.
4. A large number of wave heights are recorded and on analysis it is found that the numbers falling in various height bands are:

wave height band (m)	$0-\frac{1}{2}$	$\frac{1}{2}-1$	$1-1\frac{1}{2}$	$1\frac{1}{2}-2$	$2-2\frac{1}{2}$	$2\frac{1}{2}-3$	$3-3\frac{1}{2}$	$3\frac{1}{2}$ 4
No. of waves	5	10	20	40	55	40	25	5

   Plot these as a histogram record and deduce the normal and Rayleigh distributions which most closely fit the measured data.
5. A 20 MN ship presents a profile as below

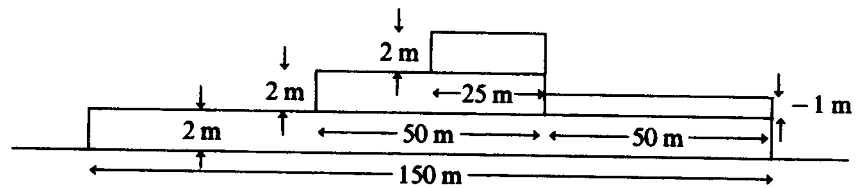


Fig. 9.31

   Assuming that the force acting on area A m² due to a wind of velocity V knots is given by

$$\text{Force} = 0.19\,V^2 A \text{ newtons}$$

   calculate the force due to a nominal wind of 50 knots (i.e. as measured 6 m above the water surface) allowing for the wind gradient as defined in Fig. 9.25. Deduce the effective average wind speed.

   Assuming the ship has a mean draught of 6 m and a metacentric height of 1.5 m calculate the approximate angle of heel due to the wind.
6. In a given wave sample, the number of waves of various heights are:

Wave height (m)	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0
No. of waves	10	39	50	35	16	5	4	1

   Calculate the average and significant wave height and the average of the $\frac{1}{10}$th highest waves.
7. Calculate the wave-lengths in metres and wave velocities in metres per second corresponding to wave periods of 1, 2, 4, 6, 8, 10, 14, 18 and 20 s.
8. The successive crests of the wave profile along the side of a ship at speed in calm water are observed to be about 100 m apart. What is the approximate speed of the ship in knots? If the ship speed is reduced to 10 knots what will the distance apart of the crests become?
9. Check the following statement:

   The orbits and velocities of the particles of water are diminished by approximately one-half for each additional depth below the mid-height of the surface trochoid equal to one-ninth of a wave-length. For example:

Depth in fraction of wave-length	0	$\frac{1}{9}$	$\frac{2}{9}$	$\frac{3}{9}$	$\frac{4}{9}$
Proposed velocities and diameters	1	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$

10. Using empirical formulae, calculate the 2-node frequency of vibration of the hull of a cargo ship to which the following data relates:

Displacement	7925 tonnef
Length	125 m
Beam	15.3 m
Depth	9.1 m
Draught	6.1 m

Allowance should be made for the entrained water.

10 Powering of ships: general principles

The power required to drive a ship through the water depends upon the resistance offered by the water and air, the efficiency of the propulsion device adopted and the interactions among them. Because there is interaction it is vital to consider the design of the hull and the propulsion device as an integrated system. When the water surface is rough, the problem is complicated by increased resistance and by the propulsion device working in less favourable conditions. Powering in waves is considered in Chapter 12. This chapter is devoted to the powering of ships in calm water and concentrates on displacement monohulls. In multi hull displacement forms there will be effects on both viscous and wavemaking resistance due to interference between the separate hulls. In planing, surface effect or hydrofoil craft special considerations apply. These can only be touched upon briefly in this book (Chapter 16).

For the merchant ship, the speed required is dictated by the conditions of service. It may have to work on a fixed schedule, e.g. the cruise ship, or as one of a fleet of ships maintaining a steady supply of material. Therefore a designer must be able to predict accurately the speed a new design will attain. The fuel bill is a significant feature in the operating costs of any ship, so the designer will be anxious to keep the power needed for the operating speed to a minimum. Oil is also a dwindling natural resource.

The speed of a warship is dictated by the operational requirements. An anti-submarine frigate must be sufficiently fast to close with an enemy submarine and destroy it. At the same time, excessive speed and fuel consumption can only be met at the expense of the amount of armament the ship carries.

In all ships, the power needed should be reduced to a minimum consistent with other design requirements to minimize the weight, cost and volume of. machinery and fuel. It follows, that an accurate knowledge of a design's powering characteristics is of considerable importance and that a fair expenditure of effort is justified in achieving it. For predicting full-scale resistance, the designer can use full-scale data from ships built over a considerable period of years, theoretical analysis or models.

Generally speaking, full-scale data is limited in usefulness because of the process of evolution to which ships are subject. To mention two factors, the introduction of welding led to a smoother hull, and ships have tended over the years to become larger. Again, the new ship is often required to go faster so that data from her predecessors cannot be used directly for assessing her maximum power. Clearly, this method is not valid when a new ship form is introduced such as the SWATH (Small Waterplane Twin Hull) ship or the trimaran.

Theory has been used as an aid to more practical methods and continues to develop. Computational fluid dynamics is a very powerful tool which is increasingly used by researchers to study problems of fluid flow, including those involving cavitation but the main contribution of theory is still generally to guide the model experimenter, providing a more rational and scientific background to his work, suggesting profitable lines of investigation and indicating the relative importance of various design parameters.

Where a methodical series of tests has been carried out on a form embracing the new design, the details should be obtained from the literature. Even without a methodical series, systematic plotting of previous data can provide a first estimate of power needs.

The main tool of the designer has been, and remains, the model with theory acting as a guide and full-scale data providing the all-essential check on the model prediction. The model is relatively cheap and results can be obtained fairly rapidly for a variety of changes to enable the designer to achieve an optimum design.

An example of the results obtainable by a judicious blend of theory and model data is provided by what is known as regression analysis. Basically, a mathematical expression is produced for the resistance of the ship, in terms of various ship parameters such as *LIB* ratio, Cp, etc. This expression is then used to deduce the required trend on these parameters to minimize resistance and produces a form superior to those currently in use.

These various considerations are developed more fully later but first it is necessary to consider some of the properties of the fluids in which the ship moves. These are fundamental to the prediction of full-scale performance from the model and for any theoretical investigation.

Fluid dynamics

There are two fluids with which the naval architect is concerned, air and water. Unless stated otherwise water is the fluid considered in the following sections. Air resistance is treated as a separate drag force. Models are used extensively and it is necessary to ensure that the flow around the model is 'similar' to that around the ship in order that results may be scaled correctly. Similarity in this sense requires that the model and ship forms be geometrically similar (at least that portion over which the flow occurs), that the streamlines of the fluid flow be geometrically similar in the two cases and that the fluid velocities at corresponding points around the bodies are in a constant ratio.

Water possesses certain physical properties which are of the same order of magnitude for the water in which a model is tested and for that in which the ship moves. These are:

the density, ρ
the surface tension, (]"
the viscosity, f1
the vapour pressure, P_v

the ambient pressure, p_∞
the velocity of sound in water, a

The quantitative values of some of these properties are discussed in Chapter 9. Other factors involved are:

a typical length, usually taken as the wetted length L for resistance work, and as the propeller diameter D for propeller design;

velocity,	V
propeller revolutions,	n
resistance,	R
thrust,	T
torque,	Q
gravitational acceleration,	g.

Dimensional analysis provides a guide to the form in which the above quantities may be significant. The pi theorem states that the physical relationship between these quantities can be represented as one between a set of non-dimensional products of the quantities concerned. It also asserts that the functionally related quantities are independent and that the number of related quantities will be three less (i.e. the number of fundamental units—mass, length, time) than the number of basic quantities.

Applying non-dimensional analysis to the ship powering problem, it can be shown that:

$$\frac{R}{\rho V^2 L^2} = F\left\{\frac{VL\rho}{\mu}, \frac{V}{\sqrt{(gL)}}, \frac{V}{a}, \frac{\sigma}{g\rho L^2}, \frac{p_\infty - p_v}{\rho V^2}\right\}$$

$$\frac{T}{\rho n^2 D^4} \quad \text{and} \quad \frac{Q}{\rho n^2 D^5} = F\left\{\frac{V}{nD}, \frac{VD\rho}{\mu}, \frac{V^2}{gD}, \frac{\sigma}{\rho g L^2}, \frac{p_\infty - p_v}{\rho V^2}\right\}$$

Expressed in another way, it is physically reasonable to suggest that if data can be expressed in terms of parameters that are independent of scale, i.e. non-dimensional parameters, the same values of these data will probably be obtained from experiments at different scales if the parameters are constant. Where the governing parameters cannot be kept constant, data will change in going from the model to full scale. The above are not the only non-dimensional parameters that can be formed but they are those in general use. Each has been given a name as follows:

$\dfrac{R}{\rho V^2 L^2}$ is termed the resistance coefficient

$\dfrac{VD\rho}{\mu}$ or $\dfrac{VL\rho}{\mu}$ is termed the Reynolds' number (the ratio μ/ρ is called the kinematic viscosity and is represented by ν)

$\dfrac{V}{\sqrt{(gD)}}$ or $\dfrac{V}{\sqrt{(gL)}}$ is termed the Froude number

$$\left.\begin{array}{ll} \dfrac{V}{a} & \text{is termed the Mach number} \\[2ex] \dfrac{\sigma}{g\rho L^2} & \text{is termed the Weber number} \end{array}\right\}$$ These two quantities are not significant in the context of the present book and are not considered further

$\dfrac{p_\infty - p_v}{\rho V^2}$ is termed the cavitation number

$$\frac{T}{\rho n^2 D^4} = K_T = \text{thrust coefficient}$$

$$\frac{Q}{\rho n^2 D^5} = K_Q = \text{torque coefficient}$$

$$\frac{V}{nD} = J = \text{advance coefficient}$$

Unfortunately, it is not possible to set up model scale experiments in which all the above parameters have the same values as in the full-scale. This is readily seen by considering the Reynolds' and Froude numbers. Since ρ, ν, and g are substantially the same for model and ship, it would be necessary for both VL and $V/\sqrt{L}$ to be kept constant. This is physically impossible. By using special liquids instead of water in which to test models, two parameters could be satisfied but not all of them.

Fortunately, certain valid results can be obtained by keeping one parameter constant in the model tests and limiting those tests to certain measurements. For example, model resistance tests are conducted at corresponding Froude number and model propeller cavitation tests at corresponding cavitation number. This means that resistance forces which depend on Reynolds' number will have to be modified in going from model to ship. It will be shown that, had this difficulty of achieving physical similarity not been present, the early experimenters would not have experienced so much difficulty in predicting the resistance of the full-scale ship.

Components of resistance and propulsion

It is necessary to provide a propulsive device to drive the ship through the water. It has been explained, that since the propulsion device interacts on the resistance of the ship the two cannot be treated in isolation. However, as a matter of convenience, the overall problem is considered as the amalgamation of a number of smaller problems. The actual divisions are largely arbitrary but are well established. In the following, it is assumed that the propulsive device is a propeller.

If the naked hull of the ship could be driven through the water by some device which in no way interacted with the hull or water, it would experience a total resistance R_T which would be the summation of several types of resistance as is explained later. The differentiation between types of resistance is necessary because they scale differently in going from model to full-scale. The product of R_T and the ship's speed V defines a horsepower which is known as the *effective power* (P_E). This e.h.p. can be regarded as the useful work done in propelling the ship.

The power actually delivered to the shafts for propelling a ship is the *shaft power* (P_S). The ratio between the shaft and effective powers is a measure of the overall propulsive efficiency achieved and is termed the *propulsive coefficient* (PC). It should be noted that some authorities take the PC as the ratio of P_{EA} to P_S, P_{EA} defined as below. The propulsive coefficient arises partly from the efficiency of the propeller, and partly from the interaction of propeller and hull. In addition, it has to be modified to make model and full-scale data compatible.

Following on from the above, four basic components of the powering problem suggest themselves:

(*a*) P_E or the hull resistance
(*b*) the propeller
(*c*) hull/propeller interaction
(*d*) ship/model correlation.

EFFECTIVE POWER

One watt is the rate of performing 1 joule of work per second. As far as propelling the ship through the water is concerned, the 'useful' or 'effective' work is that done in overcoming the resistance of the ship by its speed of advance. The resistance concerned is conventionally taken to be that of the 'naked' hull, i.e. without any appendages. This leads to the following definition:

The *effective power* of a ship is the product of the resistance of the naked hull and the speed of the hull. Therefore,

$$P_E = R_T \times V$$

A corresponding definition can be evolved using the resistance of the hull including that of appendages and this is conventionally denoted by P_{EA}.

The ratio of P_{EA} to P_E is known as the *Appendage coefficient*, i.e.

$$\text{Appendage coefficient} = P_{EA}/P_E$$

EXAMPLE 1. At 251 m/min the tow rope pull of a naked hull is 35.6 kN. Find the effective power of the hull at this speed.

Solution:

$$P_E = 251\,\text{m/min} \times 35.6\,\text{kN} \times \frac{1\,\text{min}}{60\,\text{s}} = 149\,\text{kW}$$

EXAMPLE 2. A 6000 tonnef destroyer develops a total power of 44.74 MW at 30 knots. Assuming that the effective power is 50 per cent of this total power, calculate the resistance of its naked hull.

Solution:

$$P_E = \frac{1}{2} \times 44.74 = 22.37\,\text{MW}$$

Therefore $22.37 \times 10^6\,\text{W} = (\text{Resistance})\ \text{newtons} \times \left(\frac{30 \times 1852}{3600}\right)\ \text{m/s}$

i.e.

$$\text{Resistance} = 1.449 \times 10^6\ \text{newtons}$$

TYPES OF RESISTANCE

The classical theory of hydrodynamics has shown that a body deeply immersed in fluid of zero viscosity experiences no resistance. No matter how the streamlines may be deflected as they pass the body, they return to their undisturbed state a long way downstream of the body (see Fig. 10.1) and the resultant force on the body is zero. There are pressure variations in the fluid as the streamlines are deflected and particle velocities change. In this respect, Bernoulli's theorem is obeyed, i.e. increased velocities are associated with pressure reductions. Thus, the body can be acted upon by forces of considerable magnitude but they all act so as to cancel each other out.

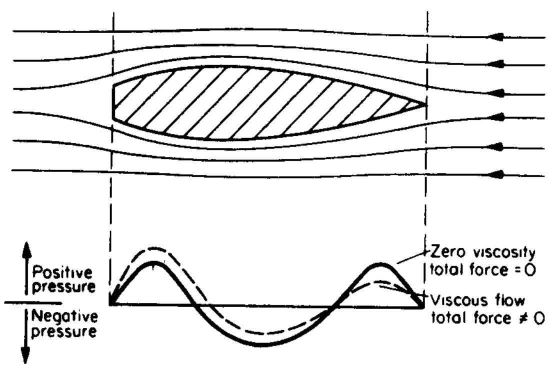


Fig. 10.1

In a practical case the fluid is viscous and a deeply immersed body would suffer a frictional drag. In addition, when the body approaches a free surface, the pressure variations around the body can manifest themselves as elevations or depressions of the water surface. That is to say, waves are formed on the surface. This process upsets the balance of pressures acting on the body which results in a drag force. The magnitude of the drag force is related to the energy of the wave system created.

The total resistance of a ship moving on a calm water surface has several components. They are: wave-making resistance; skin frictional resistance; viscous pressure resistance; air resistance; appendage resistance.

Each component can now be studied separately provided it is remembered that each will have some interaction with the others.

WAVE-MAKING RESISTANCE

It is common experience, that a body moving across an otherwise undisturbed water surface produces a wave system. This system arises from the pressure field around the body and the energy possessed by it must be derived from the body. As far as the body is concerned the transfer of energy will manifest itself as a force opposing the forward motion. This force is termed the *wave-making resistance*.

A submerged body also experiences a drag due to the formation of waves on the free surface, the magnitude of this drag reducing with increasing depth of submergence until it becomes negligible at deep submergence. This typically occurs at depths equal to approximately half the length of the body. An exception to this general rule can occur with submarines at sea if they are moving close to the interface between two layers of water of different density. In this case, a wave system is produced at the interface resulting in a drag on the submarine.

A gravity wave, length λ, in deep water moves with a velocity C defined by

$$C^2 = \frac{g\lambda}{2\pi}$$

Because the wave pattern moves with the ship, C must be equal to the ship velocity V and λ being a length measurement can, for dimensional analysis, be represented as proportional to the ship length L for a given speed.

Thus it is seen that of the non-dimensional parameters deduced earlier it is V^2/gL or $V/\sqrt{(gL)}$ which is significant in the study of wave-making resistance. As stated in the section on fluid dynamics, the quantity $V/\sqrt{(gL)}$ is usually designated the *Froude number*. In many cases, the simpler parameter $V/\sqrt{L}$ is used for plotting results but the plot is no longer non-dimensional.

Hydrodynamically, the ship can be regarded as a moving pressure field. Kelvin considered mathematically the simplified case of a moving pressure point and showed that the resulting wave pattern is built up of two systems. One system is a divergent wave system and the other a system of waves with crests more or less normal to the path of the pressure point. Both systems travel forward with the speed of the pressure point (Fig. 10.2).

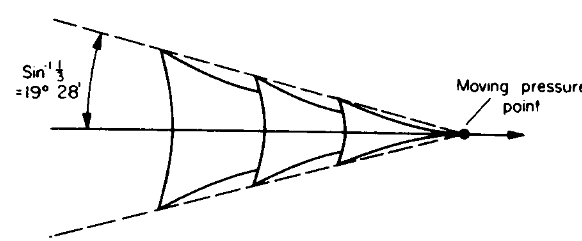


Fig. 10.2 Wave system associated with moving pressure point

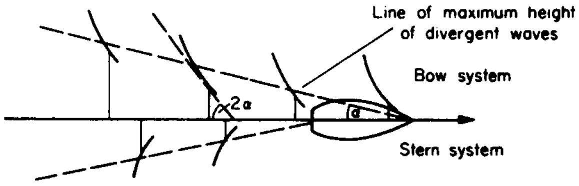


Fig. 10.3 Ship wave pattern

The wave system associated with a ship is more complicated. To a first approximation, however, the ship can be considered as composed of a moving pressure field sited near the bow and a moving suction field near the stern. The bow produces a wave pattern similar to that produced by Kelvin's pressure point with a crest at the bow. The stern on the other hand produces a wave system with a trough at the stern.

If the line of maximum height of crests of the divergent system is at *a,* then the wave crests at these positions subtend an angle of approximately *2a* to the ship middle line as in Fig. 10.3.

The two transverse wave systems, i.e. at bow and stern, have a wave-length of $27rV^2/g$. The transverse waves increase in width as the divergent waves spread out. The total energy content per wave is constant, so that their height falls progressively with increasing distance from the ship.

In general, both divergent systems will be detectable although the stern system is usually much weaker than that from the bow. Normally, the stern transverse system cannot be detected as only the resultant of the two systems is visible astern of the ship.

In some ships, the wave pattern may be made even more complex by the generation of other wave systems by local discontinuities in the ship's form.

Since at most speeds both the bow and stern systems are present aft of the ship, there is an interaction between the two transverse wave systems. If the systems are so phased that the crests are coincident, the resulting system will have increased wave height, and consequently greater energy content. If the crest of one system coincides with the trough of the other the resulting wave height and energy content will be less. The wave-making resistance, depending as it does on the energy content of the overall wave system, varies therefore with speed and also effective length between the bow and stern pressure systems. Again, the parameters V and L are important.

Froude studied the effect on resistance of the length of the ship by towing models with the same endings but with varying lengths of parallel middle body. The results are in line with what could be expected from the above general reasomng.

The distance between bow and stern pressure systems is typically *O.9L.* The condition that crests or troughs of the bow system should coincide with the first trough of the stern system is therefore

$$\frac{V^2}{0.9L} = \frac{g}{N\pi}$$

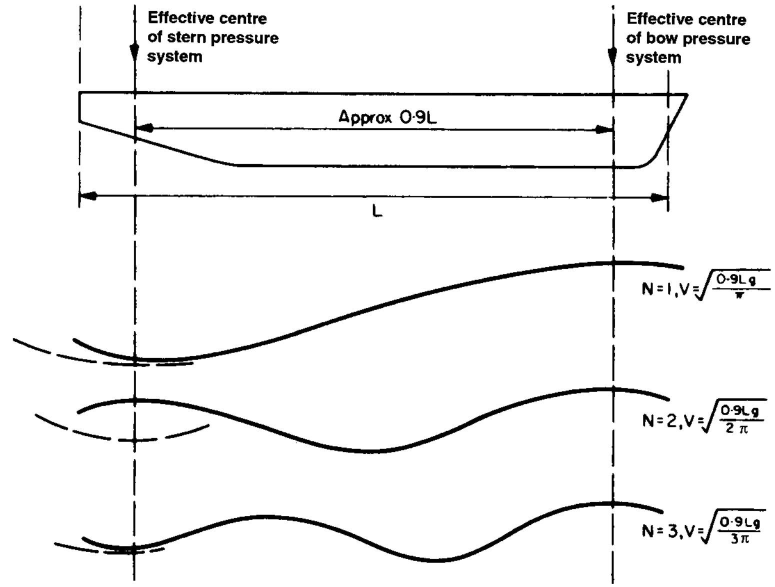


Fig. 10.4 Interaction of bow and stern wave systems

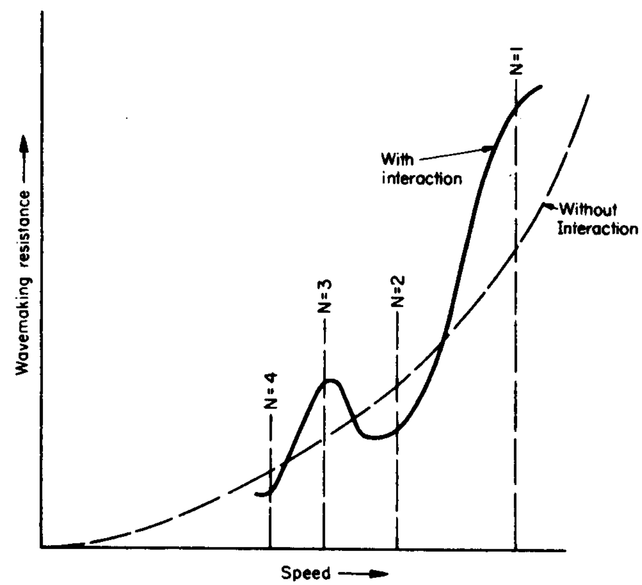


Fig. 10.5 'Humps' and 'hollows' in wave-making resistance curves

For $N = 1, 3, 5, 7$, etc., the troughs will coincide and for $N = 2, 4, 6$, etc., the crests from the bow system coincides with the trough from the after system as in Fig. 10.4.

If there were no interaction between the bow and stern wave systems, the resistance would increase steadily with speed as shown in Fig. 10.5 ('Without interaction' curve). Because interaction occurs at speeds discussed above, the actual resistance curve will oscillate about the curve as indicated.

A 'hump' occurs when N is an odd integer and a 'hollow' when N is an even integer. It is to be expected that the most pronounced hump will be at $N = 1$, because the speed is highest for this condition and this hump is usually referred to as the *main hump.* The hump associated with $N = 3$ is often called the *prismatic hump* as its influence is greatly affected by the prismatic coefficient of the form considered.

Since the Froude number $Fn = V/\sqrt{(gL)}$, the values of Fn corresponding to the humps and hollows are shown in Table 10.1.

Table 10.1

N	F_n
1	$\sqrt{\left(\frac{0.9}{\pi}\right)} = 0.54$
2	$\sqrt{\left(\frac{0.9}{2\pi}\right)} = 0.38$
3	$\sqrt{\left(\frac{0.9}{3\pi}\right)} = 0.31$
4	$\sqrt{\left(\frac{0.9}{4\pi}\right)} = 0.27$

Clearly, a designer would not deliberately produce a ship whose normal service speed was at a 'hump' position. Rather, the aim would be to operate in a 'hollow', although other considerations may be overriding in deciding on the length of the ship.

FRICTIONAL RESISTANCE

The water through which a ship moves has viscosity which is a property of all practical fluids. It was shown earlier, that when viscosity is involved the conditions for dynamic similarity are geometrically similar boundaries and constancy of Reynolds' number.

When a body moves through a fluid which is otherwise at rest, a thin layer of fluid adheres to the surface of the body and has no velocity relative to the body. At some distance from the body the fluid remains at rest.

The variation of velocity of the fluid is rapid close to the body (Fig. 10.6) but reduces with increasing distance from the body. The region in which there is a rapid change in velocity is termed the *boundary layer.*

The definition of boundary layer thickness is to some extent arbitrary since in theory it extends to infinity. It is common practice to define the thickness as the distance from the surface of the body at which the velocity of the fluid is 1 per cent of the body velocity.

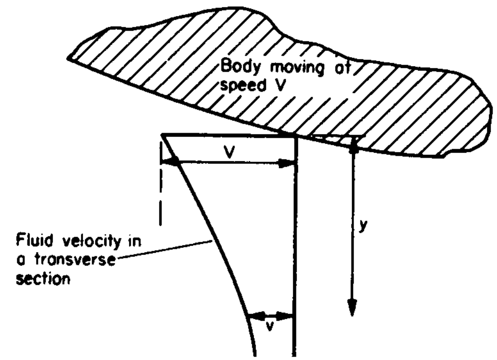


Fig. 10.6 The boundary layer

Due to the velocity gradient across the boundary layer, the fluid is in shear and the body experiences a resistance which is termed *the frictional resistance.* If the fluid velocity is v at distance y from the body the shear stress in the fluid is given by

$$\tau = \mu \frac{dv}{dy}$$

This applies to the case of *laminar flow* in which each fluid particle follows its own streamline path with no mass transfer between adjacent fluid layers. The shear in this case is due solely to molecular action. Laminar flow conditions are only likely to apply at relatively low Reynolds' numbers. At higher Reynolds' numbers the steady flow pattern breaks down and is replaced by a more confused pattern which is termed *turbulent flow.* The value of Rn at which this breakdown in flow occurs is termed the critical Reynolds' number, and its actual value depends upon the smoothness of the surface and the initial turbulence present in the fluid. For a smooth flat plate, breakdown occurs at a Reynolds' number between 3×10^5 and 10^6. In turbulent flow, the concept of a boundary layer still applies but in this case, besides the molecular friction force, there is an interaction due to the momentum transfer of fluid masp between adjacent layers. The exact mechanism of the turbulent boundary layer is incompletely understood, but it follows that the velocity distribution curve at Fig. 10.6 can represent only a mean velocity curve.

The *transition* from laminar to turbulent flow is essentially one of stability. At low Reynolds' numbers, disturbances die out and the flow is stable. At a certain *critical* value of Reynolds' number, the laminar flow becomes unstable and the slightest disturbance will cause turbulence. The critical Rn for a flat surface is a function of l the distance from the leading edge. Ahead of a point defined by l as follows:

$$(R_n)\ \text{critical} = \frac{Vl}{\nu}$$

the flow is laminar. At distance *l* transition begins and after a certain *transition region,* turbulence is fully established.

For a flat surface, the critical Reynolds' number is approximately 106. For a curved surface, the pressure gradient along the surface has a marked influence on transition. Transition is delayed in regions of decreasing pressure, i.e. regions of increasing velocity. Use is made of this fact in certain aerodynamic low drag forms such as the 'laminar flow' wing. The gain arising from retaining laminar flow is shown by the fact that a flat plate suffers seven times the resistance in all turbulent as opposed to all laminar flow.

The thickness of the turbulent boundary layer is given approximately by

$$\frac{\delta x}{L} = 0.37(R_L)^{-\frac{1}{5}}$$

where L is the distance from the leading edge and R_L is the corresponding Reynolds' number. For example, at 15 m/s, with $L = 150$ m, $\delta\tau$ is about 0.75 m.

Even in turbulent flow, the fluid particles adjacent to the body's surface are at rest relative to the body. It follows that there exists a *laminar sub-layer* although in practice this is extremely thin. It is nevertheless of importance in that a body appears smooth if surface roughness does not protrude through this sub-layer. Such a body is said to be *hydraulically smooth* and a plot of drag against Reynolds' number would be as shown by the basic curve in Fig. 10.7.

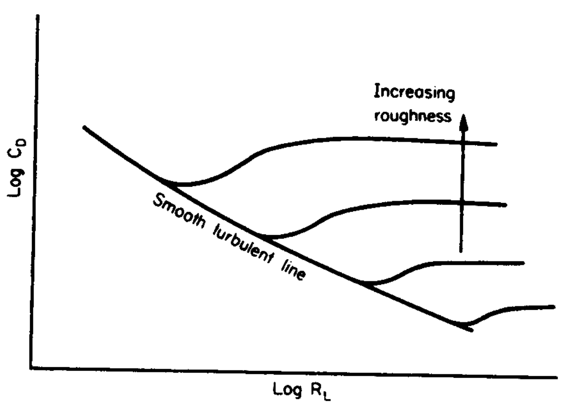


Fig. 10.7 Effect of roughness

For a rough surface, resistance follows the smooth curve as Reynolds' number is increased until a certain value and it then breaks away and eventually becomes horizontal, i.e. the drag coefficient becomes independent of *Rn* and drag varies as the square of the velocity. The rougher the surface the smaller the value of *Rn* at which the breakaway occurs.

Owing to the increase in boundary layer thickness, the ratio of roughness (i.e. effective granularity of surface) to the boundary layer thickness decreases along the length of a surface. For this reason, protrusions from a hull of a given size have less effect on resistance at the after end of a ship than they do forward.

For all practical purposes, the complete boundary layer of a ship at sea cur be regarded as turbulent. In a model in a towing tank, a portion may be laminal but the extent of this is sensitive to external conditions and it can vary con. siderably in a given model. Because of the difference in resistance associated with the two types of boundary layer, this phenomenon has led to inconsistent model test data in the past and this has caused most ship tanks to artificially stimulate turbulent flow conditions to ensure reproducible conditions. A number of devices are used to stimulate turbulence but that now most commonly used is a row of studs a short distance from the bow of the model.

For convenience, the frictional resistance of a ship is usually divided into two components. The first component is that resistance which would be experienced by a 'flat plate' of equivalent surface area. The second component is the increased frictional resistance occasioned by the actual form of the ship and this component is known as the *frictionalform resistance.*

Hull roughness is a complex subject. It depends on a wide range of features, each varying with time in a different way. The plating, as built, will have an inherent roughness, welding deposits and distortion due to fabrication. Distortions will increase in service due to water loading and damage. The paint films will gradually break down and corrosion will occur. Marine fouling can occur quite rapidly in some areas of the world. There is also the difficulty of defining the rough surface. Amplitude variations can be measured using a hull roughness gauge (that developed by the British Ship Research Association is commonly used) to give the mean apparent amplitude. But what may be termed the general 'texture' of the surface is also important.

VISCOUS PRESSURE RESISTANCE

Total ship resistance comprises the fore-and-aft component of all pressures normal to the hull. That part of the pressure resistance which manifests itself as waves has already been discussed; the remainder of the pressure resistance is due to viscous effects which inhibit that build-up of pressure around the after end of the ship predicted for a perfect fluid. Part of this resistance will be due to the generation of vortices from form discontinuities such as the turn of the bilge. Another part arises from the thickening of the boundary layer and ~ may be increased by flow separation. Because these last two elements are affected by the form of the ship they are together known as form drag or form resistance. Form drag is likely to be most significant in full bodied ships. Pressure energy lost to the sea is thus seen as waves and as eddies or vortices. Examination of the energy dissipated in the wake and in the waves may enable some of the resistance due to form to be calculated. That due to the transfer of energy between wave and wake is sometimes isolated for examination and is called wave breaking resistance.

AIR RESISTANCE

Air is a fluid, as is water, and as such will resist the passage of the upper portions of the ship through it. This resistance will comprise both frictional and eddy-making components.

In an artist's impression of a ship it is possible to depict a very smooth streamlined above water form. In practice, the weight penalty associated with such fairing and the difficulties of fabrication are not justified by the reduction in air resistance or by the relatively small gain in usable internal volume. In practice, therefore, air flowing over the superstructure meets a series of discontinuities which cause separation, i.e. streamlines break down and eddies are formed. As expected, air resistance like water eddy resistance will vary as V^2.

At full speed in conditions of no wind, it is probable that the air resistance will be some 2-4 per cent of the total water resistance. Should the ship be moving into a head wind of the same speed as the ship, the relative wind speed will be doubled and the air resistance quadrupled. Thus, clearly, in severe weather conditions such as in a full gale the air resistance can contribute materially to slowing down the ship.

APPENDAGE RESISTANCE

The discussion up to this point has been concerned mainly with the resistance of the naked hull, i.e. without appendages. Typical appendages are rudders, shaft brackets or bossings, stabilizers, bilge keels, docking keels. Each appendage has its own typical length, which is much smaller than the ship length, and accordingly is running at its own Reynolds' number. Each appendage, therefore, has a resistance which would scale differently to full-size if run at model size, although obeying the same scaling laws.

To include appendages in a normal resistance model would, therefore, upset the scaling of the hull resistance. It is for this reason that models are run naked, and the resulting total ship resistance must be modified by adding in estimates of the resistance due to each full-scale appendage.

The resistance of the appendages may be estimated from formulae based on previous experience or by running models both with and without appendages and scaling the difference to full-scale using different scaling laws from those used for the hull proper. Fortunately, appendage resistances are usually small (of the order of 10 per cent of that of the hull) so that errors in their assessment are not likely to be critical. It is usual to assume that the appendage resistance varies as V^2, so that the contribution to the non-dimensional resistance coefficient is constant.

In addition to the above resistances, the ship in service generally has her resistance to ahead motion increased by the presence of waves and spray generated by the wind. In rough weather, this effect can be of considerable magnitude and often causes a significant fall off in speed. This is discussed in Chapter 12.

RESIDUARY RESISTANCE

For the practical evaluation of ship resistance for normal ship forms, it is usual to group wave-making resistance, form resistance, eddy resistance and frictional form resistance into one force termed *residuary resistance.* This concept is not theoretically correct, but, in practice, provides a sufficiently accurate answer.

Thus the total resistance is given by

$$R_T = R_R + R_F$$

where R_R = residuary resistance, and R_F = frictional resistance of an equivalent flat plate.

Having examined how the resistance of a ship arises, it is necessary to examine the effects of the propulsion device and how consideration of the two cannot be separated. In returning to the evaluation of ship resistance in the next chapter, the resistance will then be considered as the summation of the frictional and residuary resistances.

THE PROPULSION DEVICE

The force needed to propel the ship must be obtained from a reaction against the air, water or land, e.g. by causing a stream of air or water to move in the opposite direction. The sailing ship uses air reaction. Devices acting on water are the paddle wheel, oar and screw propeller. Reaction on land is used by the punt pole or the horse towing a barge.

For general applications, the land reaction is not available and the naval architect must make use of water or air. The force acting on the ship arises from the rate of change of momentum induced in the fluid.

Consider a stream of fluid, density ρ, caused to move with velocity V in a 'tube' of cross-sectional area A. Then the mass of fluid passing any section per second = $\rho A V$ and the momentum of this fluid = $mV = \rho A V^2$. Since fluid is initially at rest the rate of change of momentum = $\rho A V^2$.

In a specific application, the force required is governed by the speed desired and the resistance of the ship. Since the force produced is directly proportional to the mass density of the fluid used, it is reasonable to use the more massive of the two fluids available, i.e. water. If air were used, then either the cross-sectional area of the jet must be large or the velocity must be high.

This explains why most ships employ a system by which water is caused to move aft relative to the ship. A variety of means is available for producing this stream of water aft, but by far the most commonly used is the screw propeller and this is dealt with first.

THE SCREW PROPELLER

Basically, the screw propeller may be regarded as part of a helicoidal surface which, on being rotated, 'screws' its way through the water driving water aft and the ship forward. Some propellers have adjustable blades-they are called *controllable pitch propellers*-but by far the greater majority of propellers have fixed blades. The ones we are concerned with first are *fixed pitch propellers.*

Propellers can be designed to turn in either direction in producing an ahead thrust. If they turn clockwise when viewed from aft, they are said to be *right-handed;* if anti-clockwise, they are said to be *left-handed.* In a twin screw ship, the starboard propeller is normally right-handed and the port propeller left-handed, i.e. they turn as in Fig. 10.8. They are said to be outward turning and this reduces cavitation which is discussed later.

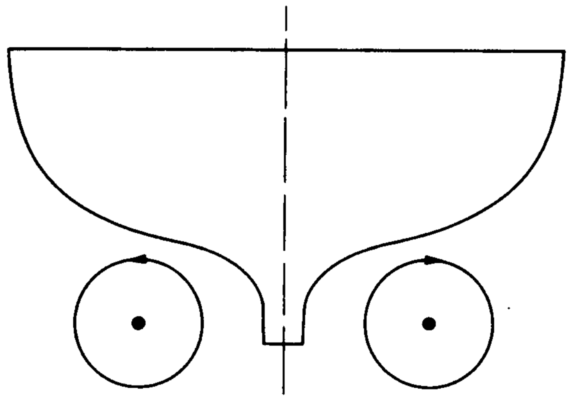


Fig. 10.8 Usual handing of propellers in a twin screw ship. Ship view from aft

Considering each blade of the propeller, the *face* is the surface seen when viewed from aft, i.e. it is the driving surface when producing an ahead thrust. The other surface of the blade is called the *back*. The *leading edge* of the blade is that edge which thrusts through the water when producing ahead thrust and the other edge is termed the *trailing edge*.

Other things being equal, the thrust developed by a propeller varies directly with the surface area, ignoring the boss itself. This area can be described in a number of ways. The *developed blade area* of the propeller is the sum of the face area of all the blades. The *projected area* is the projection of the blades on to a plane normal to the propeller axis, i.e. the shaft axis. The *disc area* is the area of a circle passing through the tips of the blades and normal to the propeller axis.

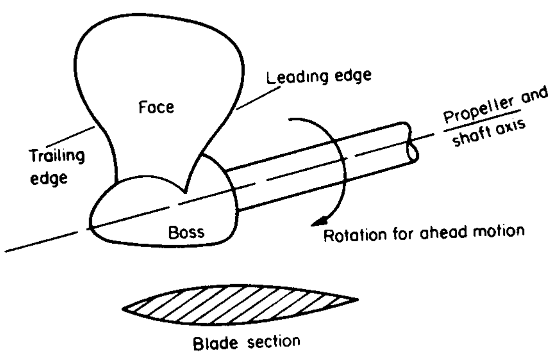


Fig. 10.9 The propeller blade

In non-dimensional work, the *blade area ratio* (BAR) is now generally used. This is the ratio of the developed blade area to the disc area, i.e.

$$\mathrm{BAR} = \frac{A_\mathrm{D}}{A_0} = \frac{4A_\mathrm{D}}{\pi D^2},\ A_\mathrm{D} \text{ obtained by drawing}$$

If the variation of helical chord length with radius is known, then the true blade area can be obtained analytically by integration. This is known as the *expanded area* and the *expanded area ratio* (EAR) is defined by

$$\mathrm{EAR} = \frac{4A_\mathrm{E}}{\pi D^2}$$

In some earlier work, the concept of a *disc area ratio* (DAR) was employed in which the developed area was increased to allow for the boss. Froude proposed a boss allowance of 25 per cent of the developed area but Gawn used 12.5 per cent.

A true helicoidal surface is generated by a line rotated about an axis normal to itself and advancing in the direction of this axis at constant speed. The distance the line advances in making one complete revolution is termed the *pitch*. For simple propellers, the pitch is the same at all points on the face of the blade. This is the *face pitch* of the propeller and the ratio of this to the propeller diameter is the *face pitch ratio*

$$\text{i.e. face pitch ratio} = \frac{P}{D}$$

The distance advanced by a propeller during one revolution when delivering no thrust is termed the *analysis pitch*. In practice, this is rather greater than the geometrical pitch of the propeller. When developing thrust, the propeller advance per revolution is less than the analysis pitch. The difference is termed the *slip*. That is,

$$\text{slip} = \text{analysis pitch} - \text{advance per revolution}$$

The ratio of the slip to the analysis pitch is correctly called the *slip ratio s*, but by common usage is often referred to simply as slip.

Most modern propellers have pitch varying with radius and to define the geometry of the propeller the variation must be specified. For convenience, a nominal pitch is often quoted which is the pitch at a radius of 0.7 times maximum radius.

The projected shape of a propeller blade is generally symmetrical about a radial line called the median. Some propellers have what is known as *skew back* and this is when the median is curved back, relative to the direction of rotation

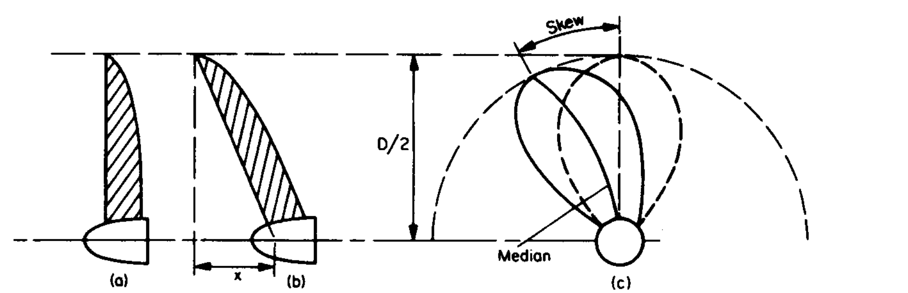


Fig. 10.10

of the propeller, as shown in Fig. 10.10(c). Skew back is defined by the circumferential displacement of the tip of the blade. It is of some advantage when the propeller is working in a flow with a pronounced circumferential variation, as not all the blade is affected at the same time and variations in thrust and torque are smoothed out.

For some applications the blade face is not normal to the propeller axis. In such a case, e.g. Fig. *10.10(b),* the blade is said to be *raked.* It may be raked either forward or aft, but generally the latter to increase the clearance between the blade tip and the hull. Referring to the figure

$$\text{Rake ratio} = X/D$$

The blade section shape at any radius is the shape of the intersection between the blade and a co-axial cylinder when the cylindrical surface has been rolled out flat. The *median* or *camber line* is the line through the mid-thickness of the blade. The *camber* is the maximum distance separating the median line and the straight line, the chord c, joining the leading and trailing edges.

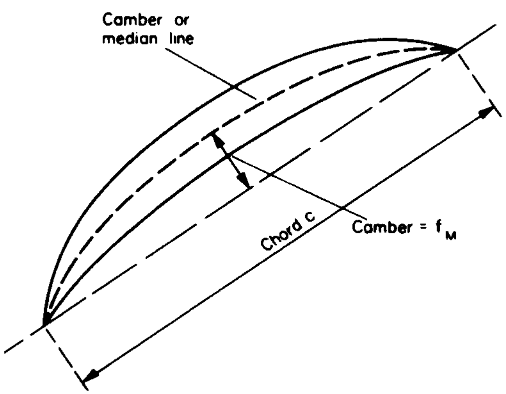


Fig. 10.11

Camber is normally expressed as the *camber ratio,* where

$$\text{camber ratio} = f_M/c$$

Similarly the *thickness ratio* of the section is t/c where t is the maximum thickness of the section. In most modern propellers, the thickness varies non-linearly with radius. The *thickness distribution* must be specified for complete definition of the propeller geometry.

SPECIAL TYPES OF PROPELLER

Most of this section deals with the fixed-pitch propeller which is the most common propulsor. Other types are:

(a) Controllable pitch propeller

In such propellers the blades can be rotated about axes normal to the driving shaft so that thrust and torque can be varied at constant shaft revolutions. This

can help match the propeller and machinery characteristics. If the blade rotation is great enough the propeller can produce astern thrust removing the need for a reversing gearbox. Manoeuvring can be faster as the blade angle can he varied more rapidly than can the shaft revolutions but there will be an optimum rate of change to produce maximum acceleration or deceleration.

Another suitable application of the CP propeller is to ships which must operate efficiently at two quite different loading conditions, e.g. the tug when towing or running free, the trawler when trawling or when on passage to or from the fishing grounds.

Limitations of the CP propeller include the power that can be satisfactorily transmitted (installations for more than 20 MW are uncommon), the complication of the mechanisms controlling the blade angle and the limitation of BAR to about 0.8 which affects the cavitation performance. The control mechanism must pass down the shaft and into the boss. The boss is enlarged to take this gear and to house the bearings for the blades. This increased boss size slightly reduces the maximum efficiency obtainable. The blade sections at the root are governed by the rotation on the boss and are poor for cavitation.

(b) Contra-rotating propeller (CRP)

In this case there are two propellers in line on a double shaft rotating in opposite directions, the forward screw on the outer shaft and the after screw on the inner shaft. Generally the two propellers will be of different diameters and rpm. The design of CRPs is more complicated because of the interactions between the screws and the need for the contra-rotating gear system, but they have higher efficiencies than conventional propellers. This is because the after screw recovers the rotational energy imparted to the wake by the forward screw. Compared with conventional propellers the CRP achieves its optimum performance at a smaller diameter for a given rpm, or at lower rpm for the same diameter. The CRP also generally has a superior cavitation performance and reduced noise emission due to the lighter blade loading.

(c) The self pitching propeller

In this type of propeller the blades are free to rotate through 360° about~ bearing axis substantially normal to the shaft axis. The blade pitch is determined solely by the action of hydrodynamic and centrifugal forces. Particular applications are for auxiliary yachts and motorsailers.

(d) The vertical axis propeller

This propeller consists basically of a horizontal disc rotating about a vertical axis. Projecting vertically down from this disc are a number of spade-like blades and these feather as they and the disc rotate. By varying the sequence in which the blades feather a thrust can be produced in any desired direction.

An obvious advantage of such a propeller is that it confers good manoeuvrability on any ship so fitted. This is touched upon in Chapter 13. With most conventional machinery units, the drive shaft is horizontal and to drive the

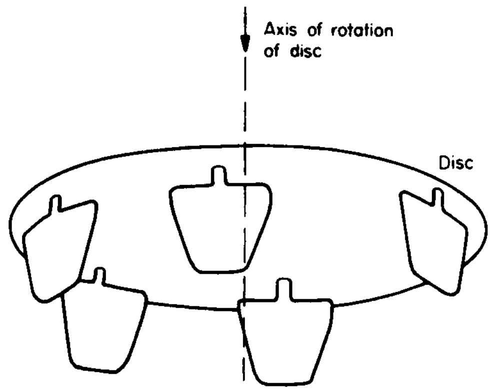


Fig. 10.12 Vertical axis type propeller

horizontal disc it is necessary to introduce a bevel gear with consequential limitations on the maximum power that can be transmitted.

(e) Ducted propellers

A typical arrangement is sketched in Fig. 10.13. Improvements over the conventional propeller performance arise from the enlargement of the tail race and the thrust that can be produced by suitable shaping of the duct to offset the drag of the shroud and its supports. Most applications have been made in ships with heavily loaded propellers, e.g. tugs, but the range of use is increasing.

Other advantages of the shroud are that it protects the propeller from physical damage and acts as a cloak masking the propeller noise.

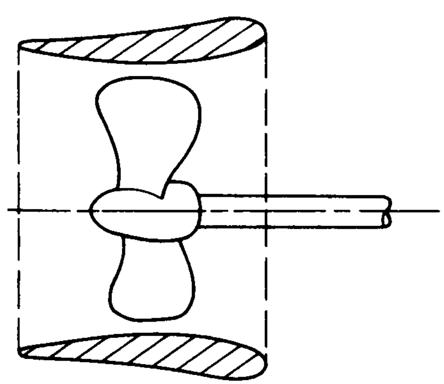


Fig. 10.13 Shrouded propeller

Van Manen gave data for various types of propeller based on open water tests as shown in Fig. 10.14. It indicates the type of propeller which will give the best efficiency for a given type of ship. Efficiency is not always the only factor to be considered, of course, in choosing the propulsion device.

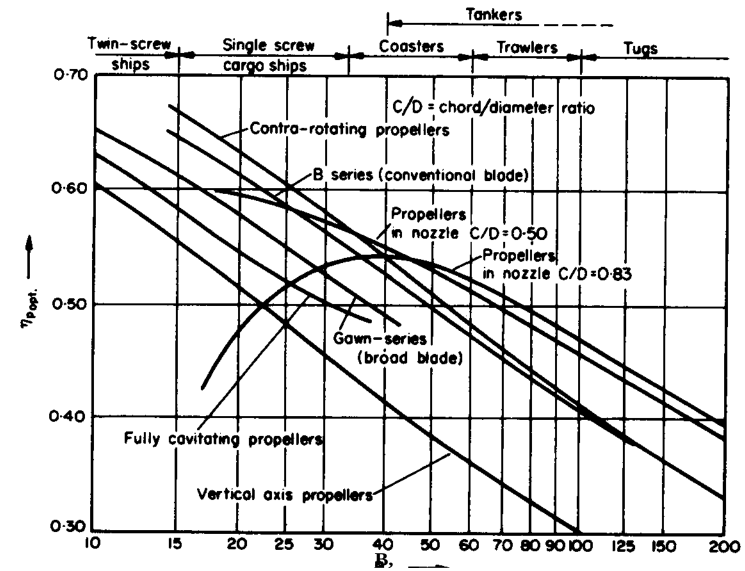


Fig. 10.14 Comparison of optimum efficiency values for different types of propulsion

(f) Pump jets

In this propulsor, a rotor with a relatively large number of blades operates between sets of stator blades, the whole being surrounded by a suitably shaped duct. When correctly designed there is no resultant heeling moment on the body being propelled, no rotational losses in the wake and no cavitation. For these reasons pump jets have been used to propel underwater vehicles.

(g) Propulsion pods

Electrically driven propellers in pods have been developed since high hysteresis motors enabled significant increases in the power that can be transmitted. The pods are on a vertical axis which enables the direction of thrust to be changed by 'azimuthing'.

ALTERNATIVE MEANS OF PROPULSION

These can only be touched upon very briefly, and the following list is by no means exhaustive:

(a) Hydraulic or jet propulsion

If water is drawn into the ship and then thrust out at the stern by means of a pump then the ship can be regarded as jet propelled.

Since the pump or impeller is basically a propeller, the overall efficiency of such a system is lower than the corresponding screw propeller, i.e. of diameter equal to the jet orifice diameter, because of the resistance to flow of water through the duct in the ship. It is attractive, however, where it is desirable to

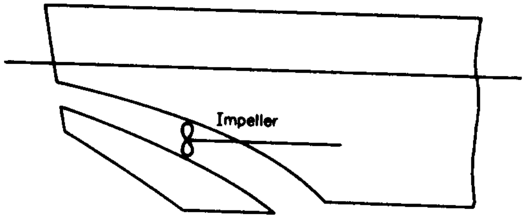


Fig. 10.15 Jet propulsion

have no moving parts outside the envelope of the main hull. This is the case of craft operating in very shallow water and a very successful class of boat has been designed using this principle for operating on shallow rivers. Many fast craft use water jet propulsion with water discharging into air and some high powered units are now available.

(b) Paddle wheels

In essence, the paddle wheel is a ring of paddles rotating about an athwartship horizontal axis.

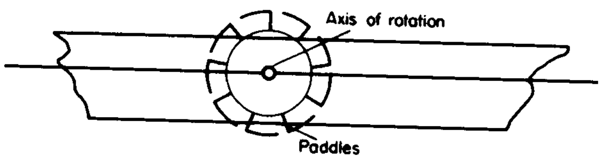


Fig. 10.16 Paddle wheel

In the simplest form, the paddles are fixed but greater efficiency is obtained by feathering them as the blades enter and leave the water. They can confer good manoeuvrability on a ship when fitted on either side amidships and for this reason many tugs have been designed using this principle. For operation in narrow waters, the large beam of this arrangement may be unacceptable and this was the consideration that led to the development of the 'stern wheeler' on the rivers of the USA.

(c) Wind/air reaction

In a sailing ship the sail, when stretched under the action of the wind, can be regarded as an aerofoil section developing lift and drag as would a solid body (see Chapter 16). The early 1980s saw a revival of interest in sails in merchant ships as an economy measure. A Japanese 1600 dwt tanker achieved 12 per cent fuel economy with a sail area of 194 m^2.

The Flettner Rotor concept gave the *Buckau* a speed of 5 knots or so in a 10 knot wind in 1925. Blown circulation control can be used to avoid the need for a rotating 'thruster' by employing a circulation controlled aerofoil device based on the phenomenon that an accelerated stream of fluid from a tangential jet tends to remain attached to a curved surface.

MOMENTUM THEORY APPLIED TO THE SCREW PROPELLER

It was shown above, that the force available for propelling a ship could he related to the momentum in the screw race. Let us now develop this idea in a little more detail. The propeller will cause water to accelerate from some distance ahead of the propeller disc and, because water is virtually incompressible, the flow of water through the disc will be as in Fig. 10.17.

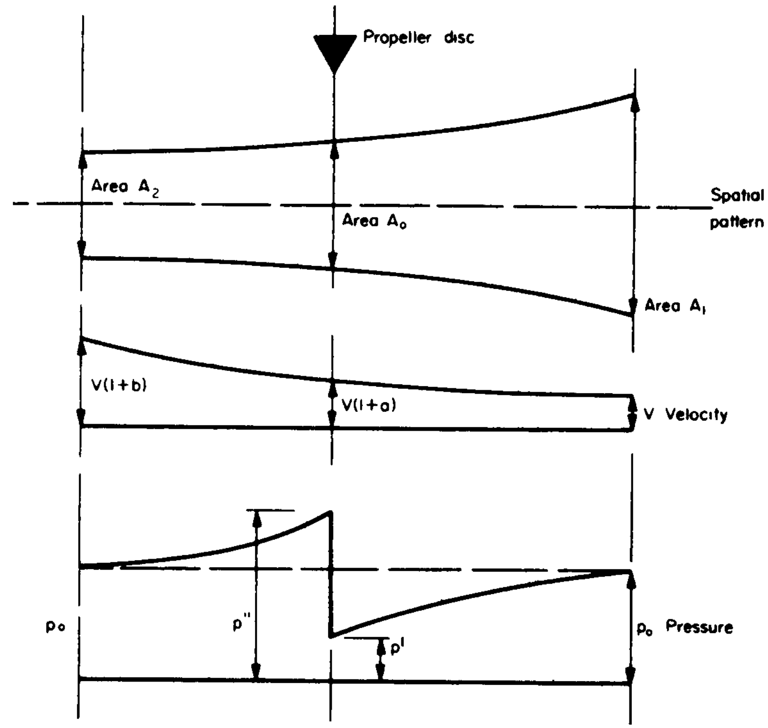


Fig. 10.17 Spatial, velocity and pressure patterns

Let A_1 and A_2 be points sufficiently ahead of and abaft the actual propeller disc so that the pressure at these points is effectively that in the free field. Due to the contraction of the screw race, the velocity will increase as shown and pressure will decrease between A_1 and the disc, suffer a jump at the disc and then decrease again between the disc and A_2. Now

$$\text{thrust on propeller} = T = A_0(p'' - p')$$

and applying Bernoulli's principle to both sides of the disc,

$$p_0 + \rho\frac{V^2}{2} = p' + \rho\frac{V^2}{2}(1+a)^2$$

$$p_0 + \rho\frac{V^2}{2}(1+b)^2 = p'' + \rho\frac{V^2}{2}(1+a)^2$$

Subtracting

$$p'' - p' = \rho\frac{V^2}{2}(1+b)^2 - \rho\frac{V^2}{2}$$

Hence

$$T = \frac{\rho}{2}A_0V^2(2b + b^2)$$

But, also, thrust = rate of increase of axial momentum

$$\therefore\ T = \rho A_0V(1+a)bV = \rho A_0V^2(1+a)b$$

Comparing these two expressions for thrust, it is seen that

$$a = b/2$$

That is to say, half the velocity increase experienced in the screw race is caused by the suction created by the propeller and takes place before the water enters the propeller disc. This factor of increase, a, is known as the *axial inflow factor*. This factor controls the propeller efficiency that can be obtained since

$$\begin{aligned}\text{Propeller efficiency} &= \frac{\text{useful work done by propeller}}{\text{power absorbed by the propeller}}\\ &= \frac{\text{thrust} \times \text{propeller speed}}{\text{overall change in kinetic energy}}\\ &= \frac{\rho A_0V^2 2a(1+a)V}{\frac{1}{2}\rho A_0V(1+a)V^2[(1+2a)^2-1]}\\ &= \frac{2\rho A_0V^3a(1+a)}{\frac{1}{2}\rho A_0V^3(1+a)(4a^2+4a)}\\ &= \frac{1}{1+a}\end{aligned}$$

This shows that even in the ideal case, high propeller efficiency is only possible with a small inflow factor, i.e. with a large diameter propeller.

In actual propellers the efficiencies will be less than this ideal due to a wide range of effects including the finite number of blades, the propeller hub, the thickness of blades, wake variations, cavitation and viscous losses.

THE BLADE ELEMENT APPROACH

The momentum theory is useful in indicating the influence of the propeller on the water ahead of its own disc, and in demonstrating that even theoretically there is a limit to the efficiency which can be achieved. It is not, however, of iirect value in assessing the torque and thrust developed in a propeller of a particular geometry.

One approach is to consider each blade of the propeller as made up of a series Jf annular elements such as the shaded portion in Fig. 10.18 which represents

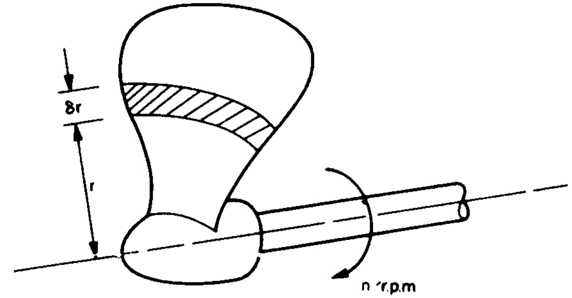


Fig. 10.18 Annular element of propeller blade

that portion of the blade between radii r and $r + \delta r$. If the propeller is turning at n r.p.m., then the element will have a tangential velocity of $2\pi rn$ besides a velocity of advance V_1 relative to the water. The element, which can be regarded as a short length of aerofoil section, will experience a relative water velocity as shown in Fig. 10.19.

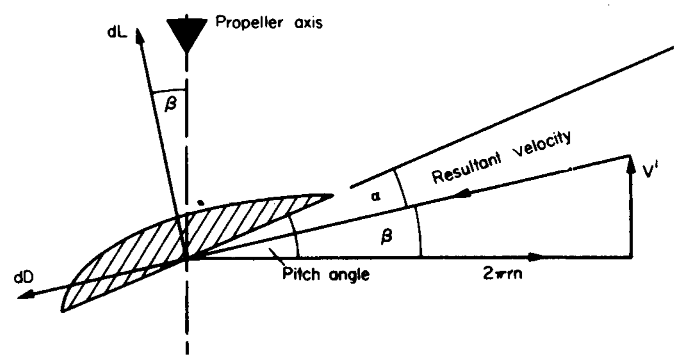


Fig. 10.19 Water flow relative to blade element

It will be seen that the blade element is at an angle α, to the resultant velocity. This angle is known as the *angle of attack*. To explain what happens now, it is necessary to introduce the concept of a *vortex*. In a potential vortex, fluid circulates about an axis, the circumferential velocity of any fluid particle being inversely proportional to its distance from the axis. The strength of the vortex is defined by the *circulation* Γ. If the blade element were in an inviscid fluid, the

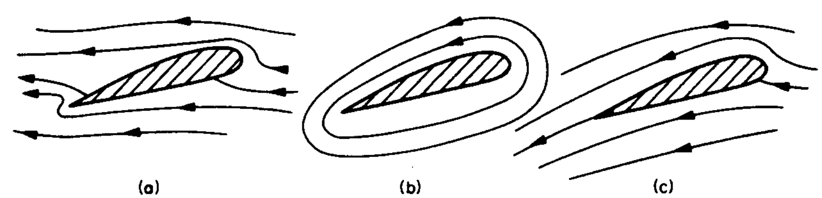


Fig. 10.20

potential flow pattern around it would be as in Fig. 10.20(a). In a real fluid, the very high velocities at the sharp trailing edge produce an unstable situation in the viscous fluid due to shear stresses. The potential flow pattern breaks down and a stable flow pattern is established as in Fig. 10.20(c). This consists of the original uniform flow with a superimposed vortex (Fig. 10.20(b)) having the foil as a core. The strength of the circulation depends upon the shape of the section and its angle of attack.

If, in an inviscid fluid, a circulation of this same strength could somehow be established, the blade element would be acted on by a lift force normal to the resultant velocity and the force would be proportional to the circulation. In a real fluid, the viscosity which gives rise to the circulation also introduces a small drag force whilst having little influence on the lift.

In Fig. 10.19, the lift and drag acting on the blade element are shown as dL and dD respectively. As already stated, the circulation depends upon the shape of the section and the angle of attack. A number of so-called aerofoil sections are available which produce high lift for small drag. If one of these sections is being used, its characteristics will be available from standard tests. Hence, the lift and drag on each element of the blade can be calculated. By resolving parallel and normal to the propeller axis, the contributions of the element to the overall thrust and torque of the propeller are

$$dT = dL\cos\beta - dD\sin\beta$$

and

$$dQ = (dL\sin\beta + dD\cos\beta) \times (\text{radius of element})$$

By repeating this process for each element and integrating over the blade, the thrust and torque on each blade and hence of the propeller can be obtained. Account can be taken of propellers in which the pitch angle varies with radius, but a really comprehensive theory of propellers must also take into account the interference between blades, and the tendency for pressures on the face and back of the blade to be equalized by flow around the tip of the blade.

In more advanced theories the lifting surfaces of the propeller are represented by lines, surfaces or panels of vortices and source-sink distributions to derive lift and drag by mathematical analysis. The theories can be used to design individual propellers or to indicate broad lines of development for methodical series. They enable the design of the blades to be optimized for the variations in water velocity at the propeller disc. This in turn has increased interest in determining these velocity distributions accurately by means of wake surveys and prediction. This is not an easy matter because the propulsor itself affects the distribution. Laser measuring techniques can be used to avoid introducing physical probes into the area which themselves would affect the flow pattern. Some averaging is necessary of the changing pattern experienced by individual blade elements as the propeller rotates.

A full treatment of these matters is not possible in a book of this nature but two major factors—cavitation and the generalized interaction between hull and propulsor—are now discussed.

CAVITATION

The thrust and torque of the propeller depend upon the lift and drag characteristics of the blade sections. The lift on the section is produced partly from the suction on the back of the blade and partly from positive pressure on the face. In this context, suction and positive pressure are relative to the free field pressure at the blade. A typical pressure distribution is shown in Fig. 10.21.

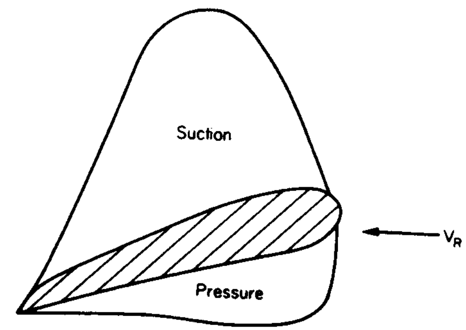


Fig. 10.21 Pressure distribution over an aerofoil section

As the pressure on the back falls lower and lower, with increasing propeller r.p.m., say, the absolute pressure will eventually become low enough for the water to vaporize and local cavities form. This phenomenon is known as *cavitation.* Since the water cannot accept lower pressures, the lift cannot increase as rapidly and the presence of cavitation manifests itself, therefore, as a fall off in thrust, torque and efficiency. There is also a marked increase in underwater noise and pressure variations in the vicinity of the propeller. Radial variation of pitch can improve cavitation performance and need have little effect on propeller efficiency. The pressure at which cavitation occurs depends upon temperature, the amount of dissolved air or other gases present and the surface tension. Without gasses in solution, the pressure might typically be of the order 3,500 N/m^2 for fresh water and the presence of air can increase this to about 10,500 N/m^2. As a corollary to the circulation around the blade, which produces the lift forces, the blade sheds trailing vortices associated with the blade tip, propeller hub and the face. The core of a vortex will be at a relatively low pressure and the first cavitation usually occurs in the tip vortex—the *tip vortex cavitation.* As σ, the cavitation number, is reduced, the cavitation spreads across the back of the blade giving the appearance of a sheet or surface of separation. This is known as *sheet cavitation* and can occur on both the face and back of the blade.

Other forms of cavitation are *bubble cavitation* which usually occurs at the thick root sections of the blades and is particularly susceptible to local irregularities in the blade surface. Unfortunately, the bubbles formed in a region of low pressure can be swept away into regions of high pressure where they collapse. This can lead to intense local pressures which may cause pitting or erosion of the propeller blades. Such pitting not only produces a weaker propeller but also increases the surface irregularities of the blades. *Face cavitation* can occur when the blade sections are working at very small or negative angles of incidence which can arise when a propeller operates in a varying velocity field. An undesirable aspect of cavitation in warships is the noise associated with the cavitation bubbles and their collapse. This can betray the ship's presence to an enemy and can attract homing weapons.

It follows from the simple discussion above of the nature of the phenomenon, that cavitation is likely to be delayed and less severe after onset if:

1. blades are lightly loaded and the pressure distribution is kept as smooth as possible. That is peaks should be avoided;
2. the pressure loading is reduced towards the blade tips;
3. the wake is kept as uniform as possible avoiding large changes in velocity and angle of inflow.

Providing high blade skew is found to be favourable for cavitation performance. In one single screw cargo ship, a tip skew of 60 degrees increased the cavitation inception speed by 3 knots compared with a straight propeller. High skew also reduces cavitation-induced vibration.

A special situation arises in propellers running at high r.p.m., as might be the case in a high speed motor boat. In these circumstances, it is impossible to eliminate cavitation but reasonable efficiency can be obtained by using a propeller designed to have the back completely covered by cavitation; although the lift generated by the back is limited by the vapour pressure, the torque component due to skin friction on the back of the blade is eliminated. Such a propeller is known as a *super-cavitating* propeller. It is very inefficient at speeds lower than 40-50 knots.

SINGING

Before the onset of cavitation, the blades of a propeller may emit a high-pitched note. This singing, as it is termed, is due to the elastic vibration of the material excited by the resonant shedding of non-cavitating eddies from the trailing edge of the blades. Heavy camber appears to be conducive to singing. Cures can be effected by changing the shape of the trailing edge of increased damping of the blade.

INTERACTION BETWEEN THE SHIP AND PROPELLER

The interaction manifests itself in the following ways:

(a) the hull carries with it a certain mass of water as was pointed out in considering the boundary layer. This means that the average velocity of water relative to the propeller disc is no longer equal to the velocity of advance of the propeller relative to still water;

(b) the water velocity will vary in both magnitude and direction across the propeller disc and the performance of the propeller will differ from that in open water even allowing for the difference in average velocity;

(c) the propeller causes variation in local pressures in the water and these will react upon the hull, leading to an effective increase in resistance.

Let us proceed to consider each of these effects in more detail.

The difference between the ship speed and the speed of the water relative to the ship is termed the *wake.* The wake is the combination of the boundary layer associated with skin friction, the flow velocities occasioned by the streamlined form of the ship and the orbital velocities of the waves created by the ship. If the water is moving in the same direction as the ship, the wake is said to be positive. If the ship speed is V and the average velocity of the water relative to the hull at the propeller position is V_1, then:

$$\text{Wake} = V - V_1$$

To non-dimensionalize this relation, the wake can be divided by either V_1 or V. The former was proposed by Froude and the latter by Taylor leading to two wake factors as follows:

$$\textit{Froude wake factor} = w_F = (V - V_1)/V_1$$
$$\textit{Taylor wake factor} = w = (V - V_1)/V$$

Clearly, these are merely different ways of expressing the same phenomenon.

Apart from this *average* flow of water relative to the hull there will be variations in velocity over the propeller disc. As the hull is approached more closely, the water moves less fast relative to the ship. Apart from this general effect of the hull there will be local perturbations due to the shaft, shaft bossings or shaft brackets and other appendages. Due to the fact that the water must 'close-in' around the stern the flow through the propeller disc will not be everywhere the same and will not, in general, be parallel to the shaft line. These effects are combined and expressed as a *relative rotative efficiency* (RRE) which is defined as

$$\text{RRE} = \eta_R = \frac{\text{efficiency of propeller behind the ship}}{\text{efficiency of propeller in open water at speed } V_1}$$

Finally, there is the influence on the hull of pressure variations induced by the propeller action. As far as the propeller is concerned it has to produce a thrust T which is greater than the resistance R of the hull without propeller.

As with the wake, there are two ways of expressing this physical phenomenon. It can be considered as an *augment of resistance*, a, where

$$a = \frac{T - R}{R}$$

or, it can be regarded as a *thrust deduction factor*, t, where

$$t = \frac{T - R}{T}$$

HULL EFFICIENCY

The thrust power (P_T), developed by the propeller is given by the product of T and V_1. On the other hand, the effective power is given by the product RV.

Now

$$P_T = TV_1 = R(1+a)\frac{V}{1+w_F} = \frac{RV(1+a)}{1+w_F} = RV(1+a)(1-w)$$

$$= RV\frac{(1-w)}{(1-t)}$$

therefore

$$\frac{P_E}{P_T} = \frac{1+w_F}{1+a} \text{ or } \frac{1-t}{1-w}$$

This ratio is known as the *hull efficiency* and seldom differs very greatly from unity.

To complete the picture of the propeller acting behind the ship, the concept of relative rotative efficiency must be added in. The three factors, augment, wake and RRE are referred to collectively as the *hull efficiency elements*. Augment and wake are functions of Reynolds' number but variation between ship and model is ignored and the error so introduced is taken account of by the trials factors.

OVERALL PROPULSIVE EFFICIENCY

The shaft power (P_S) is the power needed to propel the complete ship. The ratio between the P_E and P_S is a measure of the overall propulsive efficiency achieved and is termed the *propulsive coefficient* (PC)

$$\text{PC} = \frac{P_E}{P_S}$$

The overall efficiency can be regarded as the cumulative effect of a number of factors. Consider the following in addition to P_E and P_S

P_{EA} = power to tow hull complete with appendages,

P_T = thrust power developed by propellers = TV_1,

P_D = power delivered to propellers when propelling the ship,

P'_D = power delivered to propellers when developing a thrust T in open water at a speed V_1.

Now the propulsive coefficient can be defined as:

$$\text{PC} = \frac{P_E}{P_S} = \frac{P_E}{P_{EA}} \times \frac{P_{EA}}{P_T} \times \frac{P_T}{P'_D} \times \frac{P'_D}{P_D} \times \frac{P_D}{P_S}$$

where

$$\frac{P_E}{P_{EA}} = \frac{1}{\text{appendage coefficient}}$$

$$\frac{P_{EA}}{P_T} = \text{hull efficiency, } \eta_H$$

$$\frac{P_T}{P'_D} = \text{propeller efficiency } \eta_0 \text{ in open water at speed } V_1$$

$$\frac{P'_D}{P_D} = \text{relative rotative efficiency, } \eta_R$$

$$\frac{P_D}{P_S} = \text{shaft transmission efficiency, } \eta_S$$

That is

$$\text{PC} = \left[\frac{\eta_H \times \eta_0 \times \eta_R}{\text{appendage coefficient}}\right] \times \text{transmission efficiency}$$

It is recommended that the transmission efficiency be taken as 0.97 for ships with machinery amidships and 0.98 for ships with machinery aft. For modern warships appendage coefficients vary from about 1.05 to 1.10. In using PC it is necessary to check the definition. Some authorities use PC = P_{EA}/P_S.

The quantity in the brackets is known as the *quasi-propulsive coefficient* (QPC), η_D, and can be obtained from model results. There is some error in applying this to the full-scale ship and to allow for this and transmission efficiency and any differences between the ship and model test conditions, e.g. wind, waves, cavitation, use is made of a *QPC factor* which is defined as

$$\text{QPC factor} = \frac{\text{PC from ship trial}}{\text{QPC from model}}$$

The value to be assigned to the QPC factor when estimating power requirements for a new design is usually determined from results of a similar ship.

The National Physical Laboratory (now part of British Maritime Technology) used a *load factor* instead of the QPC factor, where

$$\text{load factor} = 1 + x = \frac{\text{transmission efficiency}}{(\text{QPC factor})(\text{appendage coefficient})}$$

In the NPL analysis, the *overload fraction* x is intended to allow for the basic shell roughness, fouling, weather conditions and depends on ship length and type. It is recommended that whatever value of x is used in estimates a standard power estimate should also be made with a load factor of unity, i.e. with x = zero, and an appendage scale-effect factor $\beta = 1$, i.e. assuming appendage resistance scales directly from the model to the ship.

SHIP-MODEL CORRELATION

The conduct of a ship speed trial is dealt with later together with the analysis by which the ship's actual speed is deduced. This demonstrates whether the ship meets its specification but does not tell the designer much about the soundness of his prediction method. If the specified speed is not reached it may be that he wrongly estimated the ship's resistance or hull efficiency elements, the propeller design may have been incorrect or the machinery may not have developed the intended power. A much more comprehensive analysis of the trials data is required by the designer to assist him with later designs. Even if the speed prediction was acceptable, it is still possible that several errors in assessing various factors cancelled each other out.

The analysis method used must depend upon the design methods to be checked. Froude developed the following method using 'circular' functions defined as below:

$$(E) = \frac{1000(P_E)}{\Delta^{\frac{2}{3}}V^3}, \quad \text{using naked model } P_E$$

$$(E)_{\mathrm{A}} = \frac{1000(P_A)}{\Delta^{\frac{2}{3}}V^3}, \quad \text{where } P_A = \text{power to tow the appendages}$$

$$(E)_{\mathrm{WP}} = \frac{1000(P_{WP})}{\Delta^{\frac{2}{3}}V^3}, \quad \text{where } P_{WP} = \text{power to overcome windage and fouling under trials conditions}$$

$$(E)_{\mathrm{T}} = (E) + (E)_{\mathrm{A}} + (E)_{\mathrm{WP}}$$

N.B. If the speed trial is carried out under good conditions, $(E)_{\mathrm{WP}}$ should be negligible.

$$(T)_{\mathrm{M}} = \frac{1000(P_T)}{\Delta^{\frac{2}{3}}V^3} = (E)_{\mathrm{T}}(1+a)(1-w)$$

$$(T)_{\mathrm{R}} = \frac{H \times 1000}{\Delta^{\frac{2}{3}}V^3} = H(T)_{\mathrm{M}}/P_T$$

where

H = thrust power from open water propeller data using the trial r.p.m. and speed and the model wake

$$(D) = \frac{1000(P_D)}{\Delta^{\frac{2}{3}}V^3}$$

$$(I) = \frac{1000(P_S)}{\Delta^{\frac{2}{3}}V^3}$$

Each of the above parameters is calculated for each run and plotted to a base of speed. The propulsive coefficient, equal to $(E)/(I)$ or $\left[(E) + (E)_{\mathrm{A}}\right]/(I)$ with alternative definition of PC is also plotted with the QPC factor which is the ratio of the PC from the ship trial to the QPC from model tests.

If the predictions from model experiments were exact the QPC factor would equal the shaft transmission efficiency, and:

$$(T)_{\mathrm{R}} = (T)_{\mathrm{M}}$$

In general, this relationship is not precise as $(T)_{\mathrm{R}}$ includes some scale effects including those due to cavitation on the ship propeller. The ratio $(T)_{\mathrm{R}}/(T)_{\mathrm{M}}$ is known as the propeller thrust correlation factor

Thus, both the QPCF and the ratio $(T)_{\mathrm{R}}/(T)_{\mathrm{M}}$ show the essential differences between the model and full-scale data. In a new design the designer uses these quantities, deduced from previous trials, to assist in scaling from the model to the ship. He would make allowance for any differences in the two designs such as different appendage coefficients.

Model testing

RESISTANCE TESTS

Many great men attempted to use models or to show how they could be used to predict full-scale behaviour, including Bouguer, Tiedemann, Newton, Chapman, Euler and Beaufoy, but it was not until the time of William Froude that full-scale prediction became a practical proposition in the late 19th century.

It was William Froude who postulated the idea of splitting the total resistance into the residuary resistance and the frictional resistance of the equivalent flat plate. He also argued that air resistance and the effects of rough water could be treated separately. By studying the wave patterns created by geometrically similar forms at different speeds, Froude found that the patterns appeared identical, geometrically, when the models were moving at speeds proportional to the square root of their lengths. This speed is termed the *corresponding speed,* and this is merely another way of expressing constancy of Froude number. He also noted that the curves of resistance against speed were generally similar' the resistance per unit displacement was plotted for corresponding speeds. Proceeding further, he found that by subtracting from the total resistance an allowance for the frictional resistance, determined from flat plates, the agreement was very good indeed.

This led to *Froude's law of comparison* which may be stated as:

> If two geometrically similar forms are run at corresponding speeds (i.e. speeds proportional to the square root of their linear dimensions), then their residuary resistances per unit of displacement are the same.

Thus the essentials are available for predicting the resistance of the full-scale ship from a model. The steps as used by Froude are still used today, refinements being restricted to detail rather than principle. For each particular value of the ship speed:

(a) measure the resistance of a geometrically similar model at its corresponding speed,
(b) estimate the skin friction resistance from data derived from experiments on flat plates,
(c) subtract the skin friction resistance from the total resistance to obtain the residuary resistance,
(d) multiply the model residuary resistance by the ratio of the ship to model displacements to obtain the ship residuary resistance,
(e) add the skin friction resistance estimated for the ship to obtain the total ship resistance.

It should be noted that any error in estimating frictional resistance applies both to the model and ship. Thus, only the effect on the difference of the two is significant.

It is now possible to see why earlier attempts to correlate the total resistance of ship and model failed. Two models with identical resistances could only represent ships with identical resistances if the ratios of their residuary and skin friction resistance were the same. In general, this could not be true unless the forms were themselves the same. Indeed, if model A had less total resistance than model B it did not even follow that ship A would be less resistive than ship B. Thus, even the qualitative comparisons made between models, used so frequently even today in many branches of naval architecture, may be invalid.

RESISTANCE TEST FACILITIES AND TECHNIQUES

With the aid of a grant from the Admiralty, Froude constructed the world's first model tank at Torquay in 1871 where R. E. Froude continued his father's work on the latter's death in 1879. The work of the Froudes proved so useful that, when the lease on the Torquay site expired in 1885, a grant was made to erect another at Haslar in 1887. This was the beginning of the Admiralty Experiment Works (now used by the Defence Evaluation and Research Agency

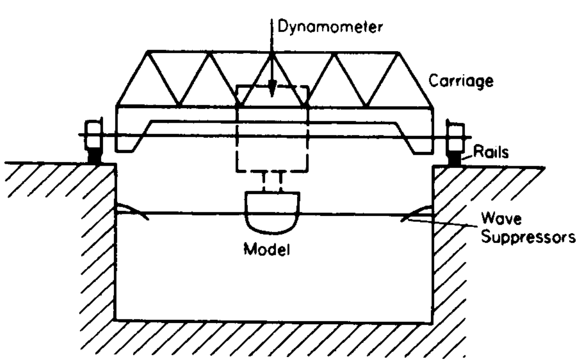


Fig. 10.22 Typical ship tank section

and British Maritime Technology), which has grown over the years and ha~ remained one of the world's leading establishments in this field.

Modern ship tanks for measuring model resistance are fundamentally the same as the first tank made by Froude. Such a facility is essentially a long tank, of approximately rectangular cross-section, spanned by a carriage which tows the model along the tank. Improvements have been made over the years in respect of the methods of propelling the carriage, in the constancy of speed holding, in the instrumentation and analysis of data. Digital recording and computers on carriages have reduced data reduction times significantly.

In a typical run, the carriage is accelerated up to the required speed, resistance records and measurement of hull sinkage and trim are taken during a period of constant speed and then the carriage is decelerated. With increasing ship lengths and service speeds, there has arisen a demand for longer and longer tanks to cope with the longer acceleration and deceleration runs.

Model test procedures today rely much on the methodical and painstaking approaches ofW. and R. E. Froude. As early as 1880, R. E. Froude was aware of unexplained variations in the resistance measured in repeat experiments on a given model. He suspected currents set up in the tank by the passage of the model and variations in skin friction resistance due to temperature changes. Methodical investigation into the first of these two features led to the adoption at AEW of small propeller type logs to record the speed of the model relative to the water. Investigation of the temperature effect led Froude to postulate that a 3 per cent decrease in skin resistance for every 10OF rise in temperature could be adopted as a fair working allowance and linked this with a standard temperature of 55of.

In the temperature experiments, R. E. Froude used the model of HMS *Iris*, a 91 m, 3760 tonnef despatch vessel, as a 'standard' model to be tested at various times throughout the year. Final proof that, even after correcting for tank currents and temperature, significant variations in resistance were occurring, came in tests on the *Iris* model in the tank at Haslar to correlate with those previously run at Torquay. This led to the application of a so-called *Iris correction* obtained by running the standard model at frequent intervals and applying a correcting factor to the resistance of a new model depending on the variation of the *Iris* resistance from its standard value. Generally, the *Iris* correction varies between **1 atfl-** 6 per cent, but during abnormal periods, commonly referred to as 'storms', the correction can be more than 10 per cent. The cause of the storms is now known to be due to the presence in the water of substances having long chain molecules. The concept of a standard model has since been adopted by other ship tanks.

The concept of deliberately introducing additives to the boundary layer in order to reduce frictional resistance has since been tried. Although substantial reductions have been achieved it is not at present an economic proposition.

MODEL DETERMINATION OF HULL EFFICIENCY ELEMENTS

Experiments must be carried out with the hull and propeller correctly combined as illustrated in Fig. 10.23.

With the model at the correct speed, corresponding to that of the ship under study, a series of runs is made over a range of propeller r.p.m. straddling the

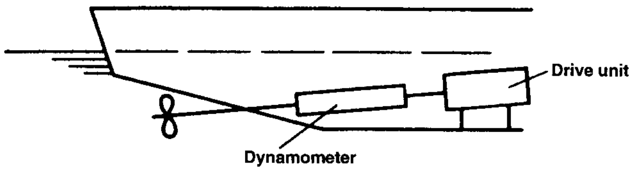


Fig. 10.23 Experimental technique

self-propulsion point of the model. Model speed and resistance are recorded together with the thrust, torque and r.p.m. of the propeller. Results are plotted to a base of propeller r.p.m., as shown for thrust in Fig. 10.24, to find the model self-propulsion point.

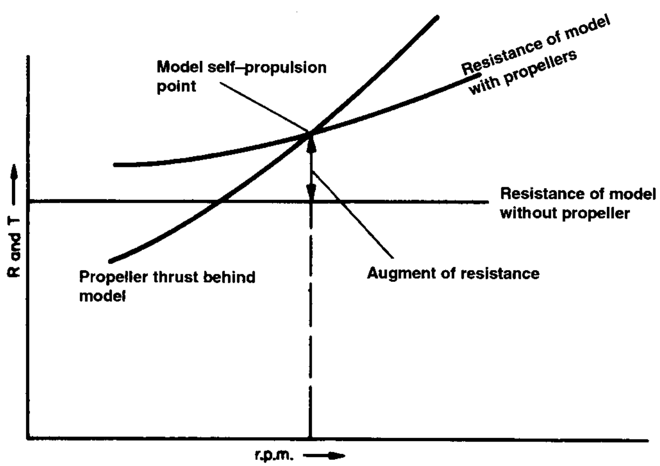


Fig. 10.24 Determination of model self-propulsion point

The model propeller then has its thrust and torque measured in open water at a speed of advance estimated to be that of the flow through the propeller when behind the hull, i.e. making allowance for the wake. By comparing this curve with that obtained in the combined experiment, the correct speed for the propeller in open water can be calculated. The difference between the model speed in the combined experiment and the corrected open water speed is the *wake.* The relative rotative efficiency follows as the ratio between the torques measured in the open water and combined experiments at self-propulsion r.p.m. The *augment of resistance* is obtained as illustrated in Fig. 10.24.

It should be noted that, although the propeller used in these experiments is made as closely representative of the ship propeller as possible, at least the first estimate of its geometry, the scale is too small to enable the thrust and torque figures to be used directly. Instead, the hull efficiency elements calculated as above are used with either methodical series data or specific cavitation tunnel measurements in order to produce the propeller design.

PROPELLER TESTS IN OPEN WATER

It is important that the designer has data available on which to base selection 0r the geometric properties of a propeller and to determine likely propeller eHi-ciency. Such data is obtained from methodical series testing of model propellers in open water. Such testing eliminates the effects of cavitation and the actual flow of water into a propeller behind a particular ship form, and makes comparisons of different propellers possible on a consistent basis.

The tests are carried out in a ship tank with the propeller mounted forward of a streamlined casing containing the drive shaft. The propeller is driven by an electric motor on the carriage. Thrust, torque, propeller r.p.m. and carriage speed are recorded and from these K_T, KQ, J and η can be calculated. Usually runs are carried out at constant r.p.m. with different speeds of advance for each run.

It will be appreciated that towing tanks can be used for a wide range of hydrodynamic tests other than those associated with resistance and propulsion. These are discussed in later chapters.

CA VIT AnON TUNNEL TESTS

It is impossible to run a model propeller in open water so that all the non-dimensional factors are kept at the same values as in the ship. In particular, it is difficult to scale pressure because the atmospheric pressure is the same for ship and model and scaling the depth of the propeller below the surface does not provide an adequate answer. If cavitation is important, the pressure of air above the water must be reduced artificially and this is the reason for using *cavitation tunnels* to study propeller performance. Such a tunnel is shown diagrammatically in Fig. 10.25, and is usually provided with means for reducing the air content of the water to improve viewing.

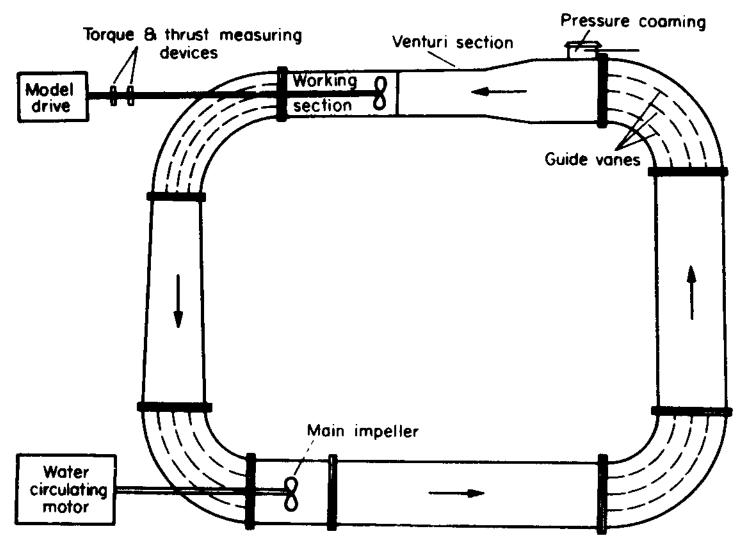


Fig. 10.25 Diagrammatic arrangement of a cavitation tunnel

In practice, experiments are usually run under the following conditions:

(a) the water speed is made as high as possible to keep Reynolds' number high to avoid serious scaling of skin friction;
(b) the model propeller is selected to have as large a diameter as is compatible with the tunnel size (tunnel wall effects must be avoided);
(c) model is run at the correct J value. This fixes the rate of propeller revolutions;
(d) the pressure in the tunnel is lowered to produce the correct cavitation number at the propeller axis.

Since the propeller revolutions are the most easily adjusted variable, it is usual to set the tunnel water speed, adjust the tunnel pressure to give the correct cavitation number and then vary the propeller r.p.m. systematically to cause a variation in the advance coefficient. The whole series can then be repeated for other ıJ values.

The tunnel shown in Fig. 10.25 is a fairly simple one and suffers from the fact that it is difficult to simulate the actual flow conditions at the after end of the ship. In some cases, attempts to reproduce this have been made using specially designed grids to control the local flow conditions. Also, the flow is from right to left in the working section so that the drive shaft on the model propeller is aft of the disc rather than forward of it as is the case for the ship. In big tunnels, both objections can be overcome by modelling the after end of the hull complete inside the tunnel and driving the propeller from inside this model hull.

A large tunnel at Hamburg, completed in 1989, has a test section 11 m long with a cross section 2.8 m x 1.6 m, water velocity up to 12 *mjs* and a pressure range 0.15 to 2.5 bar absolute. It can test integrated hulljpropulsor arrangements for surface ships, submarines or underwater weapons. Low noise levels within the facility permit the conduct of acoustical studies.

DEPRESSURIZED TOWING TANK

In the 1970s the Netherlands Ship Model Basin brought into service a towing tank in which the air pressure can be reduced above the whole water surface. It is 240 m long and 18 m wide, with a water depth of 8 m. The pressure in the tank can be lowered to 0.03 bar. The main advantage is an ability to carry out propulsion tests under more representative conditions and to study cavitation of the hull or appendages.

CIRCULATING WATER CHANNELS

It is only the relative movement of model and water that is important. Thus an alternative to towing the model through the water is to hold the model steady, whilst allowing freedom of vertical movement, and cause the water to flow past it. This is achieved in a circulating water channel (CWe) which may also be able to modify the air pressure above the water free surface. A major advantage of the CWC is the possibility of measurements being made over a much longer time span-recording time is no longer limited by the tank length. Uniform flow is difficult to achieve and for this reason smaller CWCs are often restricted

to flow visualization studies (again easier with a stationary model) and approximate force measurements.

Ship trials

SPEED TRIALS

When a ship has been completed, speed trials are carried out to confirm that the ship has met its specification as regards design speed. Such trials also provide useful data to help the designer in producing subsequent designs.

The trials are carried out over a known distance. The distance may be defined by precisely located land markers (Fig. 10.26) or in more open water by use of an accurate positional satellite navigation system. The following description relates to a land-based measured distance but the underlying principles are the same.

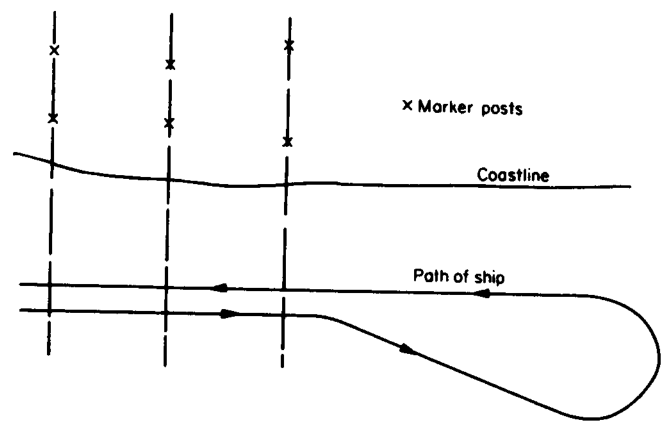


Fig.]IJ.26 Measured mile trials
(Note: Drawing not to scale. A straight approach run of about 3 miles is used.)

The ship approaches on a course normal to the lines joining corresponding pairs of 'mile' posts and sufficiently far off shore to ensure adequate dept~ water to eliminate the effect of depth of water on resistance. The time to traverse the measured distance is accurately noted together with shaft thrust, torque and revolutions. A fine day with little wind and calm seas is chosen. To reduce its effect upon resistance the use of the rudder is kept to a minimum during the run. At the end of the run the rudder is put over to a moderate angle and the ship is taken round in a large sweep, as illustrated, to provide adequate run-up for the next pass to ensure that the ship has stopped accelerating by the time it passes the first pair of posts.

The trial is carried out for a range of powers up to the maximum the machinery can generate. At each power, several runs are made in each direction to enable the effect of any tide to be eliminated. If the runs are made at regular time intervals, it is adequate to take a mean of means, i.e. by meaning each consecutive pair of speeds, taking means of consecutive pairs of results so obtained and so on. The process is illustrated by the following example in which the mean speed is 15 knots.

In order that the ship's condition may be assessed as accurately as possible, the state of the hull should be recorded at the undocking preceding the trial and the time out of dock noted. Before the ship leaves harbour, draughts and water properties should be noted to provide a measure of its displacement. The use of fuel before and during the trials will help assess the variations in the displacement and trim during the trials. During the trials the properties of the surrounding water should be recorded. Speed measurements are made of shaft rpm, torque and thrust. The meters used must be accurately calibrated.

CAVITATION VIEWING TRIALS

In warships, where noise signature is important, the first ship of a new class undertakes cavitation viewing trials. A suitable port is arranged in the hull to enable the propeller to be viewed with the ship underway. Stroboscopic lighting is used to 'freeze' the propeller so that cavitation patterns can be seen clearly and photographed. Such trials can be combined with the ship's speed trial. Barometric pressure and water properties must be recorded.

EXAMPLE 3. A ship on a measured mile course records the speeds of 14.82, 15.22, 14.80, 15.20, 14.78 and 15.18 knots for six consecutive runs at regular time intervals. Calculate the mean speed.

If runs are not carried out at regular time intervals, it is necessary to assume that the tide varies with time according to a mathematical equation such as

$$\text{Speed of tide} = v = a + a_1 t + a_2 t^2$$

where t is the time measured from the initial run made.

It is then assumed that the speed without tide would be V, say, and that the readings obtained represent $V + v$ where v is the value appropriate to the time the run was made.

Solution:

Measured Speeds (knots)	Means First	Second	Third	Fourth	Fifth
14.82					
	15.02				
15.22		15.015			
	15.01		15.010		
14.80		15.005		15.005	
	15.00		15.000		15.00
15.20		14.995		14.995	
	14.99		14.990		
14.78		14.985			
	14.98				
15.18					

Mean ship speed = 15.00 knots.

EXAMPLE 4. A ship on a measured mile course records the speeds of 15.22, 14.82, 15.20 and 14.80 at times of 1200, 1300, 1430 and 1530 hours. Calculate the speed of the ship and the equation governing the variation of the tidal current with time.

Solution: It is convenient to take the times at $t = 0$, $t = 1$, $t = 2.5$, $t = 3.5$, i.e. measuring in hours from the time of the initial run. Then, assuming that the ship speed is V and the tide is given by $v = a_0 + a_1 t + a_2 t^2$, we can write

at 1200 hrs; $15.22 = V + a_0$

at 1300 hrs; $14.82 = V - a_0 - a_1 - a_2$

at 1430 hrs; $15.20 = V + a_0 + 2.5a_1 + 6.25a_2$

at 1530 hrs; $14.80 = V - a_0 - 3.5a_1 - 12.25a_2$

Solving these equations,

$$V = 15.01 \text{ knots}$$

$$v = (0.21 - 0.028t + 0.008t^2) \text{ knots, } t \text{ in hours}$$

The difference in sign in alternate equations merely denotes that the tide is with or against the ship. Clearly, the tide is with the ship when it records its higher speeds but this is not significant to the mathematics since, if the wrong assumption is made, the tidal equation will lead to a negative tide.

SERVICE TRIALS

The above trials are carried out under calm conditions as a means of confirming that a ship meets its contractual requirement as regards speed. What is of greater interest to the owner, and designer, is the ship's performance in service under average or typical service conditions. Average figures for speed over long distances and associated fuel consumption have been extracted for many years from ship's logs. These could be related to the sea conditions as estimated by the crew. With the advent of satellites and satellite navigation systems it is possible to measure a ship's speed over shorter periods of time and link that performance with the sea system in which it was operating. Strictly the ship's speed is relative to land but this can be corrected for estimated tide and current effects. For special trials the satellite could track a buoy to provide these corrections. The specific sea conditions at any time can be obtained by retrospective analysis of wave data measured by satellite.

EXPERIMENTS AT FULL SCALE

Ship trials over a measured distance in calm water can confirm, or otherwise, the accuracy of the prediction of ship speed for a given power. They cannot, however, prove that the fundamental arguments underlying these estimates are valid. In particular, they cannot prove that the estimation of P_E was accurate because the influence of the ship propulsion system is always present.

William Froude realized this and with Admiralty assistance carried out full-scale resistance measurements on HMS *Greyhound* in 1874. More recently, full-scale resistance trials were carried out using the *Lucy Ashton* and HMS *Penelope.*

In the earlier trials, the screw sloop *Greyhound* was towed from an outrigger fitted to HMS *Active,* a vessel of about 3100 tonnef displacement. This method (Fig. 10.27) was adopted to avoid, as far as possible, any interference between the towing and the towed ship. Trials were carried out with the *Greyhound* at three displacements and covered a speed range of 3-12! knots. Some trials were with and some without bilge keels. For some runs the tow rope was slipped and the deceleration of the ship noted.

William Froude concluded that the experiments:

> ... substantially verify the law of comparison which has been propounded by me as governing the relation between the resistance of ships and their models.

In the *Lucy Ashton* trials some problems of towing a vessel were over-come by fitting the ship with four jet engines mounted high on the ship and out-board of the main hull to avoid the jet *efflux* impinging either on the hull or on the water in the immediate vicinity of the hull. Accurate measurement of thrust, totalling just over 6 tonnef from the four engines, was achieved by using hydraulic load measuring capsules. Speeds were measured over measured mile distances and special measures were taken to ensure accurate results and also to measure the surface roughness of the hull.

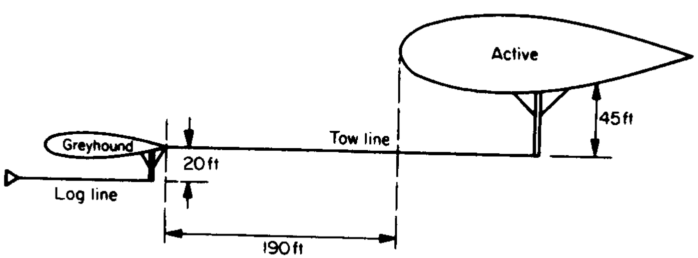


Fig. 10.27 Greyhound experiments

Resistance tests were made over a speed range of 5-15 knots with a clean naked hull with first a red oxide paint surface and then a bituminous aluminium paint. Each trial was repeated for sharp seams of plating and with the seams faired off with a plastic composition. Additional trials were run to study the effect of dummy twin-screw bossings, with twin-screw 'A' brackets and shafts and with a hull surface which had been allowed to foul for about a month.

The main purpose of the trial was to compare the various methods available for scaling model resistance to full-scale. The results indicated that Froude's law of comparison is valid for the scaling-up of wave-making resistance, but that the usual assumption that the skin friction of models and ships is the same as that of the corresponding plane surface of the same length and wetted

surface is not strictly correct. Fortunately, the error is not very important in practical calculations. The results also indicate that over the range of models tested, the interference between the skin-frictional and wave-making resistance is not significant.

The results of the trials proved that full-scale ship resistance is sensitive to small roughnesses. For instance, the bituminous aluminium paint, which was the smoother of the two surface finishes, gave about 3! per cent less total resistance which was estimated to be equivalent to about 5 per cent of the skin frictional resistance. Fairing the seams gave about a 3 per cent reduction in total resistance. The effect of 40 days fouling on the bituminous aluminium painted hull was to increase the skin frictional resistance by about 5 per cent, i.e. about kth of one per cent per day.

Trials in HMS *Penelope* were conducted by the Admiralty Experiment Works while the ship was operating as a special trials ship. *Penelope* was towed by another frigate using a mile-long nylon rope. Although the main purpose of the trial was to measure radiated noise from, and vibration in, a dead ship, the opportunity was taken to measure resistance and wake pattern of *Penelope* in calm water and in waves. For this purpose both propellers were removed and a pitot rake fitted to one shaft. Propulsion data were recorded in the towing ship also. Propulsion data for *Penelope* were obtained from separate measured mile trials with three different sets of propellers fitted.

Correlation of ship and model data showed the resistance of *Penelope* to be some 14 per cent higher than predicted over the range 12-13 knots but indicated no significant wake scale effects. The hull roughness, using a wall roughness gauge was found to be about 0.3 mm mean apparent amplitude per 50 mm. The mean apparent amplitude per 50 mm is the standard parameter used in the UK to represent the average hull roughness. The propulsion results showed that thrust, torque and efficiency of the ship's propellers were higher than predicted by model tests.

Summary

In studying the powering of ships, it is essential that the hull and propuls~ device be considered together. The shaft power required to drive a ship a'f"a given speed can be derived from a series of model tests and calculations. The basic elements in the assessment of the shaft power have been established and are summarized in Fig. 10.28.

It remains to show how model data is presented and the necessary calcula-tions carried out. This is done in the next chapter.

Problems

1. A 50 MN displacement ship, length 120 m, is to be represented by a model 3 m long. What is the displacement of the model? At what speed must it be run to represent a speed of 20 knots in the ship and what is the ratio of the ship to model effective power at this speed?

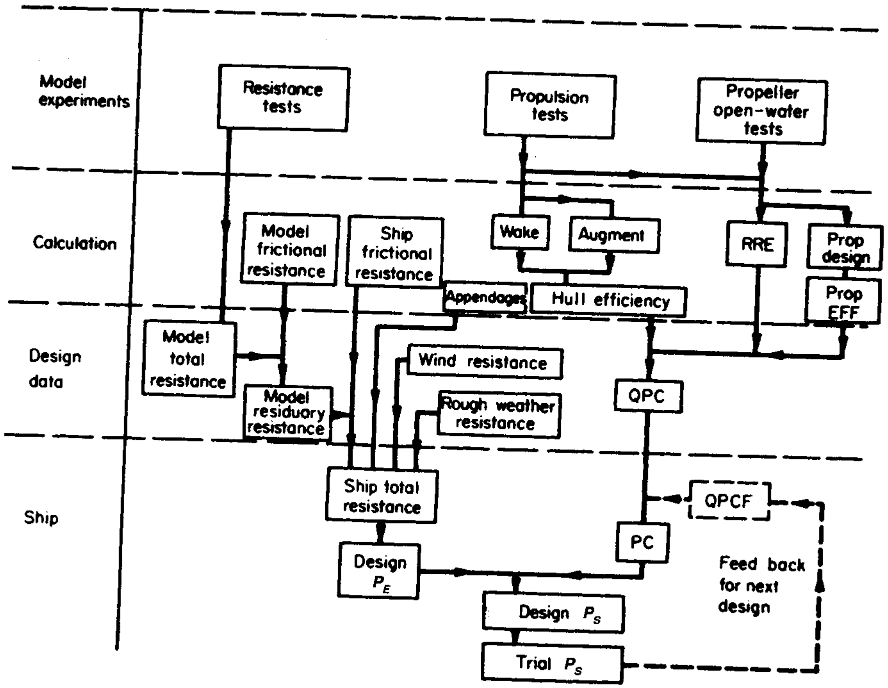


Fig. 10.28 Assessment of ship shaft power

2. Show how wave-making resistance at a given speed is affected by varying the length of parallel middle body, entrance angle and run remaining unchanged.

   On the resistance curve for a ship length 70 m with 20 m of parallel middle body, one hump occurs at a speed of 14knots and an adjacent one at V~ times this. At what speed would the main hump occur? If the length of parallel middle body is increased 15m, at what sequence of speeds will humps Occur on the resistance curve?

3. How are waves created when a typical warship form passes through the water? With the aid of sketches describe a typical ship wave pattern and explain what effect 'shoulders' on the curve of areas would have on this pattern. Draw a typical resistance curve for a ship indicating the main features of its characteristic shape and explaining why humps and hollows occur. Derive the expressions from which the position of these humps and hollows may be determined and, hence, determine the speeds at which the two most prominent humps Occur for a ship of effective wave-length 99 m.

4. Sketch and describe a typical ship wave pattern. What is meant by interference between wave systems? Show, with the use of diagrams, why humps and hollows occur in the curve of wave-making resistance against $V/\sqrt{L}$, giving approximate values of $V/\sqrt{L}$ at which the humps and hollows occur.

   A destroyer, length 122 m, is observed to be steaming at high speed. The first trough of the bow wave system is seen to coincide with the stern trough.

   Estimate the speed of the destroyer assuming that the wave system distul1l:c is $0.9L$.

5. A ship at full speed has an effective power naked of 7 x 10^6 W. The appendage coefficient is 1.15, the hull efficiency 0.98, the propeller efficiency 0.69, the RRE 0.99 and the QPCF 0.90. Calculate the propulsive coefficient, the quasi propulsive coefficient and the shaft power required.

6. A 50 MN displacement ship 100m long is towed in the naked condition on a long tow rope at a speed of 20 knots. The force in the tow rope is 1 MN. Find the effective power for the ship. Deduce the shaft power for a geometrically similar ship 120m long at 20 knots assuming that shaft power in this speed region is proportional to the cube of the speed and that:

appendage coefficient	1.20
hull efficiency	0.97
propeller efficiency	0.72
RRE	1.00
QPCF	0.95

7. Describe how speed trials are conducted, listing the items recorded. What factors would you consider important when choosing a site for a new measured mile course?

   Full-power trials of a new frigate involved five passes over the measured mile, each pass being followed by one in the opposite direction. The times of the start of each run and the speeds attained are:

Time of start	1045	1103	1127	1227	1245
Speed of run (knots)	27.59	28.66	27.64	28.60	27.69

   Making suitable adjustments to the time intervals and assuming the tide speed is given by $v = a + bt + et^2 + dt^3$, determine the true speed of the ship.

8. List the measurements which are made, during sea trials, on each run over the measured mile, explaining briefly how each measurement is made.

   The following data were obtained during progressive speed trials on a merchant ship. Assuming that the tidal velocity may be expressed in the form $v = a + bt + et^2$, calculate the true speed at each power.

Run no.	Direction	Time of day	Recorded speed (knots)	r.p.m	P_S (MW)
I	N	0830	10.35 }		
2	S	0900	9.60	83	0.86
3	N	0930	12.52 }		
4	S	1000	11.70	102	1.52
5	N	1130	14.30 }		
6	S	1200	13.96	126	2.42

9. A vessel on successive runs on the measured mile obtains the following speeds in knots:

   27.592,28.841,27.965,28.943,27.777,28.426

Calculate (i) ordinary average speed, (ii) mean of means of six runs, (iii) mean of means of first four runs, (iv) mean of means of second four runs, (v) mean of means of last four runs.

10. Assuming that the speed runs reported in the last question were obtained as a result of runs at intervals of one hour, deduce the true speed of the ship assuming that the tide is governed by an equation

$$v = a + bt + ct^2 + dt^3 + et^4$$

Determine the values of the coefficients in this equation.

11. A propeller 3 m in diameter moves ahead at 15 knots in 'open' sea water. If the propeller race has a 3 knots increase in speed, approximate by the axial momentum theory to the thrust developed.

12. The propellers of a twin-screwed ship operate in a wake of 2 knots, the ship moving ahead at 21 knots. The P_E naked is 4.47 MW, the appendage coefficient is 1.12. If the thrust developed by each propeller is 0.264 MN, calculate *(a)* the P_T of each propeller, *(b)* the hull efficiency, *(c)* the augment of resistance factor.

11 Powering of ships: application

Presentation of data

Any method of data presentation should bring out clearly the effect of the parameters concerned on the resistance of the ship. A non-dimensional form of plotting is desirable but further than this it is difficult to generalize. The best plot for a designer may not be the best for research. The best form of plotting may depend upon how data is to be processed and then the type of calculation in which it is to be used.

It was shown, in the previous chapter, how dimensional analysis can be used to derive a suitable form for non-dimensional presentation of the data involved in ship hydrodynamics.

RESISTANCE DATA

The Froude approach

It was William Froude who first postulated that a ship's resistance is made up of two main components, one due to friction and one to wavemaking. He assumed that these components followed different scaling laws and that they did not interact with each other. He further assumed that the skin friction component of resistance was the same as the resistance of a thin flat plate of the same length (model or ship) with the same wetted surface area at the same speed. He realized there was a smaller component due to eddy making but assumed this could be treated in the same way as the wavemaking resistance. The elements of resistance not due to friction he called the *residuary resistance.*

To present his work, Froude developed what is known as the *circular notation.* Although a truly non-dimensional form of presentation it looks strange ..." the modern eye. Because it was so important in developing knowledge in this field, and because a lot of data exists in this form, the *Froude notation* and its use are dealt with in the Annex at the end of the book.

ITTC presentation

The International Towing Tank Conference use the following notation

$$C = \text{Resistance coefficient} = \frac{\text{Resistance}}{\frac{1}{2}\rho S V^2}, \quad S = \text{wetted surface area}$$

Subscripts, T, V, R, F and AA are used to denote total, viscous, residuary, frictional and air resistance respectively. Further qualification is by means of subscripts S and M denoting the ship and model respectively.

The following relationships have been adopted:

$$C_{VM} = (1+k)C_{FM}$$

where k is a form factor from low-speed resistance tests

$$C_{RS} = C_{RM} = C_{TM} - C_{VM}$$
$$C_{VS} = (1+k)C_{FS} + \Delta C_F$$

where ΔC_F is a roughness allowance

$$C_{TS} = C_{VS} + C_R + C_{AAS}$$

The use of this notational method in arriving at the ship performance is discussed later. Although the ITTC presentation is now that generally used much useful data still exists within the framework of two old presentational methods described in the Annex and below.

Taylor's method

Taylor (1943) expressed resistance, both frictional and residuary, in lbf per tonf of displacement (i.e. R/Δ). For similar models at corresponding speeds such quantities are constant for resistances following Froude's law of comparison. They are compared on the basis of the following parameters:

$$\text{Speed coefficient} = \frac{V}{\sqrt{L}}, \quad V \text{ in knots}, \quad L \text{ in ft}$$

$$\text{Displacement/length ratio} = \frac{\Delta}{(L/100)^3}, \quad \Delta \text{ in tonf}$$

Prismatic coefficient

Beam/draught ratio

Taylor chose the displacement/length ratio as a quantity which is independent of displacement for similar ships. Length is used in the denominator as being the linear dimension having most influence on resistance. In this expression, ~ is in tonf of salt water whether ship or model is under consideration.

Unfortunately, this type of presentation is not truly non-dimensional and care must be taken with units in applying Taylor's data. Some typical curves are reproduced as Figs 11.1 and 11.2.

Taylor studied the influence of bow shape on resistance by considering the slope of the curve of sectional areas at the bow. This slope is expressed by a quantity t obtained as follows. Draw the tangent at the bow to the curve of sectional areas. This will cut the vertical at the centre of length, intercepting it on a certain ordinate. Then t is the ratio between this ordinate and the ordinate of the sectional area curve at the centre of length (Fig. 11.3).

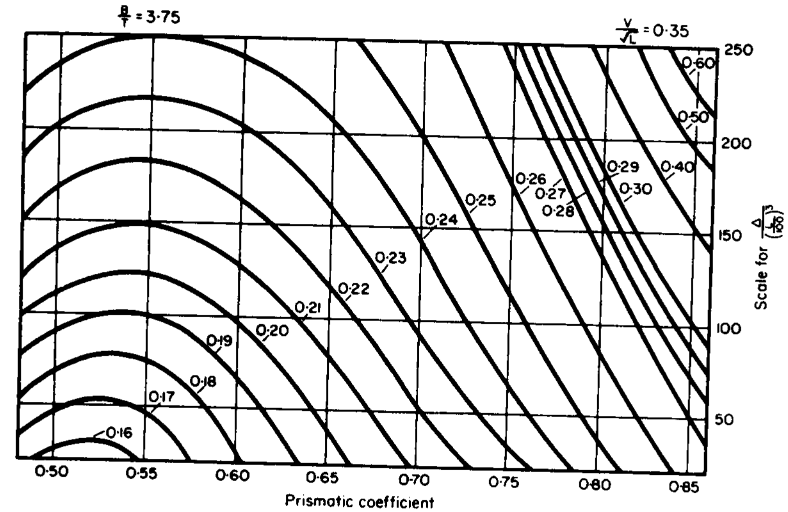


Fig. 11.1 Contours of residuary resistance in lbf per tonf of displacement

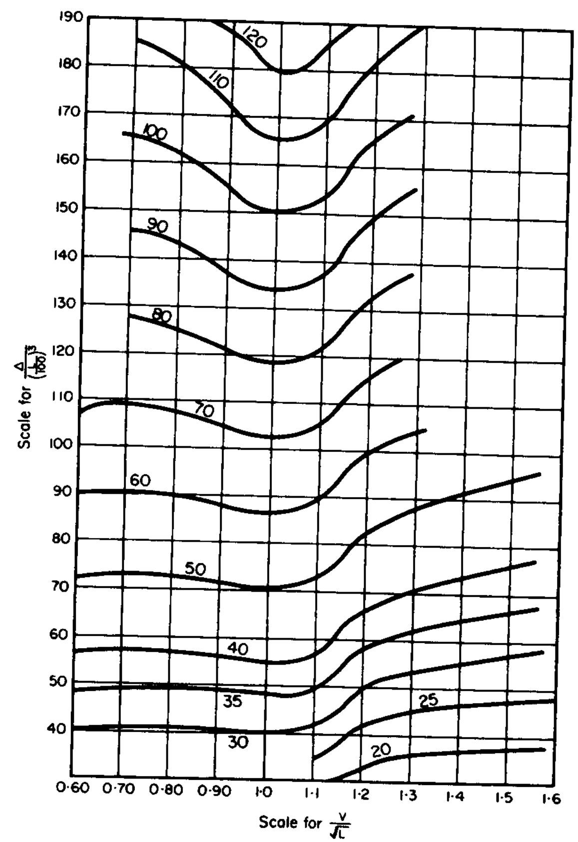


Fig. 11.2 Contours of (midship section area)/(L/100)² for minimum residuary resistance

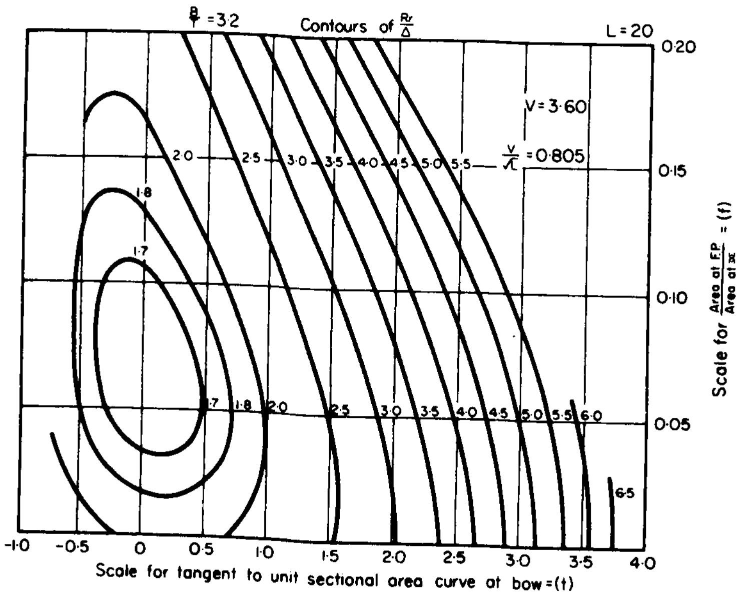


Fig. 11.3 Variation of resistance with tangent value

Taylor suggested that the wetted surface area of vessel could be obtained from the formula:

$$S = C(\Delta L)^{\frac{1}{2}}$$

where C is defined by Fig. 11.4.

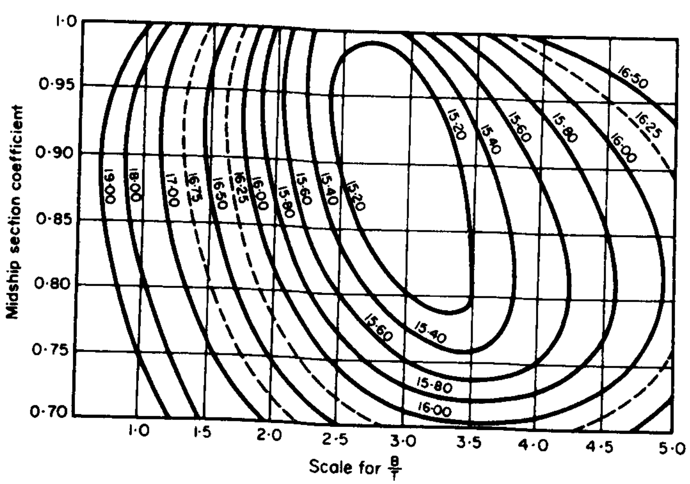


Fig. 11.4 Contours of wetted surface coefficients

Statistical analysis

A statistical analysis of resistance data for destroyers and frigates used regression equations for Ⓒ for a 4.9 m model in the form:

$$Ⓒ = b_0 + b_1x_1 + b_2x_2 + \cdots + b_nx_n$$

The first nine terms of the right-hand side of these equations represented form parameters as follows:

$x_1 = \overline{F_{WL}G}/L_{WL}$, where $\overline{F_{WL}G}$ is the horizontal distance of the c.g. from fore end of the static waterline.

x_2 = half angle of entrance.

$x_3 = B/T$

x_4 = (max. transverse sectional area below WL)/BT

x_5 = transom area as a percentage of the max. section area

x_6 = (distance of ACU from FE of static waterline)/L_{WL}

x_7 = Ⓜ

x_8 = prismatic coefficient

The remaining terms represented various combinations of these basic variates, i.e. they represented non-linear terms. The best fit was obtained with the more complex equations containing the non-linear terms. This is to be expected but it does not follow that the complex equations give better predictions for a new design. As an example, Table 11.1 gives the b coefficients for the simpler equations for four Ⓚ values.

This method of analysis can be very useful but large extrapolations from the generality of forms on which the data are derived must be treated with care. The parameters used are those which are thought to be most significant.

Table 11.1

Coefficient	Ⓚ			
	1.8	3.2	4.6	6.0
b_0	0.1032/01	0.3269/01	0.7393/01	0.4589/01
b_1	0.2798/00	−0.02364/01	−0.5395/01	−0.2332/01
b_2	0.6697/−02	0.1531/−01	−0.1972/−02	−0.6813/−02
b_3	0.3620/−01	0.8618/−01	0.1225/00	0.7545/−01
b_4	−0.4145/00	−0.1035/01	−0.5765/00	−0.1309/00
b_5	0.5635/−02	0.7369/−03	0.1501/−01	0.1200/−01
b_6	−0.1618/00	0.1293/00	0.6622/00	0.1280/00
b_7	0.5690/−01	−0.1536/00	−0.3577/00	−0.1669/00
b_8	−0.3060/00	0.1384/01	−0.5878/00	−0.6869/00

Note: The 'b' coefficients are given in exponential form, i.e. 0.1032/01 represents 1.032; −0.4145/00 represents −0.4145; 0.1200/−01 represents 0.01200, etc.

Experience may show that other parameters are more suitable. So far (M) has proved the most significant and a useful relationship is:

$$(C) = a + b/(M) + c(M)^2$$

In general it is not possible to vary one parameter without some consequential change in others. This may explain why the simple equations above indicate that an increase in transom area is always bad. This is not in accord with common experience and illustrates the danger of attributing any particular physical significance to the sign and magnitude of the regression coefficients. The interaction between various parameters also influences the extent to which optimization can be achieved. Values obtained from a mathematical optimization process may not be achievable in one form and in any case do not uniquely define the ship form. The regression equations should only be applied to forms of the same general type as those used to derive the equations.

PROPELLER DATA

In Chapter 10, dimensional analysis led to the derivation of three basic coefficients, viz.:

$$K_T = \frac{T}{\rho n^2 D^4} = \text{thrust coefficient}$$

$$K_Q = \frac{Q}{\rho n^2 D^5} = \text{torque coefficient}$$

$$J = \frac{V}{nD} = \text{advance coefficient}$$

In these coefficients, the product nD is a measure of the rotative speed of the propeller.

The other basic parameter is the propeller efficiency η which is given by

$$\eta = \frac{\text{useful output}}{\text{input}}$$

$$= \frac{TV}{Q \times 2\pi n} = \frac{K_T}{K_Q}\frac{J}{2\pi}$$

For a given advance coefficient, it is only necessary to define two of the factors *KT, KQ* and 'fl as the third follows from the above relationship. The two usually quoted are *KT* and 'fl. It was pointed out in Chapter 10 that a propeller designer makes considerable use of the results of methodical model series representing the propeller in open water. Such series are reported in various technical papers and a typical plot for a given blade area ratio is shown in Fig. 11.5. Similar plots are available other BARs.

In most design problems, the speed of advance and the power *Po* to be absorbed are known. In addition, the propeller r.p.m. are also often fixed by considerations of gear ratios and vibration. Diagrams such as that in Fig. 11.5 can be used to obtain, by interpolation, the propeller diameter for maximum

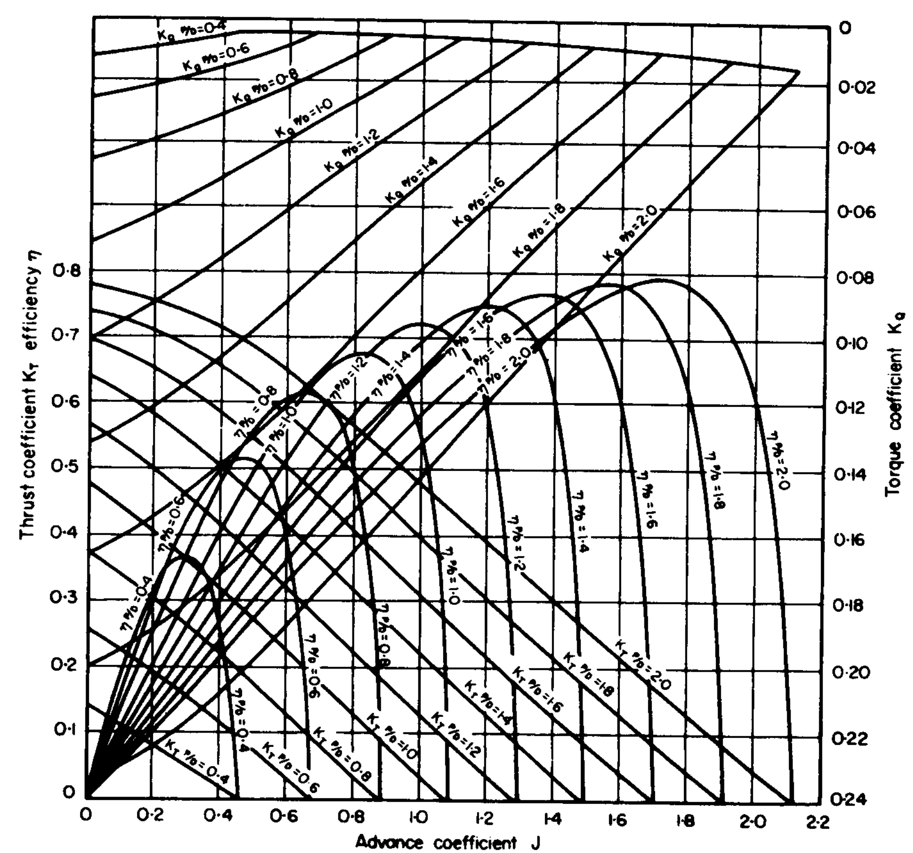


Fig. 11.5 Propeller characteristics, BAR = 0.65

efficiency. This process is described later. Another type of presentation can, however, be adopted to simplify this common type of problem.

This is a plot of B_p against δ where

$$B_p = \frac{nP_D^{\frac{1}{2}}}{V^{2.5}} = 33.08\left(\frac{K_Q}{J^5}\right)^{\frac{1}{2}}$$

$$\delta = 3.2808\frac{nD}{V}$$

where *n* is in r.p.m., *Po* is in the power, *V* is in knots, and *D* is in metres.

Such a plot is presented in Fig. 11.6.

For given values of *n, Po* and *V, Bp* is fixed, and by drawing a vertical ordinate at this value on the figure the maximum obtainable 'fl and corresponding propeller diameter can be determined. In fact, the curve for the optimum efficiency for the most favourable diameter can be plotted on the figure. It connects the points on the 'fl = constant curves at which these curves are vertical, i.e. *Bp* = constant. This line is shown dotted on Fig. 11.6. If the diameter is limited in some ways, the optimum within this limitation is readily deduced.

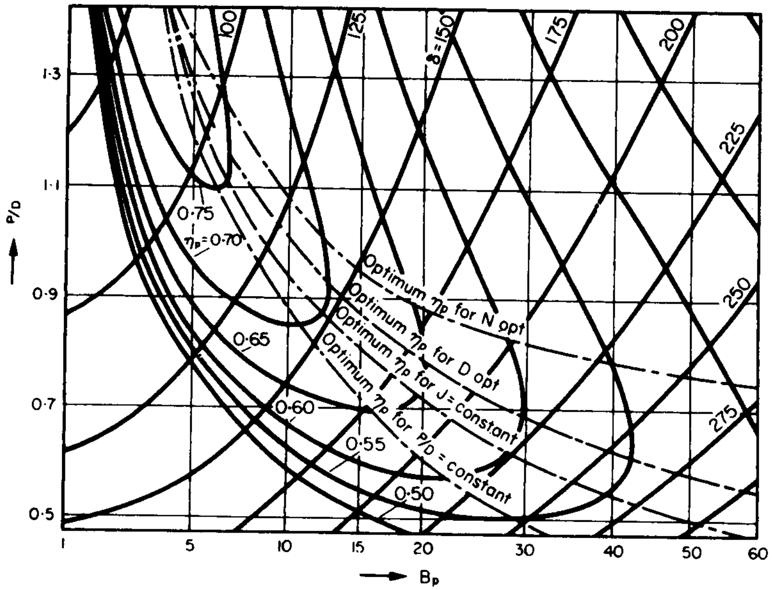


Fig. 11.6 $B_p - \delta$ *diagrams*

When cavitation occurs, as may be the case of a ship at sea or with a propeller in a cavitation tunnel, the η and K_T values are modified as shown in Fig. 11.7.

The more advanced propeller theories which were touched upon in the previous chapter give the designer the ability to design propellers suited to the specific ship wakes in which they are to operate, or to give reduced levels of noise and propeller-induced vibrations. They call for better knowledge of the flow into the propeller so that wake measurements assume greater importance.

Power estimation

RESISTANCE PREDICTION

Froude's law of comparison is the key to most forms of extrapolation from model to ship. By this law, the residuary resistance per unit of displacement is the same for model and ship at corresponding speeds. It remains then necessary to know how the frictional resistance varies with Reynolds' number to enable a plot such as Fig. 11.8 to be produced. Let AA′ represent the variation of total resistance of the model with Reynolds' number. Then, provided the skin friction line is a correct one, $\overline{A_1A_2}$ and $\overline{A_2A_3}$ are the residuary and skin friction components at a Reynolds' number $(R_n)_m$. By Froude's law of comparison, if $(R_n)_S$ is the corresponding ship R_n, $\overline{A_1A_2}$ will be equal to $\overline{B_1B_2}$. Thus, the total ship resistance curve can be obtained by drawing curves through points on the model curve parallel to the skin friction line to intersect vertical lines through the R_n values appropriate to the corresponding speeds.

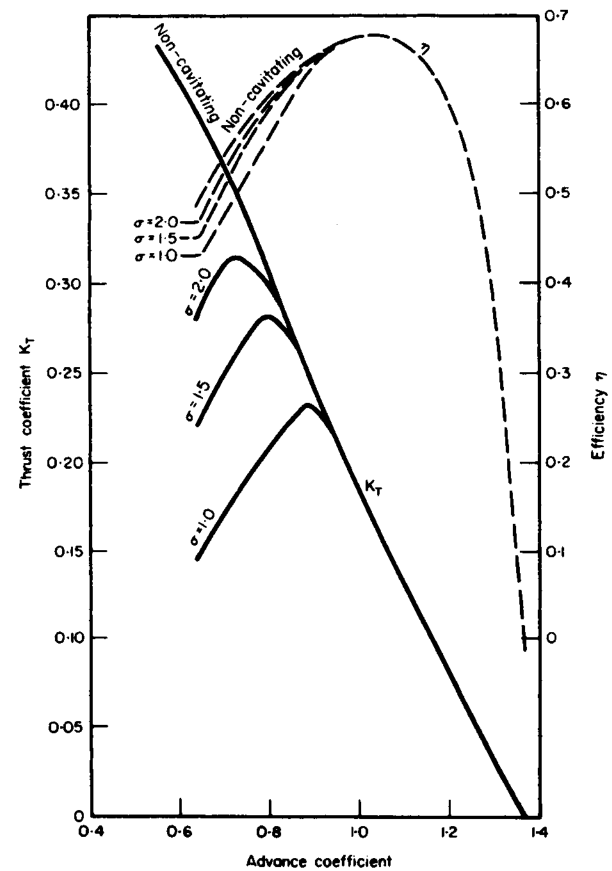


Fig. 11.7 Effect of cavitation on K_T *and* η

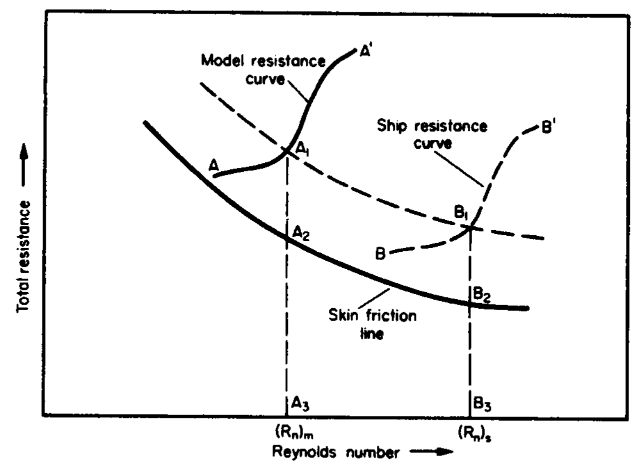


Fig. 11.8 Extrapolation from model to full-scale resistance

Clearly, the accuracy of prediction of such a method is dependent on the accuracy of the curve defining the variation of skin friction resistance with Reynolds' number. Not unnaturally, several curves have been proposed over the years each having its particular advocates. The ship resistance prediction resulting from the different skin friction curves are but marginally different.

The ITTC 1957 model-ship correlation line

Obviously it is desirable to have a single line for correlating ship and model results which is accepted by all practitioners and can be used as a 'standard'. Much effort has been devoted over the years to trying to reach agreement on such a standard line as the volumes of the Royal Institution of Naval Architects bear eloquent witness. One way of studying the relative merits of different formulations is to carry out tests on a series of models of various sizes so that a range of Reynolds' numbers is covered. By presenting all the results in a single plotting such as Fig. 11.8, the shape of the skin friction line is determined by passing curves through points on each model curve at corresponding Reynolds' numbers.

Successive International Towing Tank Conferences studied this problem and in 1957, in Madrid, agreed to a standard line. This ITTC line is defined by

$$C_F = \frac{0.075}{(\log R_n - 2)^2} = \frac{R_F}{\frac{1}{2}\rho S V^2}$$

Values of C_F for various values of R_n are given in Table 11.2.

The term 'correlation line' was used quite deliberately in recognition of the fact that the extrapolation from model to ship is not governed only by variation in skin friction.

APPENDAGE RESISTANCE

A ship has a number of appendages each of which will have associated Froude and Reynolds' numbers depending upon its characteristic length. Thus in going from the model to full scale the resistance of each appendage will scale differently from that of the main hull. Whilst an appended model can be (and is by some authorities) run and scaled as a whole this introduces an approximation which must be allowed for in some overall correlation factor. Many authorities prefer to calculate the resistance of appendages such as bilge keels, stabilizers, rudders, shaft brackets and so on. Provided a consistent method of calculation is adopted for a series of designs any errors in absolute values will be taken care of by correlation factors deduced from comparison of ship trial data with model tests.

(a) Bilge keels

Because care is taken to align the bilge keels with the flow around the hull the resistance of the keels may be taken as the skin frictional resistance of the total wetted surface, based on the characteristic length of the keels. Allowance must

Table 11.2
Coefficients for ITTC 1957 model–ship correlation line. Coefficients must be multiplied by 10^{-3}

Reynolds' number	$10^5\times$	$10^6\times$	$10^7\times$	$10^8\times$	$10^9\times$	$10^{10}\times$
1.0	8.333	4.688	3.000	2.083	1.531	1.172
1.5	7.435	4.301	2.799	1.966	1.456	1.122
2.0	6.883	4.054	2.669	1.889	1.407	1.088
2.5	6.496	3.878	2.574	1.832	1.370	1.063
3.0	6.203	3.742	2.500	1.788	1.342	1.044
3.5	5.971	3.632	2.440	1.751	1.318	1.027
4.0	5.780	3.541	2.390	1.721	1.298	1.014
4.5	5.620	3.464	2.347	1.694	1.280	1.002
5.0	5.482	3.397	2.309	1.671	1.265	0.991
5.5	5.361	3.338	2.276	1.651	1.252	0.982
6.0	5.254	3.285	2.246	1.632	1.240	0.973
6.5	5.159	3.238	2.220	1.616	1.229	0.966
7.0	5.073	3.195	2.195	1.601	1.219	0.959
7.5	4.995	3.156	2.173	1.587	1.209	0.952
8.0	4.923	3.120	2.152	1.574	1.201	0.946
8.5	4.857	3.087	2.133	1.562	1.193	0.941
9.0	4.797	3.056	2.115	1.551	1.185	0.935
9.5	4.740	3.027	2.099	1.540	1.178	0.931

be made for the fact that part of the hull is shielded from the water flow. Possible refinements are to allow for the actual mean flow velocity over the keels and for an interference drag arising from the junction between the hull and keel. An allowance for fouling can be made in line with that for the hull itself.

(b) Rudders, stabilizer fins and shaft bracket arms

These are all aerofoil type sections and drag can be deduced from the characteristics of the aerofoil section adopted. The velocity assumed can be taken as that from model flow experiments although some augmentation of the velocity used for rudders is usual (typically 10 per cent increase) to allow for the propulsor influence. An allowance for fouling can be made based on the surface area of the appendage. Shaft bracket arms will experience interference effects where they enter the hull and where they join the barrel. The resistance of the barrel itself will depend upon the projected area presented to local flow. Usually there is a significant cross-flow velocity at the shaft brackets because of the local hull shape and the need to provide good propeller/hull clearance.

(c) Large inlets

Typical of these are main condenser inlets and the resistance of these will depend upon whether a circulating pump is used. Essentially the resistance due to flow through the system is deduced from the momentum changes as the water enters, transits and then leaves the system.

1978 ITTC PERFORMANCE PREDICTION METHOD

The prediction method proposed by the ITTC in 1978 for single-screw ships follows closely the general analysis presented in Chapter 10. The steps are:

(*a*) the viscous resistance is taken as $(1+k)$ times the frictional resistance where k is determined from the model test and assumed independent of speed and scale;

(*b*) a roughness allowance is calculated from

$$\Delta C_F = \left[105\left(\frac{k_s}{L}\right)^{\frac{1}{3}} - 0.64\right] \times 10^{-3}$$

where k_s = roughness of hull = 150×10^{-6} m
L = length of waterline

(*c*) air resistance is calculated from

$$C_{AAS} = 0.001\frac{A_T}{S}$$

A_T = transverse projected area of ship above the waterline

(*d*) Taylor wake factor, w_T, and thrust deduction factor, t, are used, the latter being assumed the same in the ship as model, i.e. $t_S = t_M$;

(*e*) the thrust deduction factor, $t(= t_M = t_S)$ is obtained from the difference between the self propulsion thrust and the hull resistance without propeller, corrected if necessary for temperature differences at the time of the separate tests;

(*f*) the wake fraction is calculated from the self-propulsion tests and model propeller characteristics. From the thrust, T, and torque, Q, measured in the former

$$K_{TM} = \frac{T}{\rho D^4 n^2}, \quad K_{QM} = \frac{Q}{\rho D^5 n^2}$$

The model propeller characteristics give J_{TM} and K_{QTM} for the K_{TM} value. Hence

$$w_{TM} = \frac{V - V_1}{V} = 1 - \frac{J_{TM}Dn}{V}$$

(*g*) the full-scale wake is taken as

$$w_{TS} = (t + 0.04) + (w_{TM} - t - 0.04)\frac{C_{VS}}{C_{VM}}$$

where the value 0.04 is introduced to take account of rudder effects;

(*h*) the relative rotative efficiency is assumed the same for the ship as model

$$\eta_{RS} = \eta_{RM} = \eta_R = \frac{K_{QTM}}{K_{QM}}$$

(*i*) the total ship resistance coefficient without bilge keels is given by

$$C_{TS} = (1+k)C_{FS} + C_R + \Delta C_F + C_{AAS}$$

where

C_{FS} = frictional coefficient of ship according to the ITTC 1957 ship model correlation line

C_R = residual resistance calculated from the total and viscous resistance of the model
$= C_{TM} - (1+k)C_{FM}$

(*j*) bilge keels can be allowed for by multiplying the C_{FS} and ΔC_F terms by the ratio

$$\frac{S + S_{BK}}{S}, \quad S_{BK} = \text{surface area of the bilge keels}$$

(*k*) scale effect corrections are applied to the propeller characteristics as follows:

$$K_{TS} = K_{TM} - \Delta K_T$$
$$K_{QS} = K_{QM} - \Delta K_Q$$

where $\Delta K_T = -\Delta C_D\left(0.3\frac{P}{D}\right)\frac{cZ}{D}$

$$\Delta K_Q = \Delta C_D(0.25)\frac{cZ}{D}$$

ΔC_D = difference in drag coefficient = $C_{DM} - C_{DS}$

$$C_{DM} = 2\left(1 + \frac{2t}{c}\right)\left[\frac{0.044}{(R_{nco})^{1/6}} - \frac{5}{(R_{nco})^{2/3}}\right]$$

$$C_{DS} = 2\left(1 + \frac{2t}{c}\right)\left[1.89 + 1.62\log\frac{c}{k_p}\right]^{-2.5}$$

R_{nco} = local Reynolds' number at radius = 0.75 maximum (this value not to be less than 2×10^5 in open water test)
k_p = blade roughness = 30×10^{-6} m
c = chord length
t = maximum blade thickness
$\frac{P}{D}$ = pitch ratio;

(*l*) the load of the full-scale propeller is obtained from

$$\frac{K_{TS}}{J^2} = \frac{S}{2D^2} \times \frac{C_{TS}}{(1-t)(l - w_{TS})^2}$$

From this value J_{TS} and K_{QS} follow from the full-scale propeller characteristics. From these it follows that

$$\text{full-scale revs} = n_S = \frac{(l - w_{TS})V_S}{J_{TS}D} \text{(r.p.s.)}$$

$$\text{delivered power} = P_{DS} = 2n\rho D^5 n_S^3 \frac{K_{QTS}}{\eta_R} \times 10^{-3} \text{ (kW)}$$

$$\text{propeller thrust} = T_s = \frac{K_T}{J^2} \times J_{TS}^2 \rho D^4 n_S^2 \text{ (N)}$$

$$\text{propeller torque} = Q_s = \frac{K_{QTS}}{\eta_R} \rho D^5 n_S^2 \text{ (Nm)}$$

$$\text{effective power} = P_E = C_{TS} \times \tfrac{1}{2}\rho V_S^3 \times S \times 10^{-3} \text{ (kW)}$$

$$\text{total efficiency} = \eta_D = \frac{P_{DS}}{P_E}$$

$$\text{hull efficiency} = \eta_H = \frac{1 - t}{1 - w_{TS}}$$

EFFECT OF SMALL CHANGES OF DIMENSIONS

Froude's formula for frictional resistance may be written

$$R_F = f\Delta^{\frac{2}{3}}V^{1.825}$$

For geometrically similar ships at corresponding speeds

$$V \propto L^{\frac{1}{2}}; \ \Delta \propto L^3$$

Hence

$$R_F \propto fL^{2.9125}$$

By Froude's law of comparison the residuary resistance varies as L^3. Hence, for small changes in dimensions no large error is introduced if it is assumed that the total resistance varies in the same way, i.e.

$$R_T \propto L^3$$

Variation in residuary resistance with size and speed

At a given speed for any condition, the residuary resistance will vary with displacement and speed as follows:

$$R_R = K\Delta^m V^{n-1}$$

The power then varies as $\Delta^m V^n$. For a geometrically similar form at corresponding speed

$$R'_R = K(\Delta')^m(V')^{n-1}$$

But by Froude's law of comparison

$$\frac{R_R}{R'_R} = \frac{\Delta}{\Delta'}$$

and

$$\frac{V}{V'} = \left(\frac{L}{L'}\right)^{\frac{1}{2}} = \left(\frac{\Delta}{\Delta'}\right)^{\frac{1}{6}}$$

Hence

$$\frac{\Delta}{\Delta'} = \left(\frac{\Delta}{\Delta'}\right)^m \left(\frac{\Delta}{\Delta'}\right)^{(n-1)/6}$$

i.e.

$$m + \frac{n-1}{6} = 1$$

$$6m + n = 7$$

The value of n can be deduced from the slope of the resistance/speed curve as follows:

$$C_R = \text{Const. } \frac{R}{V^2}$$

Hence

$$C_R = \text{Const. } \frac{V^{n-1}}{V^2} = \text{Const. } V^{n-3}$$

Differentiating,

$$\frac{\partial C_R}{\partial V} = (n-3) \text{ Const. } V^{n-4}$$

Eliminating the constant in these equations

$$\frac{\partial C_R}{\partial V} = (n-3)\frac{C_R}{V}$$

Hence

$$n = 3 + \frac{V}{C_R} \cdot \frac{\partial C_R}{\partial V}$$

The value of m follows from

$$m = \frac{7-n}{6}$$

It may be necessary to depart from the form in the early design stages in a way that destroys geometric similarity. Beam may have to be increased to improve initial stability, length may have to be increased to provide an acceptable weather deck layout, and so on. Mathematically the variation of power with length, beam and draught can be expressed in terms of performance coefficients α, β and γ as

$$\frac{\partial P_E}{P_E} = \alpha\frac{\partial L}{L} + \beta\frac{\partial B}{B} + \gamma\frac{\partial T}{T}$$

The values of the performance coefficients are obtained from tests with three models—known as *triplets*. One is the parent form, one has beam changed by a small percentage and one has modified length. The change in dimension is typically 10 per cent and is a simple linear stretch in the relevant direction.

VARIATION OF SKIN FRICTIONAL RESISTANCE WITH TIME OUT OF DOCK

When a ship enters the water having been freshly cleaned and painted its resistance is a minimum. With the passage of time, seaweed and barnacles attach themselves to the surface so presenting a rougher surface to the passage of water. This roughening of the surface leads to an increase in the skin frictional resistance. It is to be expected that the amount of fouling as it is called will depend upon the area in which the ship is operating and the time spent at sea compared with the time at rest in harbour.

The MOD used to assume an increase in skin friction resistance of a quarter of one per cent per day and took as standard a 'deep and dirty' condition with the ship at deep displacement and six months out of dock. Then:

$$\text{Increase in } Ⓒ_F = \delta Ⓒ_F = \frac{365}{200} \times 0.25 Ⓒ_F = 0.456 Ⓒ_F$$

Other authorities assume a standard percentage of the available power, e.g. 20 per cent, is used up in overcoming the increased resistance due to fouling and also in overcoming the extra resistance due to running through waves. Allowances for fouling can be less with modern protection systems and more recent practice is to allow a 15 per cent margin on endurance power. The corresponding figure for the USN is 10 per cent but they use full load displacement in calculating endurance whereas the RN uses an average displacement which reflects the fact that the ship gets lighter as fuel is consumed.

EXAMPLE 1. Data for a model 5 m long of a ship 178 m long and 11,700 tonnef displacement, corrected to standard temperature, is defined by the following table.

$R_n \times 10^6$	6.600	6.894	7.188	7.522	7.817	8.111	8.445
$C_T \times 10^3$	3.944	3.944	3.951	3.972	3.979	3.982	4.014

$R_n \times 10^{-6}$	8.740	8.995	9.295	9.624	9.918	10.213
$C_T \times 10^3$	4.024	4.034	4.090	4.164	4.310	4.478

Deduce a plot, power against speed for the clean and dirty conditions assuming that the wetted surface area is 3650 m^2.

Solution: The data having been presented in the form of C_T against R_n, the 1957 ITTC analysis is applied

Reynolds' number, $R_n = \dfrac{VL}{\nu}$

The standard values (i.e. at 15 °C) for ν are

$$\nu = 1.139 \times 10^{-6}\,\text{m}^2/\text{s for fresh water}$$

$$\nu = 1.188 \times 10^{-6}\,\text{m}^2/\text{s for sea water}$$

If V is the speed of the ship, the corresponding speed for the 5 m model is $V\sqrt{(5/178)}$. Hence for the ship

$$R_n = \frac{1.139}{1.188}\left(\frac{178}{5}\right)^{\frac{3}{2}}(R_n)_{\text{model}}$$
$$= 203.7(R_n)_{\text{model}}$$

The ship speed

$$1.188 \times 10^{-6} R_n/L = 6.674 \times 10^{-9}(R_n)_{\text{ship}}\ m/s.$$

The C_T values for the model have to be corrected for the skin friction difference in going from model to ship. This correction is the difference between the ordinates of the ITTC line at the Reynolds' numbers appropriate to the model and the ship. Hence, the total resistance C_T can be deduced for the ship. Since

$$C_T = \frac{\text{resistance}}{\frac{1}{2}\rho S V^2}$$

$$\text{power} = R \times V = \tfrac{1}{2}\rho S V^3 C_T = 1.872 V^3 C_T \text{ MW}, \quad V \text{ in m/s.}$$

The increase in resistance in the dirty condition is given by

$$\delta C_F = 0.456 C_F \text{ using the ship value}$$

The calculation can now be completed in tabular form as shown in Table 11.3. The results are plotted in Fig. 11.9.

RESISTANCE IN SHALLOW WATER

Chapter 9 concentrated on waves in deep water. As can be found in any standard textbook on hydrodynamics, in water of depth, h, the speed of propagation of a wave is given by:

$$c^2/gh = \lambda/2\pi h[\tanh(2\pi h/\lambda)]$$

In deep water $[\tanh(2\pi h/\lambda)]$ tends to unity giving $c^2 = g\lambda/2\pi$. When h/λ is small $[\tanh(2\pi h/\lambda)]$ tends to $2\pi h/\lambda$ and $c^2 = gh$. This is constant for any given depth, that is it does not depend upon the wavelength. $(gh)^{\frac{1}{2}}$ is the maximum speed of a wave in shallow water and is known as the *critical speed*.

Table 11.3
1957 ITTC analysis

$(R_n)_{model}$ $\times 10^6$	$(R_n)_{ship}$ $\times 10^9$	V Ship (m/s)	$C_F \times 10^{-3}$ Model	$C_F \times 10^{-3}$ Ship	SFC $\times 10^{-3}$	$C_T \times 10^{-3}$ Model	Clean ship $C_T \times 10^{-3}$	Clean ship MW	$\delta C_F \times 10^{-3}$	Dirty ship $C_T \times 10^{-3}$	Dirty ship MW
6.600	1.344	8.97	3.229	1.476	1.753	3.944	2.191	1.534	0.673	2.864	2.005
6.894	1.404	9.37	3.204	1.468	1.736	3.944	2.208	1.817	0.669	2.877	2.368
7.188	1.464	9.77	3.180	1.461	1.719	3.951	2.232	2.082	0.666	2.898	2.704
7.522	1.532	10.22	3.154	1.453	1.701	3.972	2.271	2.423	0.663	2.934	3.131
7.817	1.592	10.63	3.132	1.446	1.686	3.979	2.293	2.736	0.659	2.952	3.557
8.111	1.652	11.03	3.112	1.440	1.672	3.982	2.310	3.100	0.657	2.967	3.982
8.445	1.720	11.48	3.090	1.433	1.657	4.014	2.357	3.566	0.653	3.010	4.554
8.740	1.780	11.88	3.071	1.427	1.644	4.024	2.380	3.991	0.651	3.031	5.082
8.995	1.832	12.23	3.056	1.422	1.634	4.034	2.400	4.390	0.648	3.048	5.575
9.295	1.893	12.63	3.038	1.417	1.621	4.090	2.469	4.975	0.646	3.115	6.277
9.624	1.960	13.08	3.020	1.410	1.610	4.164	2.554	5.716	0.643	3.197	7.155
9.918	2.020	13.46	3.004	1.405	1.599	4.310	2.711	6.639	0.641	3.352	8.209
10.213	2.080	13.88	2.990	1.400	1.590	4.478	2.888	7.723	0.638	3.526	9.429

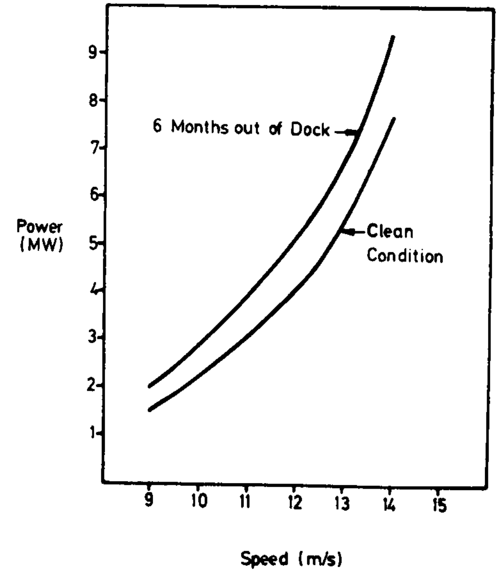


Fig. 11.9 *Plot of results*

When a ship is moving in shallow water the angle of the line of the maximum height of the diverging waves to the centreline increases, approaching 90 degrees as the ship speed approaches the critical speed. At speeds below the critical, the resistance in shallow water is greater than that in deep water and the resistance increases dramatically as the critical speed is neared. If the ship has enough power to exceed the critical speed then the resistance will fall well below that for deep water at the same speed.

CALCULATION OF WIND RESISTANCE

The fair above-water portion of the main hull experiences less resistance force per unit area than the superstructure. In fact, the resistance per unit of projected area is only about 30 per cent of that of the superstructures. Wind tunt: tests carried out by NPL showed that the resistance offered by the ship could e represented by the equation

$$\text{Resistance} = KBV^2$$

where B = projected area onto a transverse plane of the superstructure plus 30 per cent of the projected area of the above water hull, V = relative wind speed (knots), and K is a coefficient depending on the ship type and the angle of the relative wind to the middle-line of the ship.

It was found that K was fairly constant for angles up to about 15 degrees off the bow and was a maximum for an angle of about 30 degrees off the bow.

If B is in m^2 and V in knots then, approximately,

$$\text{Resistance} = 48\,KBV^2 \text{ newtons}$$

One useful parameter for comparing results is the *ahead resistance coefficient* (ARC) defined by

$$\text{ARC} = \frac{\text{fore and aft component of wind resistance}}{\frac{1}{2}\rho V_R^2 A_T}$$

In the case of a tanker, the ARC values were reasonably steady for relative winds from ahead to 50 degrees off the bow, the value varying from 0.7 in the light condition to about 0.85 in the loaded condition. Corresponding values for winds up to 40 degrees off the stern were -0.6and -0.7. Variation with relative wind direction between 50 degrees off the bow and 40 degrees off the stern was approximately linear. Two cargo ships exhibited similar trends but the values of the ARC were about 0.1 lower. The same ARC values apply in the metric system provided consistent units are used.

The results include the effect of the velocity gradient existing in atmospheric winds. (See Chapter 9.) They represent the force experienced by a ship of the size tested (tanker 169m, cargo ships 149m and passenger liner 245 m, length overall). Smaller ships will have a greater percentage of their area in the lower regions of the gradient and will suffer proportionately less force. The force can be assumed to vary as the square of the velocity. If the results are to be used to deduce the air resistance experienced by a ship moving ahead with no wind then there is no velocity gradient and the forces deduced must be increased by 25 per cent in the light condition and 40 per cent in the deep load condition for the tanker and cargo ships. The increase for a passenger ship is about 21 per cent.

Wind tunnel tests have been carried out in Japan to help assess the size of mooring rope and the amount of cable veer required by large ships in a strong wind. The problem has since acquired greater urgency due to the increase in ship size and the limitations of available anchorages.

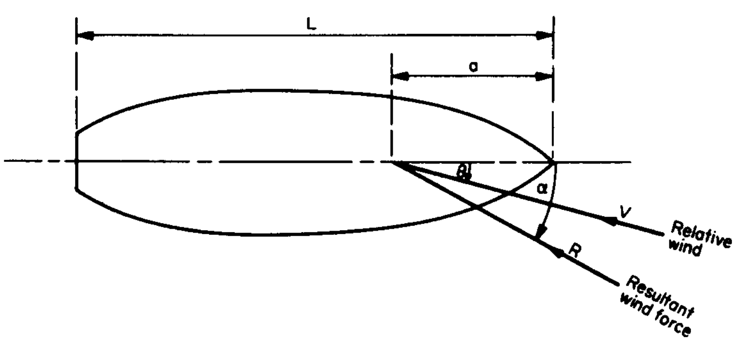


Fig. 11.10 Wind forces on a ship

The Japanese tests showed that the resultant wind force R is given by

$$R = \tfrac{1}{2}\rho C_r V^2 (A\cos^2\theta + B\sin^2\theta), \text{newtons}$$

Where ρ = atmospheric density (kgf m^{-4} s^2) = 0.123 approx., A = frontal projection area (m^2), B = lateral projection area (m^2), and V = relative wind speed (m/s).

Based on results for a number of ships, it is suggested that C_r values are given by the following relationships:

$$\text{Cargo ship: } C_r = 1.325 - 0.05\cos 2\theta - 0.35\cos 4\theta - 0.175\cos 6\theta$$

$$\text{Passenger ship: } C_r = 1.142 - 0.142\cos 2\theta - 0.367\cos 4\theta - 0.133\cos 6\theta$$

$$\text{Oil tanker: } C_r = 1.20 - 0.083\cos 2\theta - 0.25\cos 4\theta - 0.117\cos 6\theta$$

Approximate values of A and B are given by

$$A = B^2[X_A - 0.00475\,dr]. \quad \text{In this case, } B = \text{beam}$$

$$B = L^2[X_B - 0.0006\,dr]$$

where dr = percentage the actual draught is of the fully-loaded draught and values of X_A and X_B are approximately given in Table 11.4.

The distance aft from the bow at which the resultant wind force acts, expressed as a percentage of ship length, varies uniformly with wind directions between 20° and 160° off the bow and is given by

$$\frac{a}{L} = 0.291 + 0.0023\theta, \quad \theta \text{ in degrees}$$

Table 11.4
Values of X_A and X_B

Ship type	X_A	X_B
3 Island cargo-ship	1.43	0.1195
Cargo-ship with stern machinery	1.225	0.110
Oil tanker	1.095	0.099
Passenger ship	1.455	0.156

Generally, the direction of the resultant wind force, though not the same as the wind direction, changes with it as follows:

$$\alpha = \left\{1 - 0.15\left(1 - \frac{\theta}{90}\right) - 0.80\left(1 - \frac{\theta}{90}\right)^3\right\} \times 90$$

The data above were obtained from measurements on models floating on water. They are therefore subject to a certain unspecified wind gradient. If this is assumed to be comparable to that occurring in nature, then the air resistance force acting on the ship when moving ahead in calm air must be increased as proposed above for the NPL data.

Multiple regression analysis has been used to obtain expressions for the fore and aft wind force component, the lateral wind force component and the wind-induced yawing moment. Forty-nine sets of experimental data were used and the equations expressed the force or moment coefficient in the general form

$$a_0 + a_1\frac{2A_L}{L^2} + a_2\frac{2A_T}{B^2} + a_3\frac{L}{B} + a_4\frac{S}{L} + a_5\frac{C}{L} + a_6 M$$

where

L and B are overall length and beam

A_L and A_T are the lateral and transverse projected area

S is the length of perimeter of lateral projection of model excluding the waterline and slender bodies such as masts

C is distance from bow of centroid of lateral projected area

M is number of distinct groups of masts or kingposts seen in lateral projection.

Table 11.5 reproduces the values of a_0 to a_6 for the fore and aft wind force component together with the corresponding standard error.

Table 11.6 gives typical values of the independent variables to enable approximate values of wind resistance to be calculated in the early design stage.

Table 11.5
Fore and aft component of wind force

$$C_X = a_0 + a_1\frac{2A_L}{L_{0A}^2} + a_2\frac{2A_T}{B^2} + a_3\frac{L_{0A}}{B} + a_4\frac{S}{L_{0A}} + a_5\frac{C}{L_{0A}} + a_6 M \pm 1.96\text{ S.E.}$$

γ_R°	a_0	a_1	a_2	a_3	a_4	a_5	a_6	S.E.
0	2.152	−5.00	0.243	−0.164	—	—	—	0.086
10	1.714	−3.33	0.145	−0.121	—	—	—	0.104
20	1.818	−3.97	0.211	−0.143	—	—	0.033	0.096
30	1.965	−4.81	0.243	−0.154	—	—	0.041	0.117
40	2.333	−5.99	0.247	−0.190	—	—	0.042	0.115
50	1.726	−6.54	0.189	−0.173	0.348	—	0.048	0.109
60	0.913	−4.68	—	−0.104	0.482	—	0.052	0.082
70	0.457	−2.88	—	−0.068	0.346	—	0.043	0.077
80	0.341	−0.91	—	−0.031	—	—	0.032	0.090
90	0.355	—	—	—	−0.247	—	0.018	0.094
100	0.601	—	—	—	−0.372	—	−0.020	0.096
110	0.651	1.29	—	—	−0.582	—	−0.031	0.090
120	0.564	2.54	—	—	−0.748	—	−0.024	0.100
130	−0.142	3.58	—	0.047	−0.700	—	−0.028	0.105
140	−0.677	3.64	—	0.069	−0.529	—	−0.032	0.123
150	−0.723	3.14	—	0.064	−0.475	—	−0.032	0.128
160	−2.148	2.56	—	0.081	—	1.27	−0.027	0.123
170	−2.707	3.97	−0.175	0.126	—	1.81	—	0.115
180	−2.529	3.76	−0.174	0.128	—	1.55	—	0.112

Mean Standard Error 0.103

Table 11.6
Values of independent variables

Variable	2AL -L6-A	2AT -B-2	LOA B	S LOA	C LOA	Ass AL	M
Maximum	0.246	2.32	9.75	1.97	0.619	0.595	7
Minimum	0.072	0.88	4.00	1.23	0.401	0.138	1
Mean	0.143	1.78	7.39	1.51	0.506	0.246	4
Ship type							
1	0.192	1.95	7.66	1.44	0.492	0.398	2
2	0.111	1.67	7.80	1.51	0.490	0.258	4
3	0.149	2.04	7.80	1.58	0.489	0.188	4
4	0.122	1.75	7.80	1.51	0.550	0.253	5
5	0.151	2.06	7.80	1.58	0.526	0.175	5
6	0.076	1.03	7.46	1.33	0.547	0.252	3
7	0.117	1.43	7.46	1.40	0.522	0.161	3
8	0.100	1.59	7.46	1.33	0.568	0.211	3
9	0.121	1.68	7.46	1.40	0.537	0.139	3
10	0.166	1.80	6.47	1.45	0.476	0.229	2
11	0.236	1.43	4.05	1.86	0.405	0.396	1

PROPELLER DESIGN

It is possible in this book only to outline the main factors to be considered by the propeller designer. They are:

(a) Shaft revolutions. Apart from the direct influence on propeller efficiency the choice of shaft r.p.m. depends upon the gearing available, critical whirling speeds of shafts and avoidance of the fundamental frequencies of hull vibration;
(b) Number of blades which influences vibration (Fig. 11.11) and cavitation;
(c) Propeller diameter and hence clearance between propeller tips and the hull which has a marked effect on vibration;
(d) Blade area. The greater the blade area for a given thrust the less likely is cavitation;
(e) Boss diameter. Dictated mainly by strength considerations;
(j) Geometry of the blades, e.g. pitch, camber;
(g) The wake in which the propeller is to operate.

The choice of the principal propeller dimensions is an easier problem and is considered below.

CHOICE OF PROPELLER DIMENSIONS

The propeller dimensions are found using methodical series propeller data. This is adequate if the propeller is to be similar in geometry to those forming the methodical series. If it is to differ in some significant way, the dimensions are first approximated to by using the series data and then adjusted as a result of special tests on the propeller so obtained.

It is assumed that the designer has information on P_E, hull efficiency, **wake,** QPC factor and (j)R/(j)M as a result of model experiments and previous **ship**

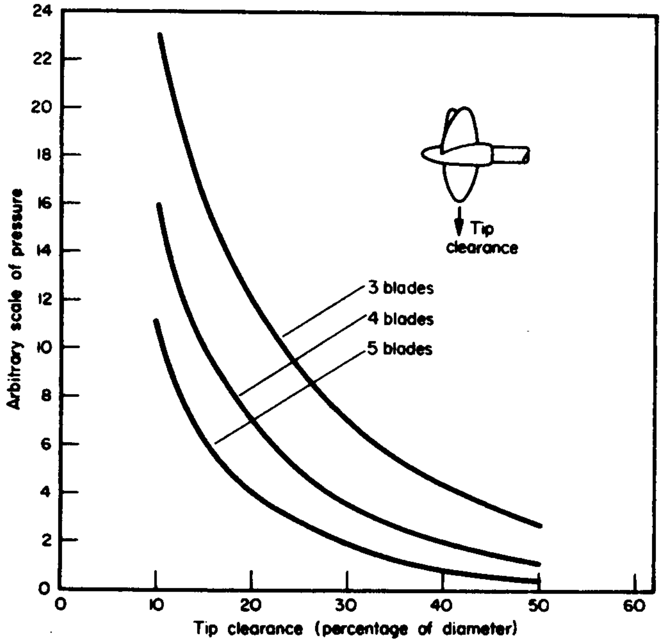


Fig. 11.11 Variation of harmonic pressure with tip clearance

trials. The speed will be stated in the requirements for the ship. The steps in the design process are now:

(*a*) thrust power in open water $= TV_1 = P_T \times \dfrac{(T)_R}{(T)_M}$

from which thrust T can be deduced;

the designer assesses an allowable pressure on the blades if serious cavitation effects are to be avoided. In the absence of other data, it is recommended that a figure of $80\,\text{kN/m}^2$ be used. From this, the blade area to provide the thrust T, and hence BAR can be obtained in terms of the diameter of the propeller;

(*c*) for values of BAR, for which methodical series data are available, deduce the propeller diameter D. We can now illustrate the method to be used by reference to Fig. 11.5 for a BAR of 0.65. The procedure has to be repeated for each BAR under consideration;

(*d*) for a series of values of J, calculate the corresponding r.p.m., n;

(*e*) for each n value calculate K_T and by cross plotting find the corresponding P/D ratio and η value;

(*f*) plot n, P and η to a base of J and note the values of n and P corresponding to the maximum η. Note this η value;

(*g*) repeat for a number of BARs and plot n, P and η to base D;

(*h*) read off the values at the optimum value of D. This will either be that for maximum efficiency or, if this diameter is too great, the diameter must be restricted to give a satisfactory tip clearance from the vibration point of view. The propeller diameter may also be limited by consideration of possible damage when the ship is docked or is coming alongside a jetty;

(*i*) the propulsive coefficient can then be deduced using the relationship

$$\text{PC} = \left[\frac{\eta_H \times \eta_0 \times \eta_R}{\text{appendage coefficient}}\right] \times \text{QPC factor, relative to } P_E$$

(*j*) calculate shaft power $= P_E/\text{PC}$

EXAMPLE 2. A 174 m long twin-screw ship of 11,684 tonnef similar to that in the example for calculation of P_E had a P_E of 23.49 MW at 28 knots in the deep and dirty condition. Model data for this design suggest the following:

Hull efficiency	= 0.98
Relative rotative efficiency	= 1.00
Wake	= 10 per cent
Appendage coefficient	= 1.06

Trial data from a similar ship suggest that with a pressure coefficient of $80\,\text{KN/m}^2$, the QPC factor is 0.92 and $(T)_R/(T)_M$ is 1.04.

Determine the dimensions and revolutions of the propeller to give maximum efficiency assuming that the maximum diameter from the point of view of docking is 4.27 m. Calculate also the propulsive coefficient at 28 knots and power required at this speed.

Solution: Following the procedure outlined above

P_E per screw = 11.74 MW

$$P_T \text{ per screw} = \frac{P_{EA}}{\text{HE}} \times \frac{(T)_R}{(T)_M} = \frac{11.74}{0.98} \times 1.04 \times 1.06 = 13.21\,\text{MW}$$

$$V_1 = \frac{28}{1.10} = 25.45\text{ knots} = 13.1\,\text{m/s}$$

Since $P_T = (\text{thrust}) \times V_1$

$$\text{Thrust per screw: } \frac{13.21}{13.1} = 1.01\,\text{MN}$$

Area of blades to restrict pressure loading to $80\,\text{kN/m}^2$

$$= \frac{1.01}{80} \times 10^3 = 12.6\,\text{m}^2$$

Hence

$$\text{BAR} = \frac{4A_D}{\pi D^2} = \frac{12.6 \times 4}{\pi D^2} = \frac{16.04}{D^2}$$

Gawn (1953) published propeller data for BARs of 0.2, 0.35, 0.5, 0.65, 0.80, 0.95 and 1.1. Corresponding values of *Dare:* 8.94, 6.75, 5.65, 4.96, 4.47, 4.10, 3.81 m.

Because of the limitation imposed on the propeller diameter, we need only consider the last three values of BAR the value 0.80 being used merely to define the trends.

$$J = \frac{V_1}{nD} = \frac{13.1}{nD} = \text{in m, s units}$$

Hence

$$n = \frac{786}{DJ}\ \text{r.p.m.}$$

Table 11.7 can now be constructed.

For each value of D and n, K_T can be calculated from

$$K_T = \frac{T}{(\rho/g)n^2D^4} = \frac{1.01\times10^6}{1025n^2D^4} = \frac{976}{n^2D^4},\ n \text{ in r.p.s.}$$

The calculation can be simplified somewhat by introducing in this expression for K_T the relationship

$$J = \frac{V_1}{nD}$$

$$K_T = \frac{976}{(V_1)^2}\frac{(J)^2(D)^2}{D^4} = \frac{976J^2}{V_1^2D^2}$$

$$= \frac{976}{(13.1)^2}\cdot\frac{J^2}{D^2} = 5.69\frac{J^2}{D^2}$$

$$= 3.64\times\frac{1}{D^2}\ \text{for } J = 0.8$$

$$= 5.69\times\frac{1}{D^2}\ \text{for } J = 1.0$$

$$= 8.19\times\frac{1}{D^2}\ \text{for } J = 1.2$$

By cross plotting the Gawn data, the values of *P/D* and 'fl can be obtained for each value of *KT*, *J* and BAR. In order to obtain a reliable value of 'fl, it is recommended that a line be drawn parallel to the base line at the appropriate *KT* value, noting the *J* value appropriate to each *P/D* and the corresponding 'fl. Plotting 'fl against *P/D* the 'fl value required can be obtained knowing *P/D*.

Now by plotting for each BAR the values of *n*, *P/D* and 'fl to a base of *J* the values of *P/D* (and hence *P*) and *n* corresponding to maximum 'fl can be read off. These results together with the 'fl value are now plotted to a base of *D*.

In this case, the limitation on propeller diameter is the overriding factor, since efficiency is still increasing as the diameter exceeds the 4.27 m value. Corresponding to the 4.27 m diameter

Table 11.7
Propeller calculation

BAR	D	$\frac{786}{D}$	J	n	K_T	P/D	η	Values for η_{max}: n	P/D	η	P
			0.8	219.9	0.183	1.11	0.658				
0.80	4.47	175.9	1.0	175.9	0.286	1.50	0.665	200	1.24	0.676	18.2
			1.2	146.6	0.410	1.96	0.638				
			0.8	239.5	0.217	1.15	0.648				
0.95	4.10	191.6	1.0	191.6	0.338	1.56	0.640	228	1.24	0.650	16.7
			1.2	159.7	0.487	2.03	0.610				
			0.8	257.8	0.250	1.16	0.615				
1.1	3.81	206.2	1.0	206.2	0.392	1.59	0.610	247	1.24	0.624	15.5
			1.2	171.8	0.563	2.06	0.585				

$$\eta = 0.66$$
$$n = 217\ \text{r.p.m.}$$
$$P = 5.21\ \text{m}$$
$$\text{BAR} = \frac{15.9}{D^2} = \frac{15.9}{(4.27)^2} = 0.872$$

Now

$$\text{Propulsive coefficient} = \text{PC} = \left[\frac{\eta_H\times\eta_0\times\eta_R}{\text{appendage coeff}}\right]\times\text{QPC factor}$$

$$= \left[\frac{0.98\times0.66\times1.00}{1.06}\right]0.92 = 0.562$$

$$\text{Shaft power} = \frac{23.49}{\text{PC}} = \frac{23.49}{0.562} = 41.8\ \text{MW}$$

This is the power which must be developed by the machinery.

PROPELLER DESIGN DIAGRAM

The example above shows that the amount of work involved in designing this one propeller is fairly lengthy, particularly when it is realized that the various cross-plottings necessary in the solution have not been reproduced.

A shorter analysis is possible if use is made of a 3-bladed propeller design diagram based directly on the data published by Gawn. This diagram, developed by the Admiralty Experiment Works is reproduced as Fig. 11.12. It makes use of the parameter K_T/J^2 which depends only on thrust, propeller diameter and speed of advance, since:

$$\frac{K_T}{J^2} = \frac{T}{\rho n^2D^4}\Big/\frac{V_1^2}{n^2D^2} = \frac{T}{\rho D^2V_1^2}$$

Thus one unknown, n, has been removed. By calculating the value of K_T/J^2 and drawing a line across the diagram at this level the values of maximum efficiency, J, and P/D can be read off directly at the appropriate BAR value.

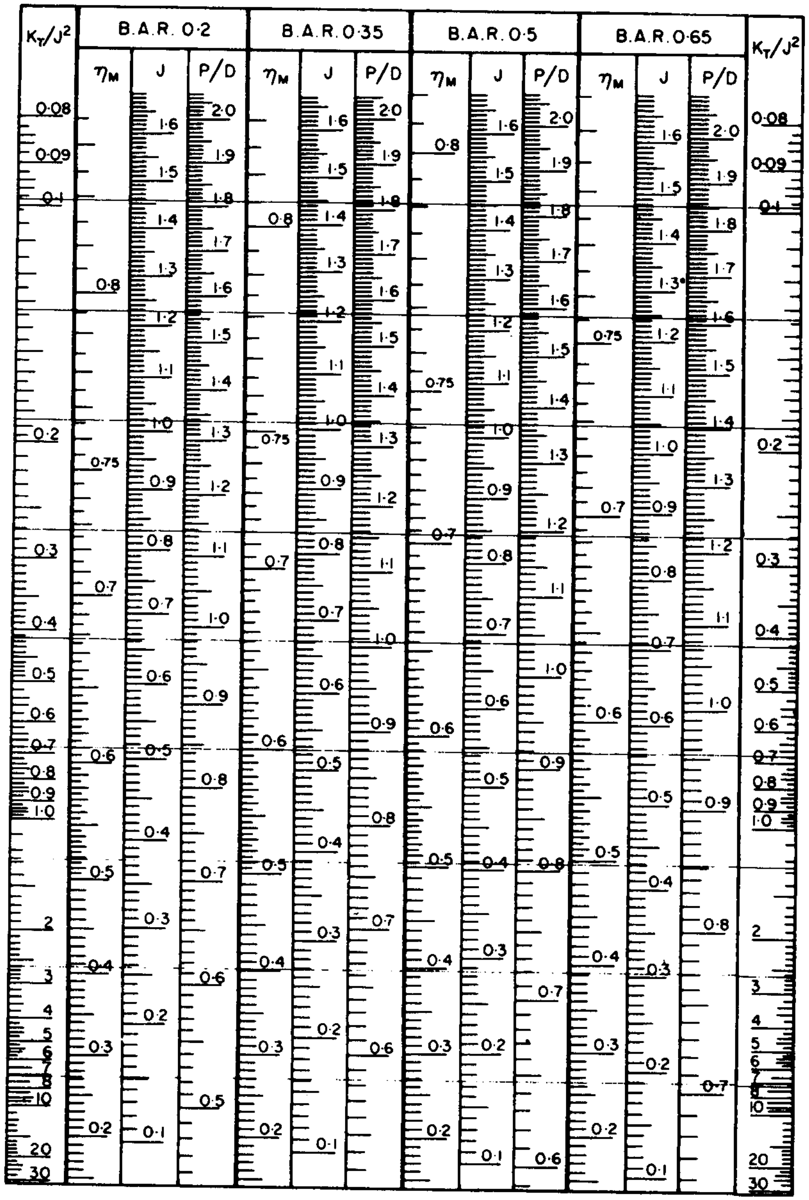


Fig. 11.12 Preliminary propeller design diagram

Fig. 11.12 continued

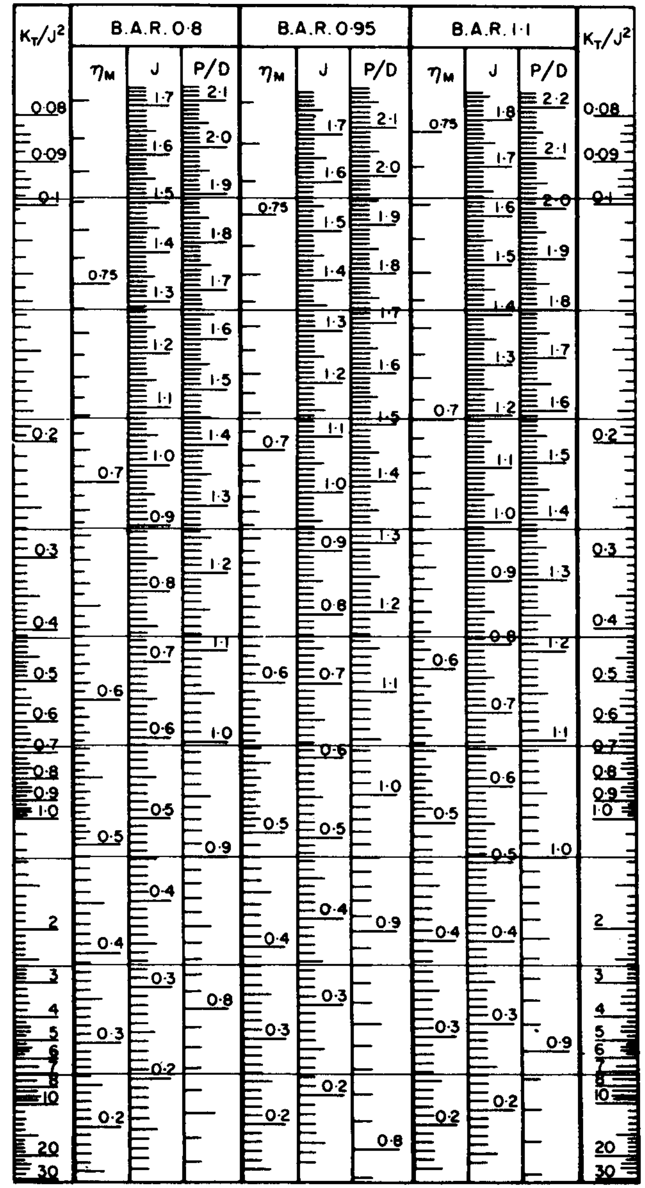


Diagram gives the maximum peller efficiency, η_M, obtainable any known value of K_T/J^2 corresponding values of advance efficient J and pitch ratio

$$\frac{K_T}{J^2} = \frac{36.13}{D^2V^2}T$$

$$J = \frac{30.88}{nD}$$

where T = thrust in tonnef
D and P are in metres
n is in r.p.m.
V is in knots

EXAMPLE 3. Use the propeller design diagram to design the propeller in the previous example.

Solution: In the previous example, it was shown that values of D corresponding to BARs of 0.80, 0.95 and 1.1 were 4.47, 4.10 and 3.81 m. Other data calculated in that example were:

Thrust per screw = 1.01 MN

$V_1 = 13.1$ m/s

Hence

$$\frac{K_T}{J^2} = \frac{T}{\rho D^2 V_1^2}$$

$$= \frac{1.01}{1025} \times \frac{10^6}{(13.1)^2} \times \frac{1}{D^2} = \frac{5.74}{D^2}$$

A table can now be constructed as in Table 11.8.

Table 11.8
Calculation for worked example using propeller design diagram

BAR	0.80	0.95	1.1
D, m	4.47	4.10	3.81
$\frac{K_T}{J^2}$	0.287	0.341	0.395
η_m	0.676	0.650	0.624
J	0.872	0.842	0.830
P/D	1.24	1.237	1.239
P	5.54	5.07	4.72
DJ	3.90	3.45	3.16
r.p.m.	202	228	249

In Table 11.8, the values of η_m, J and P/D are read directly from the propeller design diagram.

The propeller revolutions, n, can be obtained from

$$n = \frac{V_1 \times 30.88}{DJ} = \frac{786}{DJ}$$

Values of p, n and η_m can be plotted against diameter as before, showing that it is the limit on diameter at 4.27 m which is the governing factor and giving a propeller of the following characteristics

$D = 4.27$ m   PC for ship = 0.565

$P = 5.21$ m $P_S = 41.8$ MW

$n = 217$ BAR = 0.878

$\eta = 0.664$

Although it is not in general use an interesting form of propeller design chart is presented in Figs 11.13 to 11.15. Figure 11.13 uses vertical and horizontal

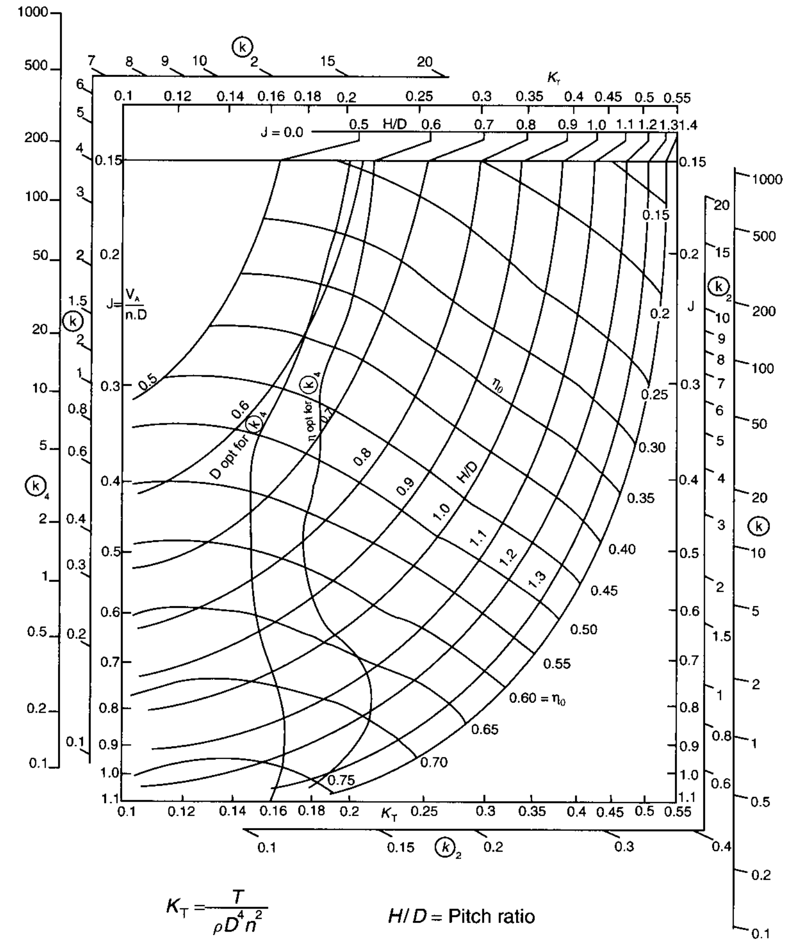


Fig. 11.13 Design diagram for given thrust power

axes of log J and log K_T. The results of any systematic series can be plotted in the form of curves of constant efficiency and pitch diameter ratio. Those shown are for a Wageningen series 4-bladed screw with a blade area ratio of 0.55. The circular notation used should not be confused with the Froude notation described earlier.

With this type of plot, if

$$(K)_n = \frac{K_T}{J^n}$$

$$\log (K)_n = \log K_T - n \log J$$

Thus plots of constant $(K)_n$ will be straight lines with a slope defined by n. The lines for values of $(K)_2$ and $(K)_4$ are represented by short inclined lines around the edges of the diagram.

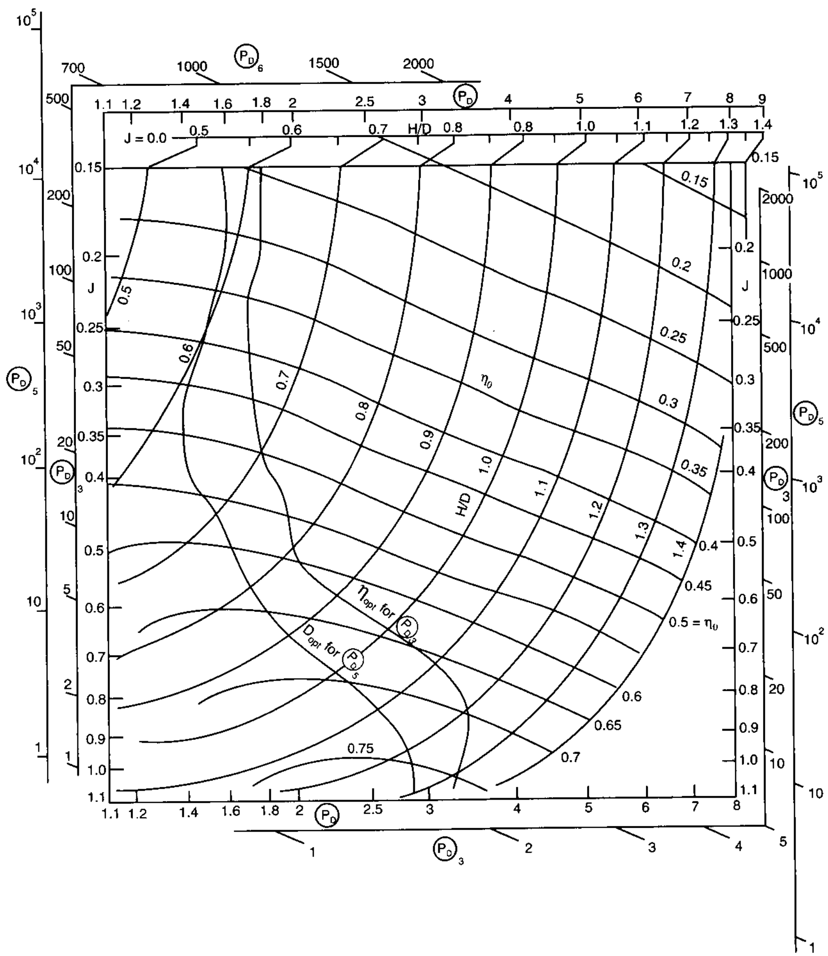


Fig. 11.14 Design diagram for given propulsion power

$$\textcircled{K}_2 = \frac{K_T}{J^2} = T/\rho D^2 V_1^2$$

$$\textcircled{K}_4 = \frac{K_T}{J^4} = Tn^2/\rho V_1^4$$

Optimum values of n and D for $\textcircled{K}_2$ and $\textcircled{K}_4$ respectively are defined by the point at which the propeller efficiency curve is tangential to the $\textcircled{K}_2$ or $\textcircled{K}_4$ curve. Thus if T, V_1 and either D or n are known the curves can be used directly to determine the optimum efficiency, pitch ratio and the n or D value.

Figure 11.14 is based on similar arguments and is used where the propulsive power P_D is known. In the diagram

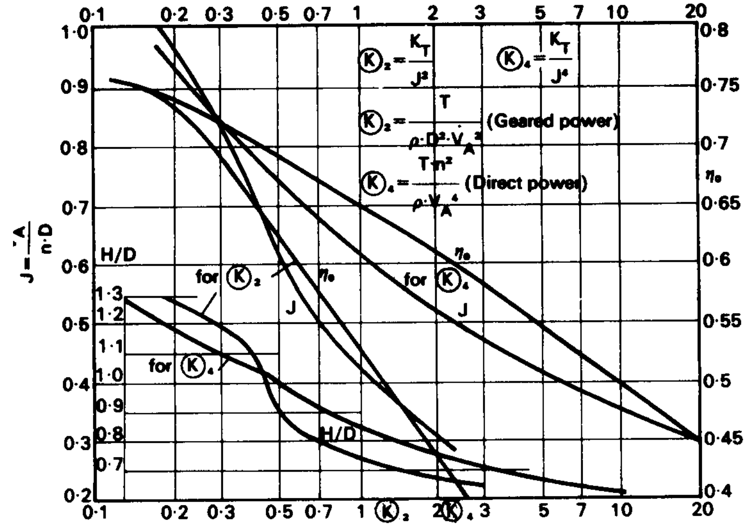


Fig. 11.15 Optimum curves given T and V_A

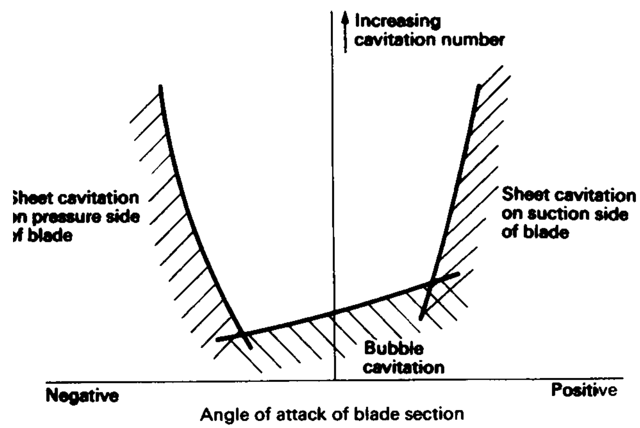


Fig. 11.16

$$\textcircled{P_D} = \frac{1000 P_D}{\rho D^5 n^3} = 83{,}776\ K_Q$$

$$\textcircled{P_D}_3 = \frac{\textcircled{P_D}}{J^3},\quad \textcircled{P_D}_5 = \frac{\textcircled{P_D}}{J^5}$$

Figure 11.15 presents optimum curves assuming T and V_1 are known together with propeller revolutions or diameter. Thus either $\textcircled{K}_2$ or $\textcircled{K}_4$ is known and the corresponding optimum values of efficiency, advance coefficient and pitch ratio are determined. The value of n or D whichever is the unknown follows from the advance coefficient.

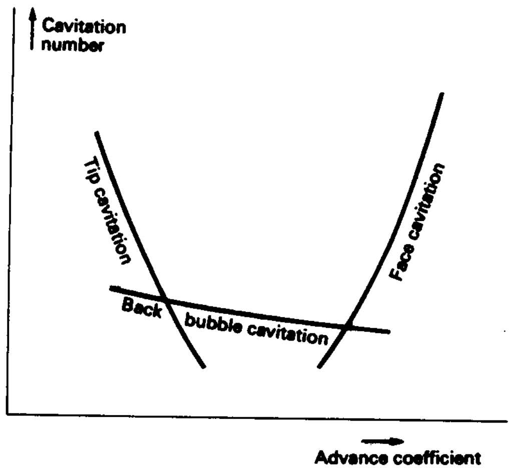


Fig. 11.17

CAVITATION

Figure 11.7 shows the influence of cavitation upon the overall propeller characteristics. In designing a propeller the naval architect needs to know the cavitation characteristics of the chosen blade section. The so called cavitation 'bucket' curve is used as in Fig. 11.16. The shape of the 'bucket' depends upon the blade thickness, camber and angle of attack.

The same characteristic of a 'cavitation bucket' is manifested in the results for the propeller itself as indicated in Fig. 11.17. The wider the bucket the greater the range of J over which the propeller can operate at a given cavitation number without cavitating in one way or another. The curves can be defined as a result of tests carried out in cavitation tunnels although, as has been pointed out in Chapter 10, these can only approximately represent the flow conditions at the propeller in the full-scale ship. Heavy cavitation can reduce the induced velocities ahead of the propeller and the thrust deduction fraction will fall. Cavitation will be more variable and noise and vibratory forces will be greater, if the blades are operating in an unsteady flow compared with the equivalent steady flow. This is in addition to the fall off in propeller efficiency.

INFLUENCE OF FORM ON RESISTANCE

It must be made clear that there is no absolute in terms of an optimum form. The designer has many things to consider besides the powering of the ship, e.g. ability to fit machinery, magazines, etc., seakeeping, manoeuvrability and so on. Even from the point of view of powering, one form may be superior to another at one displacement and over one speed range but inferior at other displacements or speed ranges.

Again, the situation is complicated by the fact that, in general, one parameter cannot be varied without affecting others. For example, to increase length,

keeping form coefficients and beam constant, will change displacement **and** draught. Thus the following comments can be regarded as being valid only in u general qualitative way. Wherever possible, reference should be made to methodical series tests.

Wetted length

Given freedom of choice of length, keeping displacement sensibly constant, a designer will choose a short form for slow speed ships and a long, slender form for high speed ships. This is because an increase in length increases the wetted surface area and hence the skin frictional resistance. At low speed this will more than offset any reduction in wave-making resistance, but for high speeds the possible reduction in wave-making resistance will be all important. Nevertheless, the variation of wave-making resistance with length does not obey a simple law as was explained in Chapter 10. Because of the interference between bow and stern wave systems, there will be optimum bands of length with intermediate lengths being relatively poor.

Prismatic coefficient

This coefficient has little influence on the skin frictional resistance but can have a marked effect on residuary resistance. Ifpossible, reference should be made to methodical series data from models similar to the design under development. Broadly, however, the optimum C_P value increases with increasing Froude number (Fig. 11.18). Since the influence of the prismatic coefficient is mainly related to the residuary resistance, it is not critical for low speed ships. In such ships, the choice of C_P value is much more likely to be governed by the cargo carrying capacity.

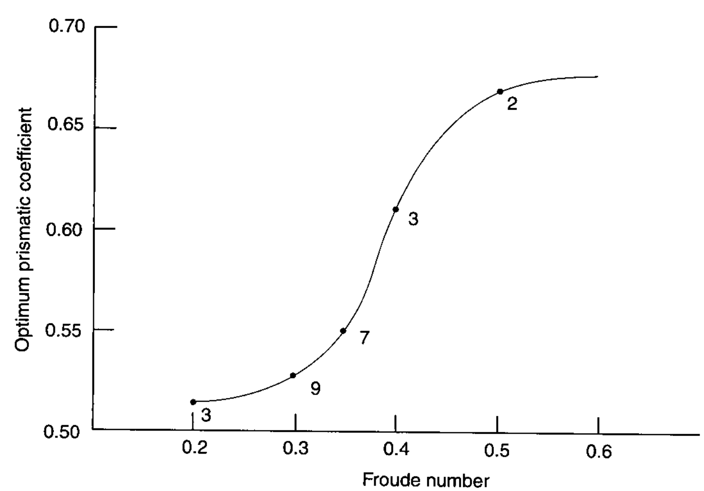


Fig. 11.18 Curve of optimum prismatic coefficient. Figures on the curve are the approximate percentage increases in resistance for ±0.05 *change from the optimum* C_P

Curve of areas

It is essential that this should be a fair curve with no sudden changes of curvature. Apart from this it is difficult and perhaps dangerous to generalize although it has been shown that very small changes in the curve of areas can produce really large changes in the residuary resistance.

Cross-sectional shapes

Generally, these are not critical but, if other ship requirements permit, U-shaped sections are to be preferred to V-shaped sections forward. This arises principally from the fact that more volume is removed from the vicinity of the waterplane and the wave-making resistance is accordingly reduced. V-shaped sections are used aft for vessels operating at high Froude number.

Centre of buoyancy position

The CB should vary from a few per cent of the length forward of amidships for slow ships to about 10 per cent of the length aft for fast ships.

REDUCING WAVE-MAKING RESISTANCE

Since, physically, wave-making resistance arises from a disturbance of the free surface, it is reasonable to expect that a lower resistance will result from concentrating displacement remote from the waterplane. That is to say, U sections are less resistful than V sections. Other generalizations are dangerous. In one case results for two forms with the same prismatic coefficient and apparently very similar curves of area show that, for one the wave-making resistance is double that of the other for $Fn = 0.23$ with substantial differences for Fn is the range 0.22 to 0.31.

Theory can help to explain why apparently small form changes can lead to large variations in wave-making resistance. It is for this reason that theory can often guide the model experimenter in the search for a better form. It should be emphasized, however, that there is no universal 'optimum' ship form giving minimum resistance at all speeds but rather a best form for a given Froude number.

It has been demonstrated that significant decreases in wave-making resistance occur when the bow and stern wave systems are out of phase. It is therefore reasonable to enquire whether a reduction can be obtained by artificially creating a wave system to interact with the ship system. In fact, this is the principle of the bulbous bow. Depending on its size, a bulb produces a wave system with crests and troughs in positions governed by the fore and aft position of the bulb relative to the bow. Unfortunately, this again can only produce a reduction in resistance over a limited range of Fn and then only at the expense of resistance at other speeds. Where a ship operates for a large percentage of its time at one speed as, for instance, is usually the case for most

merchant ships, such a device can be of great benefit and is becoming more extensively used.

Although bulbous bows were introduced to reduce wave-making resistancc it is now known that their action is more complex than the above simple explanation. They appear to modify the flow over the ship's hull generally and can lead to reduced resistance in full bodied ships at relatively low speeds. This is probably because they modify the flow over the bilges.

BOUNDARY LA YER CONTROL

Blowing, or more usually suction, offers the possibility of controlling separation of flow and vortex shedding, particularly for full hull forms and on control surfaces. This is similar to the control of flow over aircraft wings.

Another means of reducing drag in streamlined forms is the injection of polymers into the boundary layer. The effect is to reduce turbulence. Royal Navy trials on a surface ship, injecting Polyox at very low concentrations gave reductions in skin friction drag of up to 20 per cent. At 15 knots a concentration of 1.4 ppm gave an overall thrust reduction of 3.5 per cent. Work by the Russian Navy on submarines showed drag reductions of the order of 50 per cent, an increase in maximum speed of 10 per cent and a reduction in hydrodynamic noise level. The addition of the polymer did not affect the thrust deduction factor but the velocity profile in the boundary layer was smoother.

The advantages of such measures must be set against the complications of installing them, including space and weight considerations. The systems also need maintaining and the ejection slots must be kept clear of marine growth. They are most likely to be applied to military vehicles which can benefit from relatively short periods of high speed running, say a total of 100 hours during a mission.

COMPATIBILITY OF MACHINERY AND PROPELLER

Having the geometry of the propeller for the full-power condition, it follows that the thrust and torque variations with shaft revolutions are fully determined. To be satisfactory, the machinery must always be able to develop th~ torques at the various revolutions, otherwise the machinery will 'lock-up', 1.e. as speed is increased the machinery will arrive at a point where its power output is prematurely limited by the torque demanded by the propeller.

If there is no other solution available, it may be possible to solve the problem by fitting a controllable pitch propeller.

STRENGTH OF PROPELLERS

Calculations of propeller strength must take account of the torque and bending moments acting at the blade roots. Stress levels accepted must be such that the propeller will last the life of the ship and must allow for the cyclic variations in loads due to the wake and the increased forces due to ship motions and manoeuvnng.

EFFECT OF SPEED ON ENDURANCE

The rate of fuel consumption depends upon:

(a) the efficiency of the machinery at various power outputs. This feature is to some extent within the control of the machinery designer, e.g. by designing for optimum efficiency at full power, at cruising power or at some intermediate figure in order to balance the two;

(b) the power needed to supply the domestic loads of the ship such as lighting, galleys, air-conditioning, etc. This load is often referred to as the 'hotel' load and is independent of the forward speed of the ship.

The hotel load in a modern ship, particularly a passenger ship, can absorb a large proportion of the power generated at low speed. For this reason, the economical speed has tended to increase in recent years. The economic speed is a complicated thing to calculate but factors to be considered are: the fuel bill for covering a given distance at various speeds; the wages bill for the crew; the number of round voyages possible per year; special considerations depending on the payload of the ship, e.g. for a passenger ship a speedier passage may entice passengers away from the airlines. On the other hand, if the journey is between say Southampton and New York there is no attraction in arriving in one of the ports at midnight-if a faster journey is not feasible then the speed might as well be reduced.

In the following example, the influence of speed on the fuel bill is calculated for a typical steam ship of 26.25 MW.

EXAMPLE 4. The shaft power speed curve for a given ship with a total installed power of 26.25 MW, is as shown in Fig. 11.19 and the specific fuel consumption for various percentages of full power are as shown in Fig. 11.20. Calculate the endurance for 1000 tonnef of fuel over a range of speeds and the weight of fuel required for 1000 miles endurance.

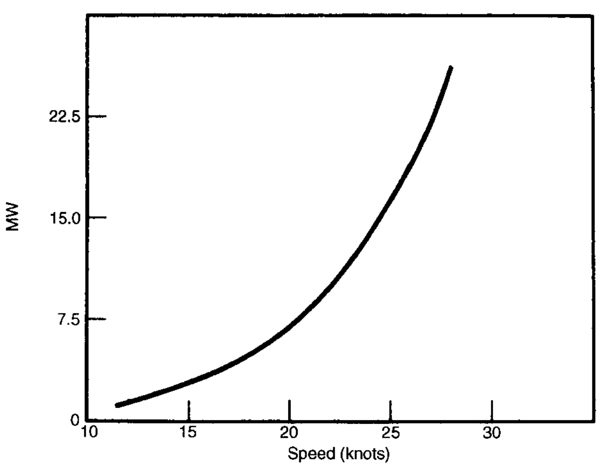


Fig. 11.19 Shaft power/speed curve

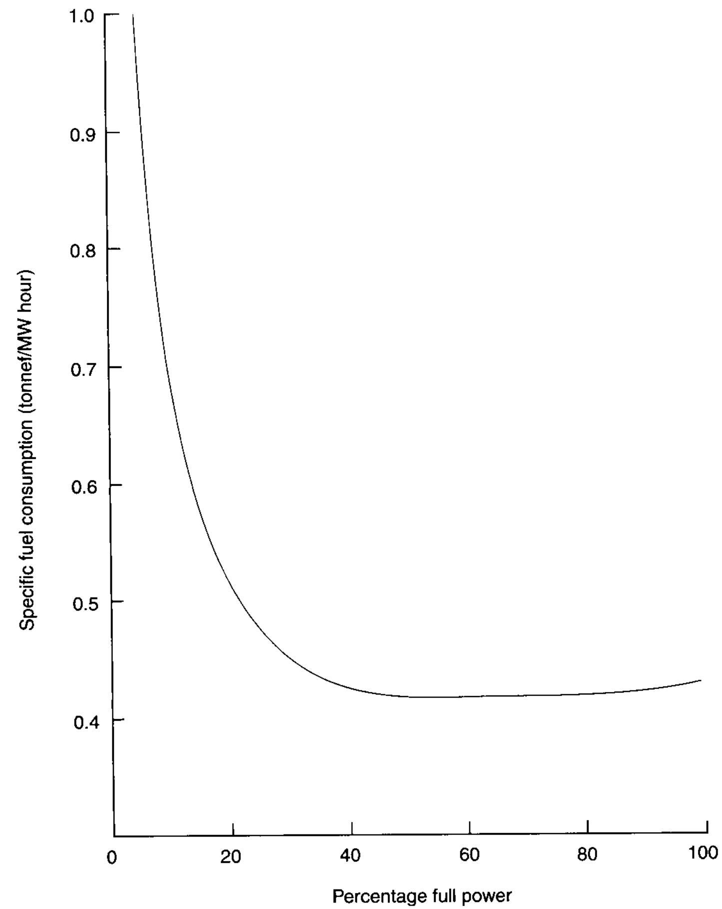


Fig. 11.20 Specific fuel consumption

Solution: At a speed of V knots, the distance travelled in 1 hr is V nautical miles. To travel 1000 miles takes $1000/V$ hr.

If the fuel consumption is S kg per MW hour, the fuel required for 1000 miles at speed V is

$$S \times (\text{power}) \times \frac{1000}{V} \text{ kg}$$

Conversely, the number of hours steaming possible on 1000 tonnef of fuel is given by

$$\text{No. of hours} = \frac{1000 \times 10^3}{S(\text{power})}$$

$$\text{Distance travelled} = \frac{10^6 \text{ V}}{S(\text{power})} \text{ nautical miles}$$

A table can now be constructed as in Table 11.9.

Table 11.9
Calculation of endurance

V (knots)	Power (MW)	% full power	SFC (tonnef/MW hour)	tonnef of fuel per 1000 miles	Endurance for 1000 tonnef fuel
12	1.43	5.4	0.97	115	8700
14	2.26	8.6	0.74	119	8370
16	3.31	12.6	0.61	126	7920
18	4.80	18.3	0.52	139	7210
20	6.90	26.3	0.46	159	6300
22	9.74	37.1	0.43	190	5250
24	13.65	52.0	0.42	239	4190
26	18.74	71.4	0.42	303	3300
28	26.25	100.0	0.43	403	2480

It will be seen that there is not a lot of penalty in increasing the cruising or endurance speed from 12 to 16knots. Indeed, economically, the lower salary bill would probably compensate for the increased fuel bill and the ship is a sounder economic proposition because of its increased mileage in a year.

Computational fluid dynamics

With the increasing power of modern computers, computational fluid dynamics (CFD) is becoming a possible tool for the naval architect although it is too complex to cover in this book in other than outline.

For surface ship calculations water can be assumed to be incompressible. This means that the volume of water entering a given elemental space in the vicinity of the hull must be matched by an equal volume of water flowing out. This leads to a continuity equation which the flow must obey. This, taken with the Navier-Stokes equations which define the conservation of momentum of flow, is adequate to characterize the physics for flow around a ship's hull. These equations are not soluble at present in their basic form because of turbulence effects and flow has to be averaged over a period of time. This time period is small compared with overall ship motions but large compared to the turbulence fluctuations. This leads to the Reynolds-averaged Navier-Stokes (RANS) equations

If viscosity, and hence turbulence, is ignored the Euler equations of motion can be used with the continuity equation, although these are not generally recommended for ship applications.

With the additional assumption of irrotational flow the problem reduces to one of potential flow. The three velocity components at any point are now related to the potential, ϕ. The derivative of ϕ in any direction gives the velocity in that direction. The continuity equation then reduces to Laplace's equation for potential flow.

Briefly, then, CFD embraces techniques for solving the equations (RANS, Euler or Laplace) by numerical methods using large numbers of elements around the ship. These techniques include:

- Boundary element methods (BEM), also known as panel methods. **Theile** can be used only for tackling potential flow problems. The surface of **the** hull and the surrounding water surface are divided into discrete elements. or panels. This method is currently the most widely used commercially, with applications such as resistance calculations.
- Finite difference methods (FDM). Unlike BEM, the whole fluid domain is divided into elements. Its name derives from the fact that the derivatives of the fluid equations are approximated by finite differences. The disadvantage of this method is that the mass and momentum of flow may not be fully conserved.
- Finite volume methods (FVM). These also cover the whole fluid domain and use finite differences, but in this case the methods used ensure that mass and momentum are conserved. Most commercial RANSE solvers use FVM. Grid generation is more onerous than for the BEM methods.

Powerful software is available commercially to compute the flow with pressure, velocity components and turbulence at the nodes or centroid of each computational cell. They can also output the forces and moments (pressure and skin friction components) on the hull and appendages. Software flow visualization tools enable visual checks to be made at each stage of calculation.

Although CFD can be used for seakeeping and manoeuvring assessments, strip methods which are now well established tend to be used in preference.

The capability of CFD is gradually improving. The techniques are not easy to apply and to date most applications have been by researchers. Early work was related to deeply submerged bodies with calculations covering the hull, control surfaces and propulsors. Deeply submerged bodies, of course, avoid the complication presented by the sea/air interface on which vessels other than deeply submerged submarines must operate. However, methods can solve the free surface interface although not yet for all hull forms.

CFD techniques can be used in the design process looking at a range of design options in conjunction with semi-empirical methods and model tests. In many cases the theoretical results need to be adjusted to fit observed trial data. Typical applications have been to simulate resistance and, increasingly, propulsion. Its use in wake prediction and propeller design is established. For ~ pellers it can predict the thrust in fully developed cavitating and non-cavitating conditions in steady and unsteady flow. It is not so good for predicting torque, the onset of cavitation or the hull/propeller interaction. More work is needed to take account of the effects of air and impurities entrained in the water.

As an example of what can be achieved, in one application MIT compared propellers behind a tapered and full stern axisymmetric submarine hull. A vortex lifting-line method was used to give a preliminary propeller design, providing three-dimensional blade geometries. Then a 3-D vortex-lattice lifting-surface propeller blade program, coupled to a RANSE solver, was used to find an optimized propeller geometry and the resulting forces for the propeller/hull combination. The combination of methods took account of the propeller/hull interactions giving thrust deduction and the effective inflow to the propeller.

Although not conclusive this study suggested that a full stern hull may have advantages.

Grid generation is a key element in CFD. To generate a good grid for dealing efficiently with any given problem is not easy at present. It demands considerable experience of the methods. Also, grid generation accounts for a major part of the time needed for a CFD investigation, perhaps 80 per cent. In future it is to be expected that more user friendly methods will become available, making use of expert systems computer techniques to guide the less experienced practitioner.

As CFD methods become easier to use, and with more experience of comparing theoretical results with model and full-scale data, it is to be expected that they will become more common as part of the design process.

Summary

The methods of presentation and calculation presented enable the general principles established in Chapter 10 to be applied to the calculation of the shaft power required to propel a new design at the required speed. Allowance can be made for air resistance and for hull fouling as a result of marine growth which increases with time out of dock. As part of the process, a suitable design of screw propeller is obtained from methodical series data.

This is, of course, only a beginning. While it enables a suitable design of hull and propeller to be effected, it does not describe many of the refinements to the process. These the student must pursue through the transactions of the learned societies.

Problems

1. A ship of 50 MN displacement is driven at a speed of 12 knots. A ship of 65 MN of similar form is being designed. At what speed of the larger ship should its performance be compared with the 50 MN ship?
2. A ship of length 64.6 m, 6.02 m beam, 1.98 m draught, wetted surface 369 m^2 and displacement 4.26 MN has a resistance of 35 kN at 15.8 knots.
   Deduce the dimensions and effective power of a ship of similar form 233 m long at the corresponding speed.
3. With the definitions of (C) and C_T given in the text show that

$$C_T = \frac{8\pi}{1000}\frac{(C)}{(S)}$$

4. A 5 m model of a 180 m long ship is towed in a ship tank at a speed of 1.2 m/sec. The towing pull is 11.77 newtons. Assuming that 60 per cent of the resistance force is due to skin friction, calculate the corresponding speed for the ship in knots and the P_E at this speed assuming a wetted surface area of 3600 m^2 in the ship.

5. A destroyer 97.5 m × 1473 tonnef has a full speed of 35 knots with clean bottom. The (C) value of a 4.88 m model of the same form is 1.75 at the corresponding speed. Estimate the proportion of the ship resistance at full speed attributable to skin friction using
   (1) Froude analysis
   (2) ITTC ship-model correlation line.
   NOTE: Use Question 3 to change (C) to C_T.
6. A model of a vessel, 122 × 20 × 7.3 m draught, of 8697 tonnef displacement, is run, and the curve of P_E on a base of speed of ship is 2.42, 3.01, 3.74, 4.62 and 5.71 MW for 16, 17, 18, 19 and 20 knots respectively. Make an estimate of the P_E of a ship of 16,250 tonnef, of similar form, for speeds of 20 and 21 knots and give the dimensions of the new ship.
7. The data below relates to the 4.88 m model of a ship 120 m long, 4064 tonnef displacement. Assuming (S) = 7.15 and a propulsive coefficient of 0.57 plot the P_S-speed curve for the ship when 6 months out of dock. Assume that the skin frictional resistance increases by $\frac{1}{4}$ of a per cent per day out of dock.
   What speed is likely in this condition with 30 MW?

(K)	1.6	2.0	2.5	3.0	3.5	4.0	4.5
(C)	1.150	1.176	1.245	1.418	1.500	1.770	2.085

8. The following data relates to a 4.88 m model of a ship 195 m long and 14,453 tonnef displacement.

(K)	1.2	1.5	2.0	2.3	2.5	3.0	3.1	3.2	3.3	3.4
(C)	1.205	1.170	1.145	1.145	1.155	1.160	1.170	1.181	1.210	1.253

Compute and plot the P_S-speed curve for the ship 6 months out of dock assuming (S) = 7.40, and a propulsive coefficient of 0.55.
What power is required for 20 and 28 knots?

9. A ship of 6096 tonnef displacement is required to have a maximum speed of 25 knots. The total effective power including appendages is 11.19 MW. Other data are:

Hull efficiency	= 1.0
Mean wake	= 1.4 per cent
Relative rotative efficiency	= 0.98
Quasi-propulsive coefficient factor	= 0.88
Pressure coefficient	= 7.65 tonnef/ m^2

Use the propeller design diagram to find the propeller diameter and shaft power required for shaft revolutions of 200 r.p.m. Calculate the efficiency, pitch and r.p.m. of a 3.96 m diameter propeller.

10. A propeller is found by calculation to have a K_T/J^2 value of 0.2225 when developing a P_T of 3.73 MW at a speed of advance through the water of 25 knots. What is the diameter of the propeller?
    Use the propeller design diagram to find the maximum efficiency possible and the corresponding pitch.
11. What is *cavitation* and what is its effect on torque, thrust and efficiency of a propeller? Explain why experiments on models to determine the effects of cavitation cannot be carried out in an open ship tank.
    A 4 m propeller has been designed for a destroyer to give a top speed of 30 knots at 250 r.p.m. It is desired to run a 50 cm model of the propeller in a cavitation tunnel at a water speed of 6 m/s. At what water pressure and r.p.m. must the model propeller be run, to simulate ship conditions? C.L. of propeller below surface = 4 m. Atmospheric pressure = $10^5\,\mathrm{N/m^2}$, water vapour pressure = $1700\,\mathrm{N/m^2}$. Froude wake factor = −0.01.
12. Estimate the r.p.m. and expected thrust and torque of a model propeller, 229 mm in diameter, fitted behind a 5.49 m model of a ship 137 m long. The model is to be run in fresh water. The following data are available for the ship (in salt water, $0.975\,\mathrm{m^3/tonnef}$):

shaft r.p.m.	= 110
ship speed	= 8.03 m/s
Froude wake factor	= 0.47
P_D	= 4.77 MW
P_T	= 2.97 MW

13. What is meant by the terms quasi-propulsive coefficient and quasi-propulsive coefficient factor? Explain how these quantities are obtained from model experiments and full-scale trials.
    The following results were obtained from experiments on a 5 m model of a new design of frigate: Length 130 m, displacement 3600 tonnef,

Ship C at 27 knots	= 1.59
Hull efficiency	= 0.95
Relative rotative efficiency	= 0.98
Openwater screw efficiency	= 0.69
Appendage coefficient	= 1.07

    On trials, the frigate achieved a speed of 27 knots at a measured shaft power of 22 MW. Calculate the quasi-propulsive coefficient and the quasi-propulsive coefficient factor.
14. Assuming that the curve of K_T against J is a straight line over the working range, show that, for a given speed of advance, the thrust (T) developed by a propeller varies with its r.p.m. (N) in the following manner:

$$T = AN^2 - BN$$

where A and B are constants for a given propeller and speed of advance.

For a particular 50.8 cm model propeller, running at 168 m/min, the constants have the following values: $A = 75.6$; $B = 289$, when N is measured in hundreds of r.p.m., and T is measured in newtons.
    Calculate the thrust of a geometrically similar propeller of 3.05 m diameter at 200 r.p.m. and a speed of advance of 12 knots in sea water.
15. Resistance experiments are to be run on a 6.1 m model of a new warship design. Estimate the model resistance in newtons at a speed corresponding to the ship's full speed, given the following design information:
    Length, 177 m; displacement, 11,379 tonnef; estimated P_S at full speed of 30 knots, 55.18 MW; estimated propulsive coefficient, 0.54; wetted surface, $3530\,\mathrm{m^2}$.
    Use the ITTC line to calculate skin frictional resistance for model and ship; applying a roughness correction of 0.0004 for the ship only.
    The appropriate kinematic viscosities are:

fresh water, $1.229 \times 10^{-6}\,\mathrm{m^2/s}$
salt water, $1.113 \times 10^{-6}\,\mathrm{m^2/s}$

16. Describe, briefly, the three main causes of 'wake' when considering a ship moving through the water.
    How is the hull efficiency related to wake and augment?
    The thrust–r.p.m. curve for a model propeller, run in open water at an advance speed of 2.5 m/s, is given by the equation:

$$T = 53\,N^2 - 225\,N.$$

where T is in newtons, N is in hundreds of r.p.m.
    This curve coincides with the 'behind thrust' curve for the propeller-ship model combination when run at a speed of advance of 2.8 m/s.
    The augmented resistance can be approximated to the straight line:

$$T = 45\,N - 145$$

and the model hull resistance, when towed without the propellers and at a speed of 2.8 m/s is 51 newtons.
    Determine the propeller revolutions for model self propulsion and hence find the wake, augment and hull efficiency for the propeller-ship combination.
17. Describe the methods by which the hull efficiency elements may be deduced from model experiments.
    Why can good comparisons be made between various sizes of propellers by only maintaining the advance coefficient constant, when running deeply submerged in open water?
    The results of tests on a 30 cm diameter propeller run at 500 r.p.m. are shown below.

What will be the maximum efficiency obtainable for this propeller, and the appropriate speed of advance?

J	0	0.2	0.4	0.6	0.8	1.0	1.2
K_T	0.715	0.62	0.50	0.37	0.25	0.14	0.03
K_Q	0.13	0.114	0.094	0.072	0.05	0.032	0.014

18. Very briefly, describe methods by which hull-efficiency elements may be determined from model experiments.

After towing a model, it is deduced that the effective power required to move the ship's hull and appendages at a speed of 28 knots would be 17.9 MW of which 39 per cent would be due to skin friction. Skin friction for the model would account for 43.5 per cent of the total resistance.

Experiments were then conducted with the model and propellers combined at a model speed of 2.44 m/s which corresponds to the required ship's speed. At self-propulsion the shaft speed was 14.0 rev/s and thrust and torque provided were 32.7 and 4.3 newtons respectively. Open water experiments with the same propellers gave the same thrust and revs when advancing at 2.33 m/s. Determine the augment and wake fractions and the propeller efficiency behind the ship.

Hence, assuming RRE and appendage coefficient to be both 1.0, deduce the P_S required for 28 knots and a QPC factor of 0.94.

19. A new design has a displacement of 115 MN and length 174 m. Corresponding values of C and K for a 4.88 m model are:

K	2.2	2.3	2.4	2.5	2.6	2.7	2.8
C	1.130	1.130	1.132	1.138	1.140	1.141	1.150
K	2.9	3.0	3.1	3.2	3.3	3.4	
C	1.153	1.156	1.172	1.193	1.235	1.283	

Assuming S $= 7.205$ and a propulsive coefficient of 0.55, plot the curve of power (MW) against speed for the clean ship. Assuming that the skin frictional resistance increases by $\frac{1}{4}$ per cent per day out of dock, plot the corresponding curve for the ship 6 months out of dock.

Assuming 33.56 MW installed power, calculate the maximum speeds (*a*) in the clean condition and (*b*) 6 months out of dock.

12 Seakeeping

Seakeeping qualities

The general term sea worthiness must embrace all those aspects of a ship design which affect its ability to remain at sea in all conditions and to carry out its specified duty. It should, therefore, include consideration of strength, stability and endurance, besides those factors more directly influenced by waves. In this chapter, the term seakeeping is used to cover these more limited features, i.e. motions, speed and power in waves, wetness and slamming.

The relative importance of these various aspects of performance in waves varies from design to design depending upon what the operators require of the ship, but the following general comments are applicable to most ships.

Motions

Excessive amplitudes of motion are undesirable. They can make shipboard tasks hazardous or even impossible, and reduce crew efficiency and passenger comfort. In warships, most weapon systems require their line of sight to remain fixed in space and to this end each system is provided with its own stabilizing system. Large motion amplitudes increase the power demands of such systems and may restrict the safe arcs of fire.

The phase relationships between various motions are also important. Generally, the phasing between motions is such as to lead to a point of minimum vertical movement about two-thirds of the length of the ship from the bow. In a passenger liner, this area would be used for the more important accommodation spaces. If it is desirable to reduce the vertical movement at a given point, then this can be achieved if the phasing can be changed, e.g. in a frigate motion at the flight deck can be the limiting factor in helicopter operations. Such actions must inev~ ably lead to increased movement at some other point. In the frigate, incre«e"d movement of the bow would result and wetness or slamming might then limit operations.

Speed and power in waves

When moving through waves the resistance experienced by a ship is increased and, in general, high winds mean increased air resistance. These factors cause the ship speed to be reduced for a given power output, the reduction being aggravated by the less favorable conditions in which the propeller is working. Other unpleasant features of operating in waves such as motions, slamming and wetness are generally eased by a reduction in speed so that an additional speed reduction may be made voluntarily.

Wetness

When the relative movement of the bow and local wave surface becomes too great, water is shipped over the forecastle. At an earlier stage, spray is driven over the forward portion of the ship by the wind. Both conditions are undesirable and can be lessened by increasing freeboard. The importance of this will depend upon the positioning of upper deck equipment and its sensitivity to salt spray. Spray rails, flare angles and knuckles may all influence the troublesome nature of spray which, in cold climates, causes ice accretion.

Slamming

Under some conditions, the pressures exerted by the water on a ship's hull become very large and slamming occurs. Slamming is characterized by a sudden change in the vertical acceleration of the ship followed by a vibration of the ship girder in its natural frequencies. The conditions leading to slamming are high relative velocity between ship and water, shallow draught and small rise of floor. The area between 10 and 25 per cent of the length from the bow is the area most likely to suffer high pressures and to sustain damage.

Ship routing

Since the ship behaviour depends upon the wave conditions it meets, it is reasonable to question whether overall performance can be improved by avoiding the more severe waves. This possibility has been successfully pursued by some authorities. Data from weather ships are used to predict the speed loss in various ocean areas and to compute the optimum route. In this way, significant savings have been made in voyage times, e.g. of the order of 10-15 hours for the Atlantic crossing.

Importance of good seakeeping

No single parameter can be used to define the seakeeping performance of a design. In a competitive world, a comfortable ship will attract more passengers than a ship with a bad reputation. A ship with less power augment in waves will be able to maintain tighter schedules or will have a lower fuel bill. In extreme cases, the seakeeping qualities of a ship may determine its ability to make a given voyage at all.

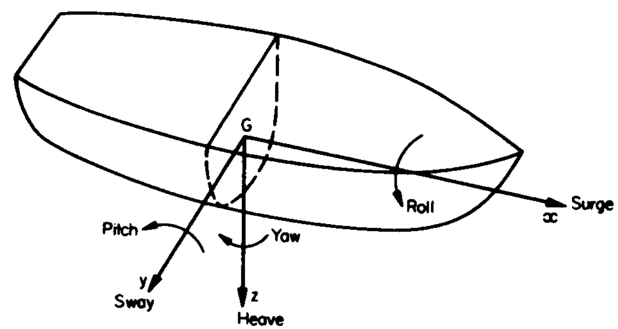
Good seakeeping is clearly desirable, but the difficulty lies in determining how far other design features must, or should, be compromised to improve seakeeping. This will depend upon each particular design, but it is essential that the designer has some means of judging the expected performance and the effect on the ship's overall effectiveness. Theory, model experiment and ship trial all have a part to play. Because of the random nature of the sea surface in which the ship operates, considerable use is made of the principles of statistical analysis.

Having improved the physical response characteristics of a ship in waves the overall effectiveness of a design may be further enhanced by judicious siting of critical activities and by fitting control devices such as anti-roll stabilizers.

As with so many other aspects of ship design a rigorous treatment of sea-keeping is very complex and a number of simplifying assumptions are usually made. For instance, the ship is usually regarded as responding to the waves us a rigid body when assessing motions and wetness although its true nature as an elastic body must be taken into account in a study of structure. In the same way it is instructive, although not correct, to study initially the response of a ship to regular long-crested waves ignoring the interactions between motions, e.g. when the ship is heaving the disturbing forces will generate a pitching motion. This very simple approach is now dealt with before considering coupled motions.

Ship motions

It was seen in Chapter 4, that a floating body has six degrees of freedom. To completely define the ship motion it is necessary to consider movements in all these modes as illustrated in Fig. 12.1. The motions are defined as movements of the centre of gravity of the ship and rotations about a set of orthogonal axes through the c.g. These are space axes moving with the mean forward speed of the ship but otherwise fixed in space.



Translation or rotation	Axis	Description	Positive sense
Translation	Along *x*	Surge	Forwards
	Along *y*	Sway	To starboard
	Along *z*	Heave	Downwards
Rotation	About *x*	Roll	Starboard side down
	About *y*	Pitch	Bow up
	About *z*	Yaw	Bow to starboard

Fig. 12.1 *Ship motions*

It will be noted that roll and pitch are the dynamic equivalents of heel and trim. Translations along the *x*- and y-axis and rotation about the z-axis lead to no residual force or moment, provided displacement remains constant, as the ship is in neutral equilibrium. For the other translation and rotations, movement is opposed by a force or moment provided the ship is stable in that mode. The magnitude of the opposition increases with increasing displacement from the equilibrium position, the variation being linear for small disturbances.

This is the characteristic of a simple spring system. Thus, it is to be expected that the equation governing the motion of a ship in still water, which is subject to a disturbance in the roll, pitch or heave modes, will be similar to that governing the motion of a mass on a spring. This is indeed the case, and for the undamped case the ship is said to move with simple harmonic motion.

Disturbances in the yaw, surge and sway modes will not lead to such an oscillatory motion and these motions, when the ship is in a seaway, exhibit a different character to roll, pitch and heave. These are considered separately and it is the oscillatory motions which are dealt with in the next few sections. It is convenient to consider the motion which would follow a disturbance in still water, both without and with damping, before proceeding to the more realistic case of motions in waves.

UNDAMPED MOTION IN STILL WATER

It is assumed that the ship is floating freely in still water when it is suddenly disturbed. The motion following the removal of the disturbing force or moment is now studied for the three oscillatory motions.

Rolling

Let ϕ be the inclination of the ship to the vertical at any instant. The moment, acting on a stable ship, will be in a sense such as to decrease ϕ. For small values of ϕ,

$$\text{moment} = -\Delta\overline{\mathrm{GM}}_\mathrm{T}\phi$$

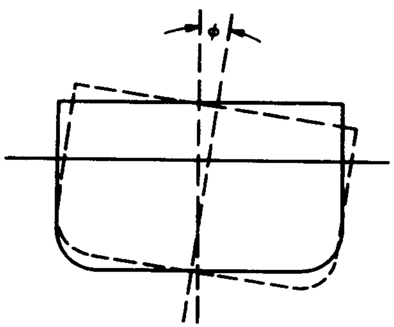


Fig. 12.2 Rolling

Applying Newton's laws of motion

$$\text{moment} = (\text{moment of inertia about } 0x)(\text{angular acceleration})$$

i.e.

$$-\Delta\overline{\mathrm{GM}}_\mathrm{T}\phi = +\frac{\Delta}{g}k_{xx}^2\frac{\mathrm{d}^2\phi}{\mathrm{d}t^2}$$

i.e.

$$\frac{\mathrm{d}^2\phi}{\mathrm{d}t^2} + \left(g\frac{\overline{\mathrm{GM}}_\mathrm{T}}{k_{xx}^2}\right)\phi = 0$$

This is the differential equation denoting simple harmonic motion with period T_ϕ where

$$T_\phi = 2\pi\left(\frac{k_{xx}^2}{g\overline{\mathrm{GM}}_\mathrm{T}}\right)^{\frac{1}{2}} = \frac{2\pi k_{xx}}{(g\overline{\mathrm{GM}}_\mathrm{T})^{\frac{1}{2}}}$$

It will be noted that the period of roll is independent of ϕ and that this will hold as long as the approximation $\overline{\mathrm{GZ}} = \overline{\mathrm{GM}}_\mathrm{T}\phi$ applies, i.e. typically up to ± 10 degrees. Such rolling is termed *isochronous*.

In practice k_{xx} must be increased to allow for what are usually termed 'added mass' effects due to motion induced in the water although this does not mean that a specific body of water actually moves with the ship. Added mass values vary with frequency but this variation can often be ignored to a first order. Typically the effect increases k_{xx} by about 5 per cent.

Hence

$$T_\phi \propto \frac{1}{(\overline{\mathrm{GM}}_\mathrm{T})^{\frac{1}{2}}}$$

Thus the greater is $\overline{\mathrm{GM}}_\mathrm{T}$, i.e. the more stable the ship, the shorter the period and the more rapid the motion. A ship with a short period is said to be 'stiff'—compare the stiff spring—and one with a long period is said to be 'tender'. Most people find a long period roll less unpleasant than a short period roll.

Pitching

This is analogous to roll and the motion is governed by the equation

$$\frac{\mathrm{d}^2\theta}{\mathrm{d}t^2} + \left(\frac{g\overline{\mathrm{GM}}_\mathrm{L}}{k_{yy}^2}\right)\theta = 0$$

and the period of the motion is

$$T_\theta = \frac{2\pi k_{yy}}{(g\overline{\mathrm{GM}}_\mathrm{L})^{\frac{1}{2}}} \text{ for very small angles of pitch.}$$

Heaving

Let z be the downward displacement of the ship at any instant. The force acting on the ship tends to reduce z and has a magnitude F_z given by

$$F_z = -\frac{A_\mathrm{W}z}{u}$$

where u is the reciprocal weight density of the water.

Hence, the heaving motion is governed by the equation

$$\frac{\Delta}{g}\frac{d^2 z}{dt^2} = -\frac{A_W z}{u}$$

or

$$\frac{d^2 z}{dt^2} + \frac{g A_W}{u\Delta} z = 0$$

from which

$$\text{period} = 2\pi\left(\frac{u\Delta}{gA_W}\right)^{\frac{1}{2}}$$

Δ may be effectively increased by a significant amount (perhaps doubled) by the 'added mass' effect.

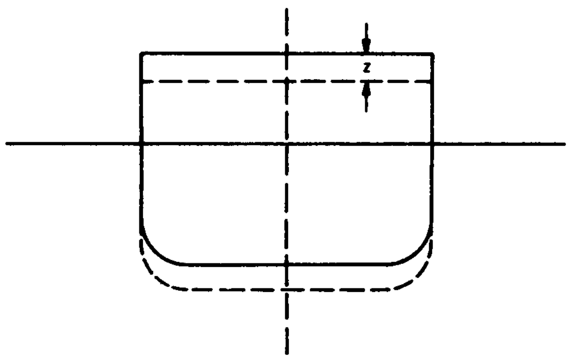


Fig. 12.3 Heaving

DAMPED MOTION IN STILL WATER

Now consider what happens when the motion is damped. It is adequate to illustrate the effect of damping on the rolling motion.

Only the simplest case of damping is considered here namely, that in which the damping moment varies linearly with the angular velocity. It opposes the motion since energy is always absorbed.

Allowing for the entrained water the equation for rolling in still water becomes

$$\frac{\Delta}{g}k_{xx}^2(1+\sigma_{xx})\ddot{\phi} + B\dot{\phi} + \Delta\overline{GM}_T\phi = 0$$

where

$$\frac{\Delta k_{xx}^2}{g}\sigma_{xx} = \text{augment of rolling inertia of ship due to entrained water}$$

$$B = \text{damping constant.}$$

This can be likened to the standard differential equation

$$\ddot{\phi} + 2k\omega_0\dot{\phi} + \omega_0^2\phi = 0$$

where

$$\omega_0^2 = \frac{g\overline{GM}_T}{k_{xx}^2(1+\sigma_{xx})} \quad \text{and} \quad k = \frac{Bg}{2\omega_0\Delta k_{xx}^2(1+\sigma_{xx})}$$

which in turn defines the effective period T_ϕ of the motion as

$$T_\phi = \frac{2\pi}{\omega_0}(1-k^2)^{-\frac{1}{2}} = 2\pi k_{xx}\left(\frac{1+\sigma_{xx}}{g\overline{GM}_T}\right)^{\frac{1}{2}}(1-k^2)^{-\frac{1}{2}}$$

When the damping is not proportional to the angular velocity the differential equation is no longer capable of ready solution.

APPROXIMATE PERIOD OF ROLL

Of the various ship motions the roll period is likely to vary most from design to design and, because of the much greater amplitudes possible, it is often the most significant. Various approximate formulae have been suggested for calculating the period of roll including:

$$T_\phi = 2\pi\frac{K}{(g\overline{GM}_T)^{\frac{1}{2}}}$$

Suggested values of K for merchant ships and warships are given by the respective expressions:

Merchant ships

$$\left(\frac{K}{B}\right)^2 = F\left[C_B C_u + 1.10C_u(1-C_B)\left(\frac{H}{T} - 2.20\right) + \frac{H^2}{B^2}\right]$$

where

C_u = upper deck area coeff. $= \frac{1}{LB}$ (deck area)

H = effective depth of ship $= D + A/L_{pp}$

A = projected lateral area of erections and deck

L_{pp} = L.B.P.

T = mean moulded draught

F = constant = 0.125 for passenger and cargo ship,

= 0.133 for oil tankers,

= 0.177 for whalers.

Warships

$$\left(\frac{K}{B}\right)^2 = F\left[C_B C_u + 1.10C_e(1-C_B)\left(\frac{H_n}{T} - 2.20\right) + \frac{H_n^2}{B_u^2}\right]$$

where

B_u = max. breadth under water;
C_e = exposed deck area coeff.;
$H_n = D + A_n/L_{pp}$;
D = depth from top of keel to upper deck;
A_n = sum of the projected lateral areas of forecastle, under bridge and gun;
F = constant ranging from 0.172 for small warships to 0.177 for large warships.

MOTION IN REGULAR WAVES

In Chapter 9, it is explained that the irregular wave systems met at sea can be regarded as made up of a large number of regular components. A ship's motion record will exhibit a similar irregularity and it can be regarded as the summation of the ship responses to all the individual wave components. Theoretically, this super-position procedure is valid only for those sea states for which the linear theory of motions is applicable, i.e. for moderate sea states. It has been demonstrated by several authorities, however, that provided the basic data is derived from relatively mild regular components, the technique can be applied, with sufficient accuracy for most engineering purposes, to more extreme conditions. Thus, the basic element in ship motions is the response of the ship to a regular train of waves. For mathematical convenience, the wave is assumed to have a sinusoidal profile. The characteristics of such a system were dealt with in Chapter 9.

In the simple approach, it is necessary to assume that the pressure distribution within the wave system is unaffected by the presence of the ship. This is one of the assumptions made by William Froude in his study of ship rolling and is commonly known as 'Froude's Hypothesis'.

Rolling in a beam sea

The equation for rolling in still water is modified by introducing a forcing function on the right-hand side of the equation. This could be obtained by calculating the hydrodynamic pressure acting on each element of the hull and integrating over the complete wetted surface.

The resultant force acting on a particle in the surface of a wave must be normal to the wave surface. Provided the wave-length is long compared with the beam of the ship, it is reasonable to assume that the ship is acted on by a resultant force normal to an 'effective wave surface' which takes into account

all the sub-surfaces interacting with the ship. Froude used this idea and further assumed that the 'effective wave slope' was that of the sub-surface passing through the centre of buoyancy of the ship.

With this assumption it can be shown that, approximately, the equation of motion for undamped rolling motion in beam seas becomes

$$\frac{\Delta}{g}k_{xx}^2(1+\sigma_{xx})\ddot{\phi} + \Delta\overline{GM}_T(\phi - \phi') = 0$$

where $\phi' = \alpha \sin \omega t$; α = maximum slope of the surface wave; ω = frequency of the surface wave.

If ϕ_0 and ω_0 are the amplitude and frequency of unresisted rolling in still water, the solution to this equation takes the form

$$\phi = \phi_0 \sin(\omega_0 t + \beta) + \frac{\omega_0^2\alpha}{\omega_0^2 - \omega^2}\sin\omega t$$

The first term is the free oscillation in still water and the second is a forced oscillation in the period of the wave train.

The amplitude of the forced oscillation is

$$\frac{\omega_0^2\alpha}{\omega_0^2 - \omega^2}$$

When the period of the wave system is less than the natural period of the ship ($\omega > \omega_0$), the amplitude is negative which means that the ship rolls into the wave (Fig. 12.4(*a*)). When the period of the wave is greater than the natural period of the ship, the amplitude is positive and the ship rolls with the wave (Fig. 12.4(*b*)). For very long waves, i.e. ω very small, the amplitude tends to α and the ship remains approximately normal to the wave surface. When the frequencies of the wave and ship are close the amplitude of the forced oscillation becomes very large.

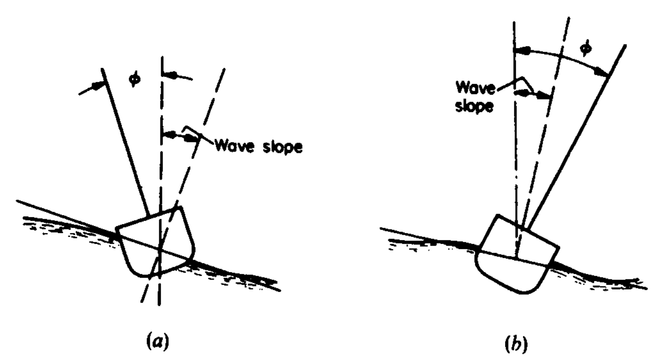


Fig. 12.4

The general equation for rolling in waves can be written as:

$$\ddot{\phi} + 2k\omega_0\dot{\phi} + \omega_0^2\phi = \omega_0^2\alpha\cos\omega t$$

The solution to this differential equation is

$$\phi = \mu\alpha\cos(\omega t - \varepsilon)$$

where

$$\tan\varepsilon = \frac{2k\Lambda}{1-\Lambda^2}$$

Λ = *tuning factor* $= \omega/\omega_0$; μ = *magnification factor* $= 1/\{(1-\Lambda^2)^2 + 4k^2\Lambda^2\}^{\frac{1}{2}}$.

Plots of the phase angle ε and magnification factor are presented in Fig. 12.5. It will be appreciated that these expressions are similar to those met with in the study of vibrations.

The effect of damping is to cause the free oscillation to die out in time and to modify the amplitude of the forced oscillation. In an ideal regular sea, the ship would oscillate after a while only in the period of the waves. In practice, the maximum forced roll amplitudes occur close to the natural frequency of the ship, leading to a ship at sea rolling predominantly at frequencies close to its natural frequency.

Pitching and heaving in waves

In this case, attention is focused on head seas. In view of the relative lengths of ship and wave, it is not reasonable to assume, as was done in rolling, that the wave surface can be represented by a straight line. The principle, however, remains unchanged in that there is a forcing function on the right-hand side of the equation and the motions theoretically exhibit a natural and forced oscillation. Because the response curve is less peaked than that for roll the pitch and heave motions are mainly in the frequency of encounter, i.e. the frequency with which the ship meets successive wave crests.

Another way of viewing the pitching and heaving motion is to regard the ship/sea system as a mass/spring system. Consider pitching. If the ship moved extremely slowly relative to the wave surface it would, at each point, take up an equilibrium position on the wave. This may be regarded as the static response of the ship to the wave and it will exhibit a maximum angle of trim which will approach the maximum wave slope as the length of the wave becomes very large relative to the ship length. In practice, the ship hasn't time to respond in this way, and the resultant pitch amplitude will be the 'static' angle multiplied by a magnification factor depending upon the ratio of the frequencies of the wave and the ship and the amount of damping present. This is the standard magnification curve used in the study of vibrations. Provided the damping and natural ship period are known, the pitching amplitude can be obtained from a drawing board study in which the ship is balanced at various points along the wave profile.

Having discussed the basic theory of ship motions, it is necessary to consider in what form the information is presented to the naval architect before proceeding to discuss motions in an irregular wave system.

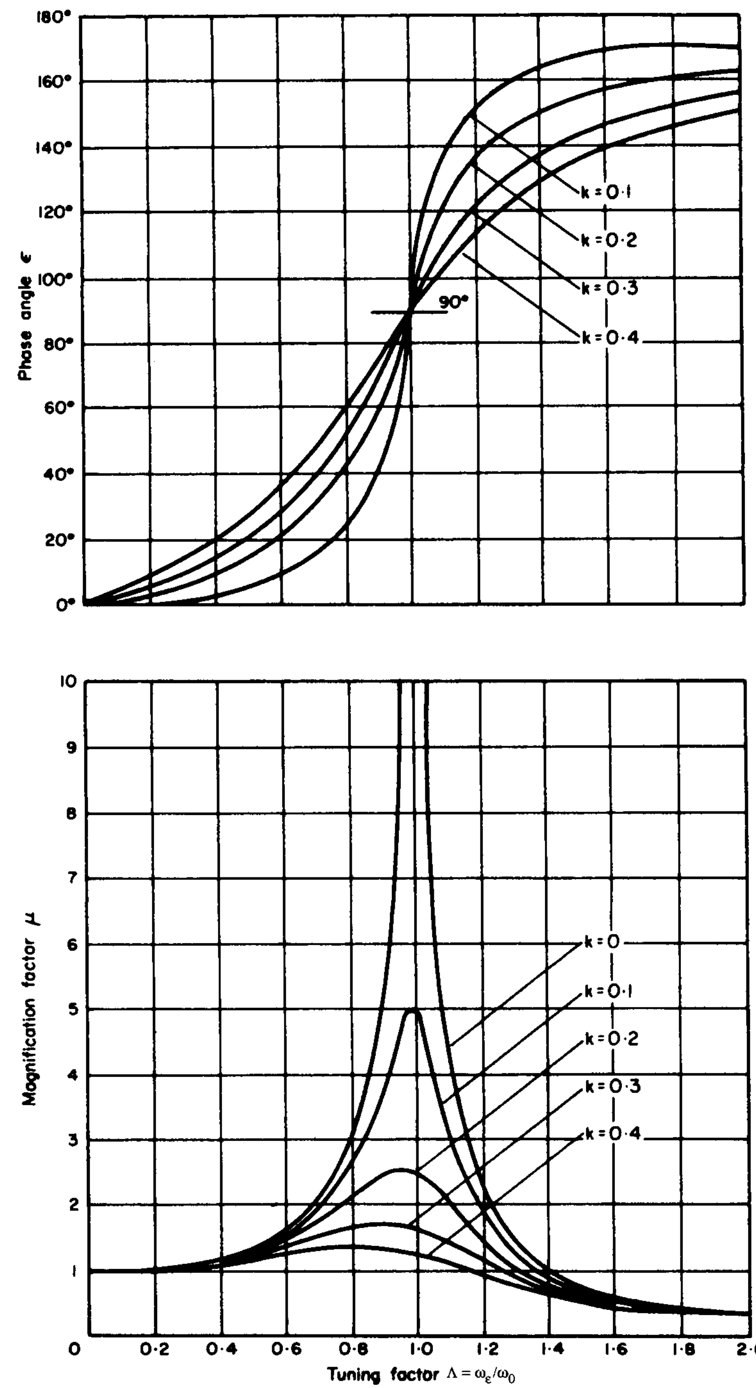


Fig. 12.5 Phase angle and magnification factor

PRESENTATION OF MOTION DATA

It is desirable that the form of presentation should permit ready application to ships of differing sizes and to waves of varying magnitude. The following assumptions are made:

(a) Linear motion amplitudes experienced by geometrically similar ships are proportional to the ratio of the linear dimensions in waves which are geometrically similar and in the same linear ratio. That is, the heave amplitude of a 200 m ship in waves 150 m long and 6 m high will be double that of a 100 m ship in waves 75 m x 3 m; $V/\sqrt{L}$ constant;
(b) Angular motion amplitudes are the same for geometrically similar ship and wave combinations, i.e. if the pitch amplitude of the 200 m ship is 2 degrees, then the pitch amplitude for the 100 m ship is also 2 degrees;
(c) For a ship in a given wave system all motion amplitudes vary linearly with wave height;
(d) Natural periods of motions for geometrically similar ships vary with the square root of the linear dimension, i.e. the rolling period of a ship will be three times that of a one-ninth scale model.

These assumptions follow from the mathematical analysis already outlined. A quite common plot for motions in regular waves, is the amplitude, expressed non-dimensionally, to a base of wave-length to ship length ratio for a series of $V/\sqrt{L}$ values. The ordinates of the curve are referred to as *response amplitude operators* (See Fig. 12.6).

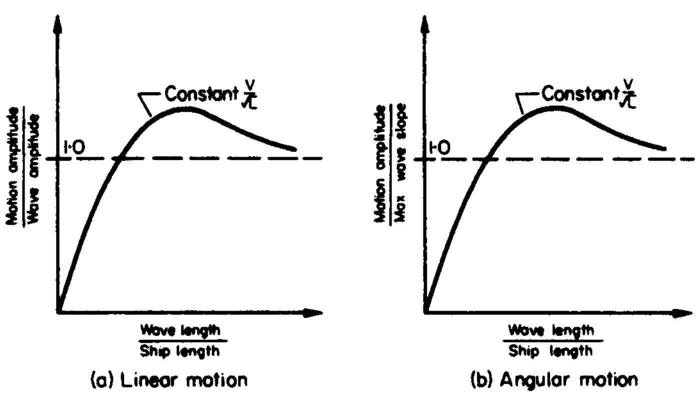


Fig. 12.6 Non-dimensional plotting

This system of plotting is non-dimensional, but a slight complication arises with angular motions when using wave spectra which are in terms of wave height. Since wave height is proportional to wave slope, the data can be presented as in Fig. 12.7 with no need to differentiate between linear and angular motions, although the curves are no longer non-dimensional.

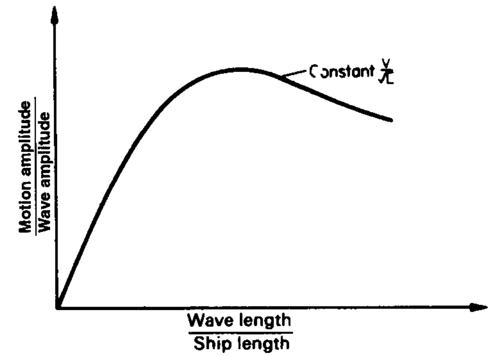


Fig. 12.7 Presentation of data for spectral analysis

Some typical response curves are reproduced in Fig. 12.8.

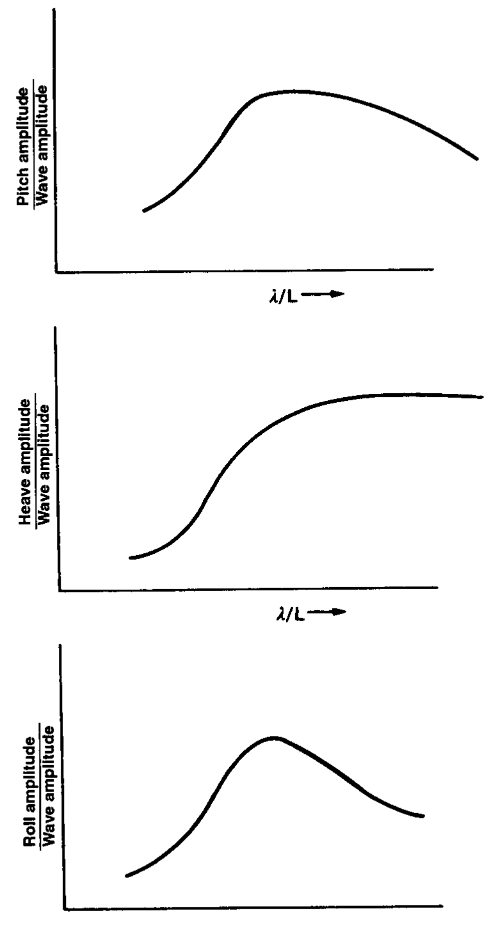


Fig. 12.8 Typical response curves

MOTION IN IRREGULAR SEAS

The foundations for the study of ship motions in irregular seas were laid in 1905 by R. E. Froude when he wrote in the context of regular wave experiments:

> 'Irregular waves such as those commonly met with at sea—are only a compound of a number of regular systems (individually of a comparatively small magnitude) of various periods, ranging through the whole gamut (so to speak) represented by our diagrams, and more. And the effect of such a compound wave series on the models would be more or less a compound of the effects proper to the individual units composing it.'

It has been seen that for regular waves the motion data can be presented in the form of response amplitude operators (RAO), for various ship speeds in waves of varying dimension relative to the ship length. Generally, a designer is concerned with a comparison of two or more designs so that, if one design showed consistently lower RAOs in all waves and at all speeds, the conclusion to be reached would be clear cut. This is not usually the case, and one design will be superior to the other in some conditions and inferior in other conditions. If it is known, using data such as that presented in Chapter 9, that on the intended route, certain waves are most likely to be met then the design which behaves better in these particular waves would be chosen.

Of more general application is the use of the concept of wave spectra. It was shown in Chapter 9 that, provided phase relationships are not critical, the apparently irregular sea surface can be represented mathematically by a spectrum of the type

$$S(\omega) = \frac{A}{\omega^5}\exp\left(-\frac{B}{\omega^4}\right)$$

where $\omega =$ circular frequency in radians per second.

A and B are constants which can be expressed in terms of the characteristic wave period and/or the significant wave height.

Since

$$\lambda = \frac{2\pi g}{\omega^2} \quad \frac{d\lambda}{d\omega} = -\frac{4\pi g}{\omega^3}$$

If λ is to be used as the base for the spectrum instead of ω then the requirement that the total spectral energy is constant leads to

$$S(\lambda) = S(\omega)\frac{d\omega}{d\lambda} = -\frac{A}{4\pi g\omega^2}\exp\left(-\frac{B}{\omega^4}\right)$$

For a known ship length the wave spectrum can be replotted to a base of λ/L to correspond to the base used above for the motion response amplitude operators. Then the wave spectrum and motion data such as that presented in Fig. 12.9 for heave can be combined to provide the energy spectrum of the motion.

Various motion parameters can then be derived from the spectral characteristics as for the waves themselves. See Chapter 9.

For example average heave amplitude $= 1.25\sqrt{m_0}$

where m_0 is the area under the heave spectrum.

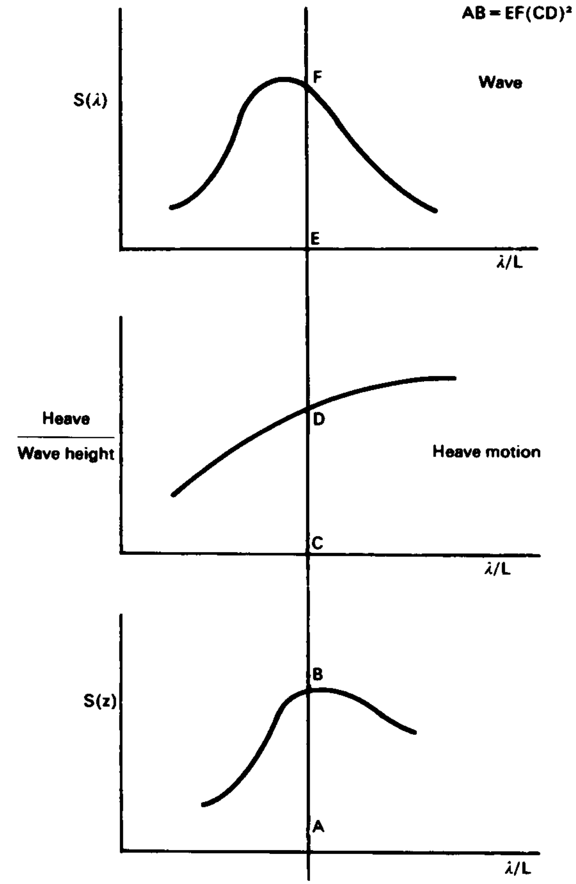


Fig. 12.9

In some cases motion data is presented to a base of frequency of encounter of the ship with the wave. The same process can be followed to arrive at the motion spectrum but noting that the wave spectrum is derived from an analysis of the variation of the surface elevation at a fixed point. In this case then it must be modified to allow for the effective or encounter spectrum as experienced by the ship.

If the ship is moving at velocity V at an angle ψ to the direction of advance of the wave system the wave spectrum as experienced by the ship is obtained by multiplying

(*a*) abscissae by $\left(1 - \frac{\omega V}{g}\cos\psi\right)$

(*b*) ordinates by $\left(1 - \frac{2\omega V}{g}\cos\psi\right)^{-1}$

When the ship is moving directly into the wave system $\cos\psi = -1$.

The effect of ship speed on the shape of the wave spectrum is illustrated in Fig. 12.10 which shows a spectrum appropriate to a wind speed of 30 knots and ship speeds of 0, 10, 20 and 30 knots.

To illustrate the procedure for obtaining the motion spectra, consider one speed for the ship and assume that the encounter spectrum for that speed is as shown in Fig. 12.11(*a*). Also, assume that the amplitude response operators for heave of the ship, at that same speed, are as shown in Fig. 12.11(*b*). The ordinate of the wave energy spectrum is proportional to the square of the amplitude of the component waves. Hence, to derive the energy spectrum for the heave motion as shown in Fig. 12.11(*c*), the following relationship is used

$$S_z(\omega_E) = [Y_{z\zeta}(\omega_E)]^2 S_\zeta(\omega_E)$$

i.e.

$$RC = (RB)^2(RA)$$

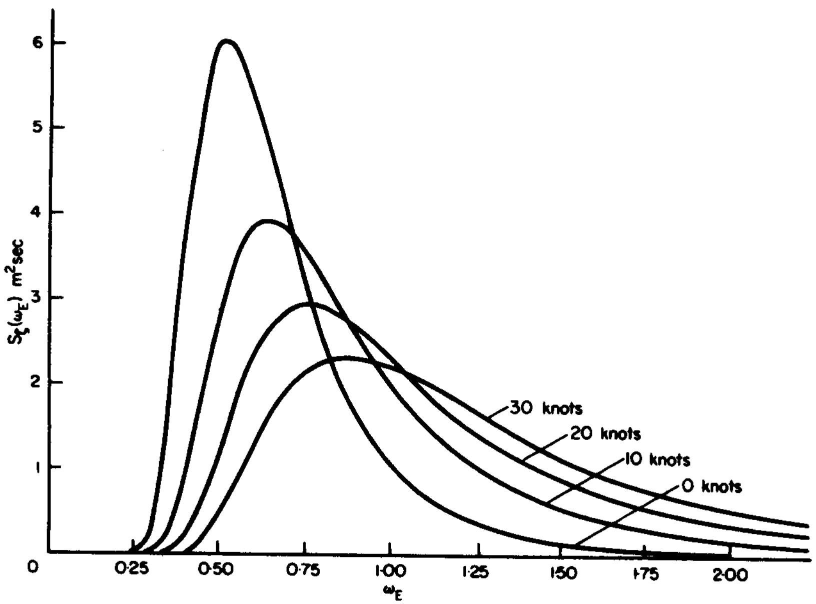


Fig. 12.10 Effect of ship speed on encounter spectrum

If the area under the motion energy spectrum is obtained by integration, the significant heave amplitude, etc., can be deduced by using the same multiplying factors as those given in Chapter 9 for waves.

For example, if m_0 is the area under the roll spectrum

$$\text{average roll amplitude} = 1.25\sqrt{m_0}$$

$$\text{significant roll amplitude} = 2\sqrt{m_0}$$

$$\text{average amplitude of } \frac{1}{10} \text{ highest rolls} = 2.55\sqrt{m_0}.$$

Any of these quantities, or the area under the spectrum, can be used to compare designs at the chosen speed. The lower the figure the better the design and the single numeral represents the overall response of the ship at that speed in that wave system. The process can be repeated for other speeds and other spectra. The actual wave spectrum chosen is not critical provided the comparison is made at constant significant wave height and not constant wind speed.

EXAMPLE 1. A sea spectrum for the North Atlantic is defined by the following table, $S_\zeta(\omega)$ being in m² s.

ω	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.1	1.2
$S_\zeta(\omega)$	0.20	2.00	4.05	4.30	3.40	2.30	1.50	1.00	0.70	0.50

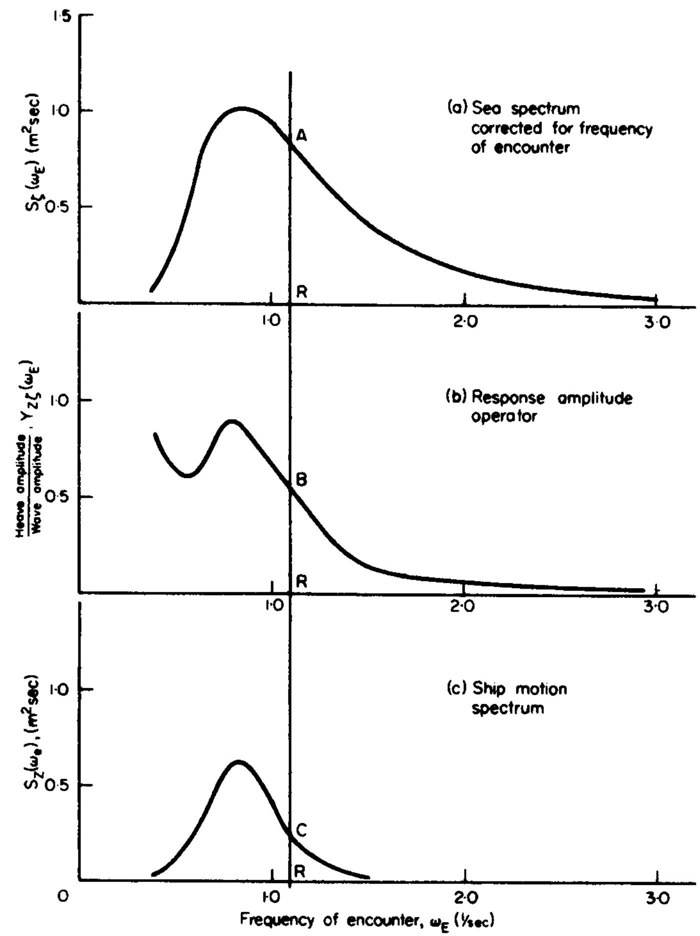


Fig. 12.11 Energy spectra and response of a ship in an irregular sea (illustrated for heave)

Calculate the encounter spectra for a ship heading directly into the wave system at speeds of 10,20 and 28 knots.

Assuming that the heave response of a ship, 175 m in length, is defined by Fig. 12.12 deduce the heave spectra for the three speeds and hence the probability curves for the motion.

Solution: It has been shown for the wave spectra, that

$$\omega_E = \omega\left(1 + \frac{\omega V}{g}\right)$$

For 10 knots;

$$V = 10 \times \frac{1852}{3600} = 5.14\,\mathrm{m/s} \quad g = 9.807\,\mathrm{m/s^2} \therefore \omega_E = \omega(1 + 0.525\omega)$$

similarly for 20 and 28 knots ω_E is equal to $\omega(1 + 1.05\omega)$ and $\omega(1 + 1.47\omega)$ respectively.

Figure 12.12 is used by calculating the wave-length appropriate to each ω value. λ and λ/L are tabulated below with the response amplitude operators from Fig. 12.12. Since curves show response at each speed the RAOs apply to the appropriate ω_E.

It has also been shown that ordinates of the spectrum must be multiplied by

$$\left(1 + \frac{2\omega V}{g}\right)^{-1} = (1 + 1.05\omega)^{-1} \text{ for 10 knots}$$
$$(1 + 2.10\omega)^{-1} \text{ for 20 knots}$$
$$(1 + 2.94\omega)^{-1} \text{ for 28 knots}$$

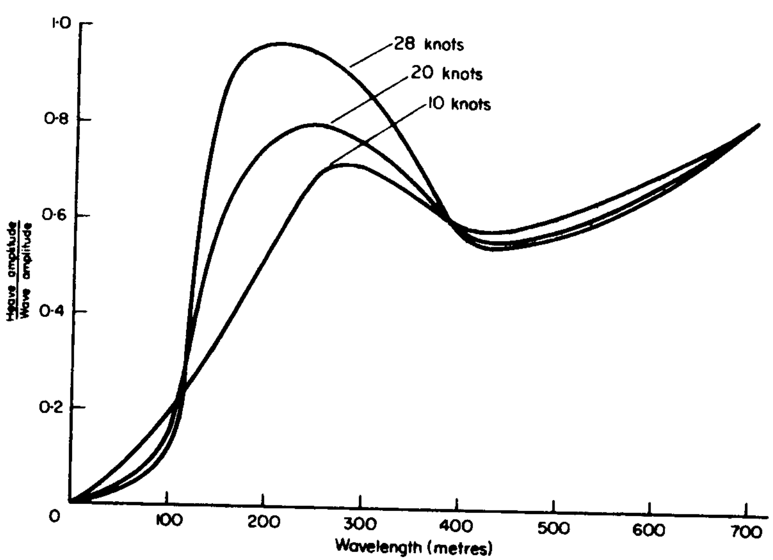


Fig. 12.12

The calculations can be carried out in tabular fashion as below for 10 knots and repeated for 20 knots and 28 knots.

ω	$1 + 0.525\omega$	ω_E	$S_\zeta(\omega)$	$1 + 1.05\omega$	$S_\zeta(\omega_E)$
0.3	1.158	0.347	0.20	1.315	0.15
0.4	1.210	0.484	2.00	1.420	1.41
0.5	1.263	0.632	4.05	1.525	2.65
0.6	1.315	0.789	4.30	1.630	2.64
0.7	1.368	0.958	3.40	1.735	1.96
0.8	1.420	1.136	2.30	1.840	1.25
0.9	1.473	1.326	1.50	1.945	0.77
1.0	1.525	1.525	1.00	2.050	0.49
1.1	1.578	1.736	0.70	2.155	0.32
1.2	1.630	1.956	0.50	2.260	0.22

			RAO		
ω	λ(m)	λ/L	10 knots	20 knots	28 knots
0.3	689	3.97	0.80	0.80	0.80
0.4	387	2.23	0.60	0.60	0.60
0.5	247	1.42	0.69	0.80	0.95
0.6	171	0.985	0.44	0.69	0.93
0.7	126	0.730	0.28	0.40	0.29
0.8	96.6	0.556	0.18	0.15	0.10
0.9	76.5	0.440	0.12	0.08	0.05
1.0	61.6	0.354	0.10	0.06	0.04
1.1	51.2	0.295	0.08	0.05	0.03
1.2	43.0	0.250	0.07	0.04	0.03

The ordinates of the heave motion spectrum at each speed are obtained by multiplying the wave spectrum ordinate by the square of the RAO as in the table below:

	10 knots			20 knots			28 knots		
ω	$S_\zeta(\omega_E)$	*RAO*	$S_Z(\omega_E)$	$S_\zeta(\omega_E)$	*RAO*	$S_Z(\omega_E)$	$S_\zeta(\omega_E)$	*RAO*	$S_Z(\omega_E)$
0.3	0.15	0.80	0.098	0.12	0.80	0.079	0.11	0.80	0.068
0.4	1.41	0.60	0.508	1.09	0.60	0.392	0.92	0.60	0.330
0.5	2.65	0.69	1.262	1.98	0.80	1.270	1.64	0.95	1.48[illegible]
0.6	2.64	0.44	0.511	1.91	0.69	0.910	1.56	0.93	1.350
0.7	1.96	0.28	0.153	1.38	0.40	0.207	1.11	0.29	0.093
0.8	1.25	0.18	0.041	0.86	0.15	0.019	0.69	0.10	0.007
0.9	0.77	0.12	0.011	0.52	0.08	0.003	0.41	0.05	0.001
1.0	0.49	0.10	0.005	0.32	0.06	0.001	0.25	0.04	—
1.1	0.32	0.08	—	0.21	0.05	—	0.17	0.03	—
1.2	0.22	0.07	—	0.14	0.04	—	0.11	0.03	—

The heave spectra can now be plotted and the areas under each obtained to give m_0. Values of m_0 so deduced are

10 knots: $m_0 = 0.37$ and $\sqrt{(2m_0)} = 0.86$

20 knots: $m_0 = 0.62$ and $\sqrt{(2m_0)} = 1.11$

28 knots: $m_0 = 0.72$ and $\sqrt{(2m_0)} = 1.20$

The probability that at a random instant of time the heave exceeds some value z is given by $P(z) = 1 - \mathrm{erf}(z/\sqrt{2m_0})$. The error function, erf, is obtained from standard mathematical tables.

MOTION IN OBLIQUE SEAS

The procedure outlined above for finding the motion spectra can be applied for the ship at any heading provided the appropriate encounter spectrum is used and the response amplitude operators are available for that heading.

In a regular wave system, as the ship's course is changed from directly into the waves, two effects are introduced, viz.:

(a) the effective length of the wave is increased and the effective steepness is decreased;

(b) the frequency of encounter with the waves is decreased as already illustrated.

An approximation to motions in an oblique wave system can be obtained by testing in head seas with the height kept constant but length increased to $\lambda/\cos\psi$; and with the model speed adjusted to give the correct frequency of encounter. This is a reasonable procedure for vertical motions but it is only an approximation.

SURGE, SWAY AND YAW

As already explained, these motions exhibit a different character from that of roll, pitch and heave. They are not subject to the same theoretical treatment as these oscillatory motions but a few general comments are appropriate.

Surge

At constant power in still water a ship will move at constant speed. When it meets waves there will be a mean reduction in speed due to the added resistance and changed operating conditions for the propeller. The speed is no longer constant and the term surge or surge velocity is used to define the variation in speed about the new mean value, Several effects are present. There is the orbital motion of the wave particles which tends to increase the speed of the ship in the direction of the waves at a crest and decrease it in a trough. In a regular wave system, this speed variation would be cyclic in the period of encounter with the waves. In an irregular sea, the height and hence the resistance of successive waves varies giving rise to a more irregular speed variation. This is superimposed upon the orbital effect which is itself irregular in this case. The propellers will also experience changing inflow conditions due to the waves and the ship's responses. The thrust will vary, partly depending upon the dynamic characteristics of the propulsion machinery and transmission system. The resulting surge is likely to be highly non-linear.

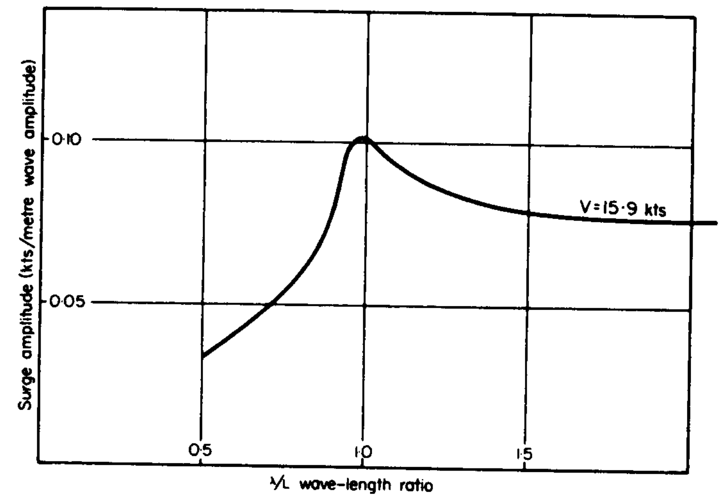


Fig. 12.13 *Surging in head sea*

The surge experienced by a vessel of length 146.15m is shown in Fig, 12.13.

The maximum response occurs in waves approximately equal in length to the ship. In waves of this length and 5 m high, the speed oscillation is about ±0.25 knots. The effect varies approximately linearly with speed.

Sway

When the wave system is other than immediately ahead or astern ofthe ship, there will be transverse forces arising from similar sources to those causing the surging motion. In a regular sea, these would lead to a regular motion in the period of encounter with the waves but, in general, they lead to an irregular athwart ships motion about a mean sideways drift. This variation about the mean is termed sway. It is also influenced by the transverse forces acting on the rudder and hull due to actions to counteract yaw which is next considered.

Yaw

When the wave system is at an angle to the line of advance of the ship the transverse forces acting will introduce moments tending to yaw the ship. Corrective action by the rudder introduces additional moments and the resultant moments cause an irregular variation in ship's heading about its mean heading. This variation is termed yawing. In a regular sea with an automatic rudder control system, the motion would exhibit a regular period depending on the period of encounter and the characteristics of the control equation (see Chapter 13). In general, however, the motion is quite irregular.

Some of the difficulty of maintaining course in rough weather is indicated in Fig. 12.14 which is for a ship of 146m.

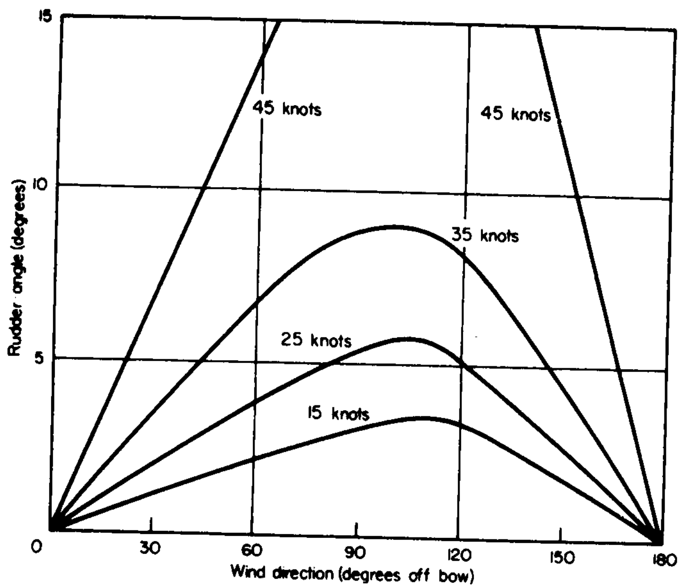


Fig. 12.14 Rudder angles for different wind speeds and directions

Large amplitude rolling

Linear theory shows that large angles of roll can occur when the wave encounter frequency of a beam sea is close to the ship's natural frequency of roll. The amplitude reached will depend upon the degree of damping and whether any stabilizing devices, such as active fins, are employed.

Linear theory asumes a steady metacentric height but when a ship is moving through waves this height is a dynamic quantity not a static one. As the wave surface moves along the length of the ship the shape of the underwater form changes, particularly at the bow and stern, an effect accentuated by heave and pitching motions. These changes lead to variations in the effective metacentric height. When a ship is in a following sea metacentric height variations are long period. Particularly in ships with flat transom sterns there may be a loss of stability and the resulting roll amplitudes can be very large.

Another non-linear effect which causes rolling occurs when the dominant encounter period approximates half the natural period of roll in head or following seas. If associated with fairly large stability variations, large roll angles can result. This phenomenon is often called *half cycle* or *parametric* rolling. It starts quite unexpectedly and quickly reaches very large amplitudes. Model tests, conducted at MARIN, on a 240m cruise ship suffered roll amplitudes of 40 degrees. It was found that below a certain wave height threshold the rolling was negligible, above the threshold a fairly regular roll motion builds up. The threshold wave height depends upon the ship's heading, the peak period and ship's speed. Zero speed proved the most severe test condition. In following seas a significant wave height as low as 2 m was sufficient to trigger

the rolling. In head seas the threshold was 2.75 m. Above the threshold th. effect of increasing wave height was dramatic. The threshold wave heigh increased with increasing ship speed.

Limiting seakeeping criteria

The ability of a ship to carry out its intended mISSIOnefficiently may be curtailed by a number of factors. There is a correspondingly wide range oj limiting seakeeping criteria. The limit may be set by the ability of the ship itself or its systems, to operate effectively and safely, or by the comfort or proficiency of passengers or crew. In so far as equipment or personnel performance is degraded when motions (e.g. vertical acceleration) exceed a certain level, careful siting of the related activity within the ship in an area of lesser motions may extend the range of sea conditions in which operation is acceptable. Other features such as slamming or propeller emergence are dependent on overall ship geometry and loading although here again the design of the ship (e.g. its inherent strength in the case of slamming) can determine the acceptable level before damage occurs or conditions become unsafe.

There is a potential danger in applying 'standard' acceptance levels of any criterion to a new design. There must be a judicious choice, both of criteria and acceptance levels, to reflect the particular design, its function and its similarity to previous designs for which operating experience is available. Thus a new design may have been specifically strengthened forward to enable it safely to withstand high slamming loads. Nevertheless guideline figures applicable to general ship types are useful in preliminary design development. Some performance parameters can be assessed in different ways. This may lead to different absolute values of criteria. Hence in using criteria values it is important they be computed for a new design using the same method as that adopted in establishing the acceptable levels.

It is now appropriate to review briefly the seakeeping parameters most frequently used as potential limiting criteria. They are speed and power in waves, slamming, wetness, propeller emergence and impairment of human performance. ~

SPEED AND POWER IN WAVES

As a wave system becomes more severe, the power needed to drive the ship through it at a given speed increases. The difference arises mainly from the increased resistance experienced by the hull and appendages, but the overall propulsive efficiency also changes due to the changed conditions in which the propeller operates. If the propulsion machinery is already producing full power, it follows that there must be an enforced reduction in speed. Past a certain severity of waves, the motions of the ship or slamming may become so violent that the captain may decide to reduce speed below that possible with the power available. This is a voluntary speed reduction and might be expected to be made in merchant ships of fairly full form at Beaufort numbers of 6 or more. **The** speed reduction lessens as the predominant wave direction changes from directly ahead to the beam (Fig. 12.15).

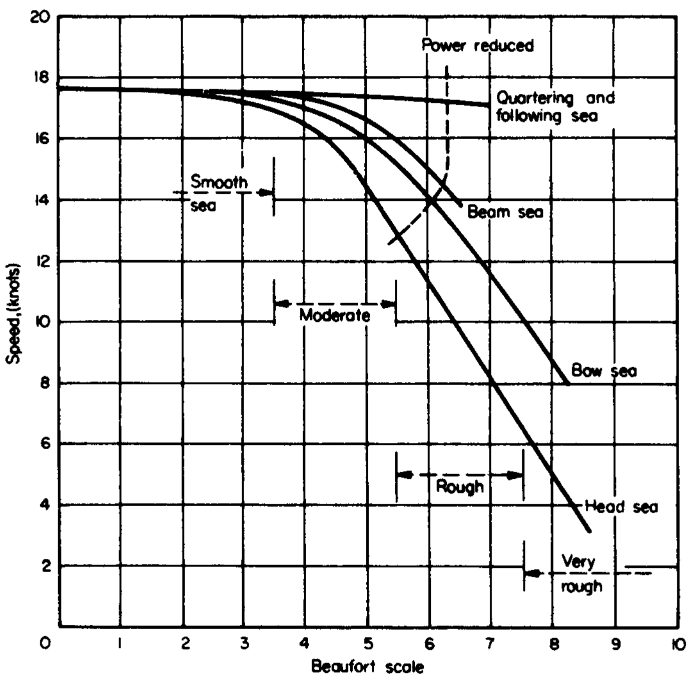


Fig. 12.15

Figure 12.16 shows how the power required for various speeds increases with increasing sea state as represented by the Beaufort number. The figure applies to a wave system 10 degrees off the bow and to a ship 150 m long with a longitudinal radius of gyration equal to 22 per cent of the length. Decreasing the longitudinal moment of inertia decreases the additional power required and also results in drier decks forward.

Figure 12.17 shows the reduction in speed which occurs at constant power (5.83 MW) for the ship in the same conditions and shows the significance of varying the longitudinal radius of gyration. The effect of the variation is less significant in large ships than in small. It is associated with a reduction in natural pitching period.

Other ship design features conducive to maintaining higher speed in rough weather are a low displacement-length ratio, i.e. $\Delta/(L/100)^3$, and fine form forward. Increased damping by form changes or the deliberate introduction of a large bulbous bow can also help. When it is realized that the passage times of ships in rough weather may be nearly doubled, it is clearly of considerable importance to design the ship, both above and below water, so that it can maintain as high a speed as possible. Wetness is a significant factor influencing the need to reduce speed, and this is dealt with later.

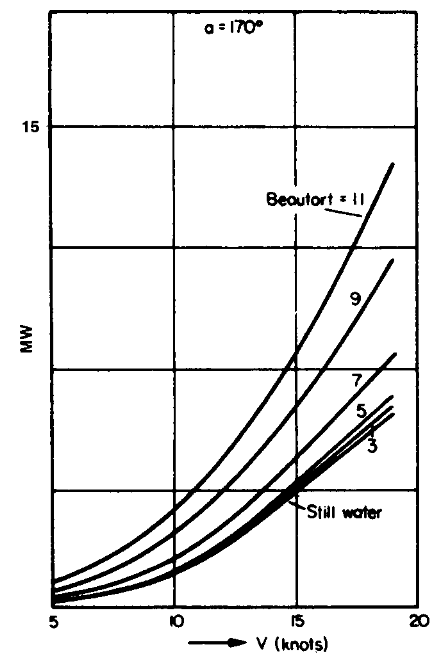


Fig. 12.16 Power in waves for a 150 m long slip

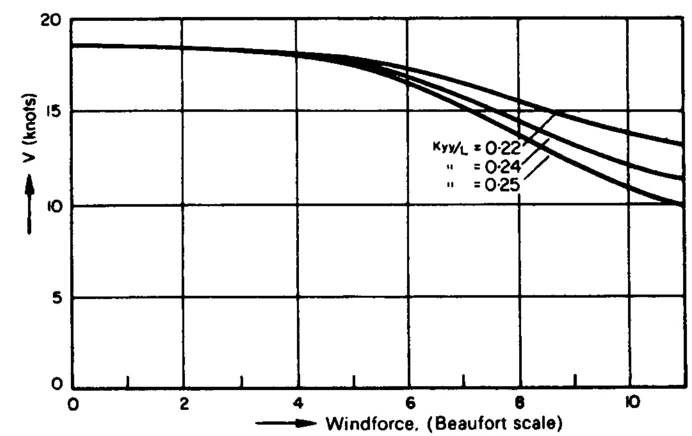


Fig. 12.17 Variation in speed at constant power

SLAMMING

Slamming is a high frequency transient vibratory response of a ship's hull to wave impact. It occurs at irregular intervals. Each blow causes the ship to shudder and is followed by a vibration of the ship's structure. The impact may be large enough to cause physical damage to the ship, the most vulnerable area being between 10 and 25 per cent of the ship's length from the bow. It is the possibility of this damage occurring that causes an experienced captain to reduce speed when his ship begins to slam badly. This always leads to reduced severity of slamming. Lightly loaded cargo ships are particularly liable to slam and the enforced speed reduction may be as much as 40 per cent. Slamming is likely to occur when the relative velocity between the ship's bottom and the water surface is large (usually when the ship is nearly level and the bow has greatest downward velocity); the bow has emerged from and is re-entering the water with a significant length of the bottom roughly parallel to the local water surface; there is a low rise of floor forward increasing the extent of the ship's bottom parallel with the local sea surface. A high relative velocity between the waves and a heavily flared bow can cause a similar, but generally less severe, effect. The heavy flare may throw the water clear of the bow reducing green seas but possibly increasing spray.

The slamming impact lasts for about 1/30 s and does not perceptibly modify the downward movement of the bow. It can be detected as a disturbance in the acceleration record and by the ensuing vibration which can last for about 30 s. Slamming pressures as high as 0.7Njmm^2 have been recorded in low speed ships. Pressures in high speed ships are generally less because of their finer form forward. The pressure can be shown to be proportional to the square of the relative velocity of impact and inversely proportional to the square of the tangent of the angle of deadrise (Fig. 12.18). This is based on analyses of the similar problem of a seaplane landing.

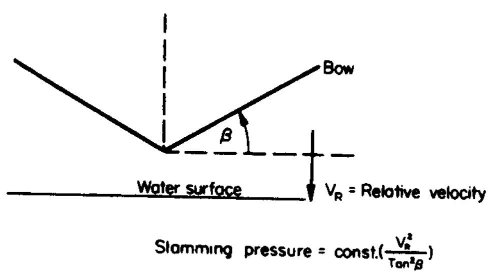


Fig. 12.18 *Slamming pressure*

When (3 is very small, the pressures obtained would tend to infinity on the above simple analysis. In practice, pressures are limited by the elastic response of the ship structure. The peak pressure moves higher in the ship section as the bow immersion increases.

If, in a given case, the conditions for high pressure apply over a limited area, the blow is local and may result only in local plate deformation. If the pressure acts over a larger area, the overall force acting on the ship is able to excite vibrations of the main ship girder. Since the duration of this vibration, typically of the order of 30 s, is long compared with the stress cycle induced by the main wave, the stress record will be as in Fig. 12.19 assuming the slam occurs at time *To*· The primary ship girder stresses may be increased by 30 per cent or more.

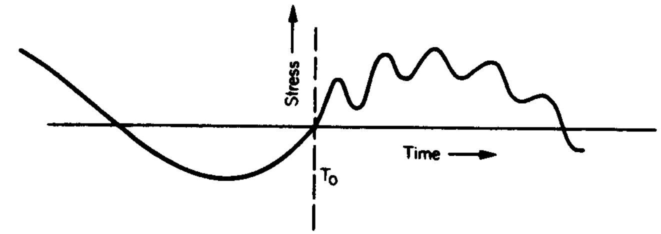


Fig. 12.19 Slamming vibration superimposed on regular stress cycle

In considering the limiting seakeeping criteria associated with slamming, it is assumed that the relative motion between the ship and wave surface is sinusoidal.

$$r = r_0 \sin(\omega t - \delta), r_0 = \text{peak relative motion}$$

differentiating

$$\dot{r} = \omega r_0 \cos(\omega t - \delta)$$

If a slam occurs at time $t = 0$

$$\delta = \sin^{-1}\left[-\frac{T}{r_0}\right], T = \text{local draught}$$

Assuming the probability of exceeding a peak relative motion of r_0 follows a Rayleigh distribution

$$P_r(r \geq r_0) = \exp\left(-\frac{r_0^2}{2m_0}\right), m_0 = \text{variance of relative motion}$$

Assuming that a peak relative motion close to r_0 occurs once in N oscillations

$$P_r = \frac{1}{N} = \frac{2\pi}{T_s\omega}, T_s = \text{arbitrary sample time}$$

$$\therefore\ r_0 = \left\{-2m_0 \log_e\left[\frac{2\pi}{T_s\omega}\right]\right\}^{\frac{1}{2}}$$

The slamming pressure is given by

$$p = \frac{1}{2}\rho\dot{r}^2 F(\beta), \beta = \text{local deadrise angle}$$

An average plot for $F(\beta)$ against β is given in Fig. 12.20. The pressure is limited to $\rho C_w \dot{r}$ where C_w is the velocity of sound in water.

The total force acting on unit length of the ship at a given point is the sum of the forces on the flat of keel and that on the two sides of the hull with deadrise. In each case the force arises from a pressure which is assumed to reach its peak

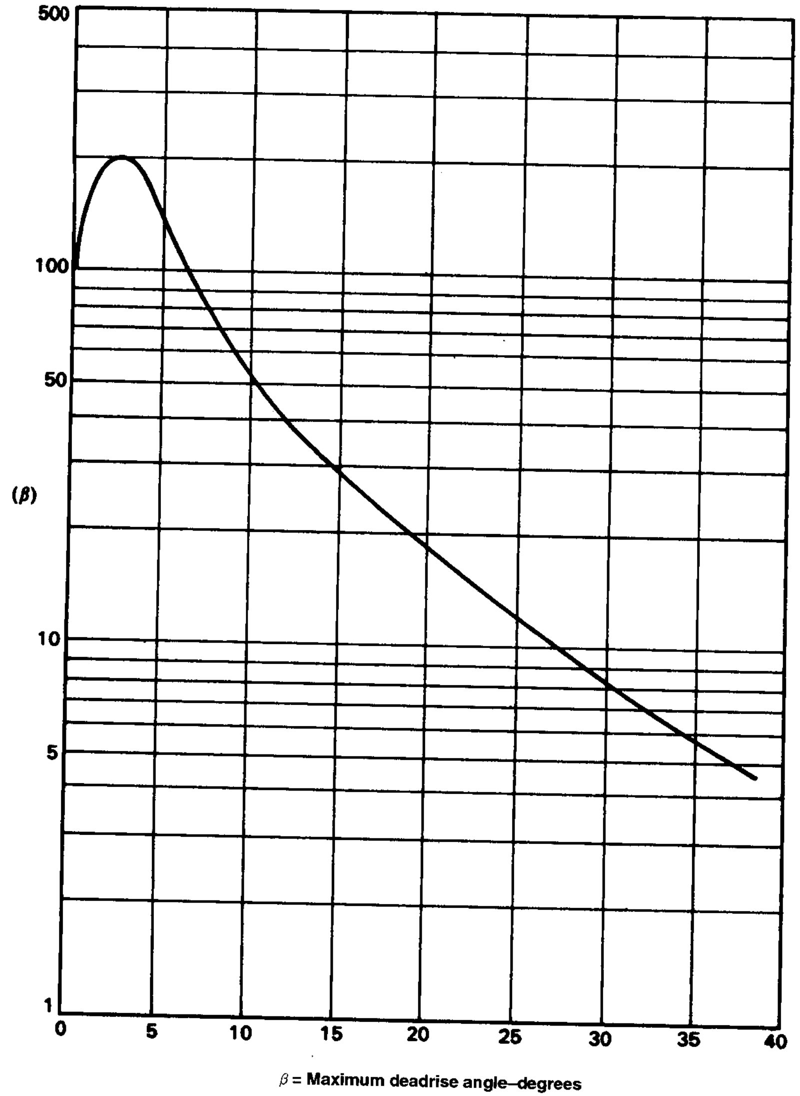


Fig. 12.20 Slamming pressure coefficient

instantaneously and then decay exponentially. Little is known about the longitudinal distribution of impact loads but a conservative approach is to assume simultaneous impact at hull sections for which keel emergence is predicted in an arbitrary sample period.

WETNESS

By wetness is meant the shipping of spray or green seas over the ship and, unless otherwise qualified, refers to wetness at the bow.

It is not possible at present to calculate wetness accurately but it may be assessed by:

(*a*) calculating the relative vertical movement of the bow and water surface and assuming that the probability of deck wetness is the same as that of the relative motion exceeding the freeboard at the stem head;
(*b*) running a model in waves and noting for each of a range of sea conditions the speed at which the model is wet and assuming that the ship will behave in a similar way. Model tests at the right F_n can represent green seas but not spray effects.

Methods based on (*a*) are usually adopted. If it is valid to assume that the relative motion between ship and waves is sinusoidal and that the probability of deck immersion follows a Rayleigh distribution:

$$P_r = \exp\left[-\frac{F^2}{2m_0}\right], F = \text{freeboard}$$

The average time interval between the deck being wet at a given station is

$$t_w = \frac{2\pi}{P_r\omega} = \frac{2\pi}{\omega}\exp\left[\frac{F^2}{2m_0}\right]$$

where ω = average frequency of the relative motion

$$= \sqrt{\frac{m_2}{m_0}}$$

m_2 = variance of the relative velocity

Besides trying to reduce the incidence of wetness the naval architect should:

(*a*) design decks forward so that water clears quickly;
(*b*) avoid siting forward any equipments which may be damaged by green seas or which are adversely affected by salt water spray.

A bulwark can be fitted to increase freeboard provided it does not trap water. The sizes of freeing ports required in bulwarks are laid down in various international regulations.

PROPELLER EMERGENCE

Using an arbitrary assumption that the propeller should be regarded as having emerged when a quarter of its diameter, *D*, is above water, criteria corresponding to those used in wetness follow, viz:

$$P_r = \exp\left[-\left(T_p - \frac{D}{4}\right)^2 \Big/ 2m_0\right]$$

T_p = depth of propeller boss below the still waterline

Average time interval between emergencies

$$t_p = \frac{2\pi}{P_r\omega}, \quad \omega = \sqrt{\frac{m_2}{m_0}}$$

DEGRADATION OF HUMAN PERFORMANCE

Besides reducing comfort, motions can reduce the ability and willingne: humans to work and make certain tasks more difficult. Thus in contro machinery, say, motions may degrade the operator's ability to decide wha should do and, having decided, make the execution of his decision r difficult. There is inadequate knowledge of the effects of motion on hu behaviour but in broad terms it depends upon the acceleration experienced its period. These can be combined in a concept of subjective motion. In this combinations of acceleration and frequency are determined at which the : jects feel the motion to have the same intensity. Denoting this level as subjec magnitude (SM) with a value of 10, other combinations of acceleration frequency are assessed as of $SM = 10n$ when they were judged to be n time intense as the original base SM. It is found that

$$SM = A(f)a^{1.43}, a = \text{acceleration amplitude in 'g'}$$

With the frequency, f, in Hz, $A(f) = 30 + 13.53(\log_e f)^2$

Assuming the sinusoidal results can be applied to random motions

$$a = \frac{2}{g}\sqrt{m_{4a}}, \quad f = \frac{1}{2\pi}\sqrt{\frac{m_{2a}}{m_{0a}}}$$

where m_{0a}, m_{2a} and m_{4a} are the variances of the absolute motion, velocity : acceleration respectively in SI units.

Then

$$SM = \left[3.087 + 1.392\left\{\log_e \frac{1}{2\pi}\sqrt{\frac{m_{2a}}{m_{0a}}}\right\}^2\right] m_{4a}^{0.715}$$

The motions experienced by any individual will depend upon their position in the ship. An overall figure for a ship can be obtained by applying a weighting curve representing the distribution of personnel in the ship. Alternatively for a localized activity (e.g. on the bridge) the SM for that one location can be obtained. Unfortunately no clear-cut limiting SM could be proposed although a figure of 15 has been suggested as an absolute maximum. Five actions a designer can take to prevent or mitigate the adverse affects of ship motion, especially sea sickness, are

(a) Locate critical activities near the effective centre of rotation.
(b) Minimize head movements.
(c) Align operator position with the ship's principal axes.
(d) Avoid combining provocative sources.
(e) Provide an external visual frame of reference.

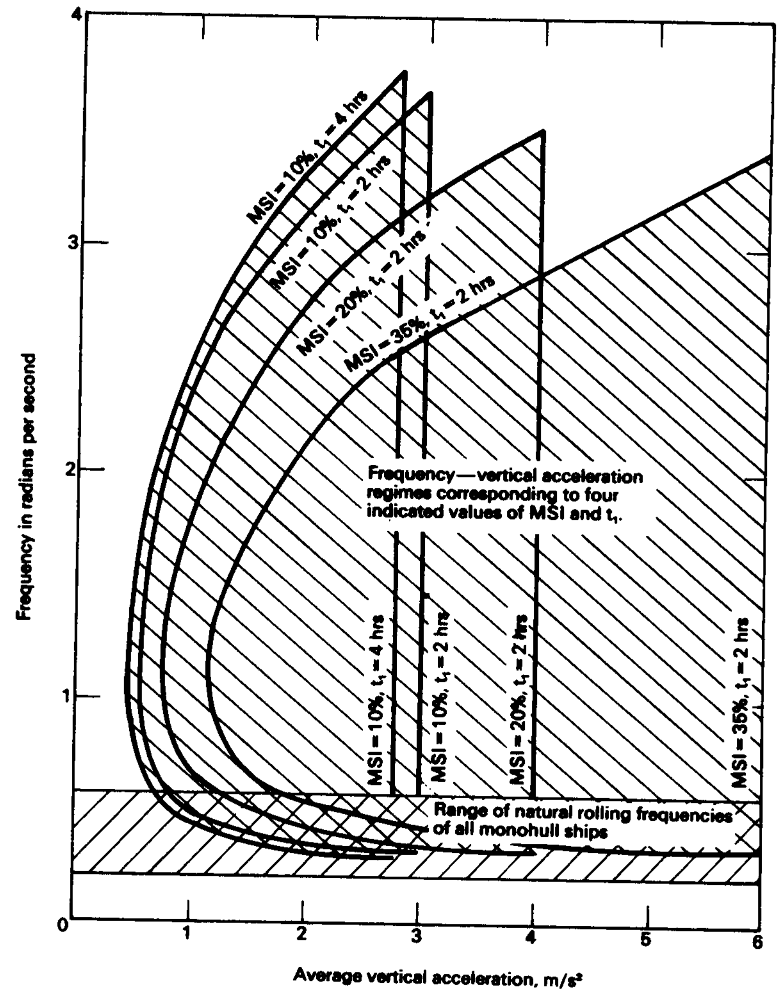


Fig. 12.21 Motion sickness index

Another approach to the effect of motions on personnel is the concept of *motion sickness incidence* (MSI). The MSI is the percentage of individuals likely to vomit when subject to the given motion for a given time. Plots can be made as in Fig. 12.21. The limitations of this approach are that the data relate to unacclimatized subjects, sinusoidal motions and the fact that human performance may be degraded long before vomiting occurs.

Overall seakeeping performance

A number of possible limiting seakeeping criteria have just been discussed. Their variety and the range of sea conditions expected in service mean that no single performance parameter is likely to be adequate in defining a design's overall seakeeping performance. This applies even within the restricted definition of seakeeping adopted in this chapter. However a methodology is developing which permits a rational approach, the steps of which are now outlined.

(a) The *sea states* in which the ship is to operate are established. The need may be specific in the sense that the ship will operate on a particular route at certain seasons of the year, or it may be as general as world-wide operations all the year round. Ocean wave statistics can be used to determine the ranges of wave height, period and direction likely to be met for various percentages of time. As described later, the technique of wave climate synthesis can be used to improve the reliability of predictions based on observed wind and wave data. This establishes the number of days a year the ship can be expected to experience various wave conditions and these can be represented by appropriate wave spectra, e.g. by adopting the formulation recommended by the ITTe.

(b) The *ship responses* in the various sea states can be assessed from a knowledge of its responses in regular waves. Even in long-crested seas the ship response depends upon the severity of the sea, the ship speed and the ship's heading relative to the wave crest line. Thus motions can be represented by a polar diagram, such as Fig. 12.22, in which contours are drawn for given values of response for each of a range of significant wave heights. Assuming a linear dependency the contours can be expressed as response operators.

If it is desired to compare designs on the basis of their relative motions at various speeds (or Froude numbers) the areas within the polar plot can be

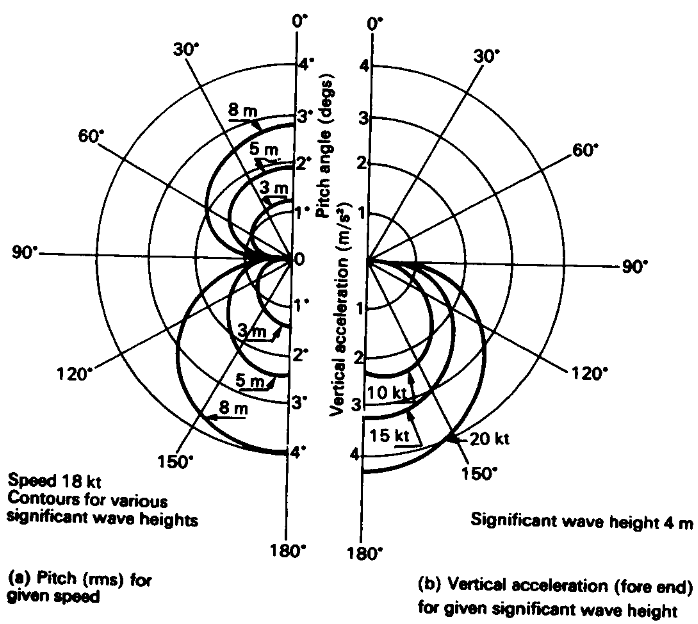


Fig. 12.22 Typical polar diagrams

used. This will average out variations with heading. If it is known that tho ship will transit on some headings more frequently than others the polar plots can be adjusted by means of suitable weighting factors. For vessels which are symmetrical about their centre-line plane in geometry and loading the polar plots will be symmetrical about the vertical axis.

(c) *Limiting conditions.* It is not usually the motion amplitudes *per se* which limit the ability of a ship to carry out its intended mission. More often it is a combination of motions and design features leading to an undesirable situation which can only be alleviated by reduction in speed or a change of heading. That is to say the ship's freedom of action is restricted.

The usual action is to reduce speed as this has the effect of avoiding synchronism with wave components other than short waves which have less effect on motions anyway. A change of course is only effective when there is a predominant wave direction and often can only be adopted for relatively short periods of time.

Various limiting seakeeping criteria have been discussed above. For any chosen criterion, the speed above which the agreed acceptable limit of the criterion is expected to be exceeded can be plotted on a polar diagram. As with motions, the area within the plot, adjusted if necessary by weighting

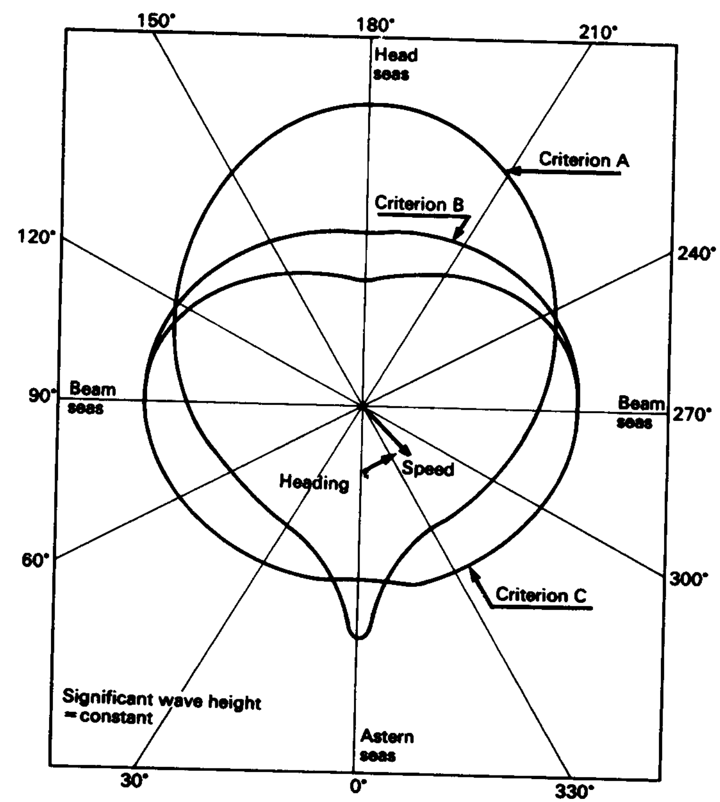


Fig. 12.23 Seakeeping speed polar diagram

factors for different headings, can be used as an overall measure of the design's performance in terms of the selected criterion in the given sea state. The greater the area the greater the range of speeds and headings over which the vessel can operate.

Plots, and hence areas, will be required for each wave spectrum of interest, each of which can be characterized by its significant wave height.

Another concept is that of a *stratified measure of merit.* The time the ship is expected to encounter various wave conditions has been established from ocean wave statistics. The product of this time and the area under the polar plot for the chosen seakeeping criterion is plotted against the significant wave height appropriate to the wave spectrum. The area under the resulting curve represents the overall performance of the design with respect to the selected criterion, over the period of time covered by the wave data used. This could be a single voyage, a year or the whole life of the ship.

(d) The *operational* ability of a design will not always be limited by the same criterion. Thus the design's overall potential must be assessed in relation to all the possible limiting criteria. If all criteria of interest are plotted on a common polar plot the area within the inner curve at each heading represents the overall limiting performance of that design in the selected sea state. The measure of merit concept can be used as for a single criterion.

This operational ability assumes a common mission throughout the life of the ship. If it is known in advance that the ship will have different missions at different times the method will need to be modified to reflect the different influence of the various criteria on the missions concerned.

In practice a captain must judge the operational importance of maintaining speed against the risk to the ship. The captain of a warship is more likely to reduce speed on a peacetime transit than in a wartime operation.

This general approach is one method of assessing the relative seagoing performance of competitive vehicle types. Other 'scoring' methods suggested are:

(a) The percentage time a given vehicle in a given condition of loading can perform its function in a specified area, in a given season at a specified speed without any of a range of chosen seakeeping criteria exceeding agreed values.
(b) The time a vehicle needs to transit between two specified locations in calm water divided by the time the vehicle would require to travel between the same locations in rough weather without any of the selected criteria value being exceeded.

Acquiring data for seakeeping assessments

It will be appreciated from the foregoing that two things are necessary to enable an assessment to be made of seakeeping performance, viz. a knowledge of:

(a) wave conditions for the area to which the assessment is related and specifically how the total energy of the wave system is distributed with respect to frequency;
(b) the responses of the ship in regular sinusoidal waves covering the neccKlmry frequency band. These responses are normally defined by the appropriutc response amplitude operators in the form ofresponse per unit wave height.

SELECTION OF WAVE DATA

Chapter 9 gave information on the type of data available on sea conditions likely to be met in various parts of the world. Much of this is based on visual observations, both of waves and winds. As such they involve an element of subjective judgment and hence uncertainty. In particular visual observations of wave periods are likely to be unreliable. Care is necessary therefore in the interpretation and analysis of wave data if sound design decisions are to be derived from them.

The National Maritime Institute, (now BMT Ltd), has developed a method, known as wave climate synthesis, for obtaining reliable long-term wave data from indirect or inadequate source information. This approach can be used when instrumented wave measurements are not available. Essentially relationships derived from corresponding sets of instrumented and observed data are used to improve the interpretation of observed data. Various sources of data and of methods of analysing it are available such as the Marine Information and Advisory Service of the Institute of Oceanographic Sciences and agencies of the World Meteorological Organisation. Much of the data is stored on magnetic tape.

The NMI analysis uses probabilistic methods based on parametric modelling of the joint probability of wave height and wind speed. Important outputs are:

(a) Wave height

When a large sample is available raw visual data provide reasonable probability distributions of wave height. However, comparisons of instrumented and visual data show that better distributions can be derived using best fit functional modelling to smooth the joint probability distributions of wave height and wind speed. This is illustrated in Fig. 12.24 for OWS *India* in which the 'NMIMET Visual' curve has been so treated.

Analysis of joint probabilities for wave height and wind speed from me'\r ured data leads to the relationship

$$\text{Mean wave height} = H_\mathrm{r} = \left[(aW_\mathrm{r}^{\mathrm{n}})^2 + H_2^2\right]^{\frac{1}{2}},\ W_\mathrm{r} = \text{wind speed.}$$

Standard deviation of the scatter about the mean is

$$\sigma_\mathrm{r} = H_2(b + cW_\mathrm{r})$$

The joint probability distribution is given by a gamma distribution

$$P(H_\mathrm{s}/H_\mathrm{r}, \sigma_\mathrm{r}) = \frac{q^{\mathrm{p}+1}}{\Gamma(p+1)} H_\mathrm{s}^{\mathrm{p}} \exp(-qH_\mathrm{s})$$

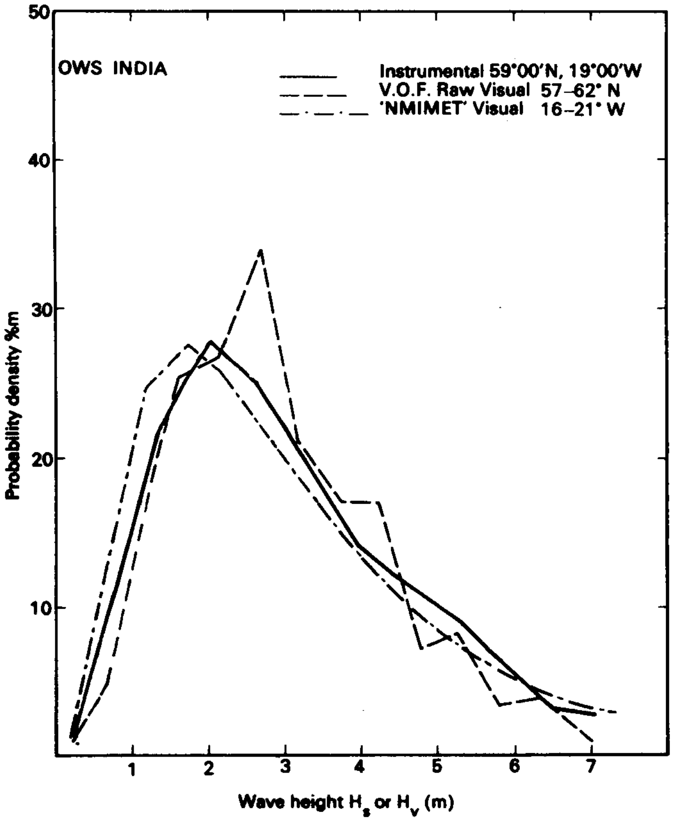


Fig. 12.24 Visual and measured wave height probabilities

$$\text{where } p = \frac{H_r^2}{\sigma_r^2} - 1$$

$$q = H_r/\sigma_r^2$$

$$H_s = \text{significant wave height}$$

H_2, a, b, c and n are the model parameters for which, in the absence of more specific data, suitable standard values may be used. The following values have been recommended on the basis of early work using instrumental data from a selection of six stations. In quoting them it should be noted that they are subject to review in the light of more recent work and meanwhile should be regarded as only valid for use with measured wind speeds up to a limit of about 50 knots. It should also be noted that the numerical values cited are to be used in association with units of metres for wave height and knots for wind speed.

	H_2 (metres)	a	b	c	n
Open ocean	2.0	0.033	0.5	0.0125	1.46
Limited fetch	0.5	0.023	0.75	0.0188	1.38

The wave height probabilities follow from the wind speed probabilities using:

$$P(H_s) = \sum_r (H_s/H_r, \sigma_r) \times P(W_r)$$

Wave directionality data can be obtained if the joint probability distributions of wave height and wind speed are augmented by corresponding joint probabilities of wave height and period and wind speed and direction.

(b) Wave periods

Reliability of visual observations of wave period is poor. NMI adopted a similar approach to that used for wave height but using a different functional representation. Based on analysis of instrumented wave height/period data, wave height and period statistics can be synthesized when reliable wave height data are available using:

$$P(T) = \sum_r P(T/H_r) \times P(H_r)$$

where

$$P(T/H_r) = F_1(\mu_h, \sigma_h, \mu_t, \sigma_t, \rho)$$

$$= [2\pi(1-\rho^2)\sigma_t^2]^{-\frac{1}{2}} \exp\left\{\frac{-1}{2(\sqrt{(1-\rho^2)}\sigma_t)^2}\left[(t-\mu_t) - \rho\frac{\sigma_t}{\sigma_h}(h-\mu_h)\right]^2\right\}$$

$$P(H_r) = F_2(\mu_h, \sigma_h)$$

$$= \frac{1}{\sqrt{2\pi}\sigma_h}\exp\left\{-\frac{(h-\mu_h)^2}{2\sigma_h^2}\right\}\left\{1 - \frac{C_s}{6}\left[3\left(\frac{h-\mu_h}{\sigma_h}\right) - \left(\frac{h-\mu_h}{\sigma_h}\right)^3\right]\right\}$$

where μ_h = mean value of h

σ_h = standard deviation of h

μ_t = mean value of $t = \ln\ \mu_T - \sigma_t^{3/2}$

σ_t = standard deviation of $t = 0.244 - 0.0225\mu_H$

ρ = correlation coefficient $= 0.415 + 0.049\mu_H$

C_s = skewness parameter $= E\left(\frac{[h-\mu_h]^3}{\sigma_h^3}\right)$

In these expressions h and t are the logarithmic values of H and T respectively. μ_h, σ_h and C_s follow from the given probability distribution of H as does μ_H the mean wave height.

$$\mu_T = 3.925 + 1.439\mu_H$$

The numerical values of the coefficients in the formulae for σ_t, ρ and μ_t were derived by regression analysis of over 20 sets of instrumental data.

(c) Extreme wave height

Sometimes the designer needs to estimate the most probable value of the maximum individual wave height in a given return period. After the probabilities of H_s are obtained the corresponding cumulative probabilities are computed and plotted on probability paper.

The methods used by NMI Ltd for analysing these cumulative probabilities for H_s differed from those described in Chapter 9 and are suitable for use when, as in the case of visual data, wave records are not available.

The data define exceedance probabilities for H_s up to a limiting level $1/m$ where m is the number of H_s values (or visual estimates of height) available. It is commonly required to extrapolate these to a level $1/M$ corresponding to an extreme storm of specified return period, R years, and duration, D hours, and in this case $M = 365 \times 24 \times R/D$.

In the NMI method this extrapolation is achieved by use of a 3-parameter Weibull distribution, the formula for the cumulative probability being:

$$P(x > H_s) = \exp - \left[\frac{(H_s - H_0)^n}{b}\right]$$

with values of the parameters n, b and H_0 determined numerically by least square fitting of the available data. The most probable maximum individual wave height H_{max} corresponding to the significant height H_{sM} for the extreme storm having exceedance probability $1/M$ is then estimated by assuming a Rayleigh distribution of heights in the storm, so that $H_{max} \doteqdot (\frac{1}{2} \ln N)^{\frac{1}{2}} H_{sM}$, where N is an estimate of the number of waves in the storm given by $N = 3600D/T$, where T is an estimated mean wave period.

OBTAINING RESPONSE AMPLITUDE OPERATORS

It has been shown that response amplitude operators are convenient both in presenting the results of regular motions in non-dimensional form and as a means of deducing overall motion characteristics in irregular seas. How are these RAOs to be obtained for a given ship? If it is a new design then calculation or model experiments must be used. If the ship exists then ship trials are a possibility. As with other aspects of ship performance the naval architect makes use of all three approaches. Theory helps in setting up realistic model tests which in turn help to develop the theory indicating where simplifying assumptions, e.g. that of linearity of response, are acceptable. Full-scale trials provide evidence of correlation between ship and model or ship and theory.

As knowledge has built up confidence in theory, and as more powerful computers have facilitated more rigorous but lengthy calculations, theory has become the favoured approach to assessing ship motions at least in the early design stages and for conventional forms. Models can be used for the final form to look at deck wetness and rolling in quartering seas for which the theory is less reliable, or to confirm data for unusual hull configurations.

Theory

To outline how theoretical predictions are made, consider the simple case of a ship heading directly into a regular series of long-crested waves at constant speed, i.e. the ship's heading is normal to the line of wave crests. If the ship is symmetrical about its longitudinal centreplane its longitudinal motions, i.e. pitch and heave, will be uncoupled from its lateral motions, sway, yaw and roll (see Fig. 12.1). It is further assumed that pitch and heave are unaffected by any surge.

The fundamental equations follow from Newton's second law of motion

$$m\ddot{z} = F$$

$$J\ddot{\theta} = M$$

In studying ship motion it is usual to consider only the changes in force and moment between the ship moving in calm water and in waves. Thus F and M are the summations of the fluid forces and moments acting on the ship due to the relative motion of ship and wave. Coupling will exist between pitch and heave and the general equations of motion will be of the form

$$(m + a)\ddot{z} + b\dot{z} + cz + d\ddot{\theta} + e\dot{\theta} + g\theta = F_0 \cos(\omega_e t + \alpha)$$

$$(I_{yy} + A)\ddot{\theta} + B\dot{\theta} + C\theta + D\ddot{z} + E\dot{z} + Gz = M_0 \cos(\omega_e t + \beta)$$

where ω_e is the frequency of encounter with the waves.

It is necessary now to obtain expressions for the various coefficients a, b, etc., and A, B, etc., in these equations. Most modern approaches are based on what is known as the 'strip theory' or 'slender body theory'.

The methods differ in detail but all make the following simplifying assumptions:

- Ship responses are small, varying linearly with wave height.
- The ship is slender, its length being much larger than its beam or draught.
- The hull is rigid.
- Ship sections are wall-sided.
- Speed is moderate so that no significant planing lift is produced.
- The water depth may be regarded as deep.
- The presence of the hull has no effect on waves. This is often referred to as the *Froude–Kriloff hypothesis*.

The vessel is considered as composed of a number of thin transverse slices or strips. The flow about each is assumed to be two-dimensional and the same as would exist if the section were part of an infinitely long oscillating cylinder of equal cross-section. This is clearly a simplification of the real state of affairs in that it ignores any interaction between the flow around adjacent sections and the three-dimensional flow that must exist particularly at the ends of the ship. In the case of a vessel pitching and heaving but with no forward velocity the assumption seems not unreasonable, with the motion of each strip considered as a combination of the two motions.

When the ship has forward motion the assumption of two-dimensional flow seems more debatable. The water mass remains approximately steady in space. Although there are orbital motions of water particles in waves the water so disturbed does not move forward with the ship. Cross-sections of the ship are taken relative to a set of axes fixed in space and account is taken of the changing form at any transverse plane due to the forward velocity of the ship. Some authorities regard the theory as unsatisfying but it has been found to provide good comparisons between theory and experiment for ships down to 40 metres in length and for that reason is widely used. Clearly accuracy depends upon, *inter alia,* the values of 'added mass' and damping assumed and much of the development has centred on such factors.

The analysis methods used begin by considering an infinitely long semi-circular cylinder heaving, swaying and rolling in a water free surface. Flow is assumed to be inviscid and incompressible so that potential theory can be used. Conformal transformation techniques are then used to extend the semi-circular results into those for realistic ship shapes. This technique was first used by Lewis (1929) for vibration studies. He found that a truncated transformation yielded acceptable results and he defined his sections in terms of beam/draught ratio and a cross-section area coefficient. The method has been expanded upon by later workers in the field.

In still water the vessel is in equilibrium with buoyancy equal to its weight since it is assumed that the hull does not develop any planing lift. Only changes relative to this still water condition are considered.

Concentrating for the moment on the vertical movement of the strip, at any instant the relative vertical displacement of a point on the hull relative to the water surface will be composed of terms due to heave, pitch and wave surface elevation, i.e.

$$z_r = z - x\theta - \zeta$$

Axes are taken with origin in the still water plane vertically above the centre of gravity and the transverse plane under consideration is a distance x from that

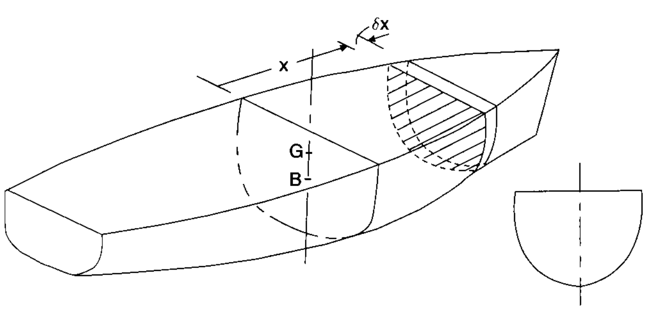


Fig. 12.25 Strip theory

origin. As described above the transverse planes are taken at fixed spatial positions. Thus they move relative to the ship based axes such that:

$$\dot{x} = -u \text{ where } u = \text{ahead velocity of ship.}$$

Differentiating z_r with respect to time

$$\dot{z}_r = w_r = \dot{z} - (x\dot{\theta} + \dot{x}\theta) - \dot{\zeta}$$
$$= \dot{z} - x\dot{\theta} + u\theta - \dot{\zeta}$$

Differentiating again

$$\ddot{z}_r = \dot{w}_r = \ddot{z} - x\ddot{\theta} - x\dot{\theta} + \dot{u}\theta + u\dot{\theta} - \ddot{\zeta}$$
$$= \ddot{z} - x\ddot{\theta} + 2u\dot{\theta} - \ddot{\zeta} \text{ since } \dot{u} = 0$$

The forces acting on a 'strip' with these motions would normally be:

(*a*) an inertial force opposing the acceleration of that section of the ship;
(*b*) hydrodynamic forces due to the relative acceleration and velocity of hull and water;
(*c*) a hydrostatic force due to the increased immersion, z_r.

The so-called 'added mass' term varies with time so that the total effect is one of momentum change

$$\frac{d(a_n w_r)}{dt} = w_r \frac{da_n}{dt} + a_n \dot{w}_r$$

The subscript n is used to denote the particular strip of the ship. If the mass of the ship itself over this strip is m_n then the vertical force δF on a strip of length δx will be of the form

$$\frac{\delta F_n}{\delta x} = -m_n(\ddot{z} - x\ddot{\theta}) - a_n\ddot{z}_r - \left(b_n + \frac{da_n}{dt}\right)\dot{z}_r - c_n z_r$$

a_n, b_n and c_n will at least depend upon hull form and the first two may also vary with motion frequency.

Additional moments, and forces, will arise from yawing and rolling.

The added mass or *acceleration* term, a_n, is obtained by reference to an oscillating circular cylinder. Such a cylinder deeply immersed in a perfect fluid and oscillating normal to its axis has an added mass equal to the displaced water mass. Hence in the more general case of a non-circular cylinder it is reasonable to use an expression of the form:

$$\text{added mass coefficient} = \rho(\text{constant})(\text{cross-sectional area})$$
$$= \rho k_2 A$$

When the cylinder oscillates in a free surface a standing wave system is formed which modifies the added mass term by a second factor, k_4, which depends upon the frequency of oscillation. In this case:

$$\text{added mass coefficient} = a_n = \rho k_2 k_4 A$$

Suitable values for k_2 and k_4 for typical ship forms may be found in published papers for a range of beam/draught ratios and for a number of section shapes at each ratio value. Values are available for rectangular and triangular shapes. Coefficient values for a ship are obtained by comparing each section with the sections presented in the reference. Hence water inertia mass per unit length is calculated for each section, the water inertia curve plotted and added to the normal weight distribution curve. The water inertia is effectively independent of forward speed.

Lewis found that due to the flat bottom amidships the greater part of the added mass is amidships. Also he found that the water inertia is nearly independent of draught so that natural frequency will vary only slowly with the displacement.

The *relative velocity* coefficient is the sum of two components, viz:

(a) a dissipative damping component, b_n, due to radiated waves

$$b_n = \rho g^2 \bar{A}^2 / \omega_e^3$$

where ω_e = frequency of radiated wave
= frequency of wave encounter.

$\bar{A}$ = ratio of amplitudes of radiated waves and the relative vertical motion. $\bar{A}$ values can be found from the literature;

(b) a dynamic damping component arising from the rate of change of a_n.

Thus this term involves the way in which the added mass coefficient varies along the length of the ship.

Havelock (1956) compared the damping coefficients obtained by strip theory and three-dimensional calculation. A completely immersed body was assumed and Havelock argued the relative magnitudes would be a guide to the relative values for a freely floating body. Vossers in discussion supported this view based on calculations for a Michell ship. Figure 12.26 gives figures for a spheroid of L/B ratio of 8. For free oscillations of typical ship forms the values calculated by the two methods are approximately equal but for forced oscillations at lower frequencies the values can differ significantly, particularly for pitch.

The *vertical displacement* term, c_n, is the added buoyancy due to increased immersion. This depends upon the waterplane area in the ship

$$c_n = \rho g \delta A_W$$
$$= \rho g B_n \text{ per unit length, } B_n = \text{local beam}$$

The value of δF acting on each element can be obtained from the above and its moment about the vessel's centre of gravity. Integrating along the length of the ship gives the overall vertical force and moment.

Strip theory is used widely for prediction of rigid body motions and gives reasonable results for most purposes except for rolling. Unfortunately the damping coefficients for rolling are non-linear and depend upon the motion

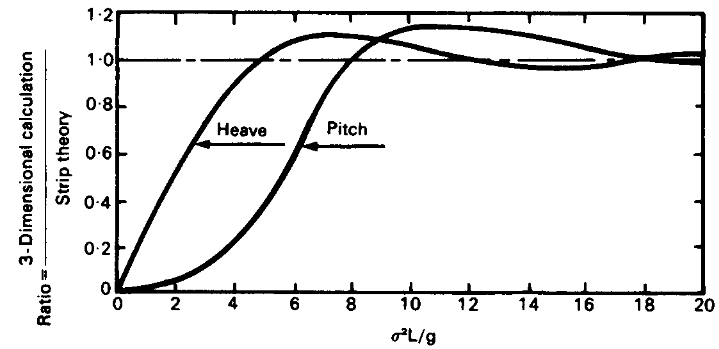


Fig. 12.26 Ratios of damping coefficients for heaving and for pitching

amplitude. This arises from the importance of eddy shedding at the bilges as the ship rolls and skin friction, which are due to viscosity, ignored in strip theory. To circumvent this problem a device can be adopted of an equivalent linear damping coefficient. The predominant rolling motions occur at the natural roll frequency. The equivalent linear roll damping coefficient is based on that frequency and the energy dissipated during the roll. It will have components due to eddy making, skin friction and drag forces on appendages.

Strip theory can be used to calculate ship motions in regular waves. The results can then be applied to motions in irregular seas by summation of the responses to a large number of regular components by which the irregular sea can be represented.

The same basic approach can be used where the ship is moving at an angle to the line of wave crests and to the other degrees of freedom. However, although early applications of strip theory to pitching and heaving were successful, predictions of sway, roll and yaw were less good. Correlation of theoretical results with model or full-scale data showed significant errors in roll, particularly for higher-speed ships. The problem arises principally from the difficulty of assessing roll damping. This has been attributed to inadequate treatment of hull appendages. Better prediction of roll damping and response can be obtained using strip theory to compute added mass, wave making dam~ and exciting forces induding:

(a) lifting surface contributions to damping and exciting forces. The various hull appendages act as lifting surfaces when the ship is underway, their contribution varying with speed. The forces depend upon the geometry of the appendages and the resulting moments upon their position relative to the centre of gravity of the ship;

(b) viscous roll damping, which has contributions from the bilge keels and other appendages, the eddy-making resistance of the hull and hull friction;

(c) hull circulatory effects.

These methods permit fair prediction oflateral motions. Studies show that there is negligible coupling of roll and yaw into sway, the sway being characterized by an exponential decay with a 90 degrees phase lead, and that yaw amplitudes are small. The theory correctly predicts the substantial reduction in roll response with increasing ship speed and the variation in roll response with ship's heading relative to the waves.

Model experiment

Several methods of model testing are in essence available, viz:

(a) measuring the response of a model in regular waves and deducing a set of response amplitude operators for use in the superpositioning theory to predict performance in an irregular sea. For reliable results such tests must be conducted in relatively moderate waves, typically with wave length to height ratio of 40:1. The test facilities can be relatively simple but a large number of runs will be required to cover adequately the range of speed and wavelength involved;

(b) running the model in a standard irregular pattern and analysing the data to provide the response amplitude operators. Several test runs will generally be necessary for each speed to obtain sufficient motion cycles to provide adequate confidence levels in the subsequent statistical analysis. The wave-makers must be capable of creating an irregular wave pattern with the desired spectral characteristics;

(c) *transient wave* or *impulse* wave testing. This method can be regarded as a special case of method (b). The wavemaker starts at high frequency, slows down and then stops. Thus it initially produces short waves which are gradually overtaken by the later, longer waves. The model starts in calm water, passes through a short wave sequence and finishes its run in calm water. By analysing the complete wave and motion records a full picture is obtained of the model responses with a considerable reduction in testing time. The range of frequencies present in the wave sequence must cover the range for which response operators are required, the waves must not become so steep that they break and the records must not be affected by reflections from the end of the tank;

(d) creating a representation of an actual (recorded on ship trial, say,) irregular wave pattern and running the model in it to record motion, wetness, power, etc. In practice it would be very difficult to ensure the model experienced the same wave pattern as the ship which would be the only way to compare directly the pattern of wetness and slamming. However, if the spectral form is correct the model and ship performance can be compared on a statistical basis and thus provide some check on the adequacy of the linear super-positioning assumption.

As with wave records discussed in Chapter 9, it must be remembered that the results of a particular series of experiments can only be regarded as a sample of all possible experimental outcomes. To illustrate this, consider a single run in an irregular wave system. The actual surface shape repeats itself only after a very long period of time, if at all. The actual model data obtained from a

particular run depends on when, during that period, the experiment is run. Tests for statistical significance can be applied.

Ship trials

In principle it is the ship trial which should provide the final check on the adequacy of theoretical and experimental predictions of ship behaviour. Unfortunately the actual ocean never exhibits a standard long-crested sinusoidal wave pattern. Thus no direct measurement of individual response operators is possible for comparison with predictions. Even truly long-crested irregular seas are never met. On occasion the sea surface may approximate to the long crested form but waiting for good conditions can be expensive of time and money. If they are met the sea and motions can be measured and response operators deduced. Comparison of ship with prediction can be made on the basis of these response operators or on the motion parameters predicted for the sea spectrum as measured.

Unfortunately, it is often the more extreme conditions that are of most concern and it is for these that the usual assumptions of linear superposition are likely to yield the greatest inaccuracies. These would be additional to those arising due to inaccuracies in recording waves and motions, sea variations over a recording period and differences in wind and tide. Thus a comparison of two ships as a result of trials conducted at different times and in different sea conditions has a number of limitations, particularly under limiting conditions for operations. Some of these are avoided, or reduced, if the two ships can go on trials in company. The sea condition will then be the same for both ships although it must be remembered that the sea condition may favour one design, particularly if the ships are of different length. It will be appreciated, therefore, that all trials data must be used intelligently with due allowance for the above factors.

The conduct of ship trials is discussed later.

Non-linear effects

Most of the remarks in this chapter, and, indeed, in the book, relate to simple linear equations of motion. This is because it is the simple approach, and

- it is a desirable step, to give a basic understanding, before going on to consider non-linear equations;
- it is an assumption that has served the naval architect with believable accuracy over many years when the tools were not available for more precise studies.

To some extent the success of the linear approach in the past has been due to the use of conservative design methods, or factors of safety that may have been unreasonably high. The key word here is 'may' because, in truth, the naval architect did not know. Designs to these methods had performed well in service but had they been over designed? Some would argue that because ships are still lost, they were inadequately designed, but perhaps there were other reasons for the losses. Or, the successful ships may have been lucky in that they did not meet the more extreme conditions for which they were designed; chance governs much of our lives.

Unfortunately, it is for the more extreme sea conditions and ship responses that non-linearities are most important. It is these conditions that are of greatest concern in safety. Extreme loading in a seaway and broaching-to are two examples of situations when it is desirable, indeed necessary, to take account of non-linear effects. How then do these effects arise?

- Some are due to the variation in the underwater form as the ship moves through, and responds to, the waves. This will arise even for moderate angles due to flare, but become severe when the deck edge becomes immersed or the bow leaves the water completely.
- Cross-coupling of motions will occur. Heave and roll will induce pitching, for example because the ship's shape and therefore forces upon it differ forward and aft.
- Some arise from the physics of the situation. Taking roll damping, the forces on appendages at high speed are proportional to roll velocity but viscous damping forces vary as the square of the roll velocity.

Software is available nowadays to enable the designer to solve non-linear, coupled, six degrees of freedom motion calculations. The exciting forces and moments arise from hydrostatics, from movement through the water (resistance and propulsion), from control surface movements (rudders and stabilizers), wind and wave forces. The computations are more time consuming and costly and should only be used where their greater accuracy is justified. The more realistic the equations the more closely a design's real-life behaviour will be predicted. Any remaining factors of safety can be more specific. For instance, factors will still be needed in structural design to allow for factors such as plate rolling tolerances, built-in distortions and defects. It is not possible in this book to cover non-linear analyses but the student should be aware of the limitations of the linear approach.

Frequency domain and time domain simulations

It has been shown how a ship's response amplitude operators can be used, in conjunction with a wave spectrum for a given sea condition, to produce a corresponding motion spectrum for each motion. This can be done for a range of ship speeds, headings and sea states. From the motion spectrum one can deduce statistical information about the motion by assuming the spectrum to be of a certain form; say, a Gaussian or Rayleigh distribution. Thus the significant pitch angle, the probability of exceeding a given roll angle, and so on, can be calculated. This is usually quite adequate for assessing the general seakeeping performance of a design. Extreme motion probabilities are difficult to assess because of doubts as to the actual form of the ends of the spectrum.

It is also possible to generate from the spectrum a time sequence for un) motion. By dividing the spectrum into a large number of small frequency intervals the motion amplitude at a large number of frequencies are obtained. These individual motions can be assumed to be sinusoidal and can be combined in random phase angle to produce the variation of the motion with time. Velocities and accelerations are obtained by differentiating with respect to time. This simulation of the motion, because it derives from the spectral frequency, is known as a frequency domain simulation. While relatively easy to generate, such simulations suffer from a number of weaknesses:

- The simulation must be continued for a long time in order to ensure that a good representation of motions in any sea state is obtained. If too short the designer may find that the simulation is of a period of relative quiet in the motion, or one of relative violence. This latter may, of course, be what the designer is seeking if it is extreme motions which are being studied.
- It is a linear representation. All motions will be proportional to wave height. This may not be too critical if the spectra used were based on motion levels typical of those being studied. It is not acceptable for many investigations into extreme motions and their effects. See the remarks on non-linear effects, above.
- Cross-coupling of motions is ignored except insofar as it is implicit in the spectra used.

The frequency domain simulation can produce reasonable representations of motions for many purposes but is inadequate for extreme conditions. One limitation is that all motions responses are assumed proportional to wave height. Another form of simulation reproduces how a ship behaves in a specific train of waves. The wave form and the ship form are defined in the *time domain.* By calculating the forces and moments on the ship as it moves through the waves the true combinations of motion can be obtained. The constantly changing shape of the immersed hull, and the forces acting on it, follow from the wave form and the motions of the ship. Cross-coupling effects will be built in. A more accurate picture of factors such as change of freeboard will result and be allowed for. Again the analysis must be continued for long enough if a full representation of the general motion is required.

The accuracy of the motions obtained will depend upon the methods used to calculate the forces involved. *Non-linear effects* due to the changing form of the immersed volume will be implicit in the method. Other non-linearities such as the variation of viscous damping with amplitude of motion (viscous roll damping is proportional to the square of the roll velocity) can be introduced into the equations used. Responses are no longer linearly related to wave height. Programs have been developed for this type of simulation. While in principle the more advanced methods of *computational fluid dynamics* could be used, in practice it has been found adequate to use potential flow methods. They do, as one would expect, require considerable computing power which is compounded by the length of simulation needed. Such computations are becoming more common as the power of computers increases and costs reduce.

Another advantage of a time domain simulation is that so-called 'memory' effects are included. These arise from the fact that the forces at any instant depend, to a degree, upon the preceding motions. Memory effects are important to some design calculations. For instance, in the design of securing systems it is necessary to know the time history of the loads as well as the maximum loads experienced. To illustrate how memory effects may be significant, consider a ship responding to rudder movements. When the rudder is put over the ship begins to turn and the flow around the hull and rudder change. The immediate effects of further rudder movement will depend upon the water flow which will be in transition until the ship takes up a steady rate of turn. In a seaway the flow around the hull is constantly changing in a pattern which does not repeat itself.

Improving seakeeping performance

It has been seen that overall seakeeping performance is limited not so much by motions *per se* as by by the interplay between motions and other design features. Thus overall performance can be improved by such actions as:

(a) siting critical activities in less-affected areas of the ship. Examples are the siting of passenger accommodation towards the position of minimum vertical motion; placing helicopter operations aft in frigates and placing only very rugged equipment forward on the forecastle;
(b) rerouting of ships to avoid the worst sea conditions;
(c) providing local stabilization for certain equipments such as radars.

To an extent these can only be regarded as palliatives and it is necessary to consider how the motions themselves can be reduced. Care is needed to ensure the reduction is 'useful'. For example, if human performance is a limiting factor significant reductions in vertical acceleration over a wide range of higher frequencies is counter-productive if it is won at the expense of even a small increase in acceleration in the frequency band critical to humans. Bearing this reservation in mind there are a number of ways open to the naval architect:

Use can be made of a radically different hull form. Examples, some of which will be discussed in a later chapter, are:

(a) the Small Waterplane Area Twin Hull (SWATH) ship. Essentially the use of a small waterplane area reduces the exciting forces and moments, the twin hull restoring the desired static stability qualities and weather deck area.
(b) the semi-submersible. The concept is similar to the SWATH. A major part of the vessel is well below the still waterplane so that the waves exert little force on it. This configuration is used for oil drilling rigs where a stable platform is essential and it must be held accurately in position over the seabed.
(c) hydrofoil craft. With suitable height sensors and foil incidence control systems a hydrofoil can provide a high speed, steady platform in sea states up to those in which the waves impact the hull. This depends upon 1'011 separation from the hull.

Special ship types can be very effective in specific applications but usually there are penalties which means that the vast majority of ships are still based on a conventional monohull. In this case the designer can either improve performance by detail form changes or stabilize the whole ship. These are now considered.

INFLUENCE OF FORM ON SEAKEEPING

It can be dangerous to generalize on the effect of varying form parameters on the seakeeping characteristics of a design. A change in one parameter often leads to a change in other parameters and a change may reduce motions but increase wetness. Again, the trend arising from a given variation in a full ship may not be the same as the trend from the same variation in a fine ship. This accounts for some apparently conflicting conclusions from different series of experiments. It is essential to consult data from previous similar ships and particularly any methodical series data that is available covering the range of principal form parameters applicable to the new design.

With the above cautionary remarks in mind the following are some general trends based on the results of methodical series data:

Length is an important parameter in its own right. This can be appreciated by considering the response of a ship to a given wave system. If the ship is long compared with the component waves present, it will pitch and heave to a small extent only, e.g. a large passenger liner is hardly affected by waves which cause a 3000 tonnef frigate to pitch violently. With most ships there does come a time when they meet a wave system which causes resonance but the longer the ship the less likely this is.

Forward waterplane area coefficient. An increase reduces the relative motion at the bow but can lead to increased vertical wave bending moment.

Length to beam ratio has little influence on motions although lower *LIB* ratios are preferable. J

Length to draught ratio. High values lead to resonance with shorter waves and this effect can be quite marked. Because of this, high *LIT* ratios lead to lower amplitudes of pitch and heave in long waves and greater amplitudes in short waves. A high *LIT* ratio is more conducive to slamming.

Block coefficient. Generally the higher the block coefficient the less the motions and the greater the increase in resistance, but the influence is small in both cases.

Prismatic coefficient. The higher the C_P value the less the motion amplitudes but the wetter the ship. High C_P leads to less speed loss at high speed and greater speed loss at low speed.

Beam to draught ratio. Higher values reduce vertical acceleration but may lead to greater slamming.

Longitudinal radius of gyration. In waves longer than the ship, a small radius of gyration is beneficial in reducing motions.

A bulbous bow generally reduces motions in short waves but can lead to increased motions in very long waves.

Forward sections. V-shaped sections usually give less resistance in waves and a larger longitudinal inertia. V-shaped sections usually produce lower amplitudes of heave and pitch and less vertical bow movement. Above-water flare has little effect on motion amplitudes but can reduce wetness at the expense of increased resistance and possible slamming effects.

Freeboard. The greater the freeboard the drier the ship.

It will be noted that a given change in form often has one effect in short waves and the opposite effect in long waves. In actual ocean conditions, waves of all lengths are present and it would not be surprising, therefore, if the motions, etc., in an irregular wave system showed less variation with form changes. Research has shown that, for conventional forms, the overall performance of a ship in waves is not materially influenced by variations in the main hull parameters. A large ship will be better than a small one.

Local form changes can also assist in reducing the adverse consequences of motion, e.g. providing finer forms forward with large deadrise angle can reduce slamming forces.

SHIP STABILIZATION

There is a limit to the extent to which amplitudes of motion can be reduced in conventional ship forms by changes in the basic hull shape. Fortunately, considerable reductions in roll amplitudes are possible by other means, roll being usually the most objectionable of the motions as regards comfort. In principle, the methods used to stabilize against roll can be used to stabilize against pitch but, in general, the forces or powers involved are too great to justify their use.

Stabilization systems

These fall naturally under two main headings:

(a) Passive systems in which no separate source of power is required and no special control system. Such systems use the motion itself to create moments opposing or damping the motion. Some, such as the common bilge keel, are external to the main hull and with such systems there is an added resistance to ahead motion which has to be overcome by the main engines. The added resistance is offset, partially at least, by a reduction in resistance of the main hull due to the reduced roll amplitude.

Other passive systems, such as the passive anti-roll tanks, are fitted internally. In such cases, there is no augment of resistance arising from the system itself.

The principal passive systems (discussed presently) fitted are:

Bilge keels (and docking keels if fitted)
Fixed fins

Passive tank system
Passive moving weight system.

(b) Active systems in which the moment opposing roll is produced by moving masses or control surfaces by means of power. They also employ a control system which senses the rolling motion and so decides the magnitude of the correcting moment required. As with the passive systems, the active systems may be internal or external to the main hull.

The principal active systems fitted are:

Active fins
Active tank system
Active moving weight
Gyroscope.

Brief descriptions of systems

The essential requirement of any system is that the system should always generate a moment opposing the rolling moment.

(a) With active fins a sensitive gyro system senses the rolling motion of the ship and sends signals to the actuating system which, in turn, causes the fins to move in a direction such as to cause forces opposing the roll. The actuating gear is usually electrohydraulic. The fins which may be capable of retraction into the hull, or may always protrude from it, are placed about the turn of bilge in order to secure maximum leverage for the forces acting upon them. The fins are usually of the balanced spade type, but may incorporate a flap on the trailing edge to increase the lift force generated.

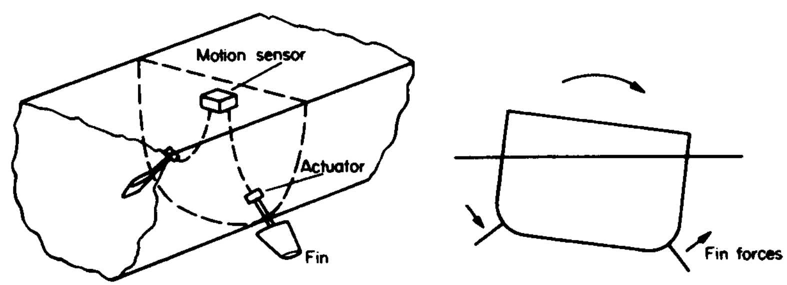


Fig. 12.27 Active fin system

The capacity of a fin system is usually expressed in terms of the steady angle of heel it can cause with the ship moving ahead in still water at a given speed. Since the force on a fin varies in proportion to the square of the ship speed, whereas the -G-Z curve for the ship is, to a first order, independent of speed, it follows that a fin system will be more effective the higher the speed. Broadly speaking, a fin system is not likely to be very effective at speeds below about 10 knots.

(b) Active weights systems take a number of forms, but the principle is illustrated by the scheme shown in Fig. 12.28. If the weight *W* is attached to H

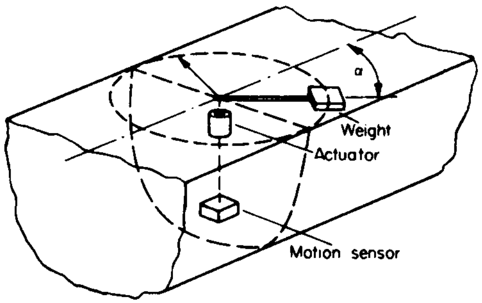


Fig. 12.28 Active weight system

rotating arm of radius R then, when the arm is at an angle α to the centre line of the ship and on the higher side,

$$\text{Righting moment} = WR \sin \alpha$$

Such a system has the advantage, over the fin system, that its effectiveness is independent of speed. It involves greater weight and power, however, and for these reasons is not often fitted.

(*c*) Active tank systems are also available in a variety offorms as illustrated in Fig. 12.29. The essential, common, features are two tanks, one on each side of the ship, in which the level of water can be controlled in accord with the dictates of the sensing system. In scheme *(a)*, water is pumped from one tank to the other so as to keep the greater quantity in the higher tank. In scheme *(b)*, the water level is controlled indirectly by means of air pressure above the water in each tank, the tanks being open to the sea at the bottom. Scheme *(b)* has the advantage of requiring less power than scheme *(a)*. In scheme *(c)*, each tank has its own pump but otherwise is similar to scheme *(a)*.

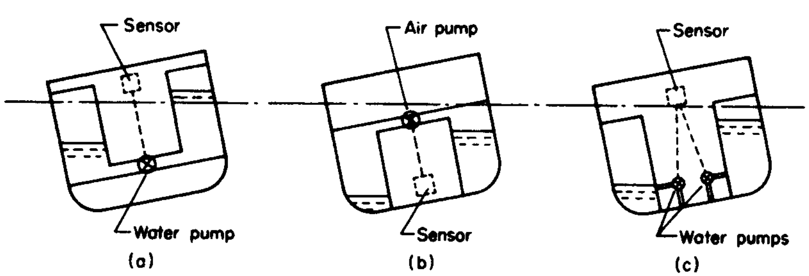


Fig. 12.29 Active tank systems

(*d*) All active stabilizing systems depend upon gyroscopes as part of their control system. If the gyroscope is massive enough, use can be made of the torque it generates when precessed to stabilize the ship. Such systems are not commonly fitted because of their large space and weight demands.

(*e*) Bilge keels are so simple and easy to fit that very few ships are not so fitted. They typically extend over the middle half to two-thirds of the ship's length at

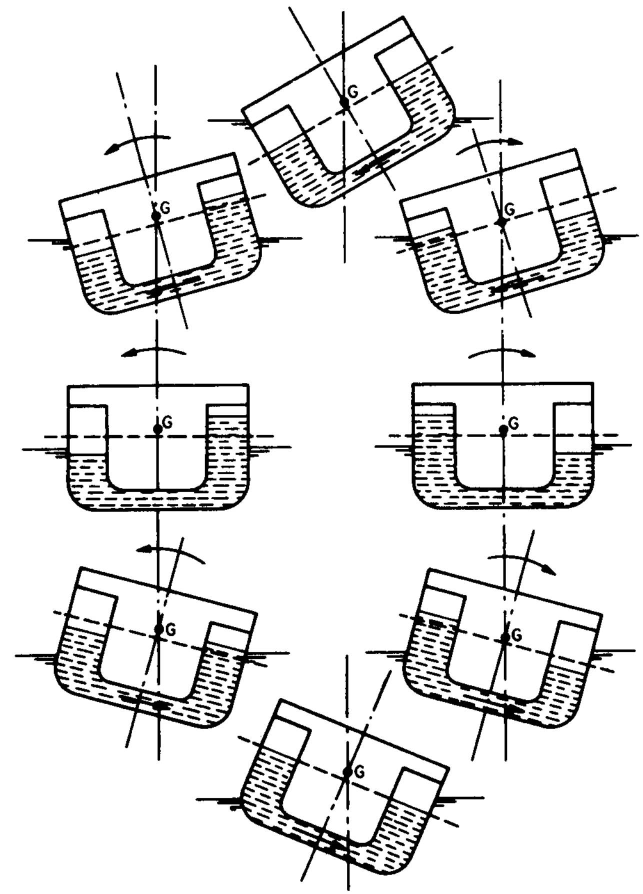


Fig. 12.30 Passive tank system

the turn of bilge. Compared with a ship not fitted, bilge keels can produce a reduction of roll amplitude of 35 per cent or more. They are usually carefully aligned with the flow around the hull in calm water so as to reduce their resistance to ahead motion. Unfortunately, when the ship rolls the bilge ke~ are no longer in line with the flow of water and can lead to significant increa in resistance. For this reason, some large ships may be fitted with a tank stabilizing system and dispense with bilge keels.

(*f*) Fixed fins are similar in action to bilge keels except that they are shorter and extend further from the ship's side. An advantage claimed for them is that, by careful shaping of their cross-section, the lift generated at a given ahead speed can be increased compared with the drag they suffer. A disadvantage is that, projecting further from the hull, they are more susceptible to damage. They are generally less effective at low speed.

(*g*) Passive tank systems use the roll of the ship itself to cause water in the tanks to move in such a way as to oppose the motion. Starting from rest with water level in the two tanks, if the ship rolls to starboard water flows from port to starboard until the maximum angle of roll is reached. As the ship now tries to recover, the water will try to return but will nevertheless lag and the moment due to the water will oppose the roll velocity. Also, if the resistance of the duct is high the water will not be able to return before the ship is rolling to port, i.e. the level of water in the tanks can be made to lag the roll motion. By carefully adjusting the resistance of the duct the system can be 'tuned' to give maximum stabilizing effect. This will be when the phase lag is 90 degrees.

One limitation of such a scheme is that the system can only be 'tuned' to one frequency. This is chosen as the natural period of roll because it is at this period that the really large angles of roll can be built up. At other frequencies the passive tank system may actually lead to an increase in roll angle above the 'unstabilized' value, but this is not usually serious because the roll angles are small anyway. A more sophisticated system is one in which the resistance in the duct can be varied to suit the frequency of the exciting waves. In this way roll damping is achieved in all wave lengths.

(h) Passive moving weight systems are similar in principle to the passive tank systems but are generally less effective for a given weight of system.

Comparison of principal systems

Table 12.1 compares the principal ship stabilizing systems. The most commonly fitted, apart from bilge keels, are the active fin and passive tank systems.

Performance of stabilizing systems

The methods of predicting the performance of a given stabilizer system in reducing motion amplitudes in irregular seas are beyond the scope of this book. A common method of specifying a system's performance is the roll amplitude it can induce in calm water, and this is more readily calculated and can be checked on trials.

When the ship rolls freely in still water, the amplitude of each successive swing decreases by an amount depending on the energy absorbed in each roll. At the end of each roll the ship is momentarily still and all its energy is stored as potential energy. If ¢I is the roll angle, the potential energy is b,-O-M-¢T. If, on the next roll, the amplitude is ¢2 then the energy lost is

$$\frac{1}{2}\Delta\overline{GM}(\phi_1^2-\phi_2^2)=\Delta\overline{GM}\left(\frac{\phi_1+\phi_2}{2}\right)(\phi_1-\phi_2)=\Delta\overline{GM}\phi\delta\phi$$

where ϕ = mean amplitude of roll.

The reduction in amplitude, $\delta\phi$, is called the *decrement* and in the limit is equal to the slope of the curve of amplitude against number of swings at the mean amplitude concerned. That is,

$$\delta\phi=\left(-\frac{d\phi}{dn}\right)_\phi$$

Table 12.1
Comparison of stabilizer systems. (Figures are for normal installations)

Type	Activated fin	Passive tank	Active tank	Massive gyro (active)	Moving weight (active)	Moving weight (passive)	Bilge keel	Fixed fin
Percentage roll reduction	90%	60–70%	No data	45%	No data	No data	35%	No data
Whether effective at very low speeds	No	Yes	Yes	Yes	Yes	Yes	Yes	No
Reduction in deadweight	1% of displacement	1–4% of displacement	Comparable with passive tank	2% of displacement	Comparable with passive tank		Negligible	
Any reduction in statical stability	No	Yes	Yes*	No	Yes*	Yes	No	No
Any increase in ship's resistance	When in operation	No	No	No	No	No	Slight	Slight
Auxiliary power requirement	Small	Nil	Large	Large	Large	Nil	Nil	Nil
Space occupied in hull	Moderate generally less than tanks	Moderate	Moderate	Large	Moderate	Less than tanks	Nil	Nil
Continuous athwartships space	No	Generally	Yes	No	Yes	Yes	No	No
Whether vulnerable to damage	Not when retracted	No	No	No	No	No	Yes	Very
First cost	High	Moderate	Probably high†	Very high	Probably high†	Probably high†	Low	Moderate†
Maintenance	Normal mechanical	Low	Normal mechanical	Probably high	Normal mechanical	Normal mechanical	Often high	Probably high

* There is an effective reduction in statical stability, since allowance must be made for the possibility of the system stalling with the weight all on one side.
† These systems have not been developed beyond the experimental stage and the cost comparison is based on general consideration.

This means that when stabilizers are rolling a ship to a steady amplitude ϕ, the energy lost to damping per swing is

$$\Delta\overline{GM}\phi\left(\frac{d\phi}{dn}\right)_\phi$$

and this is the energy that must be provided by the stabilizers.

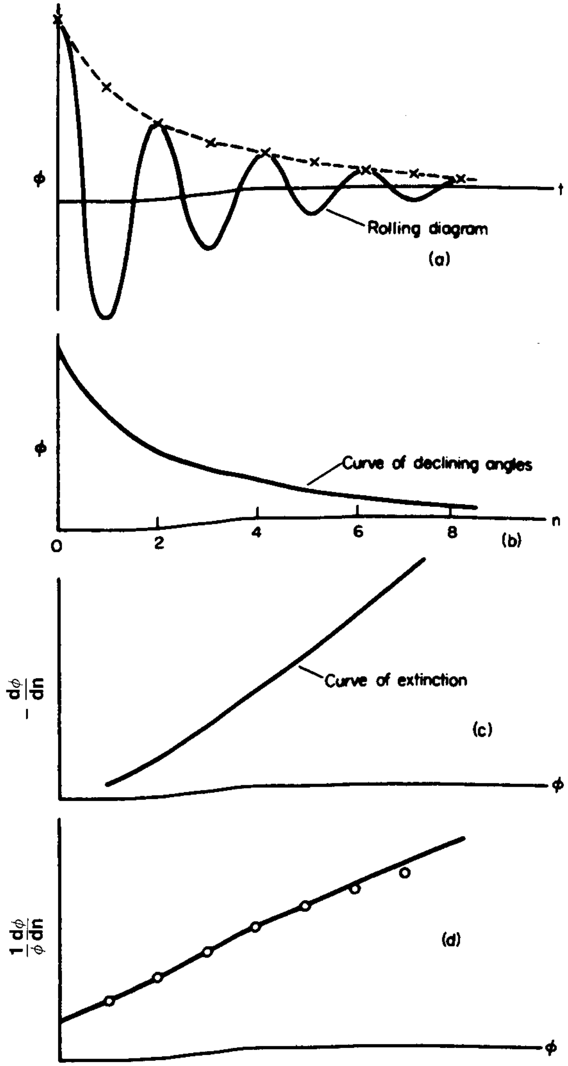


Fig. 12.31 Decrement curve

The value of $\mathrm{d}\phi/\mathrm{d}n$ can be derived from model or full-scale experiments by noting successive amplitudes of roll as roll is allowed to die out naturally in otherwise still water. These amplitudes are plotted to base n (i.e. the number of swings) and the slope measured at various points to give values of $-\mathrm{d}\phi/\mathrm{d}n$ at various values of ϕ. (See Fig. 12.31 (*a*), (*b*) and (*c*).)

In most cases, it is adequate to assume that $\mathrm{d}\phi/\mathrm{d}n$ is defined by a second order equation. That is

$$-\frac{\mathrm{d}\phi}{\mathrm{d}n} = a\phi + b\phi^2$$

or

$$-\frac{1}{\phi}\frac{\mathrm{d}\phi}{\mathrm{d}n} = a + b\phi$$

By plotting $(1/\phi)\ \mathrm{d}\phi/\mathrm{d}n$ against ϕ as in Fig. 12.31(*d*), a straight line can be drawn through the experimental results to give values of a and b.

Considering forcing a roll by moving weight, the maximum amplitude of roll would be built up if the weight could be transferred instantaneously from the depressed to the elevated side at the end of each swing as shown in Fig. 12.32.

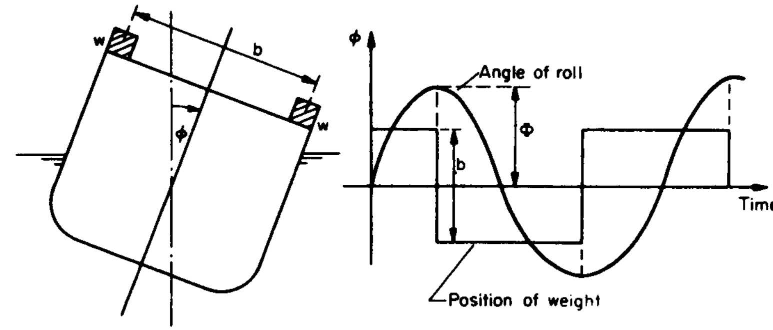


Fig. 12.32 Instantaneous weight transfer

The change in potential energy of the weight at each transfer is $wb\sin\phi$. Hence, approximately

$$wb\phi = \Delta\overline{\mathrm{GM}}\phi\left(-\frac{\mathrm{d}\phi}{\mathrm{d}n}\right)$$

or

$$-\frac{\mathrm{d}\phi}{\mathrm{d}n} = \frac{w}{\Delta}\frac{b}{\overline{\mathrm{GM}}}$$

It follows that the moving weight can increase the roll amplitude up to the value appropriate to this value of $\mathrm{d}\phi/\mathrm{d}n$.

EXAMPLE 2. Assuming that for a rolling ship, the slope of the curve of declining angles is $-\mathrm{d}\phi/\mathrm{d}n = a\phi + b\phi^2$, find the values of a and b given the following corresponding values of $-\mathrm{d}\phi/\mathrm{d}n$ and ϕ in degree units:

ϕ	5	10	15	20
$-\frac{\mathrm{d}\phi}{\mathrm{d}n}$	0.75	2.00	3.75	6.00

The above figures apply to a 3000 tonne frigate with a metacentric height of 1 m. A mass of 15 tonnes is made to move across the deck in simple harmonic motion with an amplitude of 6 m. Find the steady rolling angle which can be set up by the weight.

Solution: To find the values of a and b the values of $-(1/\phi)\ \mathrm{d}\phi/\mathrm{d}n$ must be plotted against ϕ

ϕ	5	10	15	20
$-\dfrac{1}{\phi}\dfrac{\mathrm{d}\phi}{\mathrm{d}n}$	0.15	0.20	0.25	0.30

This is a straight line and gives $a = 0.10$, $b = 0.01$, i.e.

$$-\frac{\mathrm{d}\phi}{\mathrm{d}n} = 0.10\phi + 0.01\phi^2$$

In this case, the weight is moving with simple harmonic motion. (This would be the effect if the weight were moving in a circle on the deck at constant speed.) If the angle of roll at any instant is given by

$$\phi = \Phi \sin\frac{2\pi t}{T_0}$$

then the distance of the weight from the centre line is

$$\frac{b}{2}\sin\left(\frac{2\pi t}{T_0} + \frac{\pi}{2}\right)$$

where b is the double amplitude of motion.

The movement of ship and weight must be 90 degrees out of phase for an efficient system.

The work done by the weight in moving out to out (i.e. per swing) is

$$\int_{-b/2}^{+b/2} w\phi\ \mathrm{d}\left\{\frac{b}{2}\sin\left(\frac{2\pi t}{T_0} + \frac{\pi}{2}\right)\right\}$$

$$= w\Phi\frac{b}{2}\frac{2\pi}{T_0}\int_{-T_0/2}^{T_0/2}\sin\frac{2\pi t}{T_0}\cos\left(\frac{2\pi t}{T_0} + \frac{\pi}{2}\right)\mathrm{d}t$$

$$= \frac{\pi}{T_0} wb\Phi\int_0^{T_0/2}\sin^2\frac{2\pi t}{T_0}\mathrm{d}t$$

$$= \frac{\pi}{T_0} wb\Phi\frac{1}{2}\left[t - \frac{T_0}{4\pi}\sin\frac{4\pi t}{T_0}\right]_0^{T_0/2}$$

$$= \frac{\pi}{4} wb\Phi$$

i.e.

$$\frac{\pi}{4} wb\Phi = \Delta\overline{\mathrm{GM}}\Phi\left(-\frac{\mathrm{d}\phi}{\mathrm{d}n}\right)$$

From which

$$-\frac{\mathrm{d}\phi}{\mathrm{d}n} = \frac{\pi}{4}\frac{wb}{\Delta\overline{\mathrm{GM}}} = \frac{\pi}{4}\frac{mb}{\Sigma\ \overline{\mathrm{GM}}}$$

Hence, if ϕ is the steady rolling angle produced

$$0.1\phi + 0.01\phi^2 = \frac{\pi}{4} \times \frac{15}{3000} \times \frac{2(6)}{1} \times \frac{180}{\pi}$$

whence,

$$\phi = 12.2 \text{ degrees}$$

Experiments and trials

TEST FACILITIES

Seakeeping experiments can be conducted in the conventional long, narrow ship tanks usually used for resistance tests provided they are fitted with a wavemaker at one end and a beach at the other. Unfortunately in such tanks it is only possible to measure the response of the model when heading directly into or away from the waves.

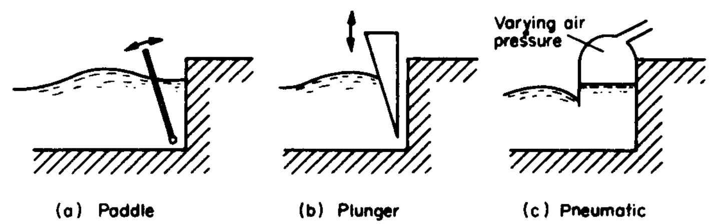


Fig. 12.33 Types of wave-maker

Various forms of wavemaker have been employed (Fig. 12.33). Beaches also take a number of forms but are essentially devices for absorbing the energy of the incident waves. They reduce the amplitude of the reflected waves which would otherwise modify the waves experienced by the model. There are a number of basins specially designed for seakeeping experiments. They permit the model to be run at any heading relative to, or to manoeuvre in, the waves. Short crested wave systems can be generated. The basin at Haslar is depicted in Fig. 12.34. It is 122m long by 61 m wide and uses a completely free remotely~controlled model. In some facilities, e.g. that at the David Taylor Model Basin, Carderoc, the basin is spanned by a bridge so that models can be run under a carriage either free or constrained. This is also a feature of the 170 m by 40 m Seakeeping and Manoeuvring Facility at MARIN which opened in 1999, in which realistic short crested wave conditions can be created.

Whilst a free model has no guides that can interfere with its motion, the technique presents many difficulties. The model must contain its own propulsion system, power supplies, radio-control devices as well as being able to record much of the experimental data. It must be ballasted so as to possess the correct stability characteristics and scaled inertias in order that its response as a dynamic system will accurately simulate that of the ship. All this must be achieved in a relatively small model as too large a model restricts the effective

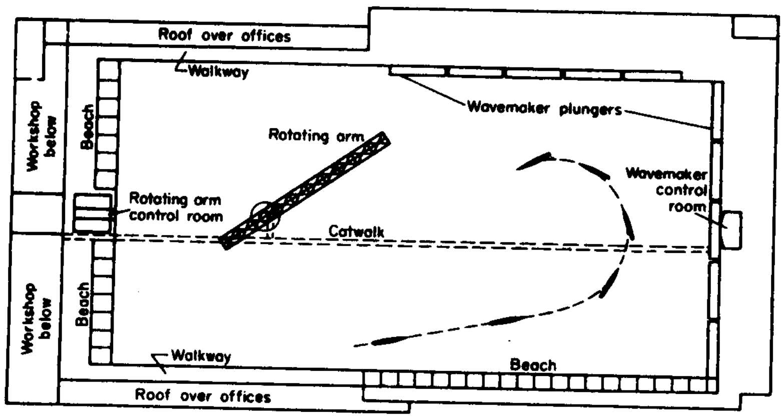


Fig. 12.34 Seakeeping basin at Haslar

length of each run. This is very important for experiments where data have to be presented statistically.

CONDUCT OF SHIP TRIALS

Ship trials are carried out for a variety of reasons, including:

(a) to confirm that the ship meets her design intention as regards performance;
(b) to predict performance during service;
(c) to prove that equipment can function properly in the shipboard environment;
(d) to provide data on which future ship designs can be based;
(e) to determine effect on human performance.

Whilst resistance and propulsion trials are usually carried out in calm water, those concerned with motions must by definition be carried out in rough water. Stabilizer performance is a special case and is discussed later. Two types of trial are possible:

(a) short duration trials in which the ship responses to a measured sea system are recorded;
(b) prolonged period trials in which statistical data is built up of ship response in a wide range of sea conditions.

The first type of trial is essential if it is wished to compare ship with model or calculated response operators over a range of ship speeds and headings. Then the likely long-term behaviour of the ship on voyage can be deduced as described earlier.

The second type of trial provides a comparison of actual and assessed behaviour during a voyage or over a period of time. Differences may be due either to the ship not responding to the wave systems as predicted or to the wave systems experienced not being those anticipated so the data is of limited

value in assessing prediction methods. The longer the time period the better the measure of a ship's 'average' performance.

Short duration trials are expensive and the opportunity is usually taken to record hull strains, motions, shaft torque and shaft thrust at the same time. Increasing attention is being paid to the performance of the people on board. The sea state itself must be recorded and this is usually by means of a wave recording buoy. For the second type of trial a simpler statistical motion recorder is used, often restricted to measurement of vertical acceleration. No wave measurements are made but sea states are observed. Statistical strain gauges may also be fitted. Satellites can be used to measure the wave system in which the trial ship is operating and can help record the ship's path.

Although various methods have been proposed for measuring a multi-directional wave system it is a very difficult task. Good correlation has been achieved between calculated and measured sea loadings in some trials by applying the cosine squared spreading function proposed by the ITTC (see Chapter 9) to a spectrum based on buoy measurements.

In the earliest trials the waves were recorded by a shipborne wave recorder but nowadays a freely floating buoy is used. Signals are transmitted to the trials ship over a radio link or recorded in the buoy for recovering at the end of the trial. Vertical motions of the buoy are recorded by an accelerometer and movement of the wave surface relative to the buoy by resistive probes. Roll, pitch and azimuth sensors monitor the attitude of the buoy. For studying complex wave systems in detail several buoys may be used.

A typical sequence for a ship motion trial is to:

(a) carry out measured mile runs at the start of the voyage to establish the ship's smooth water performance and to calibrate the log;
(b) carry out service trials during passage to record sample ship motions and propulsive data under normal service conditions;
(c) launch the recording buoy, record conditions and recover buoy, when conditions are considered suitable, i.e. waves appear to be sufficiently long-crested;
(d) carry out a manoeuvre of the type shown in Fig. 12.35 recording motions and waves for each leg.

Figures denote time in minutes spent on each leg. The accuracy of the analysis depends upon the number of oscillations recorded. For this reason, the legs running with the seas are longer than those with ship running head into the waves. The overall time on the manoeuvre has to be balanced against the possibility of the sea state changing during the trial. The two sets of buoy records and a comparison of the results from the initial and final legs provides a guide to the stability of the trial conditions. The remaining steps of the sequence are:

(e) launch buoy for second recording of waves;
(I) repeat (c), (d) and (e) as conditions permit;
(g) carry out service trials on way back to port;
(h) carry out measured mile runs on return.

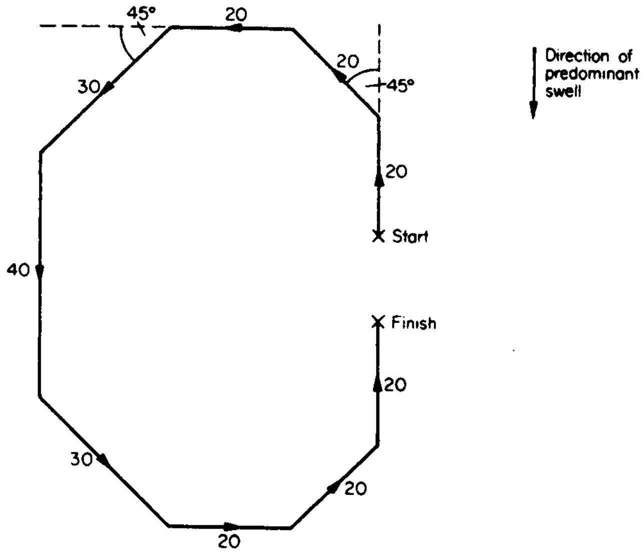


Fig. 12.35 *Typical seakeeping manoeuvre*

On return to harbour, a very lengthy analysis is required. Indeed, this type of trial would not be feasible without computer aid.

STABILIZER TRIALS

Stabilizers, fitted to reduce rolling in a seaway, can be specified directly in terms of roll under stated conditions but such performance can never be precisely proven on trial. As an alternative a designer may relate performance to the steady angle of heel that can be generated by holding the fins over in calm water at a given speed. Heel can be measured directly to establish whether or not such contractual requirements have been met. Forced rolling trials in calm water can be used to study the performance of shipborne equipment under controlled conditions. It is often difficult to distinguish the effects of the stabilizers from the cross-coupling effect of the rudders; indeed, it is possible to build up a considerable angle of roll in calm water by the judicious use of rudder alone.

Problems

1. The mast of a small floating raft, due to the passage of a train of deep water waves, is observed to oscillate with a period of 7 s and amplitude from the vertical of ±8 degrees. Find the height, length and velocity of the waves in metric units.
2. A box-shaped vessel has a length of 50m, beam 12m, draught 3m, freeboard 6m. The height of the e.g. above the bottom is 4.5 m. Assuming that the weight is uniformly distributed throughout the length and section of the vessel, and neglecting the effects of associated water, calculate the free periods of roll, pitch and heave in salt water.

3. A vessel, length 200 m, whose periods of free roll and pitch are $12\frac{1}{2}$ and 10 s respectively, is steaming at 20 knots in a sea of wave-length 250 m. Calculate the headings on which the greatest rolling and pitching are likely.
4. A ship is rolling with a constant amplitude, the rolling being maintained by moving a weight across the deck, i.e. the energy put into the ship by the moving weight just balances the energy dissipated by damping.
   Show, by general arguments, that the ideal case is one in which the weight is transferred instantly from the depressed side to the elevated side at the end of each swing.
   Compare the relative weights required to maintain a given small rolling amplitude assuming

   (*a*) instantaneous transfer
   (*b*) weight moving with constant velocity
   (*c*) weight moving with s.h.m.

   The amplitude of the movement is the same in each case.
   [*Note*: Assume rolling motion is given by $\phi = \Phi \sin(2\pi t/T)$.]
5. Show that, neglecting damping forces, the rolling of a ship to small angles in still water, without ahead motion, is simple harmonic. Hence, derive an expression for the natural period of oscillation of the ship in terms of the radius of gyration (k) and the $\overline{\mathrm{GM}}$ of the ship. What is the effect of entrained water? How do rolling considerations affect the choice of $\overline{\mathrm{GM}}$ for a passenger ship?
   During rolling trials on an aircraft carrier, a natural period of roll of 14 s was recorded. The displacement was 50,000 tonnef and the $\overline{\mathrm{GM}}$ was 2.5 m. The inertia coefficient, allowing for the effect of entrained water is 20 per cent. Calculate the radius of gyration of the aircraft carrier.
6. A ship, 4000 tonnef displacement, 150 m length, 15 m beam and 1 m metacentric height has a rolling period of 10.5 s and a decrement equation

$$-\frac{\mathrm{d}\phi}{\mathrm{d}n} = 0.20\phi + 0.15\phi^2 \ (\phi \text{ in radians})$$

   If the ship is to be rolled to an amplitude of 10° estimate the weight required to be moved instantaneously across the deck assuming that it can be moved through 12 m.
7. The differential equation for the rolling motion of a ship in regular waves can be expressed in the form:

$$\ddot{\phi} + 2k\omega_0\dot{\phi} + \omega_0^2\phi = \omega_0^2 F_0 \sin\omega_{\mathrm{E}} t$$

   Explain the significance of the terms in this equation.
   The equation of the rolling motion of a particular ship in regular waves can be expressed in the form:

$$\ddot{\phi} + 0.24\dot{\phi} + 0.16\phi = 0.48 \sin\omega_{\mathrm{E}} t$$

   where ϕ is the roll angle in degrees.

Calculate the amplitudes of roll when ω_E is equal to 0.2, 0.4 and 0.8, commenting upon their relative magnitudes. What would be the period of damped rolling motion in calm water?

8. A ship motion trial is carried out in a long-crested irregular wave system. The spectrum of the wave system as measured at a stationary point is defined by the following table:

$S_\zeta(\omega)$, (wave height)$^2/\delta\omega$ (m^{2}s)	1.2	7.6	12.9	11.4	8.4	5.6
ω, frequency (1/s)	0.3	0.4	0.5	0.6	0.7	0.8

The heave energy spectrum obtained from accelerometers in the ship, when moving at 12 knots on a course of 150 degrees relative to the waves, is defined as follows:

$S_Z(\omega_E)$, (heave)$^2/\delta\omega_E$ (m^{2}s)	0.576	1.624	1.663	0.756	0.149	0.032
ω_E, frequency of encounter (1/s)	0.4	0.5	0.6	0.7	0.8	0.9

Derive the response curve, in the form of heave/wave height, for the ship at this speed and heading, over the range of frequencies of encounter from 0.4 to 0.9.

9. The successive maximum angles in degrees recorded in a model rolling experiment are:

Port	15 (start)	10.4	7.7	5.9
Starboard	12.3	8.9	6.7	

What are the 'a' and 'b' coefficients? What maximum angle would you expect to be attained at the end of the tenth swing?

10. A vessel, unstable in the upright position, lolls to an angle α. Prove that, in the absence of resistance, she will roll between $\pm\phi$ or between ϕ and $\sqrt{(2\alpha^2 - \phi^2)}$ according as ϕ is greater or less than $\alpha\sqrt{2}$. All angles are measured from the vertical.

Explain how the angular velocity varies during the roll in each case.

11. A rolling experiment is to be conducted on a ship which is expected to have 'a' and 'b' extinction coefficients of 0.08 and 0.012 (degree units).

The experiment is to be conducted with the displacement at 2134 tonnef and a metacentric height of 0.84 m. The period of roll is expected to be about 9 s.

A mechanism capable of moving a weight of 6.1 tonnef in simple harmonic motion 9.14 m horizontally across the vessel is available.

Estimate:

(*a*) the maximum angle of roll likely to be produced,
(*b*) the electrical power of the motor with which the rolling mechanism should be fitted (assume an efficiency of 80 per cent).

12. A vessel which may be regarded as a rectangular pontoon 100 m long and 25 m wide is moving at 10 knots into regular sinusoidal waves 200 m long and 10 m high. The direction of motion of the vessel is normal to the line of crests and its natural (undamped) period of heave is 8 s.

If it is assumed that waves of this length and height could be slowed down relative to the ship, so that the ship had the opportunity of balancing itself statically to the wave at every instant of its passage, the ship would heave in the effective period of the wave and a 'static' amplitude would result. With the wave at its correct velocity of advance relative to the ship a 'dynamic' amplitude will result which may be regarded as the product of the 'static' amplitude and the so-called 'magnification factor'.

Calculate the amplitude of heave of the ship under the conditions described in the first paragraph, making the assumption of the second paragraph and neglecting any Smith correction.

The linear damping coefficient k, is 0.3.

3. A ship, 4000 tonnef displacement, 140 m long and 15 m beam has a transverse metacentric height of 1.5 m. Its rolling period is 10.0 s and during a rolling trial successive (unfaired) roll amplitudes, as the motion was allowed to die down, were:

11.3, 8.6, 6.8, 5.6, 4.5, 3.7 and 3.1 degrees

Deduce the 'a' and 'b' coefficients, assuming a decrement equation of the form

$$-\frac{d\phi}{dn} = a\phi + b\phi^2$$

4. The spectrum of an irregular long-crested wave-system, as measured at a fixed point, is given by:

$S\zeta(\omega)$, (wave amplitude)$^2/\delta\omega$ (m^2 s)	0.3	1.9	4.3	3.8
ω, (frequency) (1/s)	0.3	0.4	0.5	0.6

A ship heads into this wave system at 30 knots and in a direction such that the velocity vectors for ship and waves are inclined at 120 degrees. Calculate the wave spectrum as it would be measured by a probe moving forward with the speed of the ship.

Discuss how you would proceed to calculate the corresponding heave spectrum. Illustrate your answer by calculating the ordinate of the heave spectrum at a frequency of encounter of 0.7 s.

The relationship between amplitude of heave and wave amplitude at this frequency of encounter for various speeds into regular head seas of appropriate length should be taken as follows:

$\dfrac{\text{heavy amplitude}}{\text{wave amplitude}}$	0.71	0.86	0.92	0.95	0.96
speed (knots)	20	40	60	80	100

Assume that, to a first approximation, the heave amplitude of a ship moving at speed V obliquely into long-crested waves is the same as the heave amplitude in regular head seas of the same height and of the same 'effective length' (i.e. the length in the direction of motion) provided the speed V_1 is adjusted to give the same frequency of encounter.

15. Assuming that a ship heaves in a wave as though the relative velocity of wave and ship is very low, show that the 'static' heave is given by

$$\frac{\text{heave amplitude}}{\text{wave amplitude}} = -\frac{\sin n\pi}{\pi n}\sin\frac{2\pi t}{T_{\mathrm{E}}}$$

for a rectangular waterplane, where

$$n = \frac{\text{length of ship}}{\text{length of wave}} = \frac{L}{\lambda}$$

and the wave is defined by

$$\zeta = \frac{H}{2}\sin\left(\frac{2\pi n x}{L} - \frac{2\pi t}{T_{\mathrm{E}}}\right)$$

Show that there is zero 'static' response at $n = 1.0$.

13 Manoeuvrability

General concepts

All ships require to be controllable in direction in the horizontal plane so that they can proceed on a straight path, turn or take other avoiding action as may be dictated by the operational situation. They must further be capable of doing this consistently and reliably not only in calm water but also in waves or in conditions of strong wind. In addition, submarines require to be controllable in the vertical plane, to enable them to maintain or change depth as required whilst retaining control of fore and aft pitch angle.

Considering control in the horizontal plane, a study of a ship's manoeuvrability must embrace the following:

(a) the ease with which it can be maintained on a given course. The term steering is commonly applied to this action and the prime factor affecting the ship's performance is her directional or dynamic stability. This should not be confused with the ship's stability as discussed in Chapter 4;
(b) the response of the ship to movements of her control surfaces, the rudders, either in initiating or terminating a rate of change of heading;
(c) the response to other control devices such as bow thrusters;
(d) the ability to turn completely round within a specified space.

With knowledge on these factors the designer can ensure that the ship will be controllable; calculate the size and power of control surfaces and/or thrusters to achieve the desired standards of manoeuvrability; design a suitable control system-autopilot or dynamic positioning system; provide the necessary control equations for the setting up of training simulators.

For control in the vertical plane, it is necessary to study:

(a) ability to maintain constant depth, including periscope depth under waves;
(b) ability to change depth at a controlled pitch angle.

Submarine stability and control is dealt with in more detail in a later section.

It is clear from the above, that all ships must possess some means of directional control. In the great majority of cases, this control is exercised through surfaces called rudders fitted at the after end of the ship. In some cases, the rudders are augmented by other lateral force devices at the bow and, in a few special applications, they are replaced by other steering devices such as the vertical axis propeller. This chapter is devoted mainly to the conventional. rudder steered, ship but the later sections provide a brief introduction to some of the special devices used.

It is important to appreciate that it is not the rudder forces directly in themselves that cause the ship to turn. Rather, the rudder acts as a servo-system which causes the hull to take up an attitude in which the required forces and moments are generated hydrodynamically on the hull. Rudders are fitted aft in a ship because, in this position, they are most effective in causing the hull to take up the required attitude and because they benefit from the increased water velocity induced by the propellers. At low speed, when the rudder forces due to the speed of the ship alone are very small, a burst of high shaft revolutions produces a useful side force if the propellers and rudders are in line.

In the early days of man's movements on water, directional control was by paddle as in a canoe today. That is to say, the heading was controlled by applying a force either on the port or starboard side of the craft. As vessels grew in size, the course was changed by means of an oar over the after end which was used to produce a lateral force. Later again, this was replaced by a large bladed oar on each quarter of the ship and, in turn, this gave way to a single plate or rudder fitted to the transom. The form of this plate has gradually evolved into the modern rudder. This is streamlined in form to produce a large lift force with minimum drag and with leading edge sections designed to reduce the loss of lift force at higher angles of attack. In some cases the single rudder has given way to twin or multiple rudders.

DIRECTIONAL STABILITY OR DYNAMIC STABILITY OF COURSE

It was seen in Chapter 4, that when disturbed in yaw there are no hydrostatic forces tending either to increase or decrease the deviation in ship's head. In this mode, the ship is said to be in a state of neutral equilibrium. When under way, hydrodynamic forces act on the hull which can have either a stabilizing or destabilizing effect. However, in the absence of any external corrective forces being applied, the ship will not return to its initial line of advance when subject to a disturbance. Hence, *directional stability* cannot be defined in terms precisely similar to those used for transverse stability. A ship is said to be directionally stable when, having suffered a disturbance from an initial straight path, it tends to take up a new straight line path.

In the study of ships, directional stability usually refers to the situation without any control forces being applied. That is to say, what may be termed the inherent course stability of the ship is considered. If the degree of instability were small the vessel could probably be steered by a helmsman (or autopilot) but at the expense of continuous control surface movements which would involve greater resistance and more wear and tear on the machinery. A high degree of instability would mean the ship was uncontrollable. At the other end of the scale too high a level of course stability would make the ship hard to control because of lack of response to rudder movements.

Figure 13.1 shows an arrow with a large tail area well aft of its centre of gravity. Consider a small disturbance which deflects the arrow through a small angle ψ relative to its initial trajectory. The velocity of the arrow is still substantially along the direction of the initial path and the tail surfaces, being now at an angle of attack ψ, develop a lift force F which is in the direction

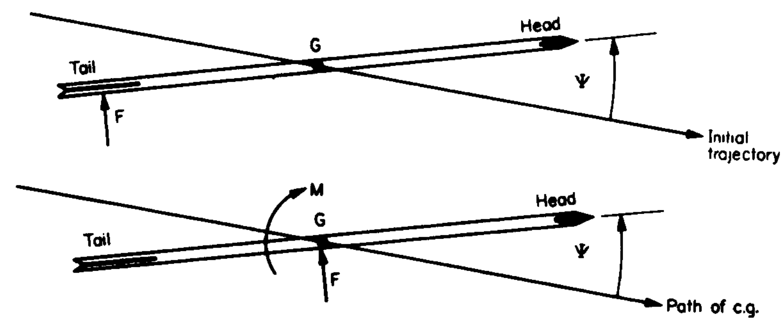


Fig. 13.1 Arrow suffering a disturbance

shown. This force is clearly acting in such a manner as to reduce ψ since it is equivalent to a side force F acting at, and a moment M acting about, the e.g. Provided the tail surfaces are large enough, the forces acting on the rest of the arrow will be negligible compared with F. Other forces will be acting on the head and shank of the arrow so that F should, more precisely, be regarded as the resultant force acting on the arrow.

The sideways force acting at G will have the effect of changing the direction of movement of the e.g. but, as M causes ψ to decrease, M and F will decrease becoming zero and then negative as the axis of the arrow passes through the path of the e.g. Thus, the arrow will oscillate a little and then settle down on a new straight path, ignoring gravitational pull. The deviation from the original path will depend upon the damping effect of the air and the relative magnitudes of M and F. It follows that to maintain a near constant path, the arrow should have large tail surfaces as far aft of G as possible. Directional stability of this very pronounced type is often referred to as 'weather-cock' stability by an obvious analogy.

Application to a ship

It is not possible to tell merely by looking at the lines of a ship whether it is direction ally stable or not. Applying the principle enunciated above, it can be argued that for directional stability the moment acting on the hull and its appendages must be such that it tends to oppose any yaw caused by a disturbance, i.e. the resultant force must act aft of G. The point at which it acts is commonly called the *centre of lateral resistance*. As a general guide, therefore, it is to be expected that ships with large skegs aft and with well rounded forefoot will tend to be more directionally stable than a ship without these features but otherwise similar. Also, as a general guide, long slender ships are likely to be more direction ally stable than short, tubby forms.

The sign convention used in this chapter is illustrated in Fig. 13.2.

It is necessary to differentiate between 'inherent' and 'piloted' controllability. The former represents a vessel's open loop characteristics and uses the definition that when, in a given environment, a ship can attain a specified manoeuvre with some steering function, that ship is said to be manoeuvrable. This ability

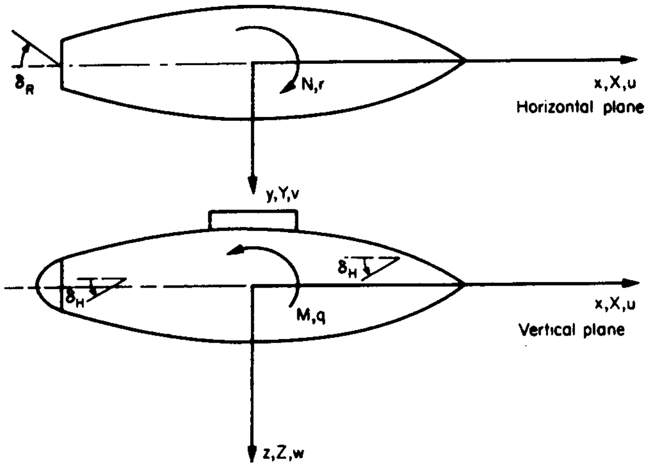


Fig. 13.2 Sign convention

depends upon the environment so that some situations could arise in which the ship becomes unmanoeuvrable. Piloted manoeuvrability reflects the ability of a ship, when controlled by a human operator or an autopilot, to perform a manoeuvre such that deviations from a preset mission remain within acceptable limits. In deciding whether a ship is manoeuvrable in this sense the mission must be specified and the limits within which it is to be achieved.

STABILITY AND CONTROL OF SURFACE SHIPS

For a surface ship we need only consider linear motions along the x and y axes and angular motion about the z axis, the axes used being body axes. If the ship is disturbed from its straight line course in such a way that it has a small sideways velocity v it will experience a sideways force and a yawing moment which can be denoted by Y_v and N_v respectively. If this was the only disturbance the ship would exhibit directional stability if the moment acted so as to reduce the angle of yaw and hence v. In the more general case the disturbed ship will have an angular velocity, angular and linear accelerations and will be subject to rudder actions. All these will introduce forces and moments. Considering only small deviations from a straight path so that second order terms can be neglected, the linear equations governing the motion become

$$(m - Y_{\dot{v}})\dot{v} = Y_v v + (Y_r - m)r + Y_{\delta R}\delta_R$$
$$(I - N_{\dot{r}})\dot{r} = N_v v + N_r r + N_{\delta_R}\delta_R$$

where subscripts v, rand bR denote differentiation with respect to the lateral component of velocity (radial), rate of change of heading and rudder angle respectively, i.e. Y_v = 8Y 18v, etc. Y denotes component of force on ship in y direction and N the moment of forces on ship about z-axis. m is the mass of the ship.

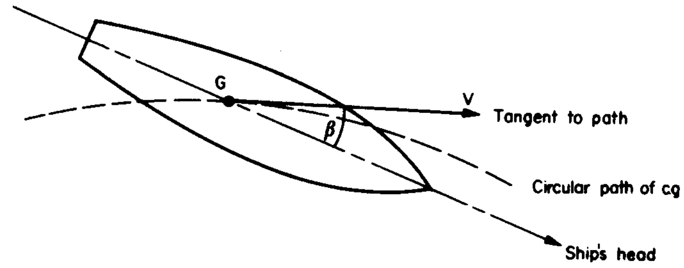


Fig. 13.3

Equations are needed only for motion along the transverse axis and about the vertical axis as it is assumed that the ship has a steady forward speed. Put into words the equations are saying no more than that the rate of change of momentum in the y axis direction is equal to the force in that direction and that that force is the sum of all such terms as (rate of change of Y with lateral velocity) $\times$ (lateral velocity).

The equations can be expressed non-dimensionally by

$$(m' - Y'_{\dot{v}})\dot{v}' = Y'_v v' + (Y'_r - m')r' + Y'_{\delta_R}\delta'_R$$
$$(I' - N'_{\dot{r}})\dot{r}' = N'_v v' + N'_r r' + N'_{\delta_R}\delta'_R$$

From Fig. 13.3 $v' = \dfrac{\dot{y}}{V} = -\sin\beta$ and the non-dimensional turn rate $r' = \dot{\psi}\dfrac{L}{V} = \dfrac{L}{R}$ where R is the radius of curvature of the path at that point. The coefficients Y'_v, N'_v, etc., are termed the stability derivatives.

As a typical example

$$Y'_{\delta_R} = \frac{1}{\frac{1}{2}\rho V^2 L^2}\frac{\partial Y}{\partial \delta_R}$$

The directional stability of a ship is related to its motion with no corrective, i.e. rudder, forces applied. In this case the equations become

$$(m' - Y'_{\dot{v}})\dot{v}' = Y'_v v' + (Y'_r - m')r'$$
$$(I' - N'_{\dot{r}})\dot{r}' = N'_v v' + N'_r r'$$

from which it follows that

$$(m' - Y'_{\dot{v}})\left[\frac{(I' - N'_{\dot{r}})\ddot{r}' - N'_r\dot{r}'}{N'_v}\right] = Y'_v\left[\frac{(I' - N'_{\dot{r}})\dot{r}' - N'_r r'}{N'_v}\right] + (Y'_r - m')r'$$

$$(m' - Y'_{\dot{v}})(I' - N'_{\dot{r}})\ddot{r}' - [(m' - Y'_{\dot{v}})N'_r + (I' - N'_{\dot{r}})Y'_v]\dot{r}'$$
$$+ [N'_r Y'_v - (Y'_r - m')N'_v]r' = 0.$$

This equation is of the form

$$\left[a\frac{\mathrm{d}}{\mathrm{d}t^2} + b\frac{\mathrm{d}}{\mathrm{d}t} + c\right]r = 0$$

which has as a general solution of the form

$$r' = r_1 e^{m_1 t} + r_2 e^{m_2 t}$$

where m_1 and m_2 are the roots of the equation

$$am^2 + bm + c = 0$$

$$m = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

In a stable ship any initial oscillation must decay to zero, which requires both m_1 and m_2 to be negative.

a and b are always positive for ships and the complex solution of the differential equation does not appear to occur. The condition for stability or stability criterion then becomes $c > 0$,

$$\text{i.e. } N'_r Y'_v - N'_v(Y'_r - m') > 0$$

$$\text{i.e. } \frac{N'_r}{Y'_r - m'} > \frac{N'_v}{Y'_v}$$

Thus the condition for stability reduces to a requirement that the centre of pressure in pure yaw should be ahead of that in pure sway.

Returning to the earlier equations, for a steady turn $\dot{v}$ and $\dot{r}$ are zero giving

$$0 = Y'_v v' + (Y'_r - m')r' + Y'_{\delta_R}\delta'_R$$
$$0 = N'_v v' + N'_r r' + N'_{\delta_R}\delta'_R$$

This leads to a relationship between r' and δ'_R as follows:

$$(N'_r r' + N'_{\delta_R}\delta'_R)Y'_v = (Y'_r - m)N'_v r' + Y'_{\delta_R}\delta_R N'_v.$$

$$[N'_r Y'_v - N'_v(Y'_r - m')]r' = [Y'_{\delta_R} N'_v - Y'_v N'_{\delta_R}]\delta'_R$$

$$\frac{r'}{\delta'_R} = \frac{Y'_{\delta_R} N'_v - Y'_v N'_{\delta_R}}{Y'_v N'_r - N'_v(Y'_r - m')}$$

It will be noted that the denominator is the stability criterion obtained above. This seems reasonable on general grounds. If the denominator in the expression for r// 8~ were zero then r// 8~ becomes infinite and the ship will turn in a circle with no rudder applied. Thus for a stable ship the denominator would be expected to be non-zero. Also by referring to Fig. 13.2 it will be seen that r'/8R must be negative for a stable ship. Following from the sign convention and the geometry of the ship, YR is positive, NR and Y~ are negative. It will also be seen later that Yeffectively acts forward of the centre of gravity so that N~is also negative. Thus the denominator must be positive for a stable ship.

An important point in directional control is the so-called *Neutral Point* which is that point, along the length of the ship, at which an applied force, ignoring transient effects, does not cause the ship to deviate from a constant heading. This point is a distance ηL forward of the centre of gravity, where

$$\eta = \frac{N'_v}{Y'_v}$$

Typically, η is about one-third, so that the neutral point is about one-sixth of the length of the ship abaft the bow.

It can be readily checked (Fig. 13.4) that with a force applied at the neutral point the ship is in a state of steady motion with no change of heading but with a steady lateral velocity, i.e. a steady angle of attack.

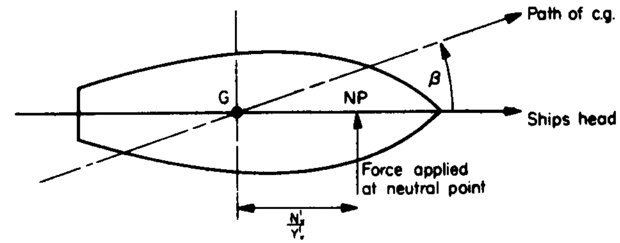


Fig. 13.4 Location of neutral point

When moving at an angle of attack β, lateral velocity $-v$, the non-dimensional hydrodynamic force and moment are vY'_v and vN'_v respectively, i.e. the hydrodynamic force effectively acts at a distance $(N'_v/Y'_v)L$ ahead of the c.g. directly opposing the applied force, so that there is no tendency for the ship's head to change. If the applied force is of magnitude F, then the resulting lateral velocity is

$$v = \frac{F}{Y_v}$$

Until the velocity has built up to this required value, there will be a state of imbalance and during this phase there can arise a change of heading from the initial heading.

It follows, that if the force is applied aft of the neutral point and acts towards port the ship will turn to starboard, and if applied in the same sense forward of the neutral point the ship turns to port. Clearly, the greater the distance of application of the force from the neutral point the greater the turning influence, other things being equal. This explains why rudders are more effective when placed aft. If $\eta = \frac{1}{3}$, then the 'leverage' of a stern rudder is five times that of a bow rudder. At the stern also, the rudders gain from the effect of the screw race.

If, in the equation above for r'/δ_R, $N'_{\delta_R} = x' Y'_{\delta_R}$ then

$$\frac{r'}{\delta_R} \text{ is proportional to } \frac{N'_v}{Y'_v} - x'$$

That is, for a given rudder angle, the rate of change of heading is greatest when the value of x' is large and negative. This again shows that a rudder is most effective when placed right aft.

THE ACTION OF A RUDDER IN TURNING A SHIP

The laws of dynamics demand that when a body is turning in a circle, it must be acted upon by a force acting towards the centre of the circle of sufficient magnitude to impart to the body the required radial acceleration. In the case of a ship, this force can only arise from the aerodynamic and hydrodynamic forces acting on the hull, superstructure and appendages. It is usual, in studying the turning and manoeuvring of ships, to ignore aerodynamic forces for standard manoeuvres and to consider them only as disturbing forces. That is not to say that aerodynamic forces are unimportant. On the contrary, they may prevent a ship turning into the wind if she has large windage areas forward.

To produce a radial force of the magnitude required, the hull itself must be held at an angle of attack to the flow of water past the ship. The rudder force must be capable of holding the ship at this angle of attack; that is, it must be able to overcome the hydrodynamic moments due to the angle of attack and the rotation of the ship. The forces acting on the ship during a steady turn are illustrated in Fig. 13.5 where F_H is the force on the hull and F_R the rudder force. F_H is the resultant of the hydrodynamic forces on the hull due to the angle of attack and the rotation of the ship as it moves around the circle.

If T is the thrust exerted by the propellers and F_H and F_R act at angles α and γ relative to the middle line plane then, for a steady turn with forces acting as shown in Fig. 13.5, these forces must lead to the radial force $\Delta V^2/R$, i.e.

$$T - \frac{\Delta V^2}{R}\sin\beta = F_H\cos\alpha + F_R\cos\gamma$$

$$-\frac{\Delta V^2}{R}\cos\beta + F_H\sin\alpha = F_R\sin\gamma$$

$$F_H\overline{GE} + F_R\overline{GJ} = 0$$

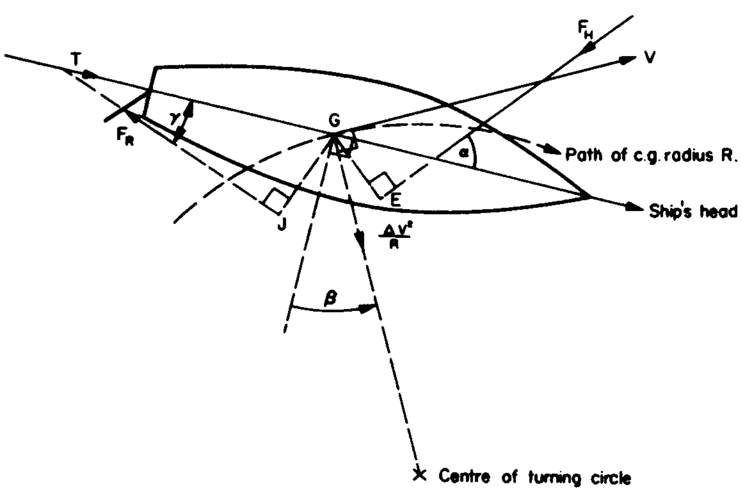


Fig. 13.5 Action of rudder in turning ship

The radial components of the forces on the rudder and the hull, F_R and F_H must have a resultant causing the radial acceleration.

LIMITATIONS OF THEORY

The student has been introduced to the simple concepts of a linear theory of motion. This is useful in providing an insight into the manoeuvring of ships but many problems are, or appear to be, highly non-linear. This has led to the introduction of higher degrees of derivative to obtain a better representation of the way forces and moments, whose deviations from a steady state condition are other than small, can vary. Such problems concern, for example, steering in a seaway (particularly in a following sea), high-speed large-angle submarine manoeuvring when the body shape may have important effects, athwartships positioning of big ships and drilling vessels. Unfortunately, such approaches are critically dependent upon the validity of the mathematical representation adopted for the fluid forces. One limitation of these analyses is that they assume the forces and moments acting on the model to be determined by the motion obtaining at that instant and are unaffected by its history. This is not true. For instance, it has been shown that when two fins (like a ship and its rudder) are moving in tandem and the first is put to an angle of attack, there is a marked time delay before the second fin experiences a change of force. This approach uses a linear functional mathematical representation which includes a 'memory' effect and shows how the results in the frequency and time domains are related. The approach is limited to linear theory but the inclusion of memory effects provides an explanation for at least some of the effects which arise in large amplitude motions.

Assessment of manoeuvrability

Assessment of manoeuvrability is made difficult by the lack of rigorous analytical methods and of universally accepted standards for manoeuvrability. The hydrodynamic behaviour of a vessel on the interface between sea and air is inherently complex. Whilst reasonable methods exist for initial estimates of resistance and powering and motions in a seaway, the situation is less satisfying as regards manoeuvring. Much reliance is still placed upon model tests and fullscale trials using a number of common manoeuvres which are outlined below.

THE TURNING CIRCLE

Figure 13.6 shows diagrammatically the path of a ship when executing a starboard turn. When the rudder is put over initially, the force acting on the rudder tends to push the ship bodily to port of its original line of advance. As the moment due to the rudder force turns the ship's head, the lateral force on the hull builds up and the ship begins to turn. The parameters at any instant of the turn are defined as:

Drift angle. The drift angle at any point along the length of the ship is defined as the angle between the centre line of the ship and the tangent to the path of the

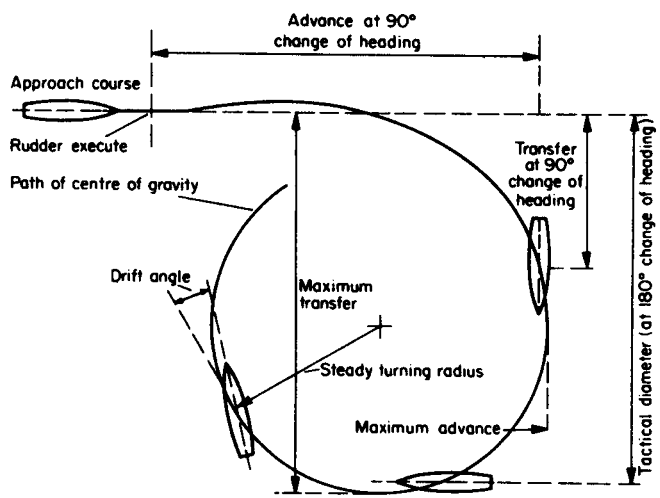


Fig. 13.6 *Geometry of turning circle*

point concerned. When a drift angle is given for the ship without any specific point being defined, the drift angle at the centre of gravity ofthe ship is usually intended. Note that the bow of the ship lies within the circle and that the drift angle increases with increasing distance aft of the pivoting point which is defined below.

Advance. The distance travelled by the centre of gravity in a direction parallel to the original course after the instant the rudder is put over. There is a value of advance for any point on the circle, but if a figure is quoted for advance with no other qualification the value corresponding to a 90 degree change of heading is usually intended.

Transfer. The distance travelled by the centre of gravity perpendicular to the original course. The transfer of the ship can be given for any point on the circle, but if a figure is quoted for transfer with no other qualification the value corresponding to a 90 degree change of heading is usually intended.

Tactical diameter. The value of the transfer when the ship's heading has changed by 180 degrees. It should be noted that the tactical diameter is not the maximum value of the transfer.

Diameter of steady turning circle. Following initial application of the rudder there is a period of transient motion, but finally the speed, drift angle and turning diameter reach steady values. This usually occurs after about 90 degrees change of heading but, in some cases, the steady state may not be achieved until after 180 degrees change of heading. The steady turning diameter is usually less than the tactical diameter.

Pivoting point. This point is defined as the foot of the perpendicular from the centre of the turn on to the middle line of the ship extended if necessary. This is not a fixed point, but one which varies with rudder angle and speed. It may be forward of the ship as it would be in Fig. 13.6, but is typically one-third to one-sixth of the

length of the ship abaft the bow. It should be noted, that the drift anglc is zero lit the pivoting point and increases with increasing distance from that point.

The turning circle has been a standard manoeuvre carried out by all ships as an indication of the efficiency of the rudder. Apart from what might be tcrmed the 'geometric parameters' of the turning circle defined above loss of speed on turn and angle of heel experienced are also studied.

Loss of speed on turn

As discussed above, the rudder holds the hull at an angle of attack, i.e. the drift angle, in order to develop the 'lift' necessary to cause the ship to accelerate towards the centre of the turn. As with any other streamlined form, hull lift can be produced only at the expense of increased drag. Unless the engine settings are changed, therefore, the ship will decelerate under the action of this increased drag. Most ships reach a new steady speed by the time the heading has changed 90 degrees but, in some cases, the slowing down process continues until about 180 degrees change of heading.

Angle of heel when turning

When turning steadily, the forces acting on the hull and rudder are *FH* and *FR.* Denoting the radial components of these forces by lower case subscripts (i.e. denoting these by *Fh* and *Fr* respectively) and referring to Fig. 13.7, it is seen that to produce the turn

$$F_\mathrm{h} - F_\mathrm{r} = \frac{\Delta V^2}{Rg}$$

where V = speed on the turn, R = radius of turn.

$$\begin{aligned}\text{Moment causing heel} &= (F_\mathrm{h} - F_\mathrm{r})\overline{\mathrm{KG}} + F_\mathrm{r}(\overline{\mathrm{KH}}) - F_\mathrm{h}(\overline{\mathrm{KE}})\\ &= (F_\mathrm{h} - F_\mathrm{r})(\overline{\mathrm{KG}} - \overline{\mathrm{KE}}) + F_\mathrm{r}(\overline{\mathrm{KH}} - \overline{\mathrm{KE}})\\ &= (F_\mathrm{h} - F_\mathrm{r})\overline{\mathrm{GE}} - F_\mathrm{r}\overline{\mathrm{EH}}\end{aligned}$$

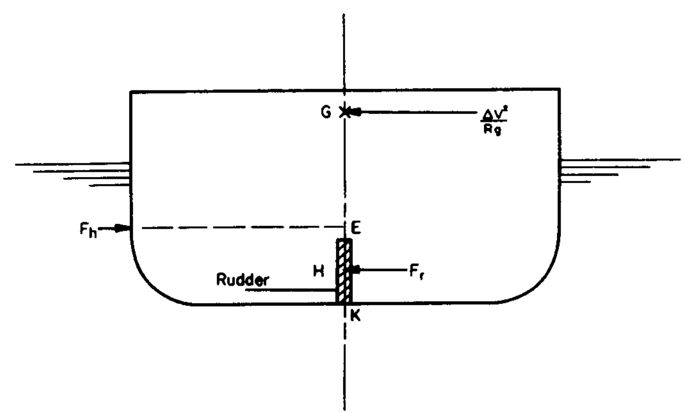


Fig. 13.7 Forces producing heel when turning

For most ships, E, the centre of lateral resistance, and H are very close and this expression is given approximately by

Moment causing heel = $(Fh - Fr)$G-E

This moment causes the ship to heel outwards during the steady turn. When the rudder is initially put over, however, Fr acts before Fh has built up to any significant value and during this transient phase the ship may heel inwards. It should also be noted that the effect of Fr during the steady turn is to reduce the angle of heel, so that if the rudder angle is suddenly taken off, the ship will heel to even larger angles. If the rudder angle were to be suddenly reversed even more serious angles of heel could occur.

It will be appreciated that Fh acts at the centre oflateral resistance only if the angle of heel is small. For large heel angles, the position of E is difficult to assess. For small angles of heel

$$\Delta\overline{\mathrm{GM}}\sin\phi = (F_{\mathrm{h}} - F_{\mathrm{r}})\overline{\mathrm{GE}}$$
$$= \frac{\Delta V^2}{Rg}\overline{\mathrm{GE}}$$

Hence

$$\frac{Rg\sin\phi}{V^2} = \frac{\overline{\mathrm{GE}}}{\overline{\mathrm{GM}}}$$

It must be emphasized, however, that the angle of heel obtained by this type of calculation should only be regarded as approximate. Apart from the difficulty of accurately locating E, some ships, particularly high speed vessels, suffer an apparent loss of stability when underway because of the other forces acting on the ship and appendages due to the flow around the ship when it is turning.

TURNING ABILITY

The turning circle characteristics are not by themselves indicators of initial response to rudder, which may be important when ships are operating in confined waters or in close company. Indeed, some factors which have a major impact on initial response have very little effect on tactical diameter. One indicator that can be used is the heading angle turned through from an initially straight course, per unit rudder angle applied, after the ship has travelled one ship length. Whilst theoretical prediction of tactical diameter is difficult because of non-linearities, linear theory can be used to calculate this initial response and it is possible to derive an expression for it in terms of the stability derivatives.

Multiple regression techniques can be used to deduce approximate empirical formulae for a design's stability derivatives from experimental data. Although they do not accurately account for all the variations in experimental data they are reproduced below as an aid to students.

$$-Y'_{\dot v}/\pi(T/L)^2 = 1 + 0.16C_B B/T - 5.1(B/L)^2$$
$$-Y'_{\dot r}/\pi(T/L)^2 = 0.67B/L - 0.0033(B/T)^2$$
$$-N'_{\dot v}/\pi(T/L)^2 = 1.1B/L - 0.041B/T$$

$$-N'_{\dot r}/\pi(T/L)^2 = 1/12 + 0.017C_B B/T - 0.33B/L$$
$$-Y'_{v}/\pi(T/L)^2 = 1 + 0.40C_B B/T$$
$$-Y'_{r}/\pi(T/L)^2 = -1/2 + 2.2B/L - 0.080B/T$$
$$-N'_{v}/\pi(T/L)^2 = 1/2 + 2.4T/L$$
$$-N'_{r}/\pi(T/L)^2 = 1/4 + 0.039B/T - 0.56B/L$$

THE ZIG-ZAG MANOEUVRE

It can be argued that it is not often that a ship requires to execute more than say a 90 or 180 degree change of heading. On the other hand, it often has to turn through angles of 10, 20 or 30 degrees. It can also be argued that in an emergency, such as realization that a collision is imminent, it is the initial response of a ship to rudder movements that is the critical factor. Unfortunately, the standard circle manoeuvre does not adequately define this initial response and the standard values of transfer and advance for 90 degrees change of heading and tactical diameter are often affected but little by factors which have a significant influence on initial response to rudder. Such a factor is the rate at which the rudder angle is applied. This may be typically 3 degrees per second. Doubling this rate leads to only a marginally smaller tactical diameter but initial rates of turn will be increased significantly.

The zig-zag manoeuvre, sometimes called a Kempf manoeuvre after G. Kempf, is carried out to study more closely the initial response of a ship to rudder movements (see Fig. 13.8). A typical manoeuvre would be as follows. With the ship proceeding at a steady speed on a straight course the rudder is put over to 20 degrees and held until the ship's heading changes by 20 degrees. The rudder angle is then changed to 20 degrees in the opposite sense and so on.

Important parameters of this manoeuvre are:

(a) the time between successive rudder movements;
(b) the *overshoot* angle which is the amount by which the ship's heading exceeds the 20 degree deviation before reducing.

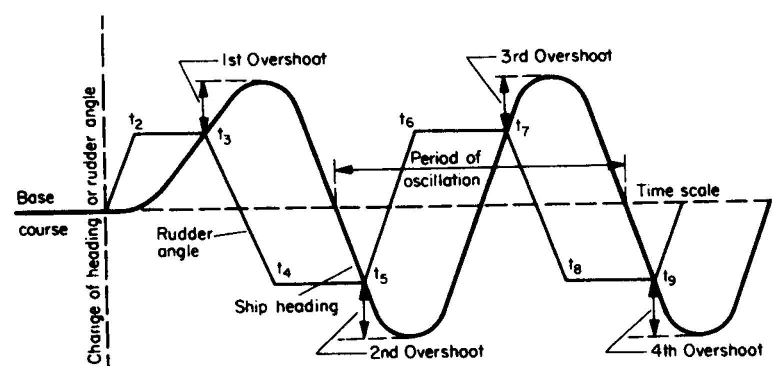


Fig. 13.8 The zig-zag manoeuvre

The manoeuvre is repeated for a range of approach speeds and for different values of the rudder angle and heading deviation.

THE SPIRAL MANOEUVRE

This manoeuvre, sometimes referred to as the Dieudonne Spiral after J. Dieudonne who first suggested it, provides an indication of a ship's directional stability or instability.

To perform this manoeuvre, the rudder is put over to say 15 degrees starboard and the ship is allowed to turn until a steady rate of change of heading is achieved. This rate is noted and the rudder angle is reduced to 10 degrees and the new steady rate of change of heading is measured. Successive rudder angles of 5°S, 0°, 5°P, 10°P, 15°P, 10°P, 5°P, 0°, 5°S, 10°S and 15°S are then used. Thus, the steady rate of change of heading is recorded for each rudder angle when the rudder angle is approached both from above and from below. The results are plotted as in Fig. 13.9, in which case *(a)* represents a stable ship and case *(b)* an unstable ship.

In the case of the stable ship, there is a unique rate of change of heading for each rudder angle but, in the case of an unstable ship, the plot exhibits a form of 'hysteresis' loop. That is to say that for small rudder angles the rate of change of heading depends upon whether the rudder angle is increasing or decreasing. That part of the curve shown dotted in the figure cannot be determined from ship trials or free model tests as it represents an unstable condition.

It is not possible to deduce the degree of instability from the spiral manoeuvre, but the size of the loop is a qualitative guide to this. Of direct practical significance, it should be noted that it cannot be said with certainty that the ship will turn to starboard or port unless the rudder angle applied exceeds δ_s or δ_p, respectively and controlled turns are not possible at low rates of turn.

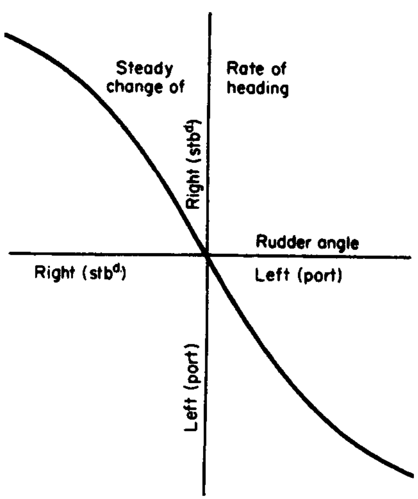


Fig. 13.9(a) Presentation of spiral manoeuvre results (stable ship)

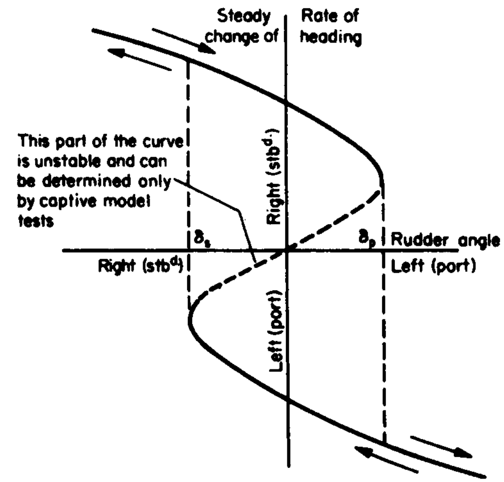


Fig. 13.9(b) Presentation of spiral manoeuvre results (unstable ship)

THE PULL-OUT MANOEUVRE

This manoeuvre is used to determine the directional stability of a ship. The rudder is put over to a predetermined angle and held. When the ship is turning at a steady rate the rudder is returned to amidships and the change of rate of turn with time is noted. If the ship is directionally stable the rate of turn reduces to zero and the ship takes up a new straight path. If the ship is unstable a residual rate of turn will persist. The manoeuvre can be conveniently carried out at the end of each circle trial during ship trials.

It has been found that for a stable ship a plot of the log of rate of turn against time is a straight line after an initial transient period.

It was shown in the section on theory that the differential equation of motion had two roots m_1 and m_2 both of which had to be negative for directional stability. It has been argued that the more negative root will lead to a response which dies

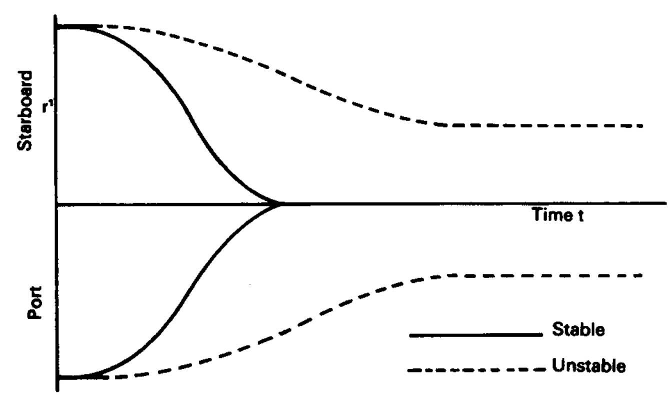


Fig. 13.10 Pull-out manoeuvre, rate of turn on time base

out during the transient phase so that the straight portion of the log rate/time curve gives the root of smaller numerical value. Thus if this root is ml,

$$r' = re^{mlt}$$

$$\log r' = \log r + mlt$$

The area under the curve of turn rate against time gives the total heading change after the rudder is centred. Thus the less the total change the more stable the ship.

STANDARDS FOR MANOEUVRING AND DIRECTIONAL STABILITY

The standards required in any particular design depend upon the service for which the ship is intended but, in any case, they are not easy to define. The problem is made more difficult by the fact that good directional stability and good manoeuvrability are to some extent conflicting requirements, although they are not actually incompatible as has often been suggested. For instance, a large rudder can increase the directional stability and also improve turning performance. Also, in a long fine form increasing draught-to-length ratio can increase stability without detriment to the turning. On the other hand, increasing beam-to-Iength ratio improves turning but reduces the directional stability. Placing a large skeg aft will improve directional stability at the expense of poorer turning ability.

For a large ocean-going ship, it is usually possible to assume that tugs will be available to assist her when manoeuvring in the confined waters of a harbour. The emphasis in design is therefore usually placed on good directional stability for the long ocean transits. This leads to less wear on the rudder gear, especially if an automatic control system is fitted, and reduces overall average resistance. The highest degree of directional stability is demanded for ships likely to suffer disturbances in their normal service such as supply ships replenishing smaller naval units at sea.

For medium size ships which spend relatively more time in confined waters and which do not normally make use of tugs, greater emphasis has to be placed on response to rudder. Typical of these are the cross channel steamers and anti-submarine frigates.

What are the parameters that are to be used to define the manoeuvring capabilities? They are those parameters measured in the various manoeuvres described in the earlier sections of this chapter. Typical values to be expected are discussed below.

Tactical diameter-to-length ratio. For ships in which tight turning is desirable this may be, say, 3.25 for modern naval ships at high speed, with conventional rudders at 35 degrees. Where even smaller turning circles are required, recourse is usually made to some form of lateral thrust unit.

A *T.D./L* value of 4.5 is suggested as a practicable criterion for merchant types desiring good handling performance. Values of this ratio exceeding 7 are regarded as very poor.

Turning rate. For very manoeuvrable naval ships this may be as high UN ~ degrees per second. For merchant types, rates of up to 1.5 degrees per second should be achieved in ships of about 100m at 16knots, but generally valucs or 0.5-1.0 degrees per second are more typical.

Speed on turn. This can be appreciably lower than the approach speed, and typically is only some 60 per cent of the latter.

Initial turning. It has been proposed that the heading change per unit rudder angle in one ship length travelled should be greater than 0.3 generally and greater than 0.2 for large tankers.

Angle of heel. A very important factor in passenger ships and one which may influence the standard of transverse stability incorporated in the design.

Directional stability. Clearly, an important factor in a well balanced design. The inequality presented earlier as the criterion for directional stability can be used as a 'stability index'. Unfortunately, this is not, by itself, very informative. A reasonable design aim is that the spiral manoeuvre should exhibit no 'loop', i.e. the design should be stable even if only marginally so. Using the pull-out manoeuvre it has been suggested that using the criterion of total heading change after the rudder is centred 15-20 degrees represents good stability, 35--40 degrees reasonable stability but that 80-90 degrees indicates marginal stability.

Time to turn through 20 degrees. This provides a measure of the initial response of the ship to the application of rudder. It is suggested that the time to reach 20 degrees might typically vary from 80 to 30 seconds for speeds of 6-20 knots for a 150m ship. The time will vary approximately linearly with ship length.

Overshoot. The overshoot depends on the rate of turn and a ship that turns well will overshoot more than one that does not turn well. If the overshoot is excessive, it will be difficult for a helmsman to judge when to start reducing rudder to check a turn with the possible danger of damage due to collision with other ships or a jetty. The overshoot angle does not depend upon the ship size and values suggested are 5.5 degrees for 8 knots and 8.5 degrees for 16knots, the variation being approximately linear with speed.

Rudder forces and torques

RUDDER FORCE

The rudder, being of streamlined cross-section, will be acted upon by a lift and drag force when held at an angle of attack relative to the flow of water. The rudder must be designed to produce maximum lift for minimum drag assuming that the lift behaves in a consistent manner for all likely angles of attack. The lift developed depends upon:

(a) the cross-sectional shape;
(b) the area of the rudder, A_R;
(c) the profile shape of the rudder and, in particular, the aspect ratio of the rudder, i.e. the ratio of the depth of the rudder to its chord length;

(*d*) the square of the velocity of the water past the rudder;
(*e*) the density of the water, ρ;
(*f*) the angle of attack, α.

Hence, the rudder force F_R, can be represented by

$$F_R = \text{Constant} \times \rho A_R V^2 f(\alpha)$$

the value of the constant depending upon the cross-sectional and profile shapes of the rudder. A typical plot for $f(\alpha)$ is as shown in Fig. 13.11.

At first, $f(\alpha)$ increases approximately linearly with angle of attack but then the rate of growth decreases and further increase in α may produce an actual fall in the value of $f(\alpha)$. This phenomenon is known as *stalling*.

Typically, for ships' rudders, stalling occurs at an angle between 35 and 45 degrees. Most ship rudders are limited to 35 degrees to avoid stall, loss of speed and large heel on turn. Stall is related to the flow relative to the rudder; in turning the water flow is no longer aligned with the ship's hull but across the stern, thereby allowing larger rudder angles before stall occurs than are possible when the rudder is first put over. This cross-flow affects also wake and propeller performance.

Many formulae have been suggested for calculating the forces on rudders. One of the older formulae is

$$\text{Force} = 577 A_R V^2 \sin(\delta_R) \text{ newtons}$$

where A_R is measured in m^2 and V in m/s,

V being the velocity of water past the rudder, allowance must be made for the propeller race in augmenting the ship's ahead speed. Typical values assumed are:

Rudder behind propeller, $V = 1.3 \times$ (Ship speed)
Centre-line rudder behind twin screws, $V = 1.2 \times$ (Ship speed).

Haslar used the following formulae for twin rudders behind wing propellers:

Force $= 21.1 A_R V^2 \delta_R$ newtons, for ahead motion, δ_R is measured in degrees.
Force $= 19.1 A_R V^2 \delta_R$ newtons, for astern motion.

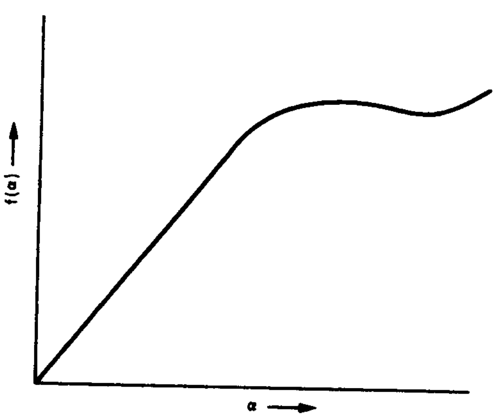


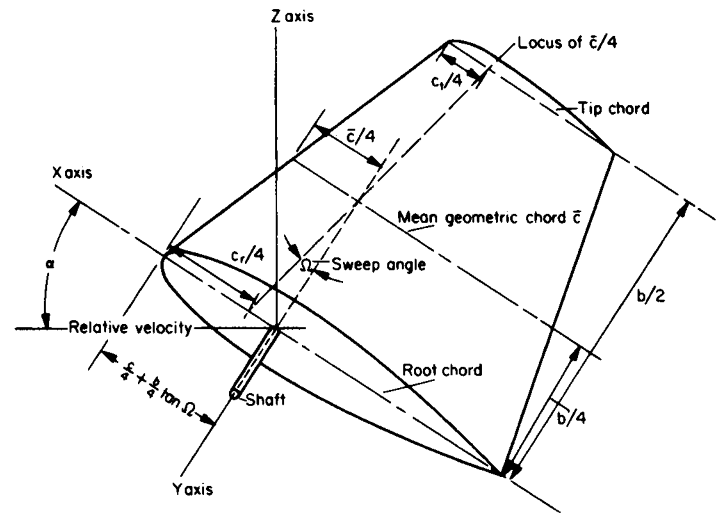
Fig. 13.11 Variation of $f(\alpha)$ with angle of attack

Using the same parameters, Baker and Bottomley have suggested that for middle line rudders behind single screws

$$\text{Force} = 18.0 A_R V^2 \delta_R \text{ newtons}$$

In these formulae V is taken as the true speed of the ship, allowance having been made in the multiplying factors for the propeller race effects.

More comprehensive formulae are given in the literature based on extensive experimental and theoretical work. It is recommended that for naval applications all-movable control surfaces be used with square tips. A good section is the NACA 0015 with a moderately swept quarter chord line. Figure 13.12 illustrates a typical control surface and gives the offsets for the NACA 0015 section, in terms of the chord c and distance x from the nose.



NACA 0015	
x/c	$\pm z/c$
0	0
0.0125	0.02367
0.0250	0.03267
0.0500	0.04442
0.0750	0.05249
0.1000	0.05852
0.1500	0.06680
0.2000	0.07170
0.2500	0.07424
0.3000	0.07500
0.4000	0.07252
0.5000	0.06615
0.6000	0.05703
0.7000	0.04579
0.8000	0.03278
0.9000	0.01809
0.9500	0.01008
1.0000	0.00158

Fig. 13.12 All-movable control surface and NACA 0015 section

The formulae recommended are:

$$C_L = \frac{\text{Lift}}{\frac{1}{2}\rho A V^2} = \left[\frac{a_0 a_e}{\cos\Omega\left(\frac{a_e^2}{\cos^4\Omega} + 4\right)^{\frac{1}{2}} + \frac{57.3 a_0}{\pi}}\right]\alpha + \frac{C_{D_c}}{a_e}\left(\frac{\alpha}{57.3}\right)^2$$

where

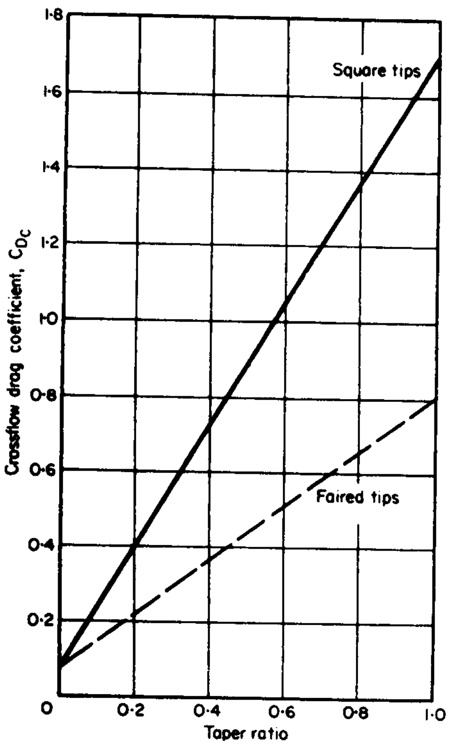
a_e = effective aspect ratio = (span)^2/(planform area)
a_0 = section lift curve slope at $\alpha = 0$
$= 0.9(2\pi/57.3)$ per degree for NACA 0015
C_{D_c} = crossflow drag coefficient (Fig. 13.13)
$= 0.80$ for square tips and taper ratio $= 0.45$

and

$$C_D = \frac{\text{Drag}}{\frac{1}{2}\rho A V^2} = C_{d_0} + \frac{C_L^2}{0.9\pi a_e}$$

where

$$C_{d_0} = \text{minimum section drag coefficient}$$
$$= 0.0065 \text{ for NACA 0015}$$



Taper ratio is the ratio of tip chord to root chord

Fig. 13.13 Crossflow drag coefficient

CENTRE OF PRESSURE POSITION

It has been seen that it is the rudder force which is important in causing a ship to turn, as the lever of the rudder force from the neutral point is not significantly affected by the position of the centre of pressure on the rudder itself. However, it is necessary to know the torque acting on the rudder to ensure that the steering gear installed in the ship is capable of turning the rudder at all speeds.

For a flat plate, Joessel suggested an empirical formula for the proportion of the breadth of the plate that the centre of pressure is abaft the leading edge and expressed it as:

$$0.195 + 0.305\sin(\alpha)$$

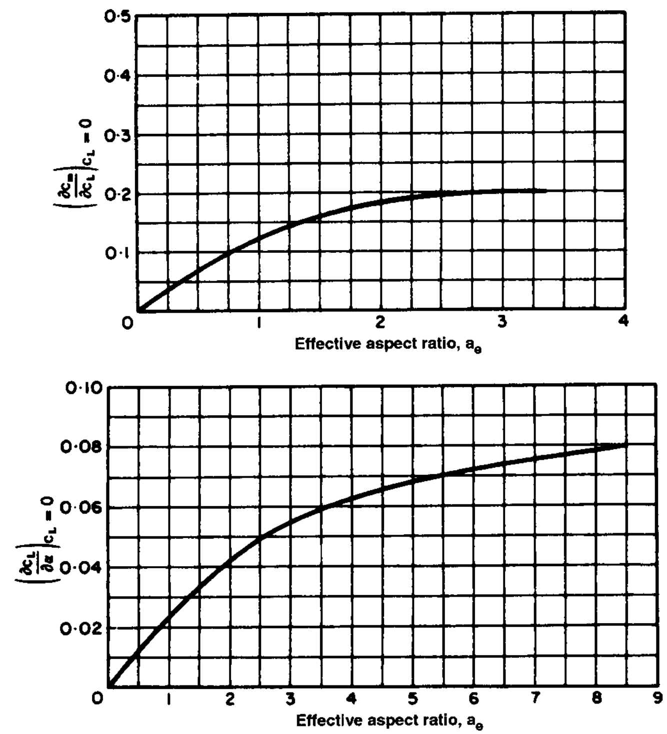


Fig. 13.14 Variation of chordwise centre of pressure and lift curve slope with aspect ratio

For rudders, the geometry of the cross-section will have an influence upon the centre of pressure position. Gawn considered that for a rectangular rudder behind a fin or skeg the centre of pressure is 0.35 times the chord length abaft the leading edge. For a rudder in open water this value is reduced to 0.31. For motion astern the rudder is always effectively in clear water and the figure of 0.31 is used in both cases and measured relative to the after edge of the rudder.

One authority recommends a torque (pitching moment) coefficient

$$C_T = \frac{\text{torque}}{\frac{1}{2}\rho A V^2 \bar{c}} = \left[0.25 - \left(\frac{\partial C_m}{\partial C_L}\right)_{C_L=0}\right]\left(\frac{\partial C_L}{\partial \alpha}\right)_{C_L=0}\alpha - \frac{1}{2}\frac{C_{D_c}}{a_e}\left(\frac{\alpha}{57.3}\right)^2$$

where

$$\bar{c} = \text{mean geometric chord} = \frac{c_t + c_r}{2}$$

$$\left(\frac{\partial C_m}{\partial C_L}\right)_{C_L=0} \quad \text{and} \quad \left(\frac{\partial C_L}{\partial \alpha}\right)_{C_L=0}$$

are defined in Fig. 13.14.

Torque is measured about the quarter-chord point of the mean geometric chord.

The centre of pressure is defined chordwise and spanwise by the following relationships:

Chordwise from leading edge at the mean geometric chord (as percentage of the mean geometric chord),

$$= 0.25 - \frac{C_T}{C_L \cos\alpha + C_D \sin\alpha}$$

Spanwise measured from the plane of the root section (in terms of the semi-span):

$$= \frac{C_L\left(\frac{4}{3\pi}\frac{b}{2}\right)\cos\alpha + C_D\left(\frac{b}{2}\right)\sin\alpha}{\frac{b}{2}(C_L\cos\alpha + C_D\sin\alpha)}$$

Typical curves for a NACA 0015 section control surface are reproduced in Fig. 13.15.

In the absence of any better guide, the figures quoted above should be used in estimating rudder forces and torques. However, because of the dependence of both force and centre of pressure on the rudder geometry it is recommended that actual data for a similar rudder be used whenever it is available. In many instances, rudders have sections based on standard aerofoil sections and, in this case, use should be made of the published curves for lift and centre of pressure positions making due allowance for the effect of propellers and the presence of the hull on the velocity of flow over the rudder.

In practice, the picture is complicated by the fact that the flow of water at the stern of a ship is not uniform and may be at an angle to the rudders when set nominally amidships. For this reason, it is quite common practice to carry out model experiments to determine the hydrodynamic force and torque acting on the rudder. As a result of such tests, it may be deemed prudent to set the rudders of a twin rudder ship at an angle to the middle line plane of the ship for their 'amidships' position.

CALCULATION OF FORCE AND TORQUE ON NON-RECTANGULAR RUDDER

It is seldom that a ship rudder is a simple rectangle. For other shapes the rudder profile is divided into a convenient number of strips. The force and centre of pressure are assessed for each strip and the overall force and torque obtained by summating the individual forces and torques.

EXAMPLE 1. Calculate the force and torque on the centre line gnomon rudder shown, Fig. 13.16, for 35 degrees and a ship speed of 20 knots. The ship is fitted with twin screws.

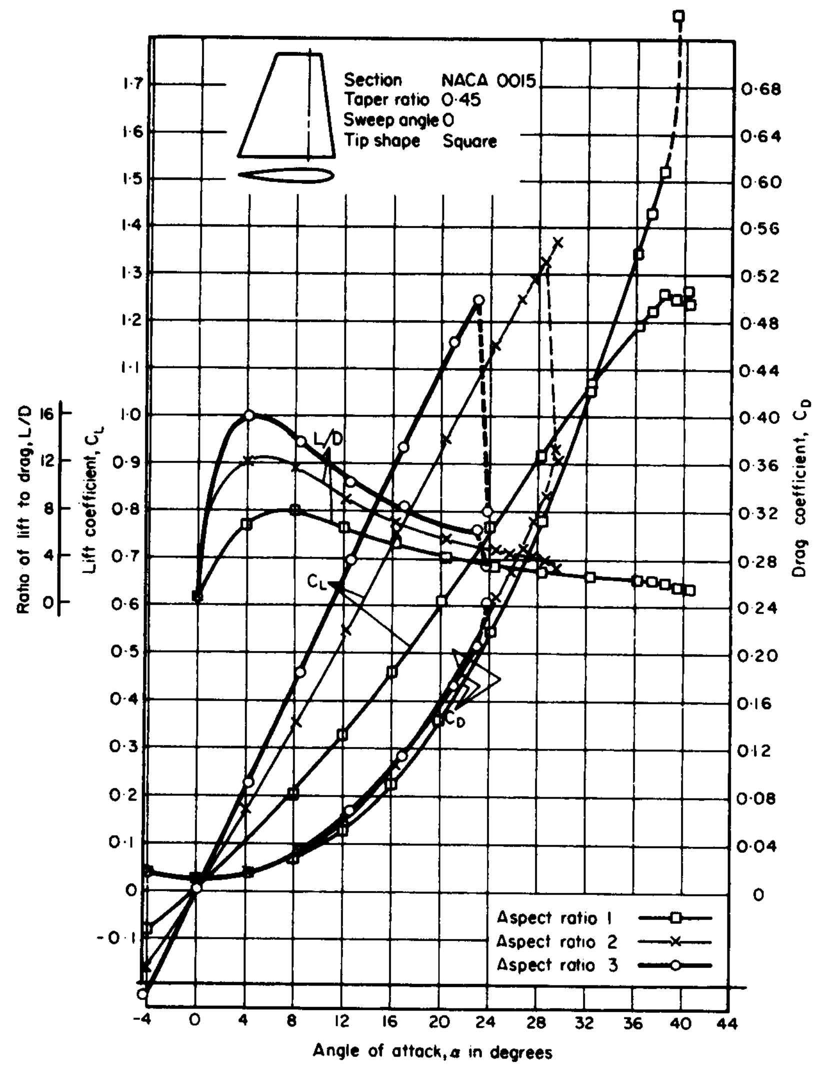


Fig. 13.15 Lift and drag for NACA 0015 sections

Solution: The rudder can conveniently be divided into two rectangular areas A_1 and A_2, A_1 being the smaller. Applying the older formula for force and Gawn's formulae for c.p. position.

Area A_1 is behind a skeg

$$\therefore \quad \text{Force on } A_1 = 557 \times 9 \times (1.2 \times 20 \times 0.51477)^2 \sin 35 = 0.4\,\text{MN}$$

$$\text{c.p. aft of axis} = 0.35 \times 3 = 1.05\,\text{m}$$

$$\text{Moment on } A_1 = 1.05 \times 0.4 = 0.42\,\text{MN m}$$

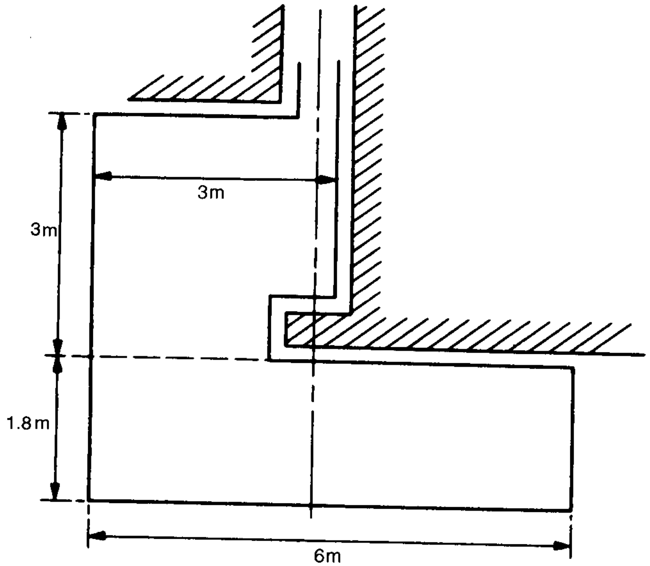


Fig. 13.16

$$\text{Force on } A_2 = 557 \times 10.8 \times (1.2 \times 20 \times 0.51477)^2 \sin 35 = 0.48\,\text{MN}$$

$$\text{c.p. aft of axis} = 0.31 \times 6.3 = -1.14\,\text{m}$$

$$\text{Moment on } A_2 = -1.14 \times 0.48 = -0.547\,\text{MN m}$$

Hence resultant force on rudder $= 0.88\,\text{MN}$

resultant moment $= -0.127\,\text{MN m}$

with c.p. forward of the axis.

EXAMPLE 2. Calculate the force and torque on the spade rudder shown in Fig. 13.17, which is one of two working behind twin propellers. Assume a rudder angle of 35 degrees and a ship speed of 20 knots ahead. If the stock is solid with a section modulus in bending of $0.1\,\text{m}^3$, calculate the maximum stress due to the combined torque and bending moment.

Solution: Assuming that the force on the rudder is given by $21.1\, A_R V^2 \delta R$ newtons, that the c.p. is $0.31 \times$ the chord length aft of the leading edge and that the force acts at the same vertical position as the centroid of area of the rudder.

$$\text{Area of rudder} = \frac{1}{3} \times 1 \times 32.5 = 10.83\,\text{m}^2$$

$$\text{c.p. aft of axis} = 5.37/32.5 = 0.165\,\text{m}$$

$$\text{c.p. below stock} = \frac{48.8}{32.5} = 1.502\,\text{m}$$

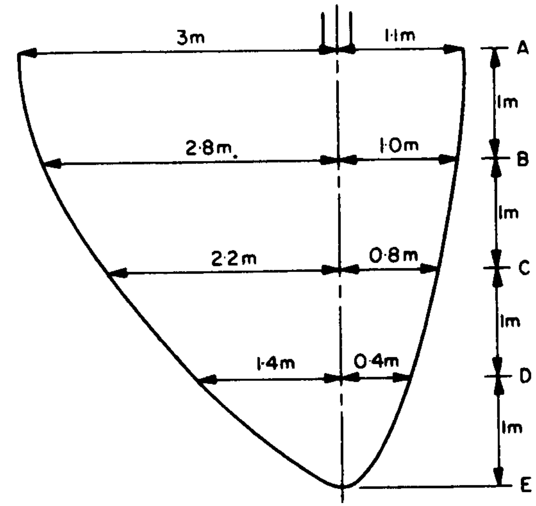


Fig. 13.17

Level	Total chord length (m)	S.M.	F(area)	Lever below stock	F(moment)	c.p. aft of leading edge	c.p. aft of axis	F(torque)
A	4.1	1	4.1	0	0.0	1.27	0.17	0.70
B	3.8	4	15.2	1	15.2	1.18	0.18	2.74
C	3.0	2	6.0	2	12.0	0.93	0.13	0.78
D	1.8	4	7.2	3	21.6	0.56	0.16	1.15
E	0	1	0	4	0.0	0.00	0.00	0.00
			32.5		48.8			5.37

$$\text{Force on rudder} = 21.1 \times 10.83 \times \left(\frac{20 \times 1852}{3600}\right)^2 \times 35 = 847{,}000 \text{ newtons}$$

$$\text{Bending moment at stock} = 847{,}000 \times 1.502 = 1.272\,\text{MN m}$$

$$\text{Torque on rudder} = 0.165 \times 847{,}000 = 140{,}000\,\text{N m}$$

The combined effect of the bending moment M and torque T is equivalent to a bending moment M' given by

$$M' = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)$$

hence $M' = 1.276\,\text{MN m}$

$$\text{Max stress} = M'/Z = 1.276/0.1 = 12.76\,\text{MN/m}^2$$

Experiments and trials

MODEL EXPERIMENTS CONCERNED WITH TURNING AND MANOEUVRING

For accurate prediction of ship behaviour, the model must represent as closely as possible the ship and its operating condition both geometrically and dynamically. It is now customary to use battery powered electric motors to drive the propellers and radio control links for rudder and motor control. The model self-propulsion point is different from that for the ship due to Reynolds' number effects, and the response characteristics of the model and ship propulsion systems differ but the errors arising from these causes are likely to be small. They do, however, underline the importance of obtaining reliable correlation with ship trials.

A number of laboratories now have large tanks in which model turning and manoeuvring tests can be conducted. The facilities offered vary but the following is a brief description of those provided at Haslar (now DERA). The basin is 122 m long and 61 m wide with overhead camera positions for recording photographically the path of the model. The models are typically 5 m long, radio controlled and fitted with gyros for sensing heel. One method used for many years to record the path of the model used lights set up on the model at bow and stern so that when the model is underway they lie in a known datum plane. These lights are photographed using a multiple exposure technique so that the path of the model can be recorded on a single negative. When enlarged to a standard scale, the print has a grid superimposed upon it to enable the co-ordinates of the light positions to be read off and the drift angle deduced. As exposures are taken at fixed time intervals, the speed of the model during the turn can be deduced besides the turning path. The heel angle is recorded within the model. The process is illustrated diagrammatically in Fig. 13.18. A number of alternative tracking methods are available.

The same recording techniques can be used to record the behaviour of a model when carrying out zig-zag or spiral manoeuvres or any other special manoeuvres which require a knowledge of the path of the model. Two points have to be borne in mind, however, if a human operator is used as one element of the control system. One, is that being remote from the model the experimenter has to rely upon instruments to tell them how the model is reacting. They cannot sense the movements of the ship directly through a sense of balance and hence their reactions to a given situation may differ from those in a ship. The second point is that the time they have to react is less. Because Froude's law of comparison applies, the time factor is reduced in proportion to the square root of the scale factor, i.e. if the model is to a scale of $\frac{1}{16}$th full size, permissible reaction times will be reduced to $\frac{1}{4}$th of those applying to the helmsman on the ship. In the same way, the rate at which the rudder is applied must be increased to six times that full scale. It follows that, whenever possible, it is desirable to use an automatic control system or, failing this, a programmed sequence of rudder orders in order to ensure consistency of results.

By using suitable instrumentation, the rudder forces and torques can be measured during any of these manoeuvres. Typically, the rudder stock is strain gauged to record the force and its line of action.

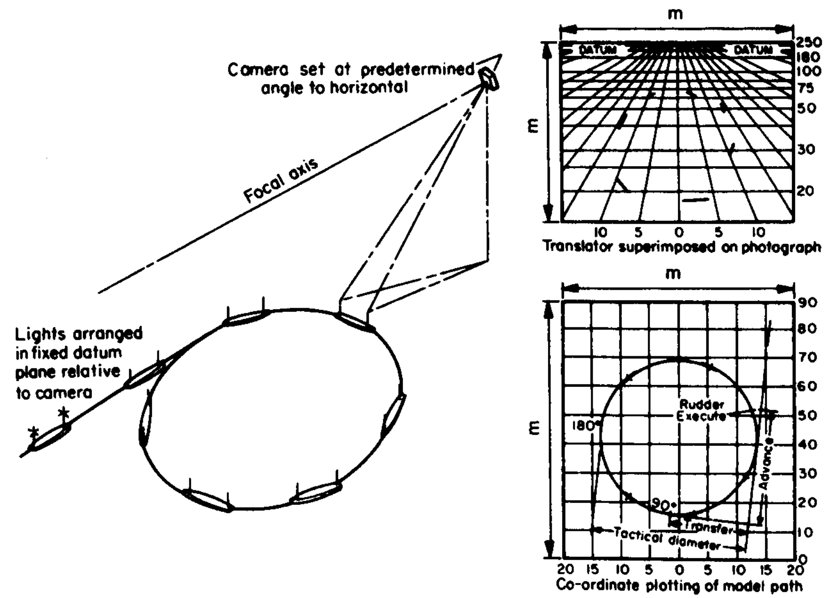


Fig. 13.18

MODEL EXPERIMENTS CONCERNED WITH DIRECTIONAL STABILITY

It has been seen that the spiral manoeuvre can indicate in a qualitative sense whether a design is directionally stable or not and that the pull-out manoeuvre can give an indication of the degree of stability. For a proper study of stability, however, it is necessary to ascertain the derivatives of force and moment as required by the theory outlined earlier. For this constrained model tests are carried out and the assumption made that the forces and moments measured on the model can be scaled directly to full scale.

It is known that there are viscous effects on forces but no suitable correction can be made. Such tests are usually carried out both with and without model propellers fitted and working at model self-propulsion revolutions, as the changed velocity distribution at the stern due to the propeller action is likely to be significant and its effect can be deduced in this way.

The derivatives of forces and moments with respect to transverse velocity (or yaw angle) can be measured in what is termed an oblique tow test in a conventional long ship tank. Measurements are made of the forces and moments required to hold the model at various yaw angles over a range of speeds. Data for the model on a curved path can be obtained using a rotating arm facility with the model at various yaw angles and moving in circular paths of different radii. Speed of advance is controlled by the arm rotational speed which is kept constant for the duration of each run. Submarine models can be run on their side to measure the derivatives with respect to vertical velocity, or trim angle.

Control surface effectiveness can be determined during the oblique tow and rotating arm experiments by measuring the variation in force and moment with control surface angle over a range of yaw angles, path curvatures and speeds of advance. These tests can be carried out over a wide range of parameter values and can thus be used to study situations in which non-linearities exist. They are, however, steady motion tests and as such do not provide any insight into acceleration derivatives.

To measure acceleration derivatives use is made of a planar motion mechanism (PMM) which can yield all the linear derivatives of both velocity and acceleration. The essence of these tests is that the model is force oscillated whilst being towed below the carriage of a conventional ship tank giving rise to sinusoidal yawing and swaying motions. The force and moment records can be analysed into phase components which yield the derivatives associated with the velocity and acceleration components of the motion. It is usual to carry out separate tests in which the model has imparted to it a pure sway and a pure yaw motion (Fig. 13.19).

The rates of turn or curvature that can be applied in PMM tests are limited and the tests do not provide good information on non-linearities and cross-coupling terms. Thus the oblique tow, rotating arm and PMM tests are complementary to one another.

The features of all three tests are incorporated in the Computerized Planar Motion Carriage (CPMC) system used at Hamburg. In this a model can be driven independently in three degrees of freedom and precise transient motions can be generated. Three independent sub-carriages are used to superimpose arbitrary surge, sway and yaw motions on the uniform forward motion of the main towing carriage which does not itself have to accelerate in order to generate transient surge motions in the model.

Besides being used to force oscillate the model the system can be used in a tracking mode to follow closely the movements of a freely manoeuvring model.

Having obtained the hydrodynamic derivatives for a new design they can be substituted in the simple formulae already quoted to demonstrate stability and

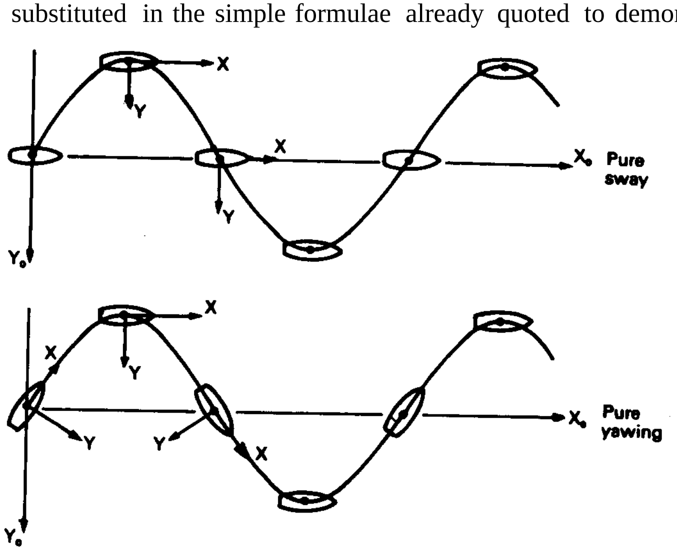


Fig. 13.19 Planar motion mechanism test

position of neutral point. More commonly, the data, including cross-couplinAi terms, derivatives from control surfaces, etc., are fed into computers which predict turning circles, zig-zag manoeuvres, etc. The validity of this approach is demonstrated in Fig. 13.20.

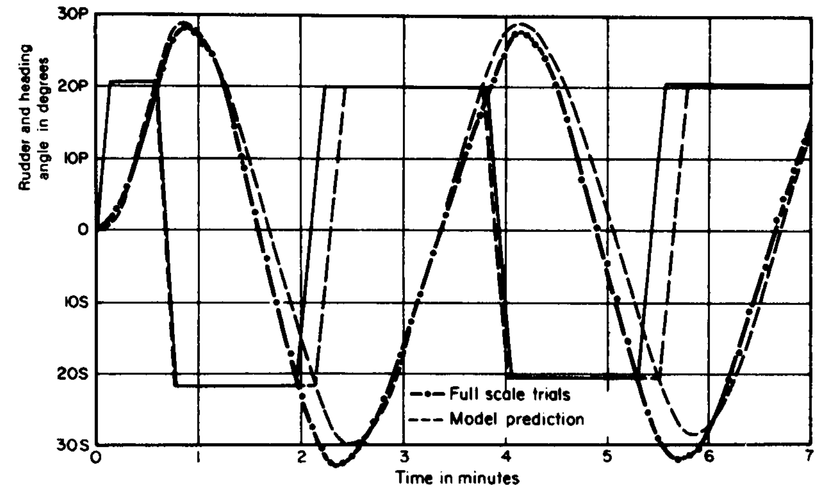


Fig. 13.20 Comparison of trial and predicted data for zig-zag manoeuvre

SHIP TRIALS

The credibility of theoretical or model experimental methods for predicting the manoeuvring characteristics of ships depends on establishing reliable correlation with the full-scale ship. Although there is not an exact correspondence of trajectories, there is sufficient correlation between model and ship to believe that the behaviour of the model represents the behaviour of the ship. Several methods can be used for recording the path of a ship at sea including

(a) a log to measure ship speed and a compass to record the ship's head. It is approximate only as most logs are inaccurate when the ship turns;
(b) use of a theodolite or camera overlooking the trial area. Limited by ~ depth of water available close in shore;
(c) the use of a navigational aid such as the Decca system to record the ship's position at known intervals of time. For accurate results the trials area must be one which is covered by a close grid;
(d) use a satellite navigation system to record the ship's path relative to land. A separate buoy could be tracked to make allowance for water movement;
(e) the use of bearing recorders at each end of the ship to record the bearings of a buoy.

To illustrate the care necessary to ensure a reliable and accurate result a system developed for the last test method is now described. A special buoy is used which moves with the wind and tide in a manner representative of the ship. Automatic recording of bearing angles is used with cameras to correct human errors in tracking the buoy and records are taken of ship's head, shaft r.p.m., rudder angle and heel angle all to a common time base which is synchronized with the bearing records.

The arrangement of the trials equipment in the ship is illustrated in Fig. 13.21.

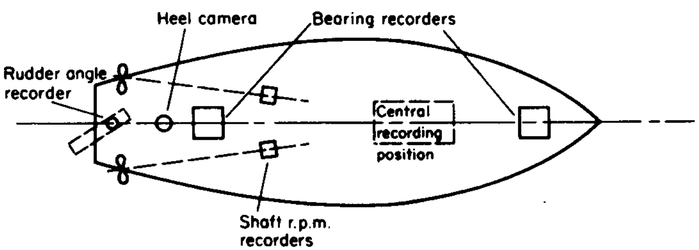


Fig. 13.21 Arrangement of turning trials equipment on board ship

Analysis of turning trials results

Having developed the film records, all of which have a common time base, the bearing of the buoy from each recorder is plotted to a base of ship heading. From this plot, the bearings at every 30 degrees (or any other desired angular spacing) can be obtained. These are then set out relative to a base line representing the distance apart of the two recorders as in Fig. 13.22, where o: and /3 are the two bearings appropriate to 120 degrees change of heading. From the intersection of the two bearing lines a perpendicular -CO- is dropped on to the base line. Then -CO- and -O-G define the position of the buoy relative to the G of the ship and the ship's centre line. Turning now to the right hand plot in Fig. 13.22, radial lines are set out from a fixed point which represents the buoy and the position of the ship for 120 degrees change of heading is set out as indicated.

This process is repeated for each change of heading and the locus of the G position defines the turning path. The drift angle follows as the angle between the tangent to this path and the centre line of the ship. Information on rates of turn is obtained by reference to the time base.

Angle of heel is recorded by photographing the ensign staff against the horizon or using a vertical seeking gyro.

Ship trials involving zig-zag, spiral and pull-out manoeuvres do not require a knowledge of the path of the ship. Records are limited to rudder angle and ship's head to a common time base. The difficulty of recording the spiral manoeuvre for an unstable ship has already been mentioned. Only the two branches of the curve shown in full in Fig. *l3.9(b)* can be defined.

Rudder types and systems

TYPES OF RUDDER

There are many types of rudder fitted to ships throughout the world. Many are of limited application and the claims for a novel type of rudder should be critically examined against the operational use envisaged for the ship. For

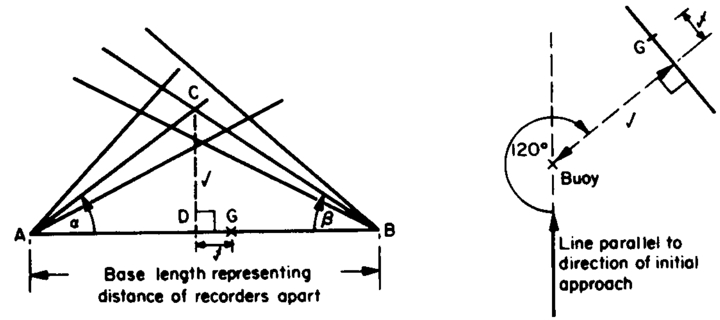


Fig. 13.22 Analysis of turning trial data

instance, some rudders are only of benefit in single screw ships of relatively low speed. It is not possible to cover all the types of rudder in a book such as this and discussion is limited to the four types illustrated in Fig. 13.23.

The choice of rudder type depends upon the shape of the stern, the size of rudder required and the capacity of the steering gear available.

The *balanced spade rudder* is adopted where the ship has a long cut up, the rudder size is not so great as to make the strength of the rudder stock too severe a problem and where it is desired to keep the steering gear as compact as possible.

The *gnomon rudder* is used where the size of rudder requires that it be supported at an additional point to the rudder bearing, but where it is

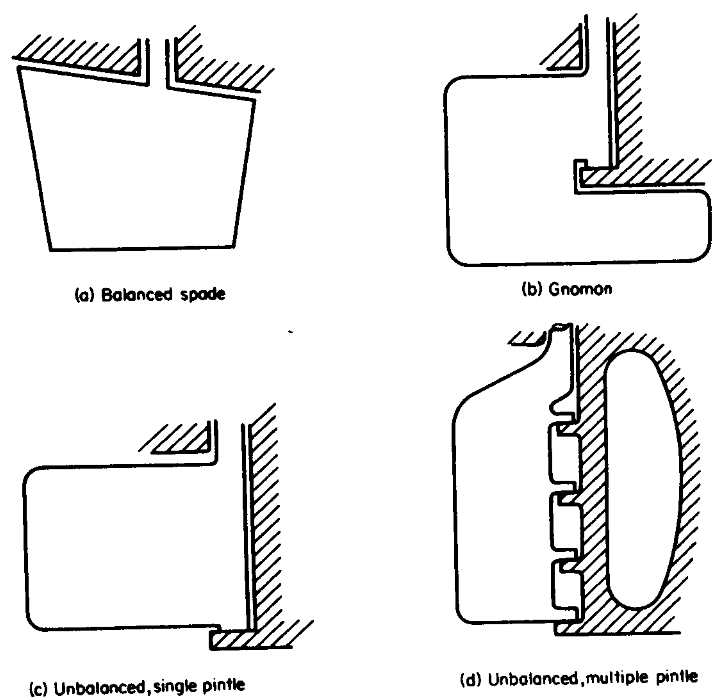


Fig. 13.23 Rudder types

still desired to partially balance the rudder to reduce the size of the steering gear.

Unbalanced rudders are used where the stem shape precludes the fitting of a balanced rudder. The number of pintles fitted is dictated by strength considerations.

BOW RUDDERS AND LATERAL THRUST UNITS

Why should it be necessary to consider bow rudders at all? It has been shown that if a lateral force is applied at the neutral point a ship follows a straight path at an angle of attack, the angle of attack depending on the magnitude of the force. Thus, it would be of considerable use if a navigator could cause a force to be applied at any selected point along the ship's length, i.e. they could control ship's head and path independently. This could be achieved if control surfaces were fitted at both ends of the ship.

Also, if the only rudders fitted are those aft there is a greater danger that damage could render the ship uncontrollable. This is particularly important in wartime when ships are liable to be attacked by weapons homing on the propellers.

Against these considerations it must be remembered that because the neutral point is generally fairly well forward rudders at the bow are relatively much less effective. Neither can they benefit from the effects of the screw race. Unless they are well forward and therefore exposed to damage, the flow conditions over the bow rudder are unlikely to be good. These factors generally outweigh the possible advantages given above and only a few bow rudders as such are fitted to ships in service.

To some extent, these disadvantages can be overcome by fitting a lateral thrust unit at the bow. Typically, such a device is a propeller in a transverse tube. They are particularly useful in ferries when speed in berthing reduces turn round time and enhances economy of operation. Model experiments have shown that the effect of these units may be seriously reduced when the ship has forward speed. The fall in side force can be nearly 50 per cent at 2 knots, 40 per cent occurring between 1 and 2 knots. Placing the unit further aft reduces the effect of forward speed. Contra-rotating propeller systems are recommended for lateral thrust units.

SPECIAL RUDDERS AND MANOEUVRING DEVICES

It has been seen that conventional rudders are oflimited use at low speeds. One way of providing a manoeuvring capability at low speed is to deflect the propeller race.

This is achieved in the *Kitchen rudder,* the action of which is illustrated in Fig. 13.24.

The rudder consists essentially of two curved plates shrouding the propeller. For going ahead fast, the two plates are more or less parallel with the propeller race causing little interference. When both plates are turned in plan, they cause the propeller race to be deflected so producing a lateral thrust. When the two plates are turned so as to close in the space behind the propeller, they cause the ahead thrust to be progressively reduced in magnitude and finally to be trans-

formed into an astern thrust albeit a somewhat inefficient one. The same principle is used for jet deflectors in modern high speed aircraft.

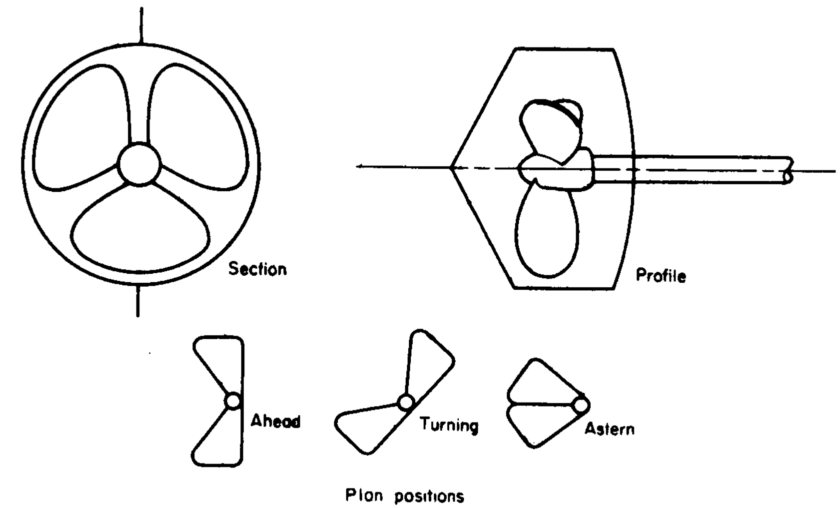


Fig. 13.24 *Kitchen rudder action*

The Kitchen rudder is used mainly for small power boats. It will be appreciated from the above, that not only can it provide lateral thrust at low ahead speed but that it can also be used to vary the magnitude and/or sense of the propeller thrust. Thus in a boat so fitted the shafts can be left running at constant speed.

An alternative to using deflector plates to deflect the propeller race would be to turn the propeller disc itself. This is the principle of the Pleuger *active rudder* which is a streamlined body actually mounted on a rudder, the body containing an electric motor driving a small propeller. To gain full advantage of such a system, the rudder should be capable of turning through larger angles than the conventional 35 degrees.

The power of the unit varies, with the particular ship application, between about 50 and 300 h.p. With the ship at rest, the system can turn the ship in its own length.

A different principle is applied in the *vertical axis propeller* such as the *VoW.,. Schneider propeller,* Fig. *13.25(b).* This consists essentially of a horizontal disc carrying a number of vertical blades of aerofoil shape.

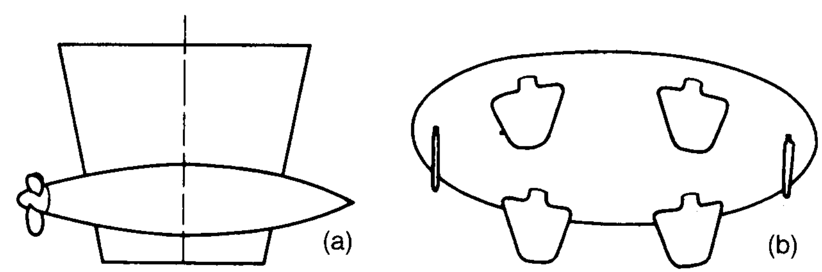


Fig. 13.25 (a) Active rudder, (b) Voith-Schneider propeller

As the harizantal disc is rotated abaut a vertical axis, a special mechanism feathers the blades in such a way as *to.* pravide a thrust in any desired directian. The thrust is caused *to.* act fare and aft far narmal propulsian and athwartships *far* steering. The limitatian 0.1' this type 0.1' prapeller is the pawer which can be transmitted *to.* the disc. In same cases, they are fitted specially far manaeuvring in canfined waters as in the case 0.1' a number *af* ferries and ather ships ape rating in canals. They are also. used where maderately large turning maments are needed at law speeds as wauld be the case with same tugs.

Many special rudder farms have been develaped aver the years. Claims are made far each type *af* special advantages aver mare canventianal rudder types. Such claims shauld be carefully examined to. ensure that the advantages will be farthcaming far the particular applicatian in mind as, in *mast* cases, this is anly *so.* if certain special canditians 0.1' speed *ar* ship farm apply.

Amangst the special types mentian can be made 0.1' the fallawing:

(a) the *flap rudder* (Fig. 13.26), in which the after partian 0.1' the rudder is caused *to.* mave *to.* a greater angle than the main partian. Typically, abaut ane-third 0.1' the tatal rudder area is used as a flap and the angle 0.1' flap is twice that 0.1' the main rudder. The effect *af* the flap is *to.* cause the camber 0.1' rudder sectian *to.* change with angle giving better lift characteristics. The number 0.1' practical applicatians is *nat* great, partly because 0.1' the *cam*plicatian 0.1' the linkage system required *to.* actuate the flap;

(b) as a special example 0.1' the flap rudder, flaps *af* quite small area at the trailing edge can be maved *so.* as to induce hydradynamic forces *an* the main rudder assisting in turning it. Such a rudder is the *Flettner rudder* (Fig. 13.27). The flaps act as a servo. system assisting the main steering gear;

(c) so-called *balanced reaction rudders* (Fig. 13.28), in which the angle of attack of the rudder sectians varies aver the depth *af* the rudder. It attempts to prafit from the ratatian *af* the propeller race, and behind prapellers *wark*ing at high slip and law efficiency is claimed *to.* produce a forward thrust;

(d) *streamlined rudders* behind a fixed streamlined skeg. This is similar in principle *to.* the flap rudder except that anly ane part moves. By maintaining a better aerafail shape at all angles the required lift farce is abtained at the expense 0.1' less drag and less rudder tarque. Such a rudder is the *oertz rudder.*

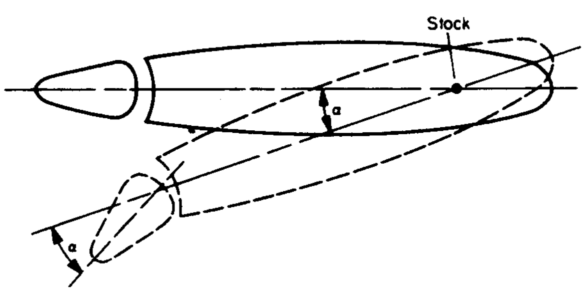


Fig. 13.26 Flap rudder

(e) *rotating cylinder rudder.* The fact that a cylinder rotating in a fluid strcmm develaps a lift narmal to its axis and the stream flow, has been knawn for It very lang time as the Magnus effect. The principle was cansidered by NPL (now BMT) *to.* imprave ship manaeuvring. Having studied several canfigurations NPL cancluded that the use 0.1' a ratating cylinder at the leading edge 0.1' the rudder was the most practical. Narmal caurse-keeping was unimpaired and far relatively law cylinder pawer, attached flaw cauld be maintained far rudder angles up *to.* 90 degrees, i.e. stall which 0.1'ten *accurs* at 35 degrees could be inhibited. An installatian prapased far a 250,000-tannef tanker had a cylinder ane metre in diameter driven at 350 r.p.m. absarbing about 400 kW. It was predicted that the turning circle diameter wauld be reduced fram abaut 800 m with 35 degrees 0.1' rudder *to.* abaut 100 m at abaut 75 degrees 0.1' rudder with the cylinder in operatian. Subsequently sea trials *an* a 200-tannef vessel were carried *aut to.* canfirm the principle and the ship cauld turn indefinitely almast in its *awn* length.

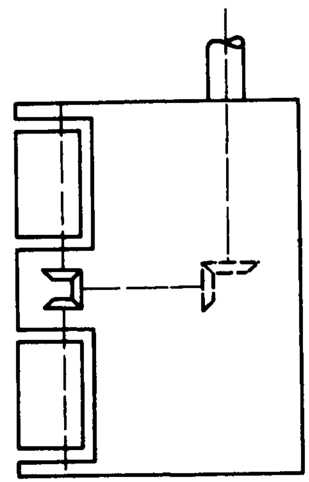


Fig. 13.27 Flettner rudder

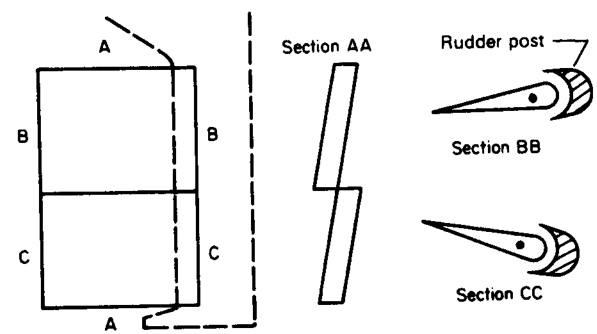


Fig. 13.28 Balanced reaction rudder

DYNAMIC POSITIONING

In ships engaged in underwater activities it may be necessary to hold the ship steady relative to some underwater datum. Typical of this situation are drilling ships and those deploying divers. If the water is shallow then it may be practical to moor the ship. In deeper water use is made of a dynamic positioning system using thrust producing devices forward and aft together with a means of detecting departures from the desired position usually using satellite navigation aids.

AUTOMATIC CONTROL SYSTEMS

Many ships, particularly those on long ocean voyages, travel for long periods of time on a fixed course, the only deviations in course angle being those necessitated by variations in tide, waves or wind. To use trained helmsmen for this type of work is uneconomical and boring for the people concerned. It is in these circumstances that the automatic control system or automatic 'pilot' is most valuable.

Imagine a system which can sense the difference, *1/Je,* between the ordered course and the actual course and which can cause the rudder to move to an angle proportional to this error, and in such a way as to turn the ship back towards the desired course, i.e.:

bR = Const. x *1/Je* = *a1/Je,* say

Then, as the ship responds to the rudder the course error will be reduced steadily and, in consequence, the rudder angle will also reduce. Having reached the desired course, the rudder angle will reduce to zero but the ship will still be swinging so that it is bound to 'overshoot'. Thus, by repetition of this process the ship will oscillate about the desired course, the amplitude of the oscillation depending upon the value of the constant of proportionality used in the control equation.

How can the oscillation be avoided or at least reduced? In a ship, a helmsman mentally makes provision for the rate of swing of the ship and applies opposite rudder before the desired course angle is reached to eliminate the swing. By introducing a rate gyro into the control system it also can sense the rate of swing and react accordingly in response to the following control equation

$$\delta_{\mathrm{R}} = a\psi_{\mathrm{e}} + b\left(\frac{\mathrm{d}\psi_{\mathrm{e}}}{\mathrm{d}t}\right)$$

By careful selection of the values of a and b, the overshoot can be eliminated although, in general, a better compromise is to allow a small overshoot on the first swing but no further oscillation as this usually results in a smaller average error. It would be possible to continue to complicate the control equation by adding higher derivative terms. A ship, however, is rather slow in its response to rudder and the introduction of higher derivatives leads to excessive rudder movement with little effect on the ship. For most applications the control equation given above is perfectly adequate.

The system can be used for course changes. By setting a new course. 1111 'error' is sensed and the system reacts to bring the ship to a new heading. Iſ desired, the system can be programmed to effect a planned manoeuvre or serics of course changes. For an efficient system the designer must take into account the characteristics of the hull, the control surfaces and the actuating systcm. The general mathematics of control theory will apply as for any other dynamic system. In some cases it may be desirable to accept a directionally unstable hull and create course stability by providing an automatic control system with the appropriate characteristics. This device is not often adopted because of the danger to the ship, should the system fail. Simulators can be used to study the relative performances of manual and automatic controls. The consequences of various modes of failure can be studied in safety using a simulator, including the ability of a human operator to take over in the event of a failure. Part-task simulators are increasingly favoured as training aids. A special example of automatic control systems is that associated with dynamic positioning of drilling or diving ships.

Ship handling

TURNING AT SLOW SPEED OR WHEN STOPPED

It has been seen that the rudder acts in effect as a servo-system in controlling the attitude of the ship's hull so that the hydrodynamic forces on the hull will cause the ship to turn. At low or zero speed, the magnitude of any hydrodynamic force, depending as it does to a first order on V^2, is necessarily small. Since under these conditions the propeller race effect is not large, even the forces on a rudder in the race are small.

The ship must therefore rely upon other means when attempting to manoeuvre under these conditions. A number of possibilities exist:

(a) A twin shaft ship can go ahead on one shaft and astern on the other so producing a couple on the ship causing her to turn. This is a common practice, but leverage of each shaft is relatively small and it can be difficult to match the thrust and pull on the two shafts. Fortunately, some latitude in fore and aft movement is usually permissible. ,

(b) Ifleaving a jetty the ship can swing about a stern or head rope. It can maKe use of such a device as a pivot while going ahead or astern on the propeller.

(c) When coming alongside a jetty at slow speed, use can be made of the so-called 'paddle-wheel' effect. This effect which is due to the non-axial flow through the propeller disc results in a lateral force acting on the stern/propeller/rudder combination in such a way as to cause the stern to swing in the direction it would do if the propeller were running as a wheel on top of a hard surface. In a twin-screw ship the forces are generally in balance. Now, consider a twin-screw ship approaching a jetty as in Fig. 13.29. If both screws are outward turning (that is viewed from aft, the tip of the propellers move outboard at the top of the propeller disc), the port shaft can be set astern. This will have the effect of producing a lateral force at the

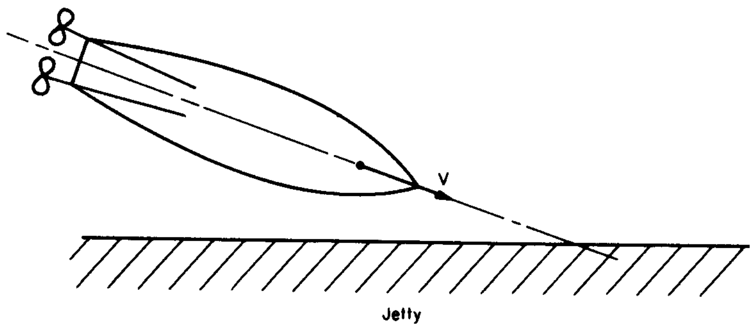


Fig. 13.29 *Paddle-wheel effect when coming alongside ajetty*

stern acting towards the jetty, besides taking the way off the ship and producing a moment on the shafts tending to bring the ship parallel to the jetty.

(*d*) The screw race can be deflected by a special device such as the Kitchen rudder or, to some extent at least, by twin rudders behind a single propeller. Clearly, unless the race can be deflected through about 90 degrees this system cannot be used without, at the same time, causing the ship to move fore and aft.

(*e*) Use one of the special manoeuvring devices described above.

INTERACTION BETWEEN SHIPS WHEN CLOSE ABOARD

Even in deep water, interaction effects can be significant when two ships are in close proximity. The pressure field created by a ship moving ahead in open water is illustrated in Fig. 13.30, its actual form depending on the ship form.

The pressure field extends for a considerable area around the ship, and any disturbance created in this field necessarily has its reaction on the forces acting on the ship. If the disturbance takes place to one side of the ship, it is to be expected that the ship will, in general, be subject to a lateral force and a yawing moment.

This is borne out by the results reproduced in Fig 13.31 for a ship A of 226 m and 37,500 tonnef overtaking a ship B of 173m and 24,000 tonnef on a parallel course. From these, it is seen that the ships are initially repelled, the force of repulsion reducing to zero when the bow of A is abreast the amidships of B. The ships are then attracted, the force becoming a maximum soon after the ships are abreast and then reducing and becoming a repulsion as the ships begin to part company.

The largest forces experienced were those of attraction when the two ships were abreast. They amounted to 26.5 tonnef on A and 35.5 tonnef on B at 10 knots speed and 15m separation. The forces vary approximately as the square of the ship speed and inversely with the separation.

When running abreast, both ships are subject to a bow outward moment but to a bow inward moment when approaching or breaking away.

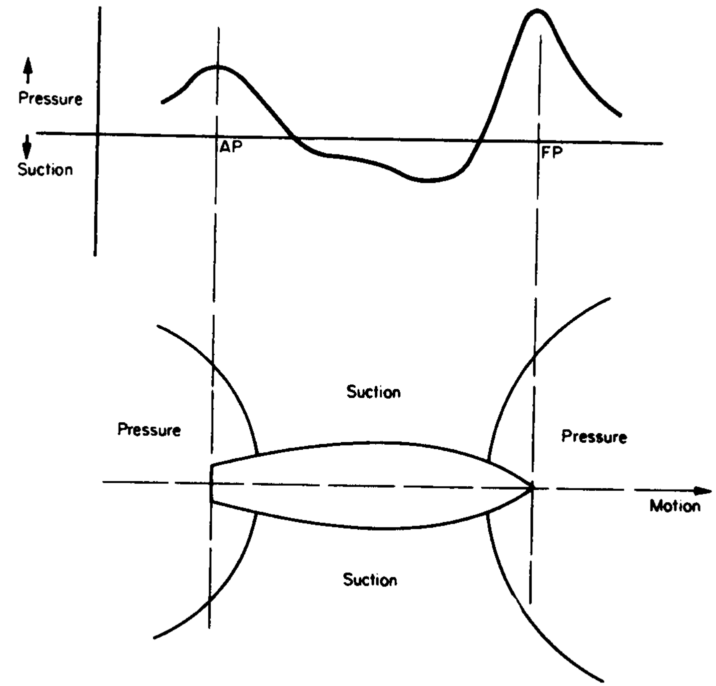


Fig. 13.30 Pressure field for ship in deep water

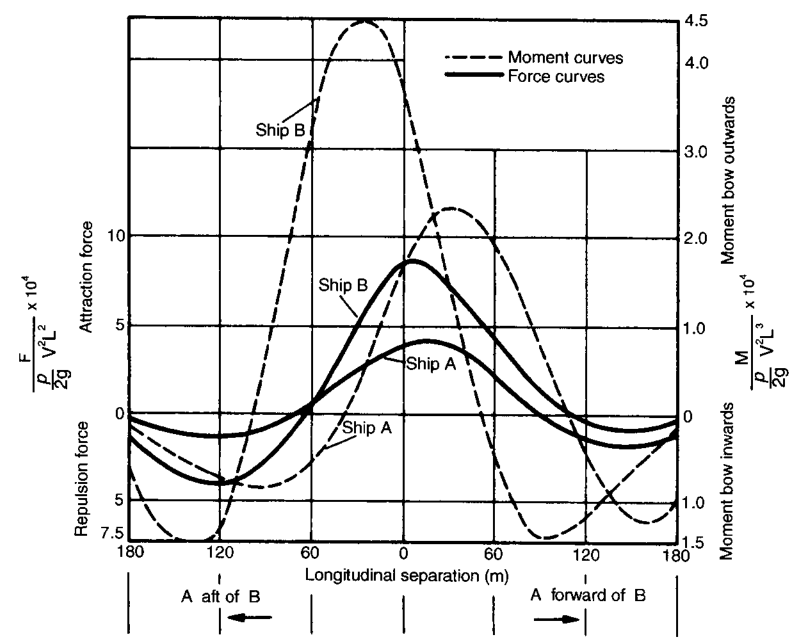


Fig. 13.31 Interaction forces and moments

To be able to maintain the desired course it is necessary to counteract these variations in force and moment, which means that the ship must not only use its rudder but must also run generally at an angle of yaw.

Ship trials show that replenishment at sea operations are perfectly feasible at speeds of up to about 20 knots and such operations are quite commonplace nowadays.

In addition to, or in place of, the disturbance created by a ship in close company, the pressure field of a ship may be upset by a canal bank, pier or by the proximity of the sea bed. In the case of a vertical canal bank or pier, the result will be a lateral force and yawing moment as for the replenishment at sea operation. By analogy, the effect of shallow water is a vertical force and trimming moment resulting in bodily sinkage of the ship and trim by the stern. This effect can lead to grounding on a sandbank which may have been expected to be several feet below the keel.

While naval architects think of these attractions in terms of Bernouilli's equation, mariners recognize the canal effect as 'smelling the ground' and the bodily sinkage in shallow water as 'squat'.

BROACHING

Broaching, or *broaching-to,* describes the loss of directional stability in waves, induced by a large yaw moment exceeding the course keeping ability of the rudders. Orbital motion of water particles in the wave can result in a zero flow past the rudders which become ineffective. This loss can cause the ship to turn beam on to the waves. The vessel might even capsize due to a large roll moment arising from the forward momentum and the large heading angle. The effect is greater because the ship's hydrostatic stability is often reduced by the presence of the waves.

Broaching is likely when the ship is running with, or being slowly overtaken by, the waves. It may be sudden, due to the action of a single wave, or be cumulative where the yaw angle builds up during a succession of waves. Although known well since man put to sea in boats, broaching is a highly non-linear phenomenon and it is only relatively recently that good mathematical simulations have been possible.

When the encounter frequency of the ship with the waves approaches zero the ship can become trapped by the wave. The ship remains in the same position relative to the waves for an appreciable time. It is then said to be *surf riding.* This is a dangerous position and broaching is likely to follow.

The Master can get out of this condition by a change of speed or direction, although the latter may temporarily result in large roll angles.

Stability and control of submarines

The high underwater speed of some submarines makes it necessary to study their dynamic stability and control. The subject assumes great importance to both the commanding officer and the designer because of the very short time available in which to take corrective action in any emergency: many submarines are restricted to a layer of water which is of the order of at most two or three

ship lengths deep. To the designer and research worker, this has meant **dirl:ctlny** attention to the change in the character of the forces governing the motion of the submarine which occurs as the speed is increased. For submarines of' orthodox size and shape below about ten knots the hydrostatic forces pn:dominate. In this case, the performance of the submarine in the vertical plane CII 11 be assessed from the buoyancy and mass distributions. Above 10 knots, however, the hydrodynamic forces and moments on the hull and control surfaces predominate.

To a certain degree, the treatment of this problem is similar to that of the directional stability of surface ships dealt with earlier. There are differences however between the two, viz.:

(a) the submarine is positively stable in the fore and aft vertical plane in that B lies above G so that having suffered a small disturbance in trim when at rest it will return to its original trim condition;

(b) the limitation in the depth of water available for vertical manoeuvres;

(c) the submarine is unstable for translations in the *z* direction because the hull is more compressible than water;

(d) it is not possible to maintain a precise equilibrium between weight and buoyancy as fuel and stores are being continuously consumed.

It follows, from *(c)* and *(d)* above, that the control surfaces or hydroplanes will have, in general, to exert an upward or downward force on the submarine. Also, if the submarine has to remain on a level keel or, for some reason, the submarine cannot be allowed to trim to enable the stability lever to take account of the trimming moment, the control surfaces must also exert a moment. To be able to exert a force and moment on the submarine which bear no fixed relationship one to another requires two separate sets of hydroplanes. Usually, these are mounted well forward and well aft on the submarine to provide maximum leverage.

Consider a submarine turning in the vertical plane (Fig. 13.32).

Assume that the effective hydroplane angle is OH, i.e. the angle representing the combined effects of bow and stern hydroplanes.

In a steady state turn, with all velocities constant, the force in the *z* direction and the trimming moment are zero. Hence 4f'''

$$wZ_w + qZ_q + mqV + \delta_H Z_{\delta_H} = 0$$

$$wM_w + qM_q + \delta_H M_{\delta_H} - mg\overline{BG}\theta = 0$$

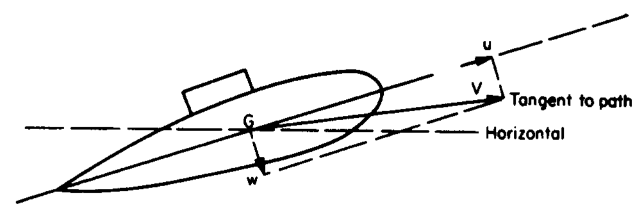


Fig. 13.32

where subscripts w, q and δ_H denote differentiation with respect to velocity normal to submarine axis, pitching velocity and hydroplane angle respectively. Compare the equations for directional stability of surface ships:

mqV is a centrifugal force term

$mg\overline{BG}\theta$ is a statical stability term

In the moment equation M_w, M_q and M_{δ_H} are all proportional to V^2, whereas $mg\overline{BG}\theta$ is constant at all speeds. Hence, at high speeds, $mg\overline{BG}\theta$ becomes small and can be ignored. As mentioned above, for most submarines it can be ignored at speeds of about 10 knots. By eliminating w between the two equations so simplified

$$\frac{q}{\delta_H} = \frac{M_w Z_{\delta_H} - M_{\delta_H} Z_w}{M_q Z_w - M_w(Z_q + mV)}$$

As with the surface ship problem the necessary condition for stability is that the denominator should be positive, i.e.

$$M_q Z_w - M_w(Z_q + mV) > 0$$

This is commonly known as the *high speed stability criterion*.

If this condition is met and statically the submarine is stable, then it will be stable at all speeds. If it is statically stable, but the above condition is not satisfied, then the submarine will develop a diverging (i.e. unstable) oscillation in its motion at forward speeds above some critical value.

Now by definition $q = \mathrm{d}\theta/\mathrm{d}t = \dot{\theta}$, so that differentiating the moment equation with respect to time

$$\dot{w}M_w + \dot{q}M_q + \dot{\delta}_H M_{\delta_H} - mg\overline{BG}q = 0$$

But in a steady state condition as postulated $\dot{w} = \dot{q} = \dot{\delta}_H = 0$. Hence $q = 0$ if $\overline{BG}$ is positive as is the practical case. That is, a steady path in a circle is not possible unless $\overline{BG} = 0$. Putting $q = 0$, the equations become

$$wZ_w + \delta_H Z_{\delta_H} = 0$$

$$wM_w + \delta_H M_{\delta_H} - mg\overline{BG}\theta = 0$$

i.e.

$$w = -\delta_H \frac{Z_{\delta_H}}{Z_w}$$

and

$$\theta = \delta_H\left(M_{\delta_H} - \frac{Z_{\delta_H}}{Z_w}M_w\right) \Big/ mg\overline{BG}$$

Now rate of change of depth $= V(\theta - w/V)$ if w is small $= V\theta - w$, i.e.

$$\frac{\text{depth rate}}{\delta_H} = \left(VM_{\delta_H} - VM_w\frac{Z_{\delta_H}}{Z_w} + mg\overline{BG}\frac{Z_{\delta_H}}{Z_w}\right) \Big/ mg\overline{BG}$$

The depth rate is zero if

$$V = -\left(mg\overline{BG}\frac{Z_{\delta_H}}{Z_w}\right) \Big/ \left(M_{\delta_H} - M_w\frac{Z_{\delta_H}}{Z_w}\right)$$

$$= mg\overline{BG} \Big/ \left(M_w - M_{\delta_H}\frac{Z_w}{Z_{\delta_H}}\right)$$

From the equation for θ, if the hydroplanes are so situated that

$$\frac{M_{\delta_H}}{Z_{\delta_H}} = \frac{M_w}{Z_w}$$

then θ is zero. The depth rate will be $\delta_H Z_{\delta_H}/Z_w$ which is not zero. The ratio M_w/Z_w defines the position of the *neutral point*. This corresponds to the similar point used in directional stability and is usually forward of the centre of gravity. A force at the neutral point causes a depth change but no change in the angle of pitch.

The equation for depth rate can be rewritten as

$$\text{depth rate}/\delta_H = \frac{Z_{\delta_H}}{Z_w}\left(1 - \frac{V}{V_c}\right)$$

where

$$V_c = mg\overline{BG} \Big/ \left(M_w - \frac{M_{\delta_H}}{Z_{\delta_H}} \cdot Z_w\right)$$

or

$$\text{depth rate}/\delta_H = \frac{Z_{\delta_H}V}{mg\overline{BG}}\left\{\frac{M_{\delta_H}}{Z_{\delta_H}} - x_c\right\}$$

where

$$x_c = \frac{M_w}{Z_w} - \frac{mg\overline{BG}}{VZ_w}$$

The first of these two expressions shows that (depth rate)/δ_H is negative, zero or positive as V is greater than, equal to or less than V_c respectively. V_c is known as the *critical speed* or *reversal speed*, since at that speed the planes give zero depth change and cause reverse effects as the speed increases or decreases from this speed. Near the critical speed the value of $(1 - (V/V_c))$ is small— hence the hydroplanes' small effect in depth changing.

It will be seen that θ is not affected in this way since

$$\frac{\theta}{\delta_{\mathrm{H}}} = -\frac{Z_{\delta_{\mathrm{H}}}}{Z_{w}}\frac{1}{V_{\mathrm{c}}} = -\frac{Z'_{\delta_{\mathrm{H}}}}{Z'_{w}}\frac{V}{V_{\mathrm{c}}}$$

The magnitude of $\theta/\delta_{\mathrm{H}}$ changes with V but not its sign. If stern hydroplanes are considered, a positive hydroplane angle produces a negative pitch angle (bow down), but depth change is downwards above the critical speed and upwards below the critical speed.

The second expression for depth change illustrates another aspect of the same phenomenon. x_{c} denotes a position $mg\overline{\mathrm{BG}}/VZ_{w}$ abaft the neutral point,

$$\frac{mg\overline{\mathrm{BG}}}{VZ_{w}} = \frac{mg\overline{\mathrm{BG}}}{\frac{1}{2}\rho L^{2}Z'_{w}V^{2}}$$

hence x_{c} is abaft the neutral point by a distance which is small at high speed and large at low speed. The critical situation is given by $x_{\mathrm{c}} = M_{\delta_{\mathrm{H}}}/Z_{\delta_{\mathrm{H}}}$, i.e. centre of pressure of the hydroplanes. The position defined by x_{c} is termed the *critical point*. Figure 13.33 illustrates the neutral and critical point positions. Figure 13.34 shows a typical plot of x_{c}/L against Froude number. The critical speed can be obtained by noting the Froude number appropriate to the hydroplane position, e.g. in the figure

$$\frac{V_{\mathrm{c}}}{\sqrt{gL}} = 0.05$$

i.e.

$$V_{\mathrm{c}} = 3\,\text{knots if } L = 100\,\text{m}$$

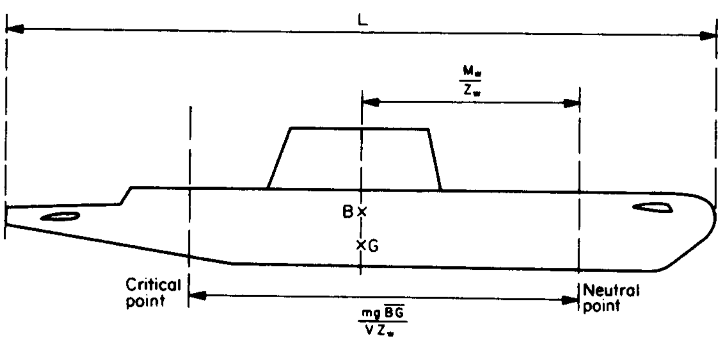


Fig. 13.33 Neutral and critical points

EXPERIMENTS AND TRIALS

As in the case of the directional stability of surface ships, the derivatives needed in studying submarine performance can be obtained in conventional ship tanks using planar motion mechanisms and in rotating arm facilities. The model is

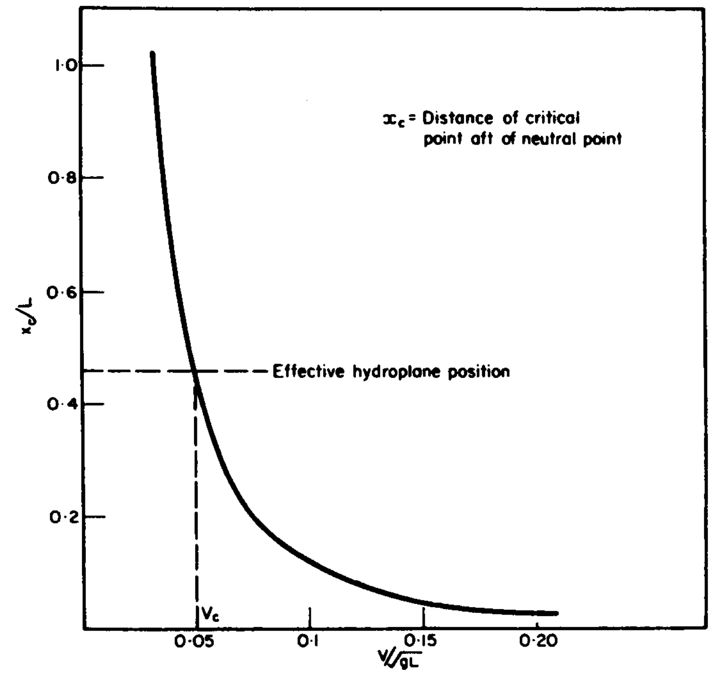


Fig. 13.34 Variation of critical point with speed

run upright and on its side with and without propellers, hydroplanes and stabilizer fins to enable the separate effects of these appendages to be studied. Data so obtained are used to predict stability and fed into digital or analogue computers. The computer can then predict the manoeuvres the submarine will perform in response to certain control surface movements. These can be used to compare with full-scale data obtained from trials. A computer can be associated with a tilting and rotating cabin, creating a simulator for realistic training of operators and for studying the value of different display and control systems.

Design assessment

MODIFYING DYNAMIC STABILITY CHARACTERISTICS

In common with most features of ship design, it is likely that the designer will wish to modify the dynamic stability standards as defined by the initial model tests. How then can the desired standards be most effectively produced?

In most cases, the basic hull form will be determined by resistance, propulsion and seakeeping considerations. The designer can most conveniently modify the appendages to change the dynamic stability. The procedure is similar for submarines and surface ships but is illustrated below for the former.

Assuming that the hydroplanes are correctly sized, the designer concentrates on the stabilizer fins (skeg for the lateral plane). If the contributions of these fins to Z'_w and M'_w, as determined from the model results with and without fins, are $\delta Z'_w$ and $\delta M'_w$ the effective distance of the fins from the centre of gravity is X_s, say, where:

$$\frac{X_s}{L} = -\frac{\delta M'_w}{\delta Z'_w}, \quad \text{i.e. } \delta M'_w = -\frac{X_s}{L}\delta Z'_w$$

The negative sign arises because the fins are aft.

The effect of the fins on the curvature derivatives can be deduced similarly or, if not available from direct model tests, it can be argued that the rotation causes an effective change of incidence at the fin, such that:

$$\delta Z'_q = \frac{X_s}{L}\delta Z'_w$$

and

$$\delta M'_q = \left(\frac{X_s}{L}\right)^2 \delta Z'_w$$

If the derivatives, as originally determined, give rise to an unstable motion, the required increase in fin area can be deduced using the above relationships and assuming that $\delta Z'_w$ is proportional to the fin area.

EXAMPLE 3. The stability derivatives found for a certain submarine, complete with all appendages are:

$$Z'_w = -0.02, \quad Z'_q = -0.01$$
$$M'_w = 0.012, \quad M'_q = -0.005$$
$$m' = 0.024$$

The corresponding figures for Z'_w and M'_w without fins are 0 and 0.022. Calculate the percentage increase in fin area required to make the submarine just stable assuming m' is effectively unaltered.

Solution: The stability criterion in non-dimensional form is

$$M'_q Z'_w - M'_w(Z'_q + m') > 0$$

Substituting the original data gives −0.000068 so that the submarine is unstable.

If the fin area is increased by p per cent then the derivatives become

$$Z'_w = -0.02 + p(-0.02)$$
$$M'_w = 0.012 + p(-0.01), \text{ in this case } X_s = -\tfrac{1}{2}L$$
$$Z'_q = -0.01 + p(-0.01)$$
$$M'_q = -0.005 + p(-0.005)$$

Substituting these values in the left-hand side of the stability criterion and equating to zero gives the value of p which will make the submarine just stable.

Carrying out this calculation gives $p = 14.8$ per cent, and the modified derivatives become

$$Z'_w = -0.023, \quad Z'_q = -0.0115$$
$$M'_w = 0.0105, \quad M'_q = -0.00575$$

EFFICIENCY OF CONTROL SURFACES

Ideally, the operator of any ship should define the standard of manoeuvrability required in terms of the standard manoeuvres already discussed. The designer could then calculate, or measure by model tests, the various stability derivatives and the forces and moments generated by movements of the control surfaces, i.e. rudders and hydroplanes. By feeding this information to a computer a prediction can be made of the ship performance, compared with the stated requirements and the design modified as necessary. By changing skeg or fin and modifying the areas of control surfaces, the desired response may be achieved.

As a simpler method of comparing ships, the effectiveness of control surfaces can be gauged by comparing the forces and moments they can generate with the forces and moments produced on the hull by movements in the appropriate plane. Strictly, the force and moment on the hull should be the combination of those due to lateral velocity and rotation, but for most purposes they can be compared separately; for example the rudder force and moment can be compared with the force and moment due to lateral velocity to provide a measure of the ability of the rudder to hold the hull at a given angle of attack and thus cause the ship to turn. The ability of the rudder to start rotating the ship can be judged by comparing the moment due to rudder with the rotational inertia of the ship. The ability of hydroplanes to cope with a lack of balance between weight and buoyancy is demonstrated by comparing the force they can generate with the displacement of the submarine. It is important that all parameters be measured in a consistent fashion and that the suitability of the figures obtained be compared with previous designs.

Effect of design parameters on manoeuvring

The following remarks are of a general nature because it is not possible t~ predict how changes to individual design parameters will affect precisely the manoeuvring of a ship.

Speed. For surface vessels increased speed leads to increased turning diameter for a given rudder angle although the rate of turn normally increases. For submerged bodies, turning diameters are sensibly constant over the speed range.

Trim. Generally stern trim improves directional stability and increases turning diameter. The effect is roughly linear over practical speed ranges.

Draught. Somewhat surprisingly limited tests indicate that decrease in draught results in increased turning rate and stability. This suggests that the rudder becomes a more dominant factor both as a stabilizing fin and as a turning device.

Longitudinal moment of inertia. Changes in inertia leave the steady turning rate unchanged. A larger inertia increases angular momentum and leads to larger overshoot.

Metacentric height. Quite large changes in metacentric height show no significant effects on turning rate or stability.

Length/beam ratio. Generally speaking the greater this ratio the more stable the ship and the larger the turning circle.

Problems

1. A rudder placed immediately behind a middle line propeller is rectangular in shape, 3 m wide and 2 m deep. It is pivoted at its leading edge. Estimate the torque on the rudder head when it is placed at 35 degrees, the ship's speed being 15 knots.
2. A ship turns in a radius of 300 m at a speed of 20 knots under the action of a rudder force of 100 tonnef. If the draught of the vessel is 5 m, $\overline{KG}$ is 6 m and $\overline{GM}$ is 2 m find the approximate angle of heel during the steady turn.
3. A rudder is shaped as shown. If it is on the middle line in a single screw ship, how far abaft the leading edge is the centre of pressure?

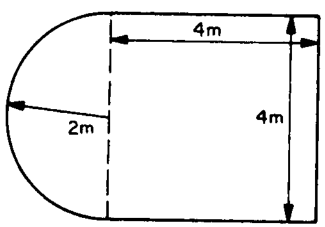


Fig. 13.35

4. A rudder has a profile as sketched in Fig. 13.36.
   Calculate the force on this rudder when operating behind a single centre-line screw at a ship speed of 20 knots ahead with the rudder at 35 degrees. Use the formula due to Baker and Bottomley.

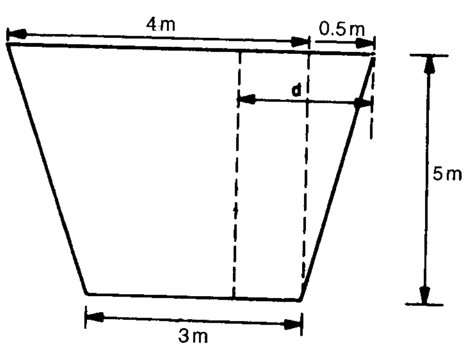


Fig. 13.36

What is the value of d in order that the torque is zero in this condition assuming the rudder is effectively in open water.

5. Calculate the force and torque on the spade rudder shown, which is one of two working behind twin propellers. Assume a rudder angle of 35 degrees and a ship speed of 18 knots ahead.

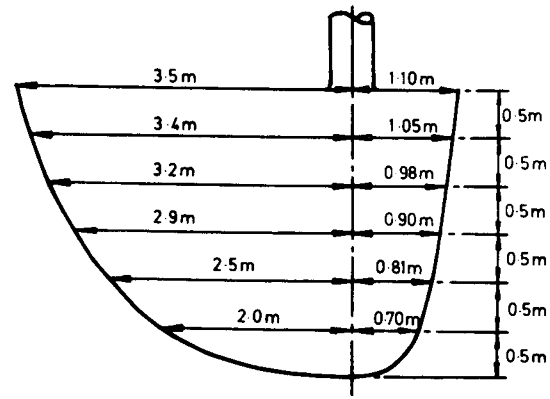


Fig. 13.37

6. The rudder sketched in Fig. 13.38 has sections similar to NACA 0015. Calculate the force and torque on the rudder for 20 knots ahead speed, with the rudder at 35 degrees, assuming no breakdown of flow occurs.

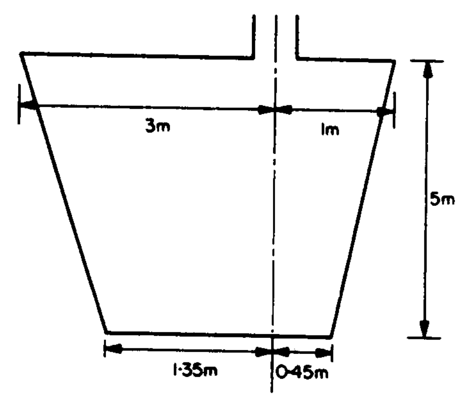


Fig. 13.38

7. Describe the action of the rudder on a ship
   (a) when it is first put over,
   (b) when the ship is turning steadily.
   Sketch a typical turning circle, giving the path of the e.g. of the ship from the point when the helm is first put over up to the point when the ship has turned through 360°. Show the position of the ship (by its centre line) at 90", 180°, 270° and 360° turn. Show on your diagram what is meant by Advance.

Transfer, Tactical Diameter and Drift Angle. Which way would you expect a submarine to heel when turning? Give reasons.

8. The balanced rudder, shown in Fig. 13.39, has a maximum turning angle of 35° and is fitted directly behind a single propeller.

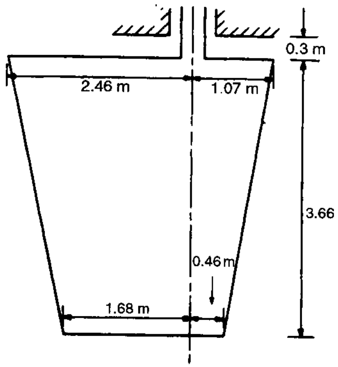


Fig. 13.39

What torque and bending moment are applied to the rudder stock at the lower end of the sleeve bearing, when the rudder is put over at a ship's speed of 26 knots?

In the force equation, $P = KAV^2\delta_R$ (P in newtons, A in m^2, V in knots), take the constant $K = 0.041$. Also, if the length of an elemental strip of the rudder surface, drawn at right-angles to the centre line of the stock, is l then assume the centre of pressure of the strip to be 0.321 from the leading edge.

9. Using Fig. 13.12, plot an NACA 0015 section with a chord length of 5 m. Calculate the area of the section, the distance of the centroid from the nose and the section modulus about each principal axis, assuming a solid section.

Take the x/c values at intervals of 0.1.

10. A twin-screw vessel has a rectangular spade rudder 1.5 m wide and 2 m deep. The axis is 0.5 m from the leading edge. If friction at the rudder stock bearings and in the stearing gear may be taken as 5000 Nm, estimate the range of possible angles which the rudder can take up if the steering gear is damaged while the ship is underway at 30 knots. Distance of centre of pressure abaft leading edge may be taken as chord $(0.195 + 0.305 \sin\theta)$.

11. Details of two ship designs A and B are given below.

	Ship A	*Ship* B
Length on WL, L (m)	215	252.5
Beam, B (m)	24	26.75
Draught, T (m)	7.625	8.0
Area of rudder, A_R (m^2)		50

Design B achieved a tactical diameter of 4.3 ship lengths at 28 knots. Assuming that tactical diameter = const. $\times L^3T/BA_R$, calculate the rudder areas necessary to give tactical diameters of 3, 3.5 and 4.0 ship lengths in design A at the appropriate speed.

12. Calculate the approximate heeling moment acting on a ship of 60 MN displacement assuming

$$\text{Force on rudder} = 21A_R V^2 \delta_R \text{ newtons, } A_R \text{ in m}^2,\ V \text{ in m/s, } \delta_R \text{ in degrees}$$
$$\text{Length of ship} = 150\,\text{m}$$
$$A_R = \text{rudder area} = 50\,\text{m}^2$$
$$V = \text{ship speed} = 18 \text{ knots on turn}$$
$$\delta_R = \text{rudder angle} = 35 \text{ degrees}$$
$$\text{Draught} = 8\,\text{m}$$
$$\overline{\text{KG}} = 10\,\text{m}$$

Height of centroid of rudder above keel = 5 m

$$\frac{TD}{L} = 3.6.$$

13. Two designs possess the following values of derivatives

	Y'_v	N'_v	Y'_r	N'_r	m'
Design A	−0.36	−0.07	0.06	−0.07	0.12
Design B	−0.26	−0.10	0.01	−0.03	0.10

Comment on the directional stability of the two designs.

Assuming both designs are 100 m long how far are the neutral points forward of the centres of gravity?

14. The directional stability derivatives for a surface ship 177 m long are:

$$Y'_v = -0.0116, \quad N'_r = -0.00166$$
$$N'_v = -0.00264, \quad m' = 0.00798$$
$$Y'_r = -0.00298$$

Y'_v and N'_v without a 9.29 m^2 skeg were −0.0050 and 0 respectively.

Show that the ship, with skeg, is stable and calculate the distance of the neutral point forward of the c.g. and the effective distance of the skeg aft of the c.g. What increase in skeg area is necessary to increase the stability index by 20 per cent?

15. A submarine 100 m long has the following non-dimensional derivatives:

$$Z'_w = -0.030, \quad M'_q = -0.008$$
$$M'_w = 0.012, \quad m' = 0.030$$
$$Z'_q = -0.015$$

Calculate the distance of the neutral point forward of the c.g. Is the submarine stable?

If $\overline{\text{BG}} = 0.5$ m and displacement is 4000 tonnef, calculate the critical speed for the after hydroplanes which are 45 m aft of the c.g.

14 Major ship design features

So far in this book we have considered the behaviour of the total ship and how that can be manipulated. Such safe and satisfactory behaviour of the ship as an entity is under the total control of the naval architect. Many professions contribute to the elements of a ship. The naval architect while not directly controlling each of them, has the responsibility of integrating them into the whole design and will usually be the Project Manager. The design must be balanced, each element demanding no more and no less than is a proper share of the total, contributing just enough to the overall performance. Standards of behaviour must be adequate but not more and facilities provided must be no more than is necessary for satisfactory functioning of the vessel. Such a balancing act among so many disparate elements is not easy. The naval architect must be in a position to judge to what extent the demands upon the ship by the individual specialist should be met. This requires enough knowledge of the specialisms to be able to discuss and cajole and, if necessary, to reject some of the demands.

Some of the specialisms are addressed in this chapter before we are able to move on to the ship design process itself. They are all bound up closely with the ship but none more so than the choice of the propulsion machinery.

Machinery

Propulsion machinery for ships used to be tailor-made to conform to the size and predominant speeds of the ship, such as top and cruising speeds. Today, even for warships, it is a question of selecting standard units and combining them in a satisfactory manner. Most merchant ships are expected to proceed at their economical speed for their entire lives so that their propulsion machinery may be optimized in a relatively straightforward manner. So dominant are economic considerations that we should begin with an examination of where the energy contained in the fuel goes. This is shown in Fig. 14.1 for a diesel-driven

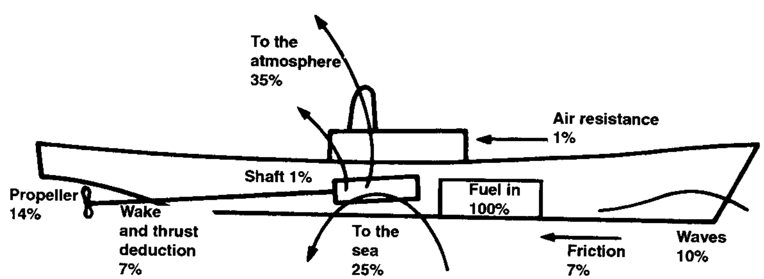


Fig. 14.1 Where the energy goes

frigate at high speed; while exact figures will differ for other ships they will be of the order of size shown. Of the total energy available in the fuel about 60 per cent is lost immediately to the sea via the condenser water and to the atmosphere via the exhaust gases. Less than 20 per cent is actually spent in overcoming the friction of the sea and in creating waves. Yet the diesel engine is the most efficient of the propulsion machinery options open to designers at present. Thermal efficiencies at maximum power of various systems are roughly:

diesels	43%
gas turbine	35%
steam turbine	20%

Relative costs of the basic fuel are currently as shown in Table 14.1 compared to a light diesel oil figure of unity. They do not remain static for long and the economic choice of machinery does require some wise foresight. Note that medium and high-speed diesels and marine gas turbines use exactly the same fuel which is standardized for NATO warships. Because coal is bulky as Table 14.1 shows and is so far associated with relatively low system efficiencies, despite the introduction of fluidized bed boilers, it has not yet found favour with many owners. A diesel engine run on heavy oil is often a preferred fit because of the relative cost advantage and as well as the large slow-running diesels, some medium-speed diesels can successfully use heavy oil. Oil producing countries are well aware of the competition from coal and a trend back to coal remains a persistent possibility. There has also been a significant trend towards high speed diesels, particularly in small vessels.

Table 14.1

	Relative cost/tonne	Relative cost/kl	Ml/kg	Stowage m^3/tonne	Stowage m^3/Ml
Light diesel oil	1.00	1.00	45	1.2	27
Heavy fuel oil	0.62	0.64	43	1.05	24
Coal	0.15	0.27	25	1.5	60

Choice, however, is not dependent entirely upon running costs. The choice of main machinery for all ships is made after an examination of many aspects: ,

(a) demands upon the ship in terms of mass and volume;
(b) overall economy in terms of procurement, installation, running and logistics costs;
(c) range of speeds likely to be needed;
(d) availability, reliability and maintainability;
(e) signature suppression, e.g. quietness, stealth and noxious efflux;
(f) vulnerability, duplication and unitization;
(g) engineering crew and automation;
(h) vibration induced in the ship.

For most merchant ships the space and mass demands upon the ship of a diesel engine installation are acceptable and they show also good overall economy and high reliability. Nor are great flexibility in speed or reduced signatures often requirements. Motor ships are therefore very common. There is a choice in the type of diesel engine. The huge diesels developed for the large tankers of the 1960s are now capable of delivering 40 MW or more at such low rotary speed that they do not need a gearbox. They are very big and heavy and unsuitable for anything but the largest ships. Medium- and high-speed diesels have also benefited from recent developments and are available in a wide range of powers. Double and selective supercharging have increased the output per tonne of diesel machinery and also ameliorated such problems as coking up at fractional power outputs. Figure 14.2 shows a very rough order of installed power necessary for merchant ships of various displacements and speeds. (It is not intended to supplant the need for proper assessment.)

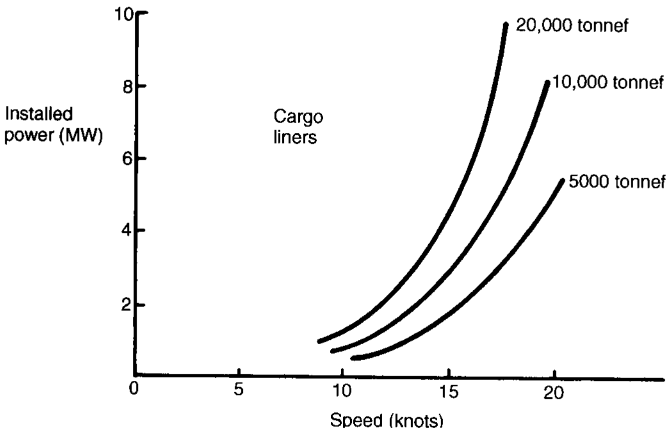


Fig. 14.2 Approximate installed power for cargo liners

Power requirements for small warships are indicated very roughly in Fig. 14.3. For these, the other factors above are often more important. It is usual for a warship for example to proceed for much of its life at an economical speed but to be capable of bursts of high speeds. Furthermore, military operations sometimes demand extreme quietness underwater for anti-submarine warfare and other signature treatments that influence the choice of main machinery. For these reasons it is usual, save in the smallest and simplest warships to arrange standard power units to combine in various ways so that different elements may be selected for each operating condition that has to be met.

Standard units are the diesel engine, steam turbine, gas turbine and electric motor which may be used alternatively or in combination. Acronyms have been developed to describe these succintly, e.g. CODOG meaning combined diesel or gas turbine and CODLAG meaning combined diesel electric and gas turbine. The switchover point from one grouping to another is important. It depends primarily upon the power delivered and the economy of running measured by the specific fuel consumption. Curves such as those shown in Fig. 14.4 are developed for many combinations which, with an assumption concerning the

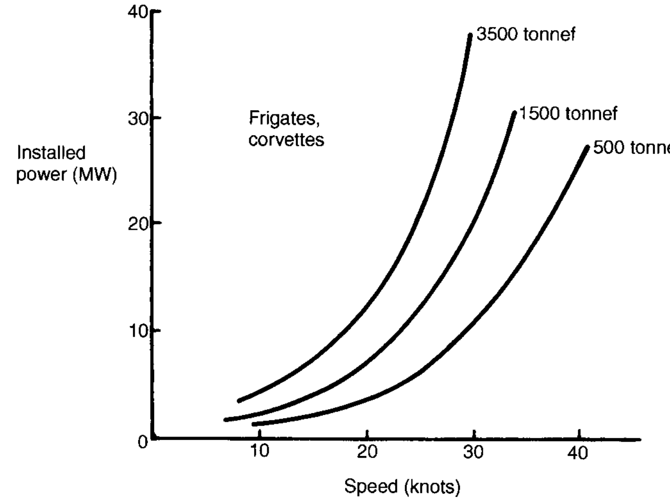


Fig. 14.3 Approximate installed power for warships

likely operating pattern, enable overall economy to be assessed. There are other considerations too such as quiet operating speed, engine wear, diesel coking and gearing. Sudden changes in efficiency or specific fuel consumption with speed are undesirable and the flat characteristics of modern diesel engines and large marine gas turbines have much eased the problems of combining units.

Many engines run with a maximum efficiency at high rotary speed so that a gearbox is necessary to reduce speeds to values acceptable to the propulsor. This is not the place to discuss gearbox design, which is an important study for marine engineers. However, there are step changes in gearbox design driven by maximum tooth loading and other factors which importantly affect the choice of machinery unit combinations. The adoption of a third gear train or epicyclic

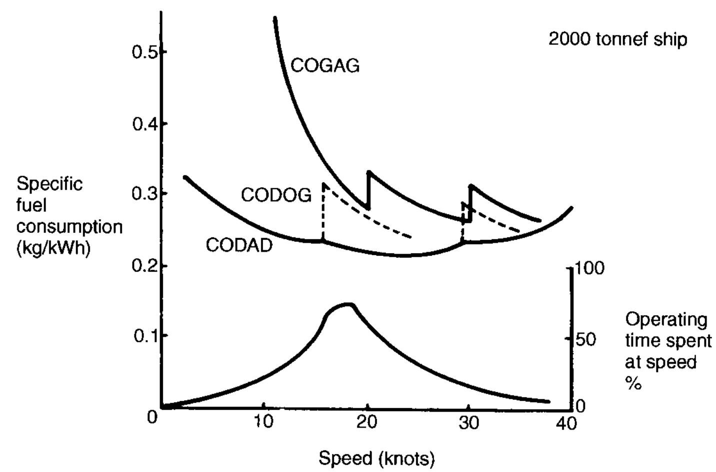


Fig. 14.4 Specific fuel consumption for combined units

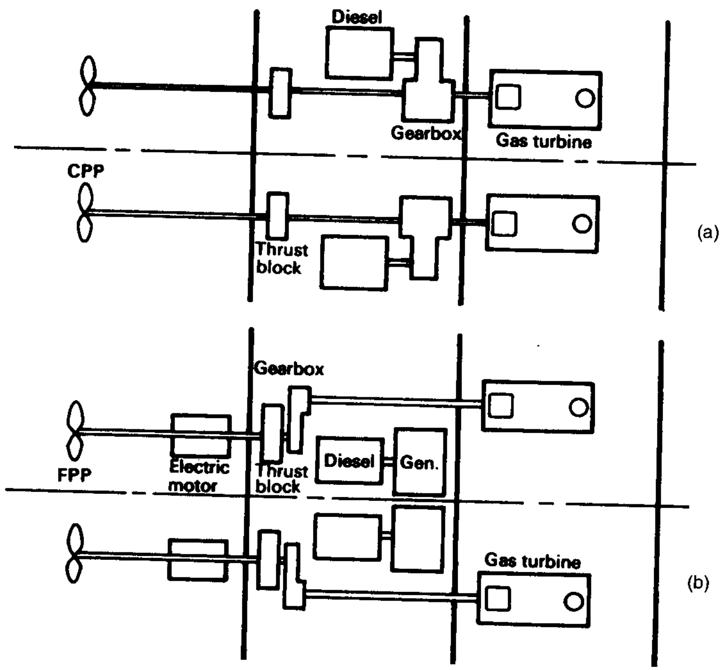


Fig. 14.5 *(a) CODOG, (h) CODLAG*

gearing may boost cost and make demands upon the ship that significantly affect the considerations concerning machinery choice. Reversing of the propulsion may be effected by the gearing or by the pitch reversal of the propulsor blade, by clutching in an auxiliary drive or by reversal of the prime mover itself.

Two of many machinery configurations are shown in Fig. 14.5.

Choice of numbers of shafts and the type of propulsors are matters of compromise. A single shaft and a large diameter slow helical propeller are propulsively the most efficient arrangement. Twin shafts in lower wake conditions and with necessarily rather smaller diameter propellers may lose 0.05 on propulsive coefficient comparatively and are slightly more expensive. However, they do allow more total power to be transmitted, they provide better standby propulsive power in case of failure and they can provide a turning moment on the ship when there is no way on the ship or when steering failure occurs. So far as the propulsor is concerned, nothing is more efficient than a well-designed helical propeller, approaching the theoretical maximum of the actuator disc (Fig. 14.6). Alternative propulsion devices need to be considered however for other reasons. Vertical axis propellers give remarkably responsive steering for such vessels as ferries and river craft and dispense with rudders; shrouded propellers may be quiet; water jets can be especially compact in small craft; paddle wheels provide good manoeuvrability; controllable pitch propellers, while a few per cent less efficient, do give a rapid means of reversing thrust and an ability to select the correct pitch for each condition of operation.

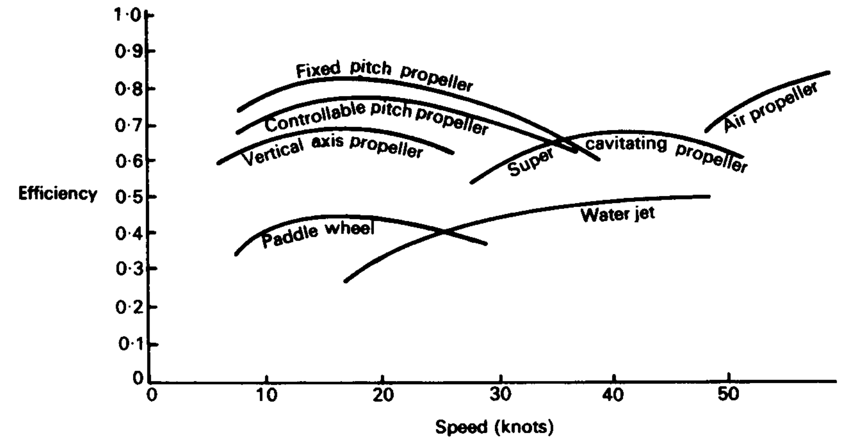


Fig. 14.6 *Approximate efficiency of propulsion devices*

In concluding this small introduction to the considerations affecting machinery choice we might cast our minds forward. With world reserves of oil running out and oil becoming increasingly uneconomical in the coming years, thoughts are already turning to alternative fuels. Wind is again being seriously studied both for main propulsion and as a means of economizing. Oil from shale rock, while currently uneconomical, will become increasingly attractive. Solar panels do not yet look very promising because of the enormous areas required to provide enough energy for a ship. Coal, on the other hand, will remain plentiful for many years yet and can be conveyed in pulverized form, pneumatically, in slurry form or even converted into oil. Nuclear propulsion has already been safely used in hundreds of vessels, mainly submarines. In merchant ships it has been successfully used although barely economically; it does give rise to social problems. While nuclear reactors are indeed exceedingly safe and reliable, the consequences of serious malfunction do give rise to understandable anxieties.

For small fast ships the diesel looks like remaining a favourite choice for a long time yet. The reasons are very clear. Despite being relatively light for their power output, gas turbines suffer three significant disadvantages. Their specifier fuel consumption is as much as double that of a comparable diesel for powers less than about 10 MW. Their power output is markedly sensitive to the temperature of the ambient air drawn in. Finally they need quantities of air three or four times those needed by diesels of similar power, so that designers of small ships find themselves constrained by large deck openings.

AIR INDEPENDENT PROPULSION (AIP)

Diesel electric submarines suffer from limited underwater endurance, particularly at high speed. While they operate their diesels when snorkelling they are vulnerable to detection at such times. The problem was overcome by applying nuclear power to produce the 'true submarine' but this solution is expensive.

It also presents problems in the disposal of nuclear waste products and eventually the boats themselves.

In the maritime field the need for a cheaper way to obtain long underwater endurance led to research into AlP systems such as the fuel cell. In other fields, for example the automotive, environmental concerns drove research for powering cars and other forms of land transport.

In this section we consider AlP systems, other than nuclear, for ship propulsion, the emphasis being on submarine applications. Fuel and an oxidant are required to generate electricity by either a heat engine (e.g. closed cycle diesel, Stirling engine, closed cycle gas turbine), or an electro-chemical cell (e.g. lead acid battery, fuel cells). Different fuels can be used. There are obvious attractions in using diesel fuel but methanol and hydrogen are also contenders. Diesel is most easily stored; methanol requires a more complex stowage and a reformer is needed to extract the hydrogen in pure form; hydrogen, for a given amount of oxygen, generates about a third more energy than diesel but it introduces safety problems if stored in liquid form; metal hydrides are compact and a safer means of storing the hydrogen.

It would be impractical to carryall the air required for normal combustion within the submarine even using cryogenics. Instead oxygen is carried in liquid form under pressure at low temperature. Including the tanks needed to provide the extra buoyancy to compensate for a heavier system, a typical AlP system requires about 15 times the displacement of the corresponding diesel oil storage of a conventional submarine.

The choice of AlP system will depend upon the power and endurance, that is the size and mission profile of the submarine. There is no single best buy. The designer must consider the impact on the total submarine system. Contenders are:

I. fuel cells with metal hydride or reformed methanol for hydrogen storage. Chemical energy is converted directly into electrical energy. Efficiencies are about 50 per cent. The Proton Exchange Membrane Fuel Cell (PEMFC) is felt to have good potential and a production model of a 300 kW PEMFC plant has been produced for a German submarine design. This PEMFC system consists of a stack of single cells separated by a bipolar plate. Between each plate is a proton exchange membrane coated with a platinum-based electro- catalyst;
2. the closed cycle diesel. The engine runs on a synthetic atmosphere in which the exhaust gases are treated by absorbing carbon dioxide and adding oxygen, and then recycled to the engine inlet. Efficiency is about 30 per cent;
3. the Stirling engine which converts fuel into heat and then into mechanical work. Efficiency is similar to the closed cycle diesel;
4. a steam turbine fed with steam generated by burning fuel with oxygen.

Efficiency figures refer to the plant itself. The high efficiency of the fuel cell arises because it converts the hydrogen and oxygen directly into electricity. When that plant is integrated with other design features to produce the total submarine, the differences in the overall effectiveness of the different systems is very much reduced.

Fuel cells are also being considered for surface ships to provide ship **service** power to reduce on-board fuel consumption and meet future, increasingly strict, emission standards. Other advantages are that they have no moving parts, are reliable and easy to maintain, and may be friendly to the environment.

ELECTRICAL GENERA nON

Depending upon the type of main propulsion machinery, one or more types of prime mover will be used for electrical generation, e.g. in a steam ship, steam turbo-alternators are fitted with back-up plant powered by diesels or gas turbines. The number of plants depends upon the total capacity required. This capacity for a ship might at first sight be thought to be the 'total connected load', i.e. the sum of all electrical demands. It is not so simple as this because, first, there are several conditions in which the ship requires different electrical equipment working and, secondly, in anyone such condition there is a diversity of equipment in use at any moment. The diversity factor is largely a matter of experience. Conditions for which electrical demands are calculated vary with types of ship but may include the following:

(a) normal cruising, summer and winter (in a cargo ship, 40-50 per cent of the normal cruising load is due to machinery auxiliaries);
(b) harbour, loading or unloading;
(c) action, all weapons in use;
(d) salvage, ship damaged and auxiliaries working to save the ship;
(e) growth during the ship's life (typically 20 per cent is allowed in merchant ships and warships),

There has been a rapid growth in the generator capacity of ships since 1939- a frigate design, for example, which might have needed I MW in 1950, needed 3 MW by 1960 and 50 MW by 2000, including 40 MW for electrical main propulsion on two shafts. Generating capacity must be provided to meet suitably any of these loads. Machines should be loaded in these conditions near maximum efficiency. Other considerations enter into the problem too-the needs of maintenance, availability of steam in harbour, break down, growth during the ship's life, capacity when damaged.

From all of these considerations, the numbers, sizes and types of generators are decided. The greatest flexibility would be provided by a large number of small capacity machines, but this is not the most efficient way. Too many generators could be very heavy and involve complex control systems. In addition to the main generators usually a salvage generator is sited remote from the primary generators and above the likely damaged waterline if possible.

Where the electric motor is part of a propulsion system some special problems arise. Switching such large power demands causes surges which need control equipment so that other parts of the ship are not deprived. Often they require their own generators and switchgear. They may, however, be incorporated into the hotel load and may even be connected into buffering arrangements using batteries and rectifier units. When it has been fully developed, superconducting electrical machinery may become attractive although it must be remembered that such a system must include also gas compressors and refrigeration machinery.

Systems

Apart from the electrical systems, over fifty different systems may be found in a major warship for conveying fluids of various sorts around the ship. Even simple ships may have a dozen systems for ventilation, fire fighting, drainage, sewage, domestic fresh water, fuel oil, compressed air, lubricating oil, etc. Because their proper blending into the ship affects very many spaces, it is desirable that the naval architect should have complete control of them, except those local systems forming part of a machinery or weapons installation, even if certain of them are provided by a subcontractor. The naval architect must be completely familiar with the design of all such fluid systems and be capable of performing the design.

ELECTRICAL DISTRIBUTION SYSTEM

The generation of electrical power has already been discussed briefly. Classification Societies allow considerable flexibility in the method of distribution of this power, but the basic design aims are maximum reliability, continuity of supply, ease of operation and maintenance and adaptability to load variation. All this must be achieved with minimum weight, size and cost. The actual system adopted, depends very much upon the powers involved. Prior to 1939, installations were small, e.g. 70 kW in a typical cargo ship. Few passenger liners had as much as 2 MW. Most installations were d.c. and it was not until after 1950 that shipowners began to require a change to a.c. generation and distribution. This change was influenced very much by the savings in weight and the reduction in maintenance effort accruing which became more important as powers increased. Typical figures are:

10,000 tonf dwt. dry cargo ship	I MW
Tankers	1.5-5 MW
Container ship (3.3 kV)	8 MW

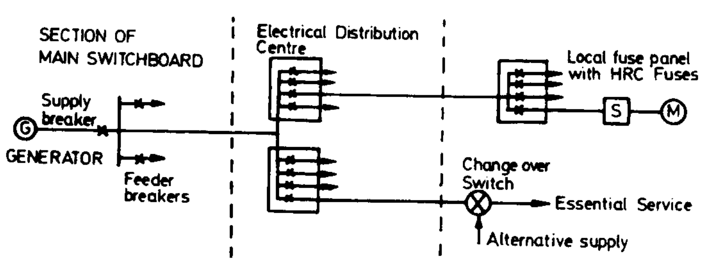


Fig. 14.7 Typical distribution system

All major warship installations in the Royal Navy are a.c. Distribution '" achieved by feeding from a small number of breakers grouped on the switchboard associated with each generator to electrical distribution centres throughout the ship.

Distribution from the EDC is by moulded case circuit breakers of 250 and 100 amp capacity. Finally power is supplied to small circuits of less than 30 A by high rupturing capacity fuses. Because of their relatively high starting currents, motors in excess of 4 kW are supplied through MCBs. Important services such as steering are provided with alternative independent supplies by well-separated cable routes from two generators feeding a change-over switch. In case of damage, a system of emergency cables is provided for rigging through the ship to connect important services to generators which are still running.

Passenger ships must have independent emergency lighting. Typical voltages of distribution are:

Merchant ships:	d.c.	220 V (power and lighting)
		110 V for some small ships
	a.c.	440 V at 60 Hz or 380 V at 50 Hz
		3.3 kV at 50 or 60 Hz generation in a few ships
		115 V or 230 V at 60 Hz (lighting)
Warships:	a.c.	440 V at 60 Hz 3-phase
		liS V at 60 Hz for lighting and domestic single phase circuits.

Most warships within NATO and some merchant ships adopt the insulated neutral earth system in order to preserve continuity of supply under fault conditions. Neutral earthing however does permit the economy of single-pole switching and fusing. Many shipowners have preferred a single solid bus bar system with all connected generators operating in parallel. This system gives maximum flexibility with minimum operating staff. The maximum installed capacities are limited by the circuit breaker designs, e.g. assuming breakers of 100 kA interrupting capacity, the system is limited to 1800 kW at 240 V d.c. and 3000 kW at 440 V a.c. Other disadvantages are that a fault at the main switchboard may cause total loss of power, and maintenance can be carried out only when the ship is shut down. The alternative is the split bus bar system whiQir gives greater security of supply and enables maintenance to be carried out by closing down one bus bar section.

Alternating current is converted to direct current by transformer rectifier units or to a.c. of a different frequency by static frequency converters. Both of these distort the sinusoidal waveforms so that spurious signals may be created within the electrical equipment throughout the ship. Design of the electrical distribution system therefore involves complicated assessments of the electromagnetic compatibilities.

PIPING SYSTEMS

Design of any liquid piping system begins by plotting on ship plans the demands for the fluid and by joining the demand points by an economical

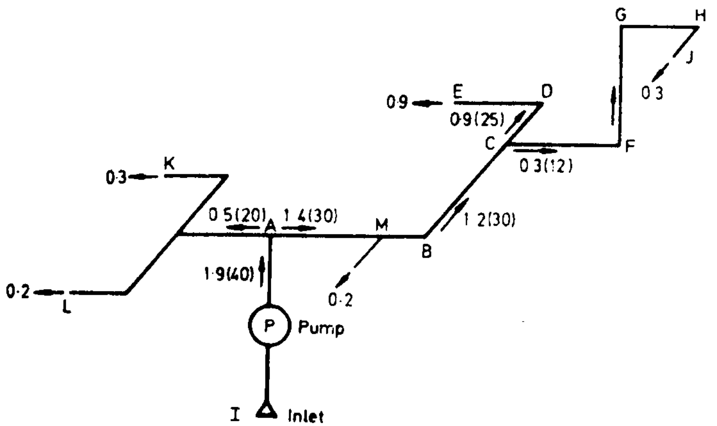


Fig. 14.8 *Typical simple open ended network*

piping layout compatible with the architecture of the ship. (A discussion of the type of piping network to be adopted Occurs later but it needs to be chosen at this stage.) Pipe sizes are then allocated on a trial basis by permitting velocities generally about 1.5-2 *mls* at which level experience has shown that erosion and noise are not excessive. It may later be found desirable to allow some stretches to run at speeds up to 3 *m/s.* From this network, must be estimated the pressure

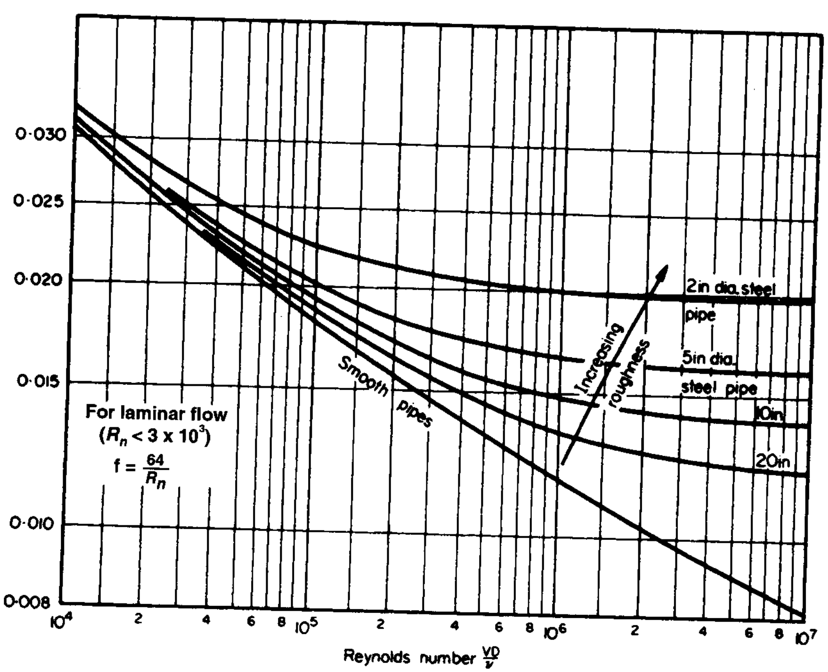


Fig. 14.9 Pipe friction coefficient

required from the pump to create this flow. Figure 14.8 shows in perspective" very simple open-ended network that might result. By working back from the presumed simultaneous demands, the flows in the different parts of the systcm can be found by simple addition and pipe sizes allocated. The pump must thcn be capable of drawing water from I and delivering it to the remotest point against *(a)* the resistance of the system and *(b)* against gravity. Resistance of the system is due to friction in the pipes and resistance (or losses) due to bends, junctions, valves, expansions, contractions, filters and nozzles. Fortunately, all of these fittings losses can be expressed in similar form

$$\text{fitting loss} = K\rho\frac{V^2}{2}$$

where K is a factor defined in Table 14.2, ρ is the mass density of the fluid, V is the velocity of flow.

Table 14.2
Losses in fittings

Fitting	$K = \dfrac{\text{loss}}{\rho V^2/2g}$	Equiv. length $\dfrac{L}{D} = \dfrac{K}{f}$
Gate valve	0.2 for $D = 25$ mm	11
	0.1 for $D = 300$ mm	9
90° Angle valve or 60° oblique valve	3.0 for $D \geq 50$ mm	190
	4.5 for $D = 12$ mm	215
Globe valve	6.0 for $D \geq 50$ mm	375
	9.0 for $D = 12$ mm	430
Oblique valve 45°	2.5 for $D \geq 50$ mm	160
	3.5 for $D = 12$ mm	150
Diaphragm valve	1.5	70
Plug or straight through cock	0.4	20
Sudden contraction	On outlet velocity	
	0.4 for $A_1/A_2 = 10$	—
	0 for $A_1/A_2 = 1$	
Sudden expansion	On inlet velocity	
	0.8 for $A_1/A_2 = 0.1$	—
	0.2 for $A_1/A_2 = 0.5$	
Inlet, smoothed entry	0	0
90° bend, radius R	0.3 for $R > 2D$	15
Outlet	1.0	50
Equal tee,		
flow past	0.3	15
flow round	1.2	60
flow from branch	1.8	90
Elbow, 90°	0.6	30
45°	0.1	12
Strainer	0.8	40

Note: Interpolate linearly. Do not extrapolate

Now, the frictional loss in a circular pipe is

$$\text{frictional loss} = f\frac{L}{D}\rho\frac{V^2}{2}$$

where f is a factor dependent on Reynolds' number and pipe roughness as given in Fig. 14.9; L is pipe length, and D is pipe bore.

If a pipe is assumed smooth, the fitting loss can conveniently be expressed in the same form as the frictional loss by calling it a number of diameters of equivalent length of pipe—a gate valve, for example (Table 14.2), is equivalent to the frictional loss due to about eleven diameters of pipe length. Ship pipes are relatively smooth with the possible exception of small diameter steel pipes and this is a common and convenient artifice; while there are clearly approximations involved, it is accurate enough for most ship systems.

The total head, therefore, required of the pump is

$$H = \sum f\frac{L}{D}\rho\frac{V^2}{2} + \sum (K_1 + K_2 + \cdots)\rho\frac{V^2}{2} + \rho h$$

where h is the vertical separation of inlet and outlet. What is important to realize is that if the pump delivers along the most resistful path, all other paths will be satisfactory. This worst path, called the index path or circuit may not always be obvious from inspection and several paths will have to be examined to find the most resistful one. Having determined this, other paths may have to be made equally resistful by the insertion of obstructions such as orifice plates to avoid too high a delivery and pressure. Let us illustrate these points by an example.

EXAMPLE I. Calculate the performance required of the salt water pump for the system shown in Fig. 14.8 which meets the demands shown in litres per second. Estimated lengths and fittings, not shown in Fig. 14.8, are given in the table below. The inlet to the pump, **I,** is at a pressure of $50\,kNjm^2$ and the vertical heights separating I and E and I and J are respectively 15 m and 20 m. Delivery is required at a pressure of $150kNjm^2$.

	Length, m	90° Elbows	45° Oblique valves
IA	10	0	0
AM	45	2	2
MB	6	1	1
BC	12	4	1
CD	20	2	2
DE	12	0	1
CF	3	1	0
FG	6	1	1
GH	6	2	0
HJ	6	0	1

Solution: It is not obvious from inspection whether IAE or IAJ is the index path, although IAK or L are clearly not. It will be assumed that the pipes are smooth. Tabular form is most convenient for this calculation to find $\Sigma\,[f(L/D) + K_1 + K_2 + \cdots]V^2$ which must then be multiplied by $\rho/2$. Take ν to be $1.05 \times 10^{-6}\,\text{m}^2/\text{s}$.

Even without gravity head, the index path is clearly IAJ. For this path the total losses are

$$\frac{1}{2} \times 632.2\frac{\text{m}^2}{\text{s}^2} \times 1025\frac{\text{kg}}{\text{m}^3} = 324{,}000\,\text{N/m}^2$$

The pump must also deliver against gravity head which is

outlet pressure + gravity head − inlet pressure

$$150{,}000\,\text{N/m}^2 + 20\,\text{m} \times 1025\frac{\text{kg}}{\text{m}^3} \times 9.807\frac{\text{m}}{\text{s}^2} - 50{,}000\,\text{N/m}^2 = 301\,\text{kN/m}^2$$

Therefore, total delivery required by the pump = 1.9 litres per second at a pressure differential of $625\,\text{kN/m}^2$.

This example illustrates the principles of pipe system design. For a given fluid, it is possible to devise charts showing frictional loss plotted against pipe velocity, diameter and quantity which speed up the calculation. The example

	Length (m)	L/D	Equiv. fitt. L/D	Total L/D	V (m/s)	$R_n = \frac{VD}{\nu}$	f	$f\frac{L}{D}V^2$
IA	10	250		310	1.5			14.3
Tee			60					
AM	45	1500	370	1885	2.0	5.7×10^4	0.0205	154.6
Branch			15					
MB	6	200	185	385	1.7	4.9×10^4	0.021	23.4
BC	12	400	275	675	1.7			41.0
Total (i)								233.3
Tee			15					
CD	20	800	370	1185	1.8	4.3×10^4	0.0215	82.5
DE	12	480	155	685	1.8			47.7
Outlet			50					
Total (ii)								130.2
Tee			60					
CF	3		30					
FG	6	1750	180	2280	2.7	3.1×10^4	0.024	398.9
GH	6		60					
HJ	6		150					
Outlet			50					
Total (iii)								398.9
(i) + (ii)								363.5
(i) + (iii)								632.2

shows also the relatively large effects of valve losses and the effects of pinching the pipe diameter. If the final 21 m of piping, for example, were to be of 18mm diameter instead of 12mm, the pressure required of the pump would be reduced by 138kN/m^2 and this would be an obvious next step in designing this particular system (when IAL would become the critical path).

Several factors affect the choice of the type of system for a particular purpose. By the nature of the demand, some systems must be closed and, once the system is primed, gravity does not influence pump characteristics (except for impeller cavitation), so that the pump delivers solely against system resistance-a hot water heating system is typical. Whether open ended, like firemain and domestic systems, or closed, the designer must consider how important is system reliability. If a simple distribution, as in the example, is adopted, pump failure will cause a cessation of supply which may be acceptable in some systems. Such a system is called a tree system. More often, a standby supply will be needed and this is achieved by cross connecting two adjacent tree systems, the whole ship being served by several tree systems each of which normally operates independently. (Fig. 14.10.) Emergency supply by one pump to two cross connected tree systems will, of course, reduce the pressure at the demand points and, unless properly designed, may result in totally inadequate supply. This case must therefore be the subject of calculation during the design stage.

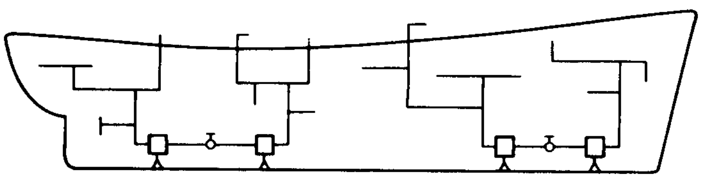


Fig. 14.10 Cross connected tree systems

Systems of importance where no interruption of supply can be tolerated are designed as ring mains with a number of pumps running in parallel. This type of system permits pumps to be rested or to break down or pipes to be damaged with a high chance of maintaining supply to equipment. Chilled water systems in warships, where deprivation of cooling water to some equipment for even a few seconds would damage performance, are typical. There are many different index paths in this case depending upon which pumps are operating or whether cross connections are opened to isolate a damaged section; moreover, the positions of null points where there is no flow in the system are not obvious. There will be a number of trial flow patterns to be tried and here again is a fruitful area for computer programming. A typical large ship chilled water ring main system is shown simplified in Fig. 14.11. There are many variations possible to this scheme and different safety and emergency devices can be incorporated; for example, automatic pressure actuated starting for alternative pumps. In a complex system, it is necessary to regulate the flow to each facility concerned by means of constant flow devices in order that the system may be balanced. Such devices meter the flow to the required amount for a wide range of pressure differential.

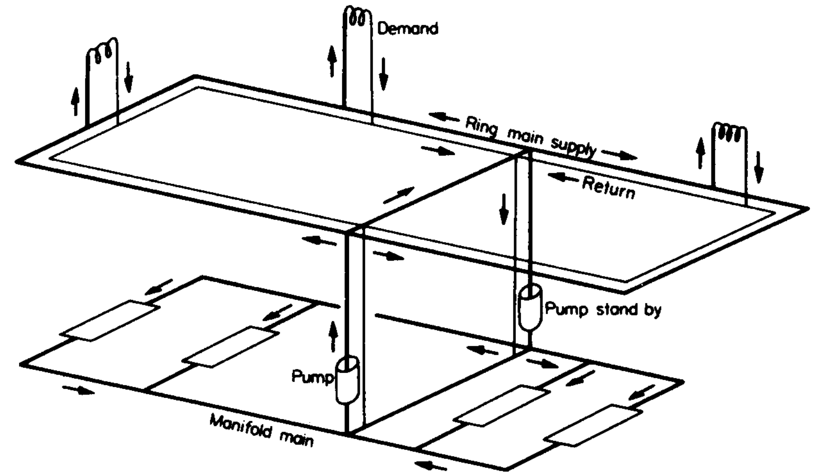


Fig. 14.11 Ring main chilled water system

In this short description of the design of piping systems, it has not been possible to cover all design features, for which separate textbooks are available. The matching of the pump characteristic normally designed by a sub-contractor, to the system demands must be considered with care to ensure stable operation. Positioning of the pump in the ship is also important to avoid a high suction demand on the pump. System priming must be considered. The need for a reservoir or gravity tank to even out the fluctuations in demand is also important. Filtering and flushing of the system must be considered and bleed valves at local high points provided to remove trapped air.

AIR CONDITIONING AND VENTILATION

It is commonly believed that ventilation in a ship is required to enable people to breathe. In fact, an atmosphere fit for breathing can be achieved on very small quantities of fresh air-a small fraction of that needed for ventilation. The major purposes of ventilation are:

(a) to remove heat generated in the ship;
(b) to supply oxygen for supporting burning;
(c) to remove odours.

For most compartments outside machinery spaces, the need to remove heat predominates. Looked at from this point of view, it is clear why normal ventilation often fails to provide comfort. Air drawn in from outside must leave hotter than it entered; a hot, muggy day outside will produce hotter. muggier conditions inside. Heat created within a compartment will be collected by the ventilation air which will be exhausted, hotter, to the atmosphere one of the two natural sinks for heat available to a ship.

Air conditioning uses the other major natural heat sink, the sea. Heat produced within the ship is ultimately exhausted to the sea and an air con-

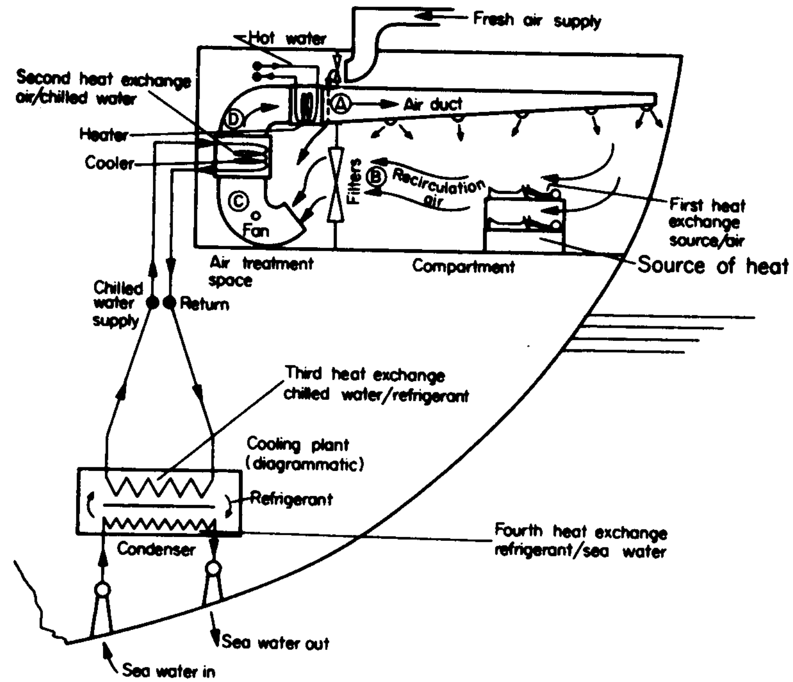


Fig. 14./2 *Typical air conditioning system, diagrammatic*

ditioning system must efficiently permit this transfer, as will presently be described (Fig, 14.12). While ventilation does remove heat and may create complete comfort for machinery, it cannot effect proper control of the three principal factors which affect human comfort, namely:

(a) air temperature;
(b) air humidity;
(c) air purity.

It is these three factors which air conditioning controls, to varying degree.

A typical air conditioning system is illustrated diagrammatically in Fig. 14.12.

The source of heat in the compartment first loses some heat to the air blown over it; this air is passed through a fan, together with a modicum of fresh air and over the surfaces of a coil heat exchanger rendered cold by chilled water, before returning to the compartment. The chilled water carries this heat to refrigeration machinery which conveys it to the sea in its condenser. Heat produced by the source must therefore be efficiently transferred from one medium to another three or four times until it reaches the sea. There are three principal components of the process; in reverse order they are:

(a) Refrigeration machinery for cooling and calorifiers for heating. Design of this is a specialized task which can conveniently be isolated from the design of the whole system, provided that the tasks required of it are adequately

defined by the system designer (i.e. pressure-quantity relationship. tel11-perature ranges, etc.).

(b) Water systems, chilled water and hot water. These are designed in the manner already described previously for piping systems. There may well be requirements for dual chilled water supply for important equipment and devices providing a constant flow are often essential.

(c) Air system. This system, involving two heat exchanges, is the crux of air conditioning and is the subject of the calculations described presently. In so doing, it is necessary to assume on the part of the reader, a familiarity with some elementary physics; some definitions of air measurement are given in Chapter 9. Basically, the air system is a recirculatory one with a small quantity of fresh air make up, sufficient only to keep bacteria levels and odours down. Physiologists recommend between 0.15 and 0.30 m^3/min, although less will limit bacteria.

All ambient air contains water. The amount carried can be measured by two thermometers, one of which is kept wet and the relationships between wet and dry bulb readings have been related by a chart known as the *psychrometric chart.* This chart relates wet and dry bulb temperatures, latent and total heats, percentage relative humidity and specific volume of air in any condition. Such a chart is shown in Fig. 14.13.

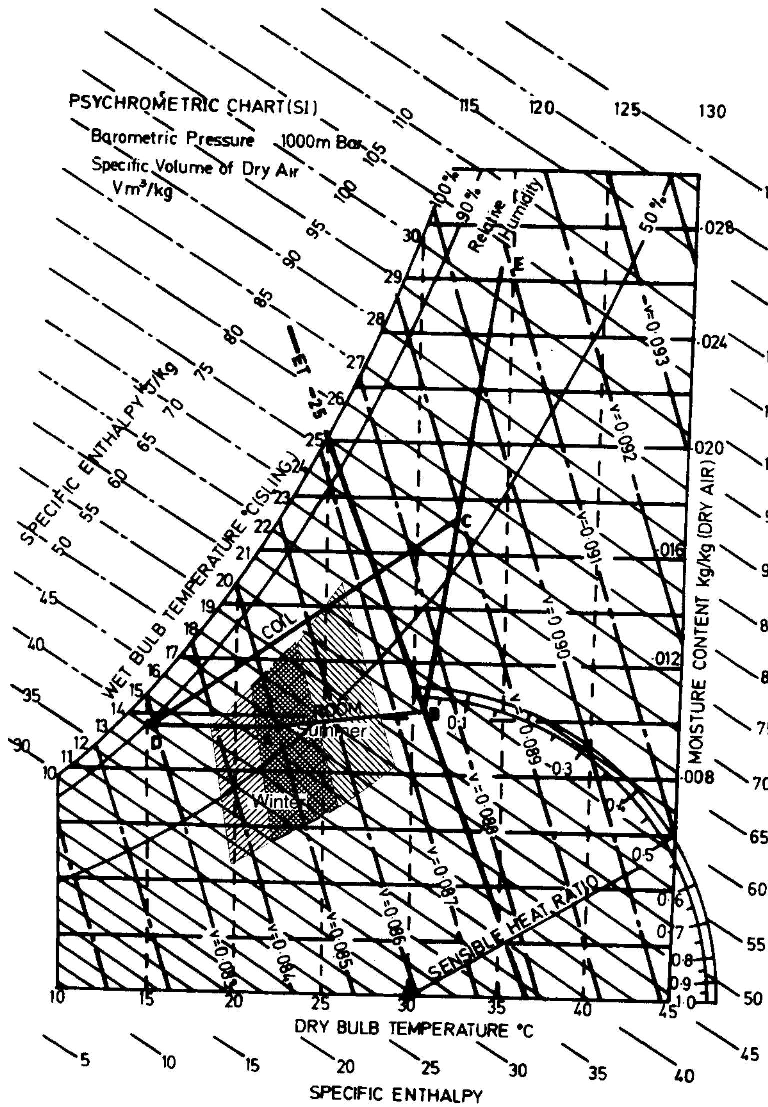
What condition of air is comfortable to personnel? Such is the accommodating nature of the human body that there is a good deal of latitude, but experiments have shown that the areas shaded in Fig. 14.13 for summer and winter are the most suitable. Any air conditioning system should therefore aim to produce communal compartment conditions in the middle of this shaded area and to enable individual cabin occupants to select conditions over such a range.

A further measure of human comfort is provided by the *effective temperature* (ET) *scale* (it is not a temperature), which is a variable ratio of wet/dry temperatures and also air velocity, found experimentally to accord a feeling of comfort. Above 25 ET discomfort increases and it is this limit, known also as the *threshold of comfort,* to which warship systems are designed to operate in extreme tropical ambients. This limit, although the extreme design condition, is infrequently met in warships which operate, like merchant ships, within t~ comfort zones.

The first step in the process of air conditioning design is the determination of the sources of heat. A given compartment gains heat by conduction through deck and bulkheads, from electrical equipment, hot pipes, lighting, the ventilation fan itself and from personnel. Each of these is a source of sensible heat and

Table 14.3
Personnel heat (Watts per person)

People in:	Sensible heat	Latent heat
Normal mess	45	135
Recreational space	45	163



Pig. 14.13 *Air cycle for system in Fig.* 14.12, *using* Sf *psychrometric chart*

)ersonnel, in addition, aJ e a source of latent heat of evaporation. Heat given off)y personnel depends o1 how heavily they are working; average figures are ;iven in Table 14.3.

Gains through bound: ries follow the law

Heat gain = U x su face area x temperature difference

J is the rate of heat tram ferred per unit area per degree and is derived from the hermal conductivity cor stant k for the material of the boundary. Values for orne materials are given in Table 14.4.

To obtain U, the insulating effect of the still air next to the boundary, **known** as *surface film resistance R*, must be added in. For air outside the ship R is about 0.039 in metric units, for air inside manned compartments it is 0.110 and for a closed air space 0.18 and for water is 0.0009, for steam 0.0001 and for Freon 0.0009.

Table 14.4
Thermal conductivity constants

Material	k(W /mK)
Standard glass fibre marine board	0.043
Cork slab	0.042
Block asbestos	0.055
Plywood, fireproofed	0.152
Teak, across grain	0.139
Cork linoleum	0.072
Aluminium	120
Steel	46
Glass (in shadow)	1.01

$$U = \frac{1}{R + \sum(x/k)}$$

where x is the insulation thickness. Furthermore, due to imperfections in fitting, the k values for insulants given in Table 14.4 are, from experience, doubled (except steel). Thus for a surface of 25 mm steel + 25 mm glass fibre + 6 mm plywood inside manned compartments.

$$U = \frac{1}{0.110 + 0.5(0.0011 + 0.5814 + 0.0395)} = 2.375\ \mathrm{W/m^2K}$$

Temperature difference is normally that of compartment temperatures but where surfaces are exposed to the sun, as in the tropics, they are assumed to acquire temperatures of 60 °C if horizontal and 49 °C if vertical. Machinery space deckheads are assumed to be at 82 °C and galleys and auxiliary machinery rooms at 49 °C.

The whole heat gain calculation is performed on standard forms which are derived

(*a*) total heat

(*b*) sensible heat ratio $= \dfrac{\text{sensible heat}}{\text{total heat}}$

A typical heat load calculation might appear as in Table 14.5. t_0, t_i and t_d are temperatures outside, inside and the difference.

Having performed the heat gain calculation, it is necessary to return to the psychrometric chart. The cycle of operation of the air in the compartment is: heat and moisture gained by recirculation air which is returned, augmented by

Table 14.5
Heat gain calculation

Source	Dimensions (m)	Type	t_0 °C	t_i °C	t_d K	Area A, m²	U W/m²K	Heat gain UAK (W)
Port bulkhead	10 × 3 25 mm	MFMB	49	29	20	30	2.38	1428
Stbd. bulkhead	10 × 3 25 mm	MFMB	34	29	5	30	2.38	357
After bulkhead	8 × 3 25 mm	MFMB	29	29	0	24	2.49	0
Fwd. bulkhead	8 × 3 30 mm	MFMB	49	29	20	24	2.18	1046
Crown	10 × 8 6 mm	lino	29	29	0	80	6.58	0
Deck	10 × 8 50 mm	MFMB	49	29	20	80	1.45	2320
								5151
Body heat, sensible	28 men × 45							1260
Lights	2000 W							2000
Equipment	5.85 kW							5850
Space heaters	2 kW							2000
Fan	1.6 h.p. × 745.7							1193
TOTAL, SENSIBLE								17,454
Body heat, latent	28 men × 135							3780
TOTAL HEAT								21,234

$$\text{SENSIBLE HEAT RATIO} = \frac{17,454}{21,234} = 0.822$$

some fresh air, to the cooling coil; at the coil it is cooled below its dew point to remove heat and moisture and, with possibly some after warming, returned to the compartment. This cycle must be constructed on the psychrometric chart as illustrated in Fig. 14.13 for the positions illustrated in Fig. 14.12.

After leaving the after warmer at A, the air picks up sensible and latent heat along AB, constructed to a slope representing the sensible heat ratio obtained from the heat gain calculation and of length given by the total heat gain. At B, the air leaves the room near the edge of the comfort zone along the 25 ET line. BC is obtained by proportioning BE in the ratio of recirculation/fresh air, E representing the condition of the intake air. (This, will generally have been pre-treated at the point where it enters the ship in order to avoid excessively hot or cold trunks between that point and the compartment.) At C, the air mixture enters the cooler. CD represents the extraction by the coil to give a point D horizontally from A since the heat added by the after warm DA is all sensible. D needs to be compatible with the available chilled water temperature and, for an efficient cooler, on the 90 per cent relative humidity line. CD then represents the total heat and sensible heat ratio of the coil (or coil slope) which permits a suitable coil to be selected from a standard range. There is a certain amount of trial and error about this process, but it is not difficult to reach a satisfactory cycle with B in a comfort zone or at least below the 25 ET line. Often, this can be achieved without after warm. The quantity of air recirculated is now obtained from the sensible heat pick up in the room over the dry bulb

temperatures represented by A and B. If this, as in the heat gain example, is 9 °C, with the specific heat of air 1009 J/kgK,

$$\text{recirculation air returned at B} = \frac{17,454}{9 \times 1009}$$
$$= 1.92 \text{ kg of dry air per second}$$

This must now be proportioned up with the fresh air to give the total air to be handled by fan and cooler. Alternatively, this may be obtained directly from the sensible heat and dry bulb readings of C and D.

This process is fundamental to good air conditioning design and represents the nub of system calculation. There are, however, many other calculations which the space here available does not permit to be described in full:

(a) Heater sizing. It is unlikely that the capacity of the heater will be determined by after warm necessary in the tropical cycle. Calculations similar to those for heat gain are performed for heat loss during the winter cycle, often on the same form. Because humidity is not involved the calculation is straight-forward.

(b) Trunk sizing calculation. Frictional losses in air ducting are similar in form to those in piping, although there is a greater variety of fittings. Pressures are an order lower and expressed normally in millibars. Velocity of air flow in the tortuous ducting expected in congested spaces is about 10 m/second. Higher speeds produce unacceptable noise except in long straight runs where speeds as high as 30 m/second permitting much smaller trunking, can be adopted. Such speeds are not uncommon in the space available in passenger liners where twin ducts, one cool and one hot, are also sometimes fitted, so permitting each passenger to adjust the temperature of the cabin.

(c) Air quantity required for the heating cycle is determined usually by the cooling cycle which is the more demanding. If not, it is found in the same way as for normal ventilation, viz.:

$$\text{Air quantity} = \frac{\text{Total sensible heat rate} \times \text{specific volume of air}}{\text{Temperature rise} \times \text{specific heat of air}}$$

If the sensible heat gain H is in Watts and the temperature rise is 3°C the air quantity Q in m³/s is given by

$$Q = \frac{H \times 0.88}{3 \times 1009} = \frac{H}{3500} \text{ approximately}$$

Finally, the purity of the air in an air conditioning system is controlled to a degree dependent on its application. Filters are fitted in most systems to trap dust, fluff and soot. Public rooms are often fitted with tobacco smoke filters.

In warships, some filters may also have to be fitted to remove radioactive and other dangerous particles from incoming fresh air. In a submarine, submerged for long periods, the purity of the air must be controlled more closely by removing carbon dioxide and hydrogen and by generating oxygen.

FUEL SYSTEMS

There are several reasons why the type of fuelling system needs to be decided early in the design. Principally, the weight and disposition of fuel have an important effect on stability and trim of the ship and the demands on space need early consideration. Being low in the ship, a large quantity of fuel has a stabilizing effect. As is explained later, special consideration must be given to stability when fuel is used up. Delivery of a satisfactory quantity offuel oil from service tank to boiler or engine, while important, is a simple matter, effected by a simple local system. An important influence on the ship design, the whole system is considerably more extensive in order to achieve some or all of the following:

(a) to store sufficient fuel to enable the ship to achieve its required range in rough weather;

(b) to accept high pressure re-fuelling without spillage or structural damage to the ship;

(c) to provide to the service tank fuel of sufficient quality;

(d) to ballast empty fuel tanks with sea water and to discharge such water overboard without polluting coastal waters;

(e) to accept fuel at positions consistent with the supply ship supply points.

Stowage of oil is usually effected in double bottoms and in deep tanks right forward and right aft in cargo ships. An adequate margin of quantity-often 20 per cent-should be provided over that needed for the planned route before refuelling in port or at sea, to allow for rough weather and emergencies.

Warships alongside oilers provide easy targets and need therefore to accept fuel at high speed and pressure. Also, it is desirable that escorts should spend a minimum of time off the screen. Bunkering may be a critical part of the turn round time for a merchant ship. When replenishing at sea through a 15 cm hose pressures may be as high as 1 MPa and rates 400 to 600 tonnef per hour. A typical system is shown in Fig. 14.14.

To avoid an excessively high beam-to-draught ratio, and yet retain adequate transverse stability, it is becoming common in warships to adopt sea water replacement of fuel. Submarines employ sea water displacement of diesel oil. Oil tankers may ballast their cargo tanks with sea water for the return journey to the oil ports to ensure seaworthiness. This use of sea water in fuel tanks creates two basic problems:

(a) fuel must be provided to the service tank in an uncontaminated state;

(b) oily ballast must not be discharged in many areas of the world unless containing less than 100 parts per million of oil. This requirement by IMO to reduce beach pollution, has been ratified by most seafaring countries and incorporated into their maritime laws. In the UK it is the Merchant Shipping (Dangerous goods and marine pollutants) Regulations.

Oil and water separate naturally for the most part and (a) is achieved by providing deep settling tanks which permit this to occur in time. The tanks are not completely emptied to avoid using any emulsified mixture at the oil/water interface, which is removed by the stripping system when the tank is

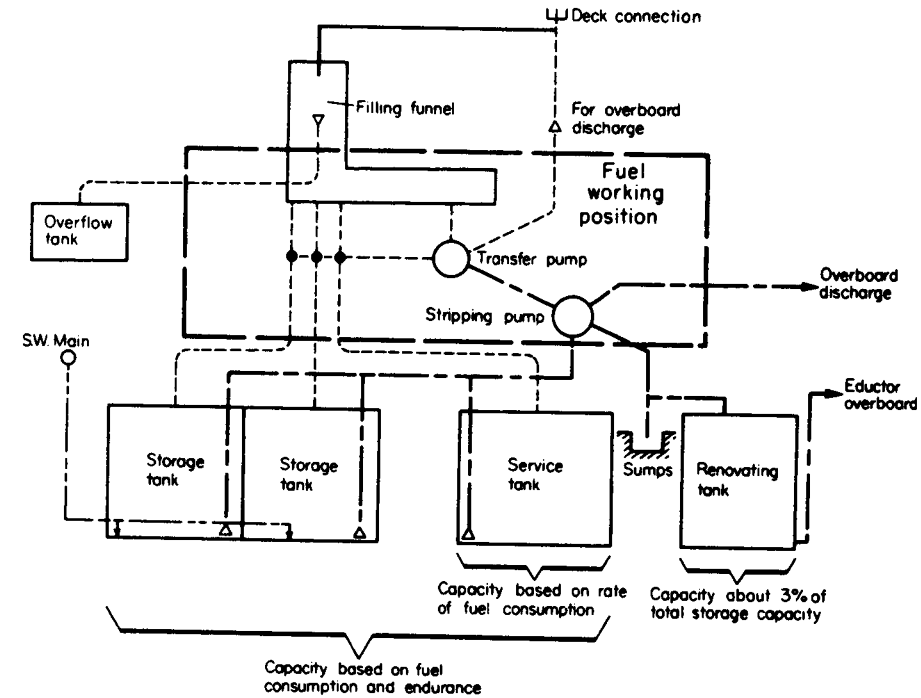


Fig. 14.14 Diagrammatic fuel system

being rested. Similar natural separation occurs in water ballasted fuel or cargo tanks where the oil collects on the surface and clings to the structure. After a period of settling, most of the ballast water can be pumped overboard and the residue pumped by the stripping system to a fuel renovating tank. There it is sprayed by a chemical additive or de-mulsifier which assists separation and the water can again be discharged. Oil centrifuges can be fitted to speed separation.

Delivery from the service tank to engines sensitive to water contamination may be achieved as shown in Fig. 14.15 incorporating centrifuge, prefilter and separator. Also shown is an emergency fuel tank which supplies fuel by.

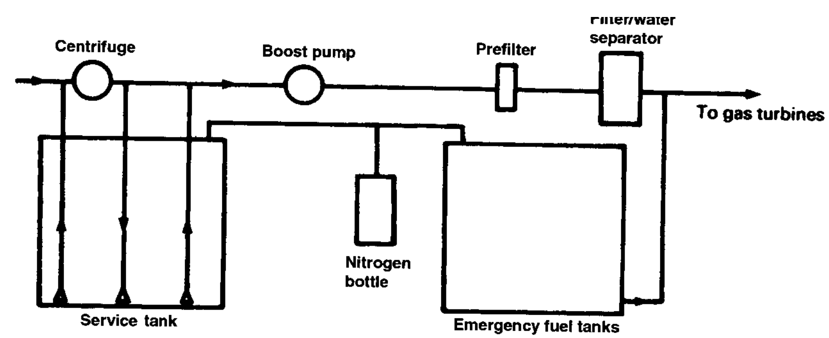


Fig. 14.15 Service and emergency fuel supplies (no valves shown)

pressurizing with nitrogen from a bottle. With such an arrangement gravity plays no part so that emergency and ready-use tanks can be kept low in the ship. Spillage during action damage is consequently less dangerous.

There are four systems associated with fuelling and ballasting, some using common piping. They are fuel filling which delivers oil to tanks fitted with air escapes and sounding tubes; fuel transfer transferring fuel from low in the storage tanks to service tanks; sullage stripping to remove residues, etc.; de-ballasting.

MARINE POLLUTION

After many years of endeavour following some disastrous spills into the seas, IMO has succeeded in producing a complex set of regulations for ships known as MARPOL 73/78. These became internationally accepted in 1983. They now deeply affect the design and construction of ships and their use to control pollution of the seas. Discharge of oily water, sewage, waste and chemical dumping are now all closely controlled and a world exchange of information via satellites about breaches of the codes of behaviour is possible.

It is essential to consult the latest regulations concerning what mayor may not be permitted. At the time of writing the broad limitations were:

(a) Raw sewage cannot be discharged at less than 12 NM (nautical miles) from the nearest land.

(b) Macerated and disinfected sewage cannot be discharged at less than 4 NM.

(c) At less than 4 NM and in harbour discharge is only permitted from approved sewage treatment plants.

(d) For discharge of sewage from holding tanks a ship must be moving at 4 knots at least and be further than 12 NM from the nearest land.

(e) No plastic may be dumped at sea.

(f) Dunnage must not be dumped less than 25 NM from the nearest land.

Consequences upon both ships and ports are considerable. Crude oil washing of the heavy oil deposits in bulk carrier oil tanks and separate exclusive water ballast tanks are now common. Steam cleaning of oil tanks is slowly being discontinued. Levels of pollution of all effluents from ships are required to be very low and, in some cases are barely attainable. At the ports, new facilities are being built up to receive different *effluents* and deal with them more easily than is possible in a ship. Thus, ships are needing to be made more capacious to incorporate segregated ballast and hold tanks for discharge in port.

Sewage can be dealt with in several ways. Heat processes can be used to produce a dry flammable product which is then burnt. Chemical treatment tends to leave a hard deposit which is difficult to remove and dispose of. Bacteria are most often used to break down the solid materials but the bacteria must be fed and special steps have to be taken if the throughput of the system falls below about a quarter of the designed capacity. The problem is further exacerbated by the big variation in loading throughout a 24 hour period. Systems utilize a vacuum collection system or rely upon gravity. Whichever system is used safeguards must be provided to prevent unpleasant, and potentially lethal, fumes penetrating to other parts of the ship. Flap valves in a vacuum sewage system have been known to stick open and allow hydruen sulphide to flow back into living spaces with fatal consequences. To **meet** demands for discharge close to land the final effluent must

(a) have a suspended solids content not more than 50 mg/I above that of the flushing water;

(b) not disco lour the surrounding water;

(c) have a coliform bacteria count not exceeding 250 per 100ml;

(d) have a five-day biochemical oxygen demand of not more than 50 mg/l;

(e) be disinfected, but with residual chlorine level not greater than 10 mg/I.

The safest way to deal with sewage is to hold it until the ship enters a port with satisfactory reception facilities. Such hold tanks are not small even if much of the liquid is first removed in settling tanks and they do need to be of adequate dimensions and in sensibly accessible places. Sewage mains in the ship, if kept under a slight suction head, do not need to rely so much on gravity and are not so sensitive to offensive leakage. Wastes from bathrooms, laundries and scuppers tend now to be kept separate from sewage because overboard discharge of washing water remains acceptable.

In a warship the average daily arisings of garbage are 0.9 kg/person/day food waste and 1.4 kg/person/day other garbage. This is dealt with by a selection of

(a) incinerators: modern equipment can burn all types of garbage, both wet and dry, including plastics and waste oil; the residual ash is sterile.

(b) pulpers: mainly food waste but can handle paper and cardboard.

(c) shredders: primarily for cans, bottles, wood packing, thick card and miscellaneous metal items up to 20 litre drums.

(d) compactors: capable of compacting all waste products into securely sealed leak-proof cardboard containers.

CATHODIC PROTECTION

Electrochemical corrosion occurs when two dissimilar metals are present in an electrolytic medium. Sea water is an efficient electrolyte. Different parts of the

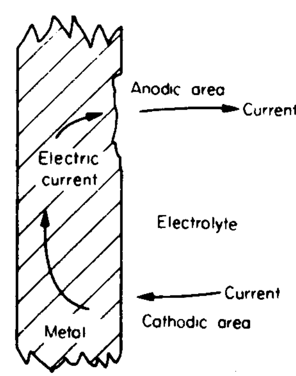


Fig. 14.16 Electrolytic corrosion

same metal made dissimilar, by treatment, or a metal and its oxide are sufficiently dissimilar to create such corrosion as shown in Fig. 14.16. An anodic area, such as iron oxide, is eaten away creating more rust, while an electric current is created, leaving the metal at the anodic area and entering it at the cathodic area, where no corrosion occurs. Painting, if perfect, increases electrical resistance and retards the process but at any imperfection in the paint, deep pitting may be caused by concentrating the electrolytic effects.

Average values of the electrical potential for a number of different metals against the same standard in sea water at 25°C are given in the electro-chemical scale of Table 14.6. Where the difference between two potentials exceeds about 0.25 volts, appreciable corrosion of the metal with the higher potential (the anode) will occur, if the junction is moist.

Table 14.6
Electro-chemical table

Material	Potential, volts
Magnesium alloy sheet	-1.58
Zinc base die casting	-1.09
Galvanized iron	-1.06
Aluminium alloy (14% Zn) casting	-0.91
Aluminium alloy (5% Mg)	-0.82
Cadmium plating	-0.78
Aluminium alloy extrusion	-0.72
Mild steel	-0.70
Cast iron, grey	-0.70
Duralumin (Al/Cu) alloy	-0.60
Chromium plating on mild steel	-0.53
Brass	-0.30
Stainless steel, austenitic	-0.25
Copper	-0.25
Gunmetal	-0.24
Aluminium bronze	-0.23
Phosphor bronze	-0.22
Millscale (Fe_3O_4)	-0.18
Monel (nickel alloy)	-0.16
Nickel plating	-0.14
Silver plating	-0.01
Platinum	+0.20
Graphite	+0.30

Another metal, higher up the electro-chemical scale, placed nearby in the electrolyte, transforms the whole of the first metal of Fig. 14.16 into a cathode if the effect is sufficient to overwhelm the local action and no corrosion occurs there (Fig. 14.17). All corrosion occurs at the new metal which is the sacrificial anode. Such protection is commonly applied to small static objects such as buoys, and individual piles and materials used for the anodes include very pure zinc, magnesium or aluminium. The basic principle is to swamp the local corrosion currents by imposing an opposing current from an external source.

A more effective system is that using an impressed current (Fig. 14.18). The potential of all areas of metal must be depressed to a value more negative than

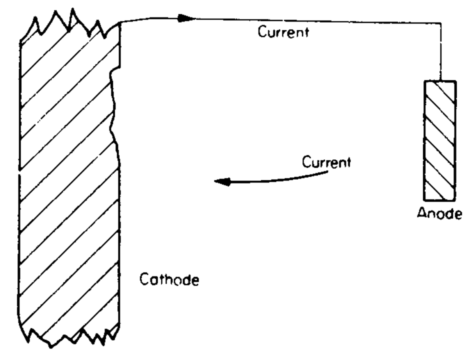


Fig. 14.17 *Sacrificial anode*

any naturally anodic area. This can be measured against a standard reference electrode in the sea water. The current densities (mA/m2) required are about 32 for painted steel, 110 for unpainted steel, 150 for non-ferrous metal and 540 for propellers. They vary with factors such as ship speed, condition of paintwork, salinity and temperature of sea water. The system can be used for ships building or laid up, as well as those in service. It can be used in large liquid cargo tanks. Automatic control units are needed to give this adjustment relative to a half cell datum fixed to the hull. A suitable permanent anode is provided by a plate of platinum-covered titanium fixed to an area of the hull covered with an epoxy resin insulant designed to spread the protective effects. Sacrificial impressed current anodes of trailing aluminium wire are not uncommon.

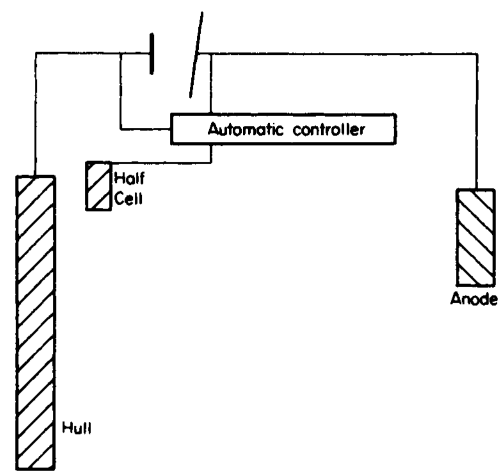


Fig. 14.18 *Impressed current cathodic protection*

There are several reasons why a decision needs to be made on whether cathodic protection is to be adopted before the design of the ship commences. Space and power demands need to be known but primarily, corrosion **margins** have to be decided. Classification societies permit a reduction of about 10 per cent in section structural modulus of ships protected by an approved process of cathodic protection. Certainly a substantial reduction in corrosion margin is justified.

As well as the external surfaces of the hull and the internal surfaces of liquid cargo tanks, cathodic protection is applicable to piping and machinery systems.

Some adverse effects of cathodic protection have been reported. Alkaline by-products of the sacrificial anodes can, over a long period, degrade timbers. Similar action can occur through any stray electrical currents. Prevention of this problem can be effected by protection of the timbers by impregnation and by judicious positioning of the minimum number of zincs.

Equipment

Layout of the weatherdeck is an important aspect of ship design and needs to be performed early. Equipment that needs to be on the weatherdeck requires some preliminary examination because interferences are inevitable. Space and geometrical demands will occur for moving equipment, arcs of fire and angles of view, free movement of personnel and for satisfactory wind flow to assure safe landing of aircraft. There will also be direct conflict for the best siting of ladderways, boats and davits, cranes, replenishment at sea gear, hatch coamings, anchors and cables, capstans, ventilation inlets and outlets, fairleads and hawser routes and, of course, aerials, armament, masts, funnels and superstructure. As always in ship design, compromise among the competing demands has to be achieved through understanding discussion with the seamen who will run the ship. The design of the bridge itself is now subject to considerable study by ergonomists as well as experienced pilots and seamen.

CARGO HANDLING

Not so long ago all cargo was loaded and unloaded by union purchase and large gangs of dockworkers manually organizing matters. Union purchase is the use of two derricks in concert to hoist cargo and transfer it simultaneously. It was an expensive and time consuming process keeping ships tied up alongside for long periods when they might have been at sea transporting their goods.

To effect greater economy in cargo handling, it was first arranged methodically in packages, in pallets or in containers of standard sizes which could be split up ashore and consigned to road or rail transportation without delaying the ship. By achieving a higher utilization of the ship and fewer handling costs, the whole transport system efficiency was improved. Container terminals had to be specially built and great gantry cranes were installed to transfer containers rapidly and efficiently from ship to shore. Container ships which ply between smaller ports not so equipped must rely upon dockside cranes or their own derricks and cranes to load and unload containers or their palletized cargo.

When trade is slack, overhead costs at container ports continue; cranes must be maintained and handling personnel paid to be on standby. On the other hand, when the terminal is full, queueing for facilities occurs. This has caused some owners to reconsider their total reliance upon container terminals and to install gantries and cranes on each of their ships. There is a general trend towards such self-unloaders for many types of cargo. Slim centreline cranes or gantries spanning the whole ship with cantilever extensions plumbing the dockside are now common on container ships.

The ultimate integration of sea and land transportation is, perhaps, the Roll-on Roll-off ferry whereby lorries, often carrying standard containers, are driven on and off ships with large garage decks which mate with ramps at their terminals. These are discussed further in Chapter 16.

Bulk carriers have also benefited from developments in rapid cargo handling. Many cargoes are suitable for movement within the hold by gravity through gates on to conveyor belts to a point where the cargo is raised by bucket elevator or rotating helices to deck level and then to a horizontal conveyor belt, usually contained within a large tube to discharge points overboard. There are many variations. Residues in holds may still have to be cleared by bulldozers or grabs but the process can be very fast.

What is clear is that the processes of handling cargo must be considered at the earliest stages of the design. It is intimately related to the type of trade, to ports of call, to company policy and to the economics of trade.

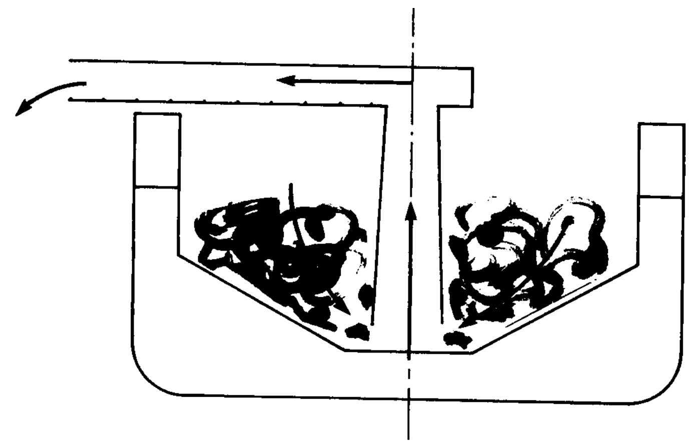


Fig. 14.19 *Overboard discharge of bulk cargo*

REPLENISHMENT OF PROVISIONS

For a passenger ship or warship carrying 2000 people, the problem of transporting twenty tonnes of victuals a day is large. Ideally, the designer will group storerooms and cold and cool rooms in a block, served by a lift or conveyor to the preparing spaces and to the weather deck, close to the point where, in port, the provisions are brought on board. Additional mechanical handling equipment may be provided for horizontal transfer of provisions.

The rig for replenishment at sea comprises, basically, a kingpost or high point on each ship between which is stretched a jack stay or highwire which is

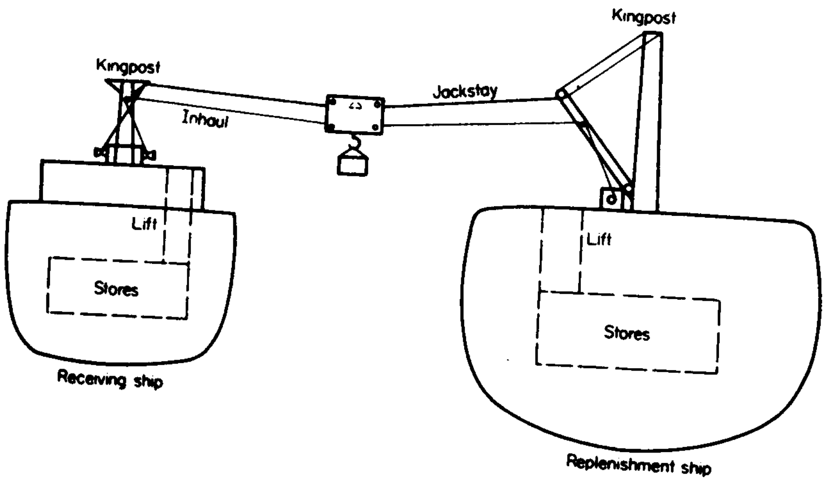


Fig. /4.20 Rig for replenishment at sea

kept at a constant tension by a self rendering winch. A traveller on pulleys is then run along the jackstay pulled either on a closed loop by one ship or by each ship in turn on an inhaul. To land the load on the deck, tension on the jackstay is slackened off. In fair weather a heavy jackstay can transfer 40 loads (each up to 2 tonnef) per hour with the ships 30 m apart. In heavy weather 25 loads with ships 45 m apart are more typical figures. Liquids are transferred by hose. Rates are limited by hose size and the filling systems on the receiving ship. Realistic rates are 450 tonnef/hour through a 5 cm hose or 150 tonnef/hour by 7.5 cm hose.

On the market today there are proprietary replenishment at sea rigs which introduce, at some expense, more automation and control. Jackstay and inhaul are connected at the kingpost to a jockey which is able to slide vertically up and down the kingpost, capturing the incoming loads and placing them with care on the deck or on to a fork-lift truck. These may permit more delicate loads, like guided weapons, to be transferred in higher sea states.

LIFE SA VING APPLIANCES

Cargo ships are required now to carry sufficient totally enclosed motor driven lifeboats for 100 per cent of the crew each side of the ship. Inflatable liferafts sufficient for all of the crew must be carried in addition. If the lifeboats are free-fall they must be fitted at the stern, sufficient for all of the crew and inflatable liferafts for all of the crew must also be carried each side of the ship. In addition, there are requirements for rescue boats and special demands on oil tankers, chemical carriers and gas ships.

Passenger ships on long or short international voyages must carry partially enclosed motor lifeboats for 50 per cent of all personnel on board each side of the ship. Inflatable liferafts for 25 per cent of personnel must be carried in lddition. Alternatively, they must carry lifeboats sufficient for 37.5 per cent and iferafts sufficient for 12.5 per cent each side. Inflatable liferafts sufficient for 25 per cent must also be carried. These figures also vary for two-compartmcnl ships on short voyages and for small ships.

An important development for some passenger ships, notably large capm:ity ferries, has been the fitting of complete marine escape systems. Similar in principle to aircraft escape, they comprise long chutes to sea level where they mate with large liferafts. They are not cheap and, if fitted, must be exercised regularly at some expense and inconvenience.

The demands of evacuation upon the architecture of the ship are substantial and should be considered as a complete system within the overall safety case for the ship. Boats and davits are but one element of an evacuation system that involves consideration of communications, alarms, drills, mustering, free passage for alarmed people, the probable environment external to the ship and within the ship at the time.

Creating a fighting ship

GENERAL

The ship weapon-system must be conceived and developed as an integrated whole. The ship must have a low susceptibility to detection by the enemy and be capable of sustaining some degree of damage whilst remaining a viable fighting unit. Even with modern detection systems a considerable degree of stealth can be conferred on a ship by reducing its radar reflectivity by avoiding flat vertical surfaces and its various signatures: acoustic, magnetic, infra-red and pressure. This makes it harder to detect and classify in the first instance, makes it more difficult for weapons to home in on it and makes its own decoys and sensors more effective.

Helicopters and other aircraft can dramatically increase the area a ship can monitor and over which it can interact with an enemy. There are other ways of extending the Command's knowledge and control of its environment. Bathythermographs can probe the depth to establish thermal layers, important in hunting submarines; sensors can be towed at a distance from the ship, e.g. variable depth sonars and towed arrays; remotely operated vehicles can hunt for mines and destroy them by setting charges near to them.

WEAPONS AND FIGHTING CAPABILITIES

Any weapon system contains four main elements: surveillance; guidance, tracking and illumination; data handling and mechanical handling (Fig. 14.21).

The surveillance system awakens the ship to a potential threat. It may be a long range radar high in the ship, a long range search sonar below the keel or information from an aircraft or other external source. Pride of place must be reserved for this system, the eyes and ears of the ship. Information on the threat is passed to the tracking device which locks on to the echo and provides range and bearing information and guidance to the weapon, the target sometimes being illuminated by electromagnetic waves or pulses to assist the process. The tracking and guidance device may be narrow beam radar or sonar, visual or television sight.

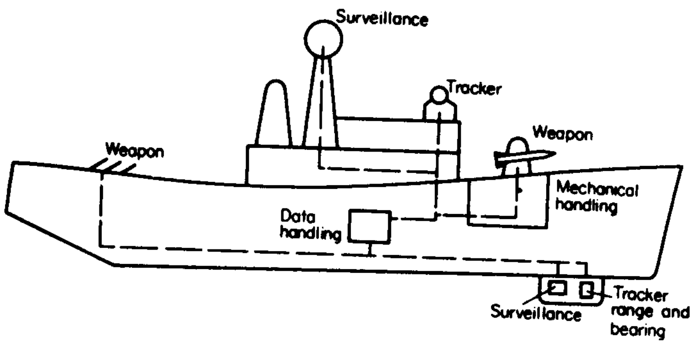


Fig. 14.21 *Typical weapon system*

The information received from sensors and other sources is processed by computer and passed to the Command and the mechanical handling system, launcher and weapon, the launcher foliowing its directions until the weapon is fired. Once fired the weapon may receive further guidance to direct it to the target. It may ride a radar beam, receive commands from the ship or sateliite or use an in-built homing system.

Within the ship the data is transmitted using a common data highway or bus which circulates information packaged electronically by a process calied multiplexing. Modern warships can look significantly different to their predecessors. Revolving radars are being replaced with static electronic scanning systems; frequency agility is used to avoid jamming; silo launchers are replacing rotating mounts.

The overall ability and effectiveness of a warship is best considered in terms of its fighting capabilities. These can include detection and classification of enemy targets; destruction of enemy submarines; preventing enemy air-launched missiles hitting own ship; and so on. These must be couched in operational terms by the Naval Staff. Thus it may be necessary to detect enemy aircraft presenting a minimum radar cross-section out to a stated range in certain ambient conditions with a given probability of success.

It may be possible to provide some capabilities in more than one way in which case one or more systems may be fitted. Thus preventing enemy air-launched missiles striking one's own ship can be achieved by destroying the missile in flight (a hard kili solution), by jamming its homing system, by seducing it with a decoy or by taking rapid avoiding action. Clearly the decisions the Command must take are complex and must be made rapidly.

A layered defence is adopted so that some sensor/weapon systems will be long range, for example surveillance radars which may operate out to ranges of several hundred miles. Others will be designed to deal with targets which evade the longer range systems. Finaliy short range systems are primarily for self defence. The combination of capabilities specified for a given design wili depend upon its intended role. A primarily anti-submarine vessel would expect to be able to detect and destroy submarines at some distance from the ship but also have a self defence capability against air attack.

INTEGRA nON OF SHIP, SENSORS AND WEAPONS

The designer, working closely with the naval staff and weapon engineers 111 list integrate the various equipments into the design so as to maximize the fighting capabilities in the undamaged state. Maximum areas of view or fire will be provided; surveillance radars wili be placed high up; missile launchers weli clear OI obstructing superstructure (arcs of fire must aliow for deviations in missile path, possible aerodynamic interference effects and relative movements due to ship motion); systems may be double headed to enable two targets to be engaged simultaneously; interference between different elements wili be minimized, for example the effects of telecommunications on radar reception, of self noise on sonar performance or of electro-magnetic radiation on missile control and firing systems.

To minimize degradation of capabilities after damage the designer wili reduce the area of the ship over which a hit can immobilize a system; duplicate important items, locate critical areas deep in the ship or provide local protection.

Whilst much of this is common sense, a methodical approach is needed. For this the designer can produce dependency diagrams showing which physical elements contribute to each capability. Then, within the context of the ship environment the effectiveness of each element and hence the overall capability can be assessed. The effect on level of capability due to certain design changes can then be studied, e.g. modifying upper deck layout to improve arcs of fire, introducing a second sensor to give all round vision, duplicating vital elements to make the system more robust. This produces trade-offs for discussion with the naval staff which wili be expressed in terms of overali ship performance in likely operational scenarios and in statistical terms based on the probabilities of the ship meeting certain combinations of enemy forces and weather conditions and with more or less friendly forces in support.

Parts of modern weapon systems are delicate and must be protected from shock and vibration (Chapter 9). Alignment of parts of a system may be critical and the natural elasticity of the ship's structure must be allowed for. If a modular approach is used then modules can be developed, tested and repaired in controlied factory conditions. Usualiy now each weapon system has its own computer rather than relying on a central ship's computer which can make the ship vulnerable to a single hit. These concepts can also ease the problems ofup-da_ weapon fits during refit or fitting weapons into ships taken up from trade.

An influence diagram iliustrating the influences governing the shipfitting of a weapon system, is shown in Fig. 14.22.

Accommodation

Demands on a warship for crew accommodation are so great that they comprise a major design feature of the ship and the size of the ship is profoundly affected by its complement. Part of the growth in size of warships since the second world war is traceable to:

(a) an increase in standards of accommodation;
(b) an increase in complements.

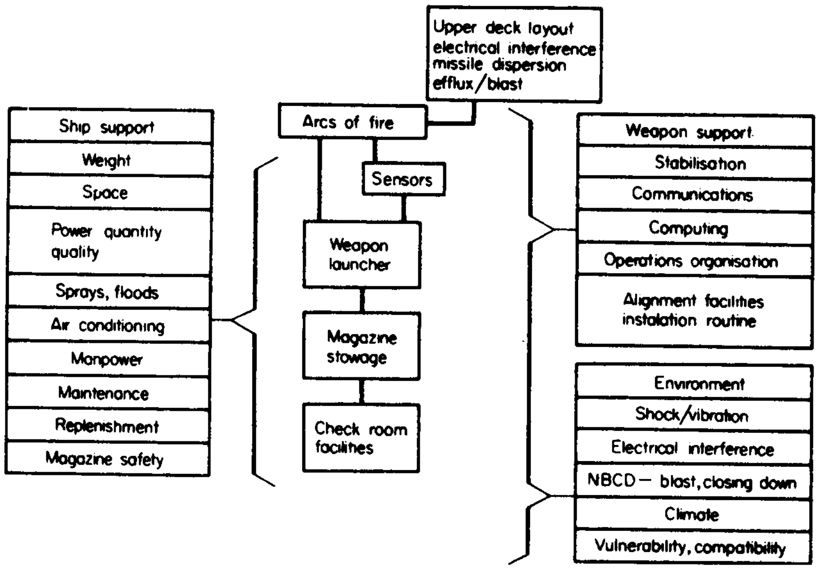


Fig. /4.22 Shipfitting influences on weapon systems

The first results from the social changes ashore, which have also resulted in women forming part of a warship's complement, while the second is due to the greater complexity of ships and a higher maintenance load. Each person requires not only space in a cabin and recreation space but also a small proportion of galley, heads, provision rooms, air conditioning, etc. All these enlarge the ship, requiring bigger machinery, more fuel and so on. Standard weight allowances in warships are fourteen people to the tonnef alone and seven people to the tonnef with their immediate effects, but the overall effect of an addition to complement is much more than this for the reasons stated.

Spaces in a warship coming under the general heading of accommodation are cabins, sea cabins, wardroom, anteroom, messdecks, galleys, serveries, vegetable preparing spaces, refrigerated spaces, aircrew refreshment bar, bakery, pantries, drying rooms, dining halls, sculleries, incinerators, bathrooms, heads, laundries, cinema, sound reproduction room, television studio, chapel, schoolroom, library, recreation spaces, canteen, bookstall, barber's shop, ice cream bar, tobacco kiosk, sick bay, dental clinics and prisons. Not all ships have all these spaces. Layout of the ship to effect a suitable juxtaposition of all of these and other spaces is a considerable task and requires a familiarity with all features of life on board ship. The crew does not like noise or smell, they do not like sleeping where they eat, they are happier if they can see daylight, they like opportunities for privacy as well as communal activities, they need to be reasonably close to adequate bathrooms and toilets, their sleep should be uninterrupted by others changing watch and their food should be well served,

Table 14.7
Initial accommodation space estimates for warships

	Officers	Warrant and Chief Petty Officers	Petty Officers	Junior Rates
Cabins	4.0 to 8.5	4.2 (WO) 2.4 (CPO)	2.1	1.6
Wardroom/anteroom	2.3 to 2.5			
Recreation area		0.8	0.8	0.55
Dining area		0.5	0.5	0.4
Bathrooms, etc.				
Showers per person	1 to 8	1 to 15	1 to 20	1 to 25
Washbasins	1 to 3	1 to 5	1 to 8	1 to 10
WCs	1 to 6	1 to 10	1 to 15	1 to 15
Urinals	1 to 18	1 to 30	1 to 45	1 to 45
Laundry	0.1 to 0.3	depending on complement		
Galley	0.1 to 0.4	depending on complement		
Canteen	0.04 to 0.07	depending on complement		

Notes:
1. All figures in square metres per person. Separate allowance to be made for access passageways, hatches, etc.
2. Senior officers have larger cabins with their own bathrooms.
3. Warrant officers, chief petty officers and petty officers are in single, double and quadruple cabins respectively.
4. Junior rates are in six berth cabins in new construction and in 18-42 person messes (multiples of 6) in older ships.

diverse and dependable. The naval architect has done well if all these are achieved without compromising the ship's fighting ability.

In the early stages of design, space must be allocated to each feature of the ship, in order to estimate the ship's size. As a first guide for warships, the allowances of Table 14.7 are useful. The figures given are nett floor areas inside stiffeners or linings. Gross figures, 20 per cent larger may suffice initially. All space allocation depends much on the shape available, and rough layouts of selected spaces should be made to confirm the values used. In arranging tip layout of compartments, dining halls must be arranged adjacent to serveries with a suitable flow of traffic for self service without cross flow or congestion; cold and dry store rooms should be readily accessible to the preparing spaces for daily supplies, lifts being provided where possible.

Because the tasks are quite different, the magnitude of the accommodation problem in a merchant ship (except passenger ships) is much smaller and it is doubtful whether the size of a cargo ship is appreciably affected by its complement. Large aircraft carriers may have complements of 3--4000, a 5000 tonnef guided missile destroyer may carry 500 people while a 200,000 tonnef dead·weight oil tanker may carry less than 30. Accommodation standards in merchant ships, as a result, are relatively much higher, minimum standards being enforced by the laws of the country of registry.

For merchant ships, crew's accommodation is generally grouped as follows:

(a) Deck and engineer officers. In single or double cabins. Bathroom with one bath or shower and one washbasin for every six persons. Separate smoke room and dining saloon;

(b) Petty officers. Cabins and washing facilities as for officers. Separate messroom. Messrooms based on 1 m² per man;

(c) Engine room hands. Separate sleeping and dining accommodation. Bathrooms as for officers. The ILC recommended minimum floor area per person in sleeping rooms as:

3.75 m^2 in ships 1000-3000 tonnef
4.25 m^2 in ships of 3000-10,000 tonnef
4.75 m^2 in ships 10,000 tonnef or over.

Where two ratings share one room the figures are reduced by 1 m^2 per person.

(d) Deck hands. As for engine room hands. Bathrooms as for officers. Crew's smoke room shared with (c).

Clothes washing facilities are required with drying and ironing facilities. The whole accommodation should be sited above the summer loadline, be provided with natural light, ventilation, artificial light and heating. Passengers are to be totally segregated from the crew. The rules are variable depending on size and type of ship and reference should be made to the appropriate legislation.

Measurement

There are two measurements of a merchant ship's earning capacity which are of fundamental importance to its design and operation. These are *deadweight* and *tonnage.* Deadweight is related to the weight of cargo and tonnage is related to the volume of cargo. *Deadmass* is now increasingly used.

The deadweight of a ship is the difference between the load displacement up to the minimum permitted freeboard and the light displacement. The lightweight comprises hull weight and machinery. Deadweight is therefore the weight of all cargo, oil bunkers, fresh and feed water, stores, crew and effects. The weight of the cargo alone is called the *cargo deadweight.* A ready reckoner of deadweight against draught for a particular ship is often supplied to the ship's master in the form of a *deadweight scale.* This shows diagrammatically for salt and fresh water, relative to the load line markings, the draught in metres and feet, the deadweight and displacement in tonnef, the TPC and MCT one cm. It provides a ready means for the master to estimate the change of draughts, or permissible load, when loading in water of density different to that of sea water. The extra permissible draught in fresh water is indicated on the load line markings.

In many ships the data is provided in tabular form or as a computer program.

A coefficient used in the early estimation of dimensions and the study of economics is the ratio of deadweight to deep displacement; this is called the

Table 14.8
Typical values of deadweight coefficient

Ship	Deadweight coefficient $= \dfrac{\text{Deadweight}}{\text{Deep } \Delta}$
50 m, Coaster	0.62
80 m, Shelter deck and one deck	0.70
100 m, Three island and two decks	0.74
150 m, Refrig. shelter deck and two decks	0.58
200 m, Container ship	0.60
250 m, Bulk carrier	0.82
300 m, Oil tanker	0.86

deadweight ratio or *deadweight coefficient* and normally refers to summer load draught. The deadweight coefficient for the type ship will be a guide for the new design; typical values are given in Table 14.8 and in Chapter 15, although, since there is considerable variation in apparently similar ships, these should be treated with caution.

The volume of a ship is expressed in tons of 100 ft³ (2.83 m^3) and is referred to as its *tonnage.* On its tonnage are based the charges for berthing the ship, docking, passage through canals and locks, and for many other facilities. It is often used as a coarse measure of a ship's size and is also confused, by the layman, with displacement.

Records of measurement of a ship's size in this manner can be traced back in the United Kingdom to the thirteenth century for the carriage of wine. A standard size of barrel, called a 'tun', was decreed in the fifteenth century for this purpose and taxes and harbour dues were based upon a ship's 'tunnage'. Over the centuries, various rules for assessing this tonnage, as it became, were devised, the most influential being the Moorson system of 1853.

Tonnage regulations have not always led to safe design. In 1773, the formula on which tonnage was assessed was $0.515(L - 0.6B)B^2$, where L is the length and B the breadth of the ship. Because this did not include draught, owners required small beam and large draught to reduce tonnage; this, howevtp created poor stability and many ships were lost as a result. More recently, in order to exempt the 'tween decks and still permit them to carry cargo, 'the shelter deck was 'opened' by the provision of a tonnage hatch in the upper deck and openings in main transverse bulkheads resulting in minimal safety standards; the regulations were, in fact, formed to permit this exemption and were responsible for a huge number of shelter deck ships constructed in this artificial manner which, nevertheless, became very efficient and safe cargo ships later.

By the middle of the twentieth century, many different methods of assessment of tonnage existed; among the important tonnage regulations were the International, British, United States, Suez and Panama. Efforts by the International Maritime Organization (IMO) were directed in the 1960s towards producing an internationally agreed system of tonnage measurement, preferably by the application of simple formulae and this proved successful in 1969 when an international conference adopted the formulae. Assessment of Suez and Panama tonnages were not changed at that time however.

There are two tonnages of primary interest no matter which authority is measuring or registering the ship. These are *gross tonnage* and *register tonnage (net).* The regulations governing their measurement are complicated but are summarized for the British Tonnage Regulations below. Other regulations differ only in detail.

The *tonnage deck* is the upper deck in vessels having one deck and the second deck in all other cases. Spaces above are 'tween decks and superstructures.

The *underdeck tonnage* is the total volume in tons of 2.83 m^3 of the ship below the tonnage deck to the inside of frames, underside of deck plating and above the inner bottom. This is obtained by detailed calculations, in a manner somewhat similar to that for displacement calculations and varies in detail for individual regulations; tonnage measured according to the tonnage regulations may, therefore, be slightly different from the actual volume of the spaces below the tonnage deck calculated for, say, cargo capacity.

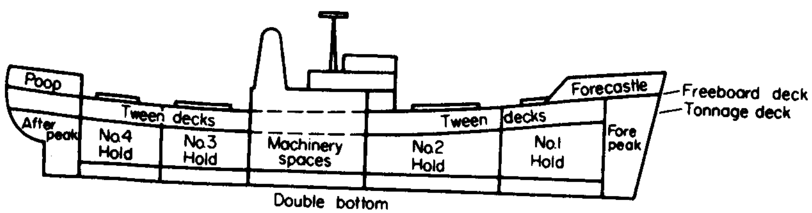


Fig. 14.23 *General cargo ship*

The *overdeck tonnage* is the volume to the inside of frames and deck plating of the 'tween decks, poop, bridge, forecastle, deckhouses and erections above the tonnage deck less the exempted spaces. Spaces *exempted* include dry cargo space (unless in a break in the deck) and certain closed-in spaces associated with machinery, safety equipment, navigation, galleys, washrooms, water ballast and workshops. *The gross tonnage* is the sum of the underdeck and overdeck tonnages plus volumes of hatchways (less one-half per cent of gross tonnage computed without hatchways), plus light and air spaces which are included at the owner's request in the measurement of the machinery spaces.

The register tonnage is the gross tonnage less the following deductions:

(i) Master's accommodation, crew's accommodation and provision storerooms (limited to a maximum of 15 per cent of the total tonnage of Master's and crew's accommodation) but not fresh water.

(ii) Spaces below deck allocated exclusively to steering, navigation, safety equipment and sails (for ships propelled exclusively by sail, up to a maximum of 2! per cent of gross tonnage).

(iii) Spaces below deck allocated, with certain provisos, to workshops, pumps, donkey engine and boiler.

(iv) Spaces below deck used exclusively for water ballast up to a maximum nf' 19 per cent of the gross tonnage, including exempted spaces such as the double bottom and capacity below line of floors containing water ballast. oil fuel, etc.

(v) Spaces below the upper deck occupied by propelling machinery. In ships propelled by screws, if the volume of this space is between 13 and 20 per cent of the gross tonnage, the deduction is 32 per cent of the gross tonnage; if the space occupied is less than 13 per cent of the gross tonnage, the deduction is in direct proportion to 32 per cent; if the volume of the propelling machinery is in excess of 20 per cent of the gross tonnage, the propelling power deduction is 1 times the tonnage of the propelling space.

Happily, this complicated assessment has now disappeared for new ships. The new formula tonnage regulations came into force in 1982 and are applicable to all new and converted ships and, at the owner's request, to existing ships. The formulae now applied to ascertain the gross and net tonnages are as follows:

Gross tonnage, $\mathrm{GT} = K_1 V$

Net tonnage, $\mathrm{NT} = K_2 V_c \left(\dfrac{4d}{3D}\right)^2 + K_3\left(N_1 + \dfrac{N_2}{10}\right)$

where $K_1 = 0.2 + 0.02 \log_{10} V$

$V =$ total volume of all enclosed spaces, m^3

$K_2 = 0.2 + 0.02 \log_{10} V_c$

$V_c =$ total volume of all cargo spaces, m^3

$d =$ moulded draught amidships, m

$D =$ moulded depth amidships, m

$K_3 = 1.25\left(1 + \dfrac{\mathrm{GT}}{10{,}000}\right)$

$N_1 =$ number of passengers (in cabins with not more than 8 berths)

$N_2 =$ number of passengers not included in N_1

note that: $\left(\dfrac{4d}{3D}\right)^2$ must not be taken as greater than unity

$K_2 V_c\left(\dfrac{4d}{3D}\right)^2$ must not be taken as greater than 0.25 GT

NT must not be taken as less than 0.30 GT

if $N_1 + N_2$ is less than 13, N_1 and N_2 shall be taken to be zero

d is the assigned summer loadline draught or, for passenger ships, the deepest subdivision loadline draught

V and V_c are calculated for metal ships to the inside of shell plating and are to include appendages

While these new regulations are considerably simpler than the old ones, they still require a great deal of precise definition of geometry and phrases such as 'enclosed spaces' and 'excluded spaces'. National regulations should be consulted for the definitions.

The measurements of ships described above are statutory, i.e. required by the law of the country of registry. Two measurements not required by law but of use, where pertinent, to the owner and designer are *grain capacity* and *bale capacity*. Grain capacity is the cargo volume in cubic metres out to the bottom and deck plating, excluding space fiIled by frames and other structure. Bale capacity is the cargo volume to the inside of the frames or sparrings on frames and beams.

Problems

I. Discuss the bases on which machinery may be chosen for a merchant ship. How do the arguments differ for a warship? What types of main propulsion machinery are available? Discuss the main properties of each.

2. Write a short description of the problems of

(a) nuclear main propulsion;
(h) all gas turbine propulsion;
(c) diesel-electric propulsion.

3. What factors affect the size and type of electrical generation in a warship? Describe briefly how electrical distribution may be achieved in merchant ships and warships.
4. Compare the properties of tree systems and ring main systems for fluid distribution in a ship. How are the calculations performed for each?
5. What does air conditioning seek to achieve that normal ventilation does not? Describe the series of heat exchanges in a ship's air conditioning system and how each is efficiently secured.
6. How does the choice of fuel system affect a ship design? Describe a system for rapid fueIling.
7. How do the laws on oil and sewage poIlution affect the design of a ship?
8. Describe the principles of cathodic protection and how they are put into effect.
9. What is a container ship? Discuss the various ways of mechanical handling of cargo into and out of general cargo carriers.
10. Describe the various elements of a ship-weapon system and how they interact. What problems face the naval architect in siting these elements and what effects has the ship on the design of the weapon system?
11. A pump is required to deliver 400 tonnes of fresh water an hour at a pressure of 150 kN $1m^2$ along 50 m of straight horizontal steel piping containing two diaphragm valves and a strainer. Estimate the pump delivery pressure required for *(a)* l5cm diam. and *(b)* 20cm diam. piping. Ambient temperature is 20DC.
12. A lubricating pump is sited in one side of a square network of 12 mm bore smooth piping circulating 0.25 m^3 of oil per minute in closed circuit. Each side of the square is 6 m long and there are two 45 degree oblique valves

and two strainers in the complete circuit. If the loss in the equipment hcinil supplied is negligible, calculate the pressure differential at the pump and the corresponding power required. The kinematic viscosity and specific gravity of the oil are respectively 5.1 x 10^{-4} $m^2 Is$ and 0.90. What would be the figures if the piping were increased to 36 mm?

13. The following table represents an open ended main salt water service in a merchant ship. If a delivery at a pressure of 0.55 MPa is required at the remote end of the system and there is positive pressure of 0.10 MPa at the drowned pump suction, estimate the performance required of the pump. If the overall efficiency of the pump is 0.72, what steady power is required of the electrical supply to the pump motor?

Leg	Length (m)	Bore (mm)	Fittings	Delivery (litres/min)
PA	15	102	I strainer 3 easy bends I globe valve	182 at A
AB	18	76	I 900 angle valve 5 easy bends	136 at B
BC	12	51	2 90° angle valves 4 easy bends	182 at C
CD	49	51	I globe valve 7 easy bends I plug cock	546 at D

14. A package protecting a guided weapon is 4.88 m long and 1.22 x 1.22 m in section and is constructed of 1.6 mm thick aluminium. It is taken from an air conditioned magazine where the dry bulb temperature is 29 DC to the upper deck where it is 43 DC in the shade. What is the rate of heat gain through the box? What would it be with 19mm glass fibre all round?
15. The refrigerated hold of a cargo ship is 15m x 15m in plan and 8m high above the tank top which is 1 m above the keel. The draught of the ship is 5 m in the Red Sea. On the after side of the hold there is a machinery sp~ whose temperature is 50 DC and on the forward side a hold at 30 DC. ea temperature is 25 DC, the deck head is at 35 DC and the sides 30 DC on the cold side and 45 DC on the sunny side.

AIl surfaces are lined with 1 cm of plywood and 15cm of cork slab. AIl steelwork is 1 cm thick.

Calculate the capacity required for refrigeration machinery to maintain a temperature of -10 DC in the hold.

16. The package of question 14 is loaded on deck off Singapore in an ambient of 31/25.5 DC before being sealed. How much can the package be cooled before condensation occurs inside? If taken into a magazine at 20 DC, how much moisture will coIlect in the package? The missile occupies 20 per cent of the space.
17. Recirculation air is required to leave a room at 26.7/20 DC and be mixed with an equal quantity of fresh air at 32/29 DC before being passed to a coil

having a slope of 0.42. The sensible heat ratio of the room is 0.65. If the air leaves the cooler at 90 per cent relative humidity, estimate how much heat per kg of air must be supplied by the after warmer.

18. Heat gain calculations for a messdeck for sixty men show the total heat to be 44,000 Watts of which 40 per cent is latent. Fresh air is drawn in at the rate of $0.005\,m^3/sec$ per man and is at 34/31 °C. Conditions in the room should not exceed 25 ET. Avoiding the need for after warm, construct a suitable psychrometric cycle, stating the performance required of the coil and the air quantity needed. Air should leave the coil at 90 per cent relative humidity.

19. A frigate design is to have a complement of fourteen officers, fifty chief petty officers, fifty-four petty officers and 206 junior ratings. Make a first estimate of the space which would be needed for living spaces, toilet facilities, laundry, galley and sick bay.

20. A warship has a length of 110m and a beam of 13m. The only decks suitable for accommodation are No.2 deck and the forward half of No.3 deck, each of which is expected to have a waterplane coefficient of 0.70. Estimate what proportion of these spaces should be given over to the accommodation of the previous question, assuming that the laundry is sited on No.4 deck and that sewage is discharged directly overboard. What would be this proportion if a sewage system enabled all heads and bathrooms to be sited on No.4 deck?

21. A cargo ship has a moulded depth and assigned freeboard of 20 m and 4 m respectively. Its gross volume is $10^5\,m^3$ and cargo volume is 78 per cent of this.

Calculate the formulae gross and net tonnages. What would the net tonnage become *(a)* if freeboard could be reduced to 3 m or *(b)* if 15 cabin passengers were carried?

15 Ship design

Design is a creative iterative process serving a bounded objective.

This definition of design brings out the four essential features of engineering design which serve well as the divisions of this chapter. Objective there must be, even if like Michaelangelo's it is simply 'To please God'. Naval architects require the discipline of a well-defined objective, if their creation is to answer properly the owner's need. Moreover, it has to be bounded; that is to say the limits to which the designer may go need stating. It has become common to address a system of which the ship is but a part; a science known these days as systems engineering. Having defined what it is the designer is to address and the limits within which to work, the creative activity can start. This is normally a circular process, a first shot, corrected and re-created often many times until it satisfies the objective. It is iterative, as will later be clear.

Students must not be dismayed that they are yet denied the chance to apply their hard-won skills to this satisfying aspect of their profession. Such skills must be applied in a constructive and disciplined manner if they are not to be wasted. Students will find that a disciplined framework will not only enrich their natural creative ability but provide an opportunity for the computer to assist them. Data bases, full of successful history can provide them with a firm start to the process of converging upon the best solutions.

It is first necessary to be clear about the role of the naval architect throughout the design process, because it changes with time. The earliest stages are generally a debate with the owner, proposing various ways in which the owner's wishes could be fulfilled, matching the operations envisaged to the investment that would be necessary to perform them. In the case of maritime trading t~ would be suggesting many different possible transportation systems and asse'rs~ ing their profitability and chance of success. In the case of military operations it would consist of proposing many different ways of achieving offensive or defensive operations and the cost and effectiveness of each solution. These early studies comprise concept design or conceptual design, sometimes called pre-feasibility. They consist of a lively debate among all those who have a contribution to make, designers, operators, economists and many others all directing their creative thoughts towards an objective declared by the owner who may be a trader or a military commander representing a government. The result of this debate is in outline a few promising ideas which lead into a pha!ie of development usually known as feasibility study.

This second phase of the work is directed more clearly at the engineering and the management. Its aim is to identify the whole system and to quantify its profitability, its material elements and the risks attendant upon its development.

It attempts to establish the viability of the best of the promising concepts. At the completion of this phase the owner chooses the way in which to proceed. This is an important decision because it commits significant funds in the succeeding phase.

That phase is the full design worked out in every necessary detail so that material may be ordered and construction may begin. The design team is not necessarily the same as the one which completed the first two phases and the nature of the work has changed significantly. It is now directed towards the definition of the ship-and other parts of the transportation system-for contract and production. Indeed at the latter end of the full design phase the means of supporting the ship during its life are produced by the designers.

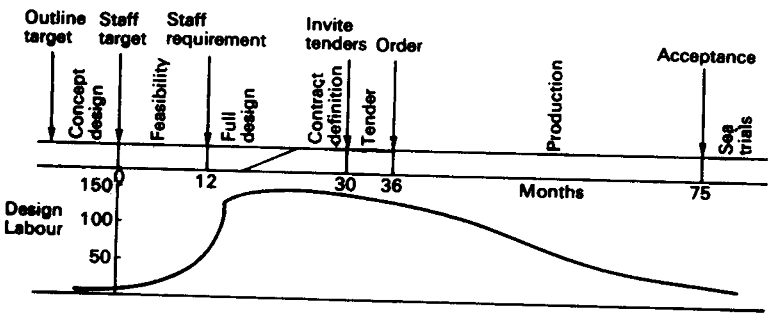


Fig. 15.1 *Phases of warship design*

Phases of design of a ship are shown in Fig. 15.1. While they may appear under different names (in the USA, concept and feasibility are reversed) their character is the same throughout the world both for merchant ships and warships. The figure also shows the times likely for each phase of a major warship design (excluding the time taken by government approval to proceed) and the design staff required for a frigate. For a destroyer the effort required to define a building contract properly now amounts to about 50,000 man-days compared with 5000 in 1960, some of this dramatic increase being due to complexity.

Objectives

It is not sufficient to say that a ship must be designed to meet the requirements of the owner until the earlier phases of design have defined what they should best be. The fundamental aim of an owner or potential owner of a merchant ship is, of course, to make a profit on the investment. For example, a tramp steamer may be ordered to ply for trade along a chosen route, picking up and delivering goods that other people want moved. It may be necessary to borrow money to buy the ship, pay taxes, pay the crew, meet bunkering costs and to lrrange the flow of cash to maximize profit. There will be constraints of cegulation, social pressures and competition affecting the choices so that there ""ill be a limit to profit which a budget will reveal. In more complex cases, nvestors will wish to consider the total distribution of goods in a fleet of ships, the means of handling, the choice of taxes and regulations to which they wi"h tu be subject, the pattern of trade and their cash flows. The transportation system to be designed in that case needs to address an objective of optimum profitability in the predicted trading conditions and environment over many years. Characteristics of the ship and all other parts of the transportation system emerge from the economics of the venture which now requires deeper study.

ECONOMICS

As a first step a potential owner decides on the sort of trading in which to indulge. From market research comes an estimate of the amount of goods that would have to be transported annually on the chosen routes and how quickly they would have to be delivered. This annual transport capacity might be met by a few large ships or many small ships and it is necessary to carry out various economic examinations to decide not only the best size of ship and its speed but the profitability over a period of time. The most important element of this calculation is the freight rate, which is the rate payable in the free markets of the world for the transport of units of specific goods. It is the subject of some competition, it fluctuates in accordance with market pressures and rates are published regularly. Sometimes such rates are determined by conferences; although the Conference System does smack of price fixing, it does give a degree of stability to trading.

Freight rate determines the flow of cash into a company. There may also be inward cash flow through Government subsidies and loans. Outward cash flow is caused by:

(a) down payment and instalments on building costs;
(b) loan repayments and interest;
(c) running costs for bunkering, crew, port charges, etc.;
(d) maintenance and repair;
(e) profit;
(I) corporation and other taxes.

Note that depreciation is not a cash flow although it may have to be calculated for tax purposes.

Now it is necessary to compare on the same basis the profitability of the various possible schemes of trading. This basis is the compound interest that would be earned by an investment P_0 at an interest rate r after a period of years i:

$$P_i = P_0(1 + r)^i$$

Inverting this, the present sum P_0 which would produce P_i in i years' time is

$$P_0 = P_i(1 + r)^{-i}$$

Over a period of n years, the annual cash flow A_i in each year i has a net present value

$$\mathrm{NPV} = \sum_{i=0}^{i=n} A_i(1 + r)^{-i}$$

This is the basis of comparison; the higher the positive net present value, the better the investment. Arguments concerning the correct discount rate rare difficult but are based upon either the assessment of alternative opportunities when it is called the opportunity cost rate or upon a personal preference for speedy profit when it is known as the time preference rate.

There are several important variations upon net present value preferred by some economists. Yield or Required Rate of Return (RRR) is that discount rate r which gives a zero NPV. Required Freight Rate (RFR) or Shadow Price is the minimum cargo rate which the shipowner has to charge the customer just to break even.

EXAMPLE 1. A ship is estimated to have a capital cost of £1M. A useful life of 20 years is predicted with a scrap value of £50,000. It is expected that the ship will earn £0.5M for each full year's operations but that due to special survey requirements this will be reduced by £60,000 and £100,000 in the 12th and 16th years. The running costs in each year, after allowance for tax, are estimated to be as in column 3 of the table below. Calculate the net present value assuming a discount rate of 7 per cent.

Solution

Year (i)	Cash flows after tax (+)ve	(−)ve	Nett = A_i	Discount factor $(1+0.07)^{-i}$	Discounted cash flows $A_i(1+0.07)^{-i}$
1	500,000	300,000	200,000	0.93458	187,000
2	500,000	50,000	450,000	0.87344	393,000
3	500,000	305,000	195,000	0.81630	159,000
4	500,000	310,000	190,000	0.76290	145,000
5	500,000	320,000	180,000	0.71299	128,000
6	500,000	315,000	185,000	0.66634	123,000
7	500,000	320,000	180,000	0.62275	112,000
8	500,000	325,000	175,000	0.58201	102,000
9	500,000	330,000	170,000	0.54393	92,000
10	500,000	320,000	180,000	0.50835	92,000
11	500,000	330,000	170,000	0.47509	81,000
12	440,000	335,000	105,000	0.44401	47,000
13	500,000	315,000	185,000	0.41496	77,000
14	500,000	330,000	170,000	0.38782	66,000
15	500,000	335,000	165,000	0.36245	60,000
16	400,000	350,000	50,000	0.33874	17,000
17	500,000	320,000	180,000	0.31657	57,000
18	500,000	345,000	155,000	0.29586	46,000
19	500,000	360,000	140,000	0.27651	39,000
20	550,000	370,000	130,000	0.25842	34,000
				Total	£2,057,000
				Less initial cost	£1,000,000
				NPV	£1,057,000

Note: Variations in negative cash flows reflect costs of surveys, varying earnings and allowances on capital cost. This latter leads to the very low figure for year 2, it being assumed that tax allowances suffer an effective delay of 1 year.

While this has demonstrated the basic elements of economic assessment. there is rather more to it. At the end of these economic exercises the owner will know the cargo carrying capacity required of the ship and probably the speed. This is the start required by the naval architect and marine engineer in their design activities. It will also indicate characteristics of other parts of the system such as the cargo handling and support facilities. They constitute subsidiary elements of the overall objective to which we shall return presently in discussing boundaries.

It is clear that the effective assessment of the economic aspects of ship design depend vitally upon a good prediction of procurement costs. Technical cost estimating draws upon the history of similar activities and procurement of equipments in the elements which have been found conducive to extrapolation into the future. Labour, materials and overheads are the three basic elements, subdivided into shipyard and manufacturing industry. Shipyard costs are subdivided into steelwork, outfitting, pipework, cabling and many hundreds of other elements, each of which is found to be governed by different variables, e.g. steelwork costs by weight, main cabling by power, paint by $L(B+D)$. These algorithms by which technical cost estimating is performed are usually kept covert but occasionally an interesting paper on this subject appears.

COST EFFECTIVENESS

The basic objective of a warship designer is to provide an advantage to a government in military action against a potential enemy. There are three elements to the value or effectiveness of a warship--or other military artefacts:

(a) capability;
(b) availability;
(c) military worth.

Capability is a measure of the offensive ability of the ship. There are very many parts to such an ability, e.g. speed, detection, range of sensors, accuracy of delivery of a missile, crew efficiency, signature suppression, reaction time to a threat. Moreover, the measures will vary with the environment in which ~ ship is working, e.g. the level of electromagnetic countermeasures, the deployment of an enemy, the climate and geography, all put together into various scenarios of operation. The assessment is the task of the operational analyst, who attempts to produce, for each scenario and for each postulated military mission, the probability that the ship will succeed. The measure of capability can therefore be expressed as a matrix of probabilities based upon many different assumptions. When the moment for action arrives, the necessary devices must of course be ready to use. This second element to the effectiveness of a warship, availability, can also be quantified in terms of probability as is discussed in the next section. Availability depends on the intrinsic likelihood that a device will work when called upon to do so, called the reliability; and the likelihood that, should it break down, it can be repaired in an acceptable timescale, called maintainability. The measures again depend upon the mission and the environment, so that availability can be constructed as a matrix of probabilities.

The third element, military worth, is an assessment of the military advantage over a potential enemy which the possession of the warship confers. Not only does this depend upon the postulated scenarios and the specific missions but upon the intelligence of an enemy's future abilities. While theoretically also a matrix, quantification is exceedingly difficult and often a matter of judgement.

Thus system effectiveness is a conjunction of three matrices, often very large and complicated.

$$\text{S.E.} = CAW$$

The primary constraint upon maximizing effectiveness is cost. A useful measure to which warship designers may apply optimization techniques is therefore

$$\text{Cost effectiveness} = \frac{\text{S.E.}}{\text{Cost}} = \frac{\mathbf{C\,A\,W}}{F}$$

In short this represents an expression of value for money. This must not be regarded as a simple fraction even though it might, in exceedingly simple cases, be possible to quantify. It is nevertheless a useful discipline to remind designers of limitations to their desire to increase effectiveness. Incidentally, cost should not be discounted unless value or military worth is also discounted because the value of an artefact which is not available for use because there is no money to repair it is zero. Value, or military worth, is also likely to degrade with time as a potential enemy's technology advances.

This measure is applicable also to mercantile operations even though availability is often very high. In that case, capability is a measure of transport capacity while military worth is replaced by utility. Utility is then a measure of the correctness of the predicted assessment of market forces or even the gamble that the owner is prepared to accept. Only later will it be known whether decisions had been correct! Utility theory is an important aspect of the study of economics.

It is clear that each element of these objectives may be traded off against another. Endurance of a warship, for example, may be increased at the expense of weapons payload; reliability of a weapon system may be increased by redundant equipments but at a higher cost; the chance of encountering a defeating situation may be low enough to accept so that a weapon system may be omitted to the benefit of other aspects of the ship; increased vulnerability may be acceptable to reduce initial costs. These trade-offs or compromises are many and complex and occupy a great deal of effort in the early design stages. The requirements for a ship consequently have to evolve until the process of design development of the ship itself is worthwhile.

Much the same occurs with merchant ships where the owner's requirements evolve steadily with the trade-off exercises. There the trade-offs would be related to such issues as choice of flag, variability of terminal facilities, bunkering positions, insurance rates.

Boundaries

The bald statements of objectives so far considered are inadequate. A designer does not have a free hand to change the world and must be constrained within boundaries that have to be defined by the owner or the government at the outset. Some of the boundaries and constraints will be quickly settled, others will depend on profitability. We will consider them very briefly under three headings:

economic, ethical and social
geographical, organizational and industrial
time and system

ECONOMIC, ETHICAL AND SOCIAL BOUNDARIES

Boundaries of the economic system as discussed earlier have a profound effect upon the ship and its supporting system. Flags of convenience and an imposed condition or preference for the carriage of goods only by national flag ships are important considerations which affect tax, construction standards, subsidies and loans. The wisdom of easily obtained credit by linking a shipbuilding industry to a merchant bank has been observed by several nations. Easy credit, i.e. low interest rates and long repayment periods, are often made available for political or social reasons by governments especially in times of recession to keep their industries alive. Lower insurance premiums are sometimes now available for ships designed to a higher degree of safety.

Of course, designers' activities must be bounded by the law, maritime law, health and safety law, consumer protection law and civil damages law. They must adhere to professional standards of conduct prescribed by their peers and in default may expect retribution. Even within the law, however, there remain choices that will be determined by ethical standards for which conduct may be judged by society at large.

The economic boundaries available will be much determined by the client and relationships with the finance houses. Registration in some flags of convenience nations might attract low fees and minimal corporation tax but engender standards which place at risk crew, innocent third parties and the environment through pollution, inadequate surveyor corner cutting. Poor quality classification societies may appear financially attractive. The lette~ the law might be served rather than its spirit so encouraging rule-cheating. These are ethical issues which deserve deep consideration by owners, insurers and ship designers at all levels. Naval architects will be wise to record carefully the discussions that they have with clients or superiors in their employment and the standards of safety that they have agreed to observe. Enquiry into a subsequent accident may absolve nobody who has made a contribution to the cause and employers cannot indemnify their employees against criminal charges.

An owner's attitude to social questions, conditions on board, care of family. conditions of employment and many others demonstrate company image and affect crew efficiency in both mercantile and military vessels. They must be chosen with deliberation.

GEOGRAPHICAL, ORGANIZATIONAL AND INDUSTRIAL BOUNDARIES

Not all ships are designed for a specific route. If they are, then the constraints upon dimensions imposed by geographical boundaries will be clear. Length, beam and draught constraints are imposed by canal passage and upper works overhang problems apply particularly to the Panama Canal for such ships as aircraft carriers. Bars at harbour entrances, depths of water at berths and in confined waterways, heights of quays, cranes and travellers all may affect the limits to dimensions. RoRo and container ships are especially affected by port facilities while warship length may be constrained by base port docks.

Organizational and industrial boundaries that will require definition may follow from company policy on its links with other companies and whether it intends to charter, whether it hires crew as it requires or keeps a nucleus in-house. Training and recruitment may be an important factor affecting the design of ships and their supporting organization and may persuade owners to standardize equipment and machinery. Bunkering places and methods, victualling, financing, repairing, agencies and communication policies should all be known to the designer. The most important boundaries however are those concerned with the transportation system and with time.

TIME AND SYSTEM BOUNDARIES

A ship designed to a minimum procurement cost may not appeal to many potential owners. Designed to minimum ownership costs it may be a very different ship. It is thus important to know the period of time over which the profitability of ownership is to be assessed.

Through-life costs over 25 years of a frigate and annual costs of a RoRo cargo liner are shown in pie chart form in Fig. 15.2. It will be seen that a frigate costing £300M to build may cost another £500M to keep for 25 years. This may encourage a designer to invest, for example, in a labour-saving device in order to save one member of the crew for 25 years which will allow the ship to be a shade smaller, saving fuel and onboard facilities. These life cycle cost trade-offs are important once the intended life of the ship is known. For example,

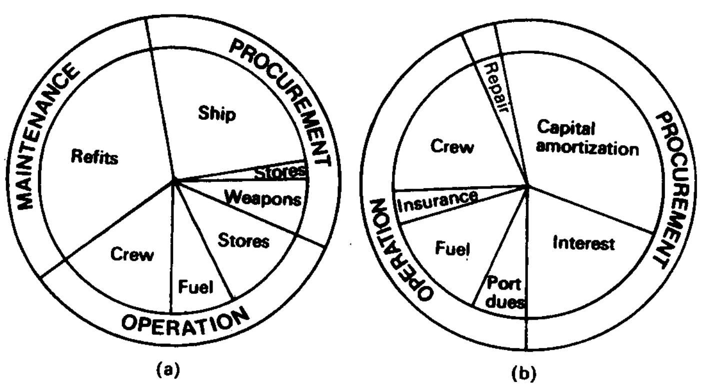


Fig. 15.2 (a) Frigate life costs, (b) RoRo annual costs

adoption of non-corrosive piping systems, fastidious paint preparation. cuth· odic protection, etc., may save many times the initial investment on maintenance costs. There is no point in thus economizing, however, if the cost of maintenancc do not fall upon the buyers of the ship because they intend to sell early. A designer must, therefore, know the boundaries of the owner's intentions and, if relevant, the character of the whole of the support system.

Ownership costs associated with warships exceed the annual expenditure on new construction many-fold. Support of the Fleet is organized on a worldwide basis to provide fuel, food, clothing, stores, ammunition, spares, repair and regular maintenance. There are standardization policies, standards of fuel quality, problems of transporting ammunition and missiles, availability of tools and facilities, coding and documentation and many other factors that must be known to designers if they are to devise a ship which is to be compatible with the existing organization.

Merchant ships face similar but rather less complicated problems. One of their most important interfaces is with the port facilities and it is this interface which has caused a revolution in merchant ship design in the last few years.

Table 15.1
Annual cost of cargo liners UK-Far East

	Conventional	Container	RoRo
Crew	4	2	5
Insurance	4	4	5
Fuel	40	50	43
Port charges	2	4	4
Cargo handling	20	7	6
Capital (ship)	30	25	30
Capital (units)	0	8	7
	100%	100%	100%

Table 15.1 compares the running costs of a conventional cargo liner on a long voyage with similar capacity container and RoRo ships. Among the striking features of this table is the dramatic reduction in cargo handling charges when the cargo is packaged into units. This is an excellent example~ extending the boundaries of economic interest beyond the ship, embracing e port handling facilities and, even, the road transport system. It is for this reason that we have been referring to transportation system design instead of simply to ship design. The economy of such systems is significantly different and the ship itself is changed as a result. Craneage on board may not be necessary at all and such is the automated container handling at the container depots that the crew may be minimized and port turn-round time halved.

Creativity

We have now to create a design which will satisfy the owner. Earlier chapters largely dealt with what might be called the *attributes* of a shipo-its stability, strength, seaworthiness and so on. Now it is necessary to consider the ship a whole, possessing these attributes to an agreed standard and able to meet 1 needs of its owner. As far as that owner is concerned the ship must be capa1 of doing various things. That is to say it must possess a number of *capabilit* within agreed operational scenarios. Contributing to these capabilities will b< range of systems and sub-systems, some devoted to one capability and oth(contributing to several. Yet others will support most or all of the capabilitie To illustrate this:

- In a single-shafted ship the shaft supports the ship's capability to move. In multi-shaft ship it will also support its ability to manoeuvre.
- A radar system will support an ability to navigate safely. In warships it w also support a number of fighting capabilities such as detecting, trackir and destroying enemy aircraft or missiles.
- An air conditioning system will be necessary to support most of a ship capabilities.

One way, then, of regarding a ship for design purposes is as a series (systems. In this context the term system is given a broad meaning in order tha all elements are contained in one system or another. Thus the main hull of th ship is a system supporting the ability of the ship to float. Most systems wii include equipments, piping, cabling and some supporting structure.

For each capability, or sub-capability, a diagram can be produced showin) how individual ship elements contribute to that capability. Such diagrams an called *dependency diagrams* (see Fig. 15.18).

Some elements will be in series and others in parallel and the effect of losin! anyone element (by failure, accident or enemy action) can be assessed. FOI example:

- The loss of one main propulsion diesel out of a total of four, will mean that only 75% power is available.
- The loss of a propeller in a single shaft ship means the loss of all propulsive power.
- The loss of one steering motor, where two are in parallel, will not affect the immediate performance but leaves the ship vulnerable to the loss of the second motor.

As is discussed elsewhere, dependency diagrams are fundamental to a systematic evaluation of ship availability. In this context they are usually termed *availability diagrams.* They are also used for vulnerability studies when they are called *vulnerability diagrams.*

Iteration in design

Any rational creative endeavour is iterative. It begins with a guess; this is tested against the criteria imposed by the objective, like maximum economic yield; and, found wanting, the guess is modified so that it can be re-tested and so on.

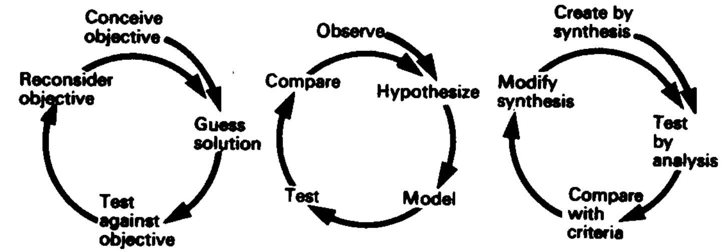


Fig. 15.3

In other circles this is called the scientific method; observe, hypothesize, model, test, compare with observations, modify hypothesis, etc.

So far as a ship is concerned, the initial guess is made easier by experience which may be in two forms:

(a) a basis ship which may be manipulated into a new ship;
(b) basic parameters derived from large numbers of previous ships.

The criteria against which the newly created ships are tested are derived not only from the prime objective but from the boundaries within which the design is to be derived such as national safety regulations, international law and reliability. Thus there will be standards which must be reached in respect of stability, structural behaviour, pollution control, seakeeping, habitability, manoeuvring, internal environment and hazard protection as well as speed and payload.

Table 15.2
Major design stages

Concept design	Choose prime parameter Make first estimate of size, cost, speed Re-assess objective Re-estimate Check other prime parameters Repeat as necessary
Feasibility design	Assess payload characteristics Develop dimensions Assess power requirements Define propulsion machinery options Sketch first layouts Consider vulnerability, safety Embark on spiral convergence, repeating Establish validity of design
Full design	Establish design management Develop each element of the ship, checking behaviour iteratively: geometry stability and flotation architecture dynamics propulsion structure

DESIGN PHASES

The amount of work involved in the design process will depend upon the size and complexity of the ship--a liquefied gas carrier will be more complicated than a simple tanker-and how closely the new ship follows the pattern of a previous, successful design. The division of work between the owner's naval architects and those of the main contractor will vary. This does not change the sum total of the technical work but can impact upon the management of the process.

Typically the naval architect's involvement can be split into seven stages although with different people at different stages. Terminology will vary from organization to organization but the basic ideas are the same.

First stage. Usually called the concept or feasibility stage.

In this the customer's requirements are established together with the criteria for customer acceptance of the total system. The overall system can then be broken down into a number of sub-systems, or functions, which can be further sub-divided to create a description of all systems.

The designer then considers various ways of meeting the needs of each element of the design and hence, the feasibility of meeting the overall requirement. Preferred options can be costed and a broad solution agreed with the customer.

Second stage. The feasibility or functional design.

The systems and sub-systems selected from the first stage are designed and the interfaces established. It is important to define accurately all the interfaces so as to avoid duplication or omissions. This is a time of progressive refinement using a wide range of inputs-design reviews, analyses, specialist advice, modelling, information from equipment suppliers and feed back of experience from sea.

Third stage. The full design.

In this stage the detailed layout of compartments and the sizing and routing of all cabling, trunking and piping is undertaken. Again it is a matter of iteration, compromise and final acceptance. The information obtained in this phase enables a list of all the material needed to build the ship can be defined.

Fourth stage.

During this phase the information-drawings, computer inputs and so on-to support manufacture and assembly is produced. This must be associated with a build plan to ensure that material and information are available in a timely fashion to support the planned building sequence.

Fifth stage. The build stage.

This is the stage that covers the manufacture and assembly of the ship and all its associated systems and equipment.

Sixth stage. Testing and commissioning.

As sections of the ship are completed they must be tested to ensure the requirements have been met, including those of safety. Early tests will indicate the air or water tightness of structure. Later will come tests to show that systems have been installed correctly. Finally the overall performance will he established by trials in the basin or at sea.

Seventh stage. In-service.

The owner will be concerned with support of the ship throughout its life. There will be initial periods which will be covered by warranty. The owner may also make the original contractor responsible for extended periods of operation. This encourages the production of a reliable product.

The first three stages are the true design phases and they are summarized in Table 15.2. Each is iterative in nature.

PRIME PARAMETERS

There are only three prime parameters by which the first estimate of size of a ship may be made. They are volume or mass or linear dimension. One of these will dominate the choice of size of ship to carry the required payload. Volume-limited ships (or capacity carriers) are those which when full are not down to their minimum freeboard. A mass-limited ship is one where there will be unused volume when the mass required is carried. Dimensionally-limited ships are those whose minimum dimensions are determined by the dimensions of their payload. It is useful to anticipate what sort of prime parameter is relevant because it is possible then to converge more quickly upon the solution.

Typical of mass-limited ships are ore carriers and the armoured battleships of old; passenger ships and many light warships are typical of volume-limited ships. Some warships, container ships and river boats are typical of ships which are wholly or partially constrained by particular dimensions; the standard container must fit into the dimensions of container ships while river boats often have a maximum draught limitation. Most modern warships are volume-limited but may be constrained or even dominated by weather deck layout (see Fig. 15.6).

So important are these three prime parameters that we must study each in some detail.

Volume

The volumes given over to various functions in a frigate and in a cargo ship are shown in Fig. 15.4. While there are variations depending mainly on enduranc,,-these values are surprisingly constant for many countries of origin.

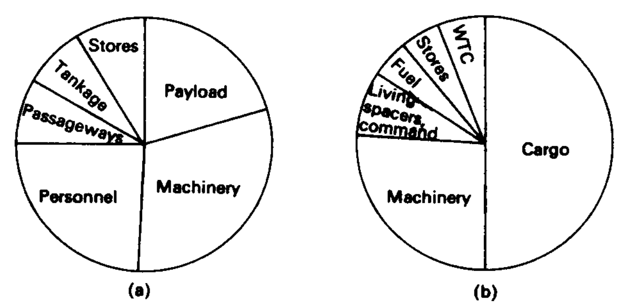


Fig. 15.4 Division by volume: (a) Frigate, (b) Cargo liner

Most destroyers and frigates devote about 20 per cent of their enclosed volume to armament and about 5 per cent to tankage. As a starting point a pseudo block coefficient is often useful; this is defined as

$$C_B' = \frac{V_T}{L.B.D.} = \frac{\text{totally enclosed volume}}{\text{length o.a.} \times \text{beam W.L.} \times \text{depth}}$$

which for frigates and destroyers is about 0.9. Thus, by knowing the armament to be carried, its estimated volume may be multiplied by 5 to give a first shot at the totally enclosed volume from which, using the ratios later discussed an estimate of dimensions may be derived. Displacement can be derived from the enclosed volume by a 'density' defined as the total enclosed volume divided by the light displacement. Figures for this density vary from 4.5 m^3/tonne for rather densely packed designs to 5.5 m^3/tonne for the more spacious.

Cargo ships of some 20,000 tonnef deadweight would normally expect to devote at least 50 per cent of their volume to cargo and 25 per cent to machinery. The breakdown of volume can be made quite easily however for a wide range of ships by rules of thumb. Not all of the moulded internal cargo volume is available for cargo because of the shape of the cargo. Percentages available are roughly:

grain 98
bale 88
insulated 72

Of course, most naval architects prefer to work from a basic reliable proven ship design if there is one available so that they can converge more quickly than is possible from the parameter approach. However the new ship is derived, these first crude steps will need refinement and variational analysis as later described.

Mass

Whether or not volume is more dominant than mass, an early estimate of displacement is necessary. Typical values of the deadweight ratio or, more correctly, deadmass ratio are:

	Deadmass/L;
Passenger liner	0.35
Container ship	0.60
Liquid gas carrier	0.62
General cargo liner	0.67
Ore carrier	0.82
Large tanker	0.86

Historical data may be used to give estimates of each weight group of various nerchant ships.

Some typical percentages of the deep displacement for various elements in nerchant ships are given in Table 15.3.

Table 15.3
Typical mass group percentages

Group	16kt Cargo ship L = 150m	22kt Passenger ship L = 240m	16kt Oil tanker L = 200 m
Net steel	21	36	17
Outfit	5	16.5	1.6
Hull systems	1.5	6	1.6
Propulsion machinery	5	5.9	2.3
Light mass	32.5	64.4	22.5
Crew and passengers	0.2	0.6	1.6
Fuel	11	14	2.6
Fresh water	0.3	13	1.3
Dry cargo	35	8	
Liquid cargo	21		72
Deadmass	67.5	35.6	77.5
Deep displacement	100	100	100
Deadmass ratio	0.675	0.356	0.775

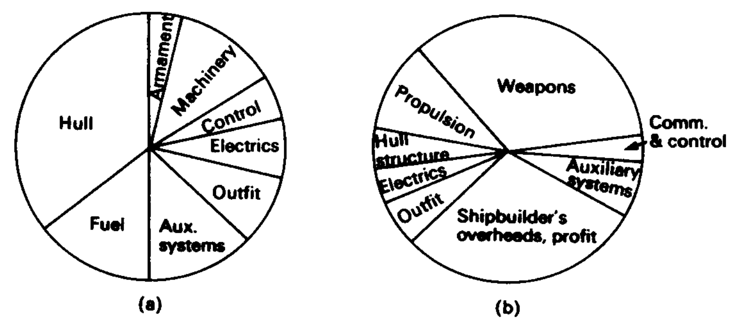


Fig. 15.5 Division offrigate (a) by mass, (b) by procurement cost

The percentages of deep displacement mass given over to various functions in a frigate are shown again in pie chart form in Fig. 15.5(a). This figure contrasts with Fig. 15.4. Payload, accounting for 20 per cent of the internal volume needs less than 10 per cent of the displacement; structure accounting for 36 per cent of the mass needs little volume. An even greater contrast occurs in the percentages of ship cost. While the figures do vary quite a lot and depend much on the particular accounting conventions adopted, Fig. 15.5(b) does give some idea of the relative costs of elements of a frigate. The hull, for example, now demands less than 10 per cent of the cost even with its slice of overheads while the weapons payload, including the helicopter but without ammunition rates very high, as might be expected.

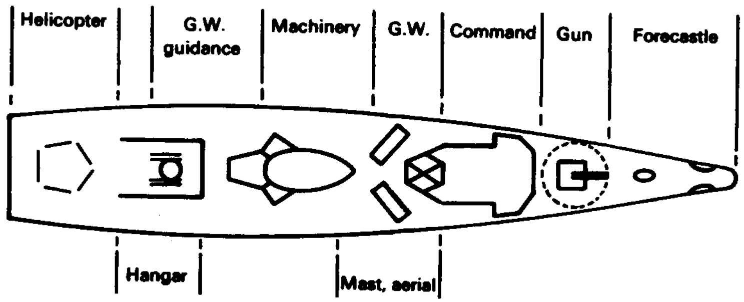


Fig. 15.6 *Frigate weatherdeck layout*

Linear dimensions

Obvious limitations to linear dimensions occur through such needs as crossing the bar over a harbour or dock entrance, passing under a bridge, docking in a dock of a particular length, operating in shallow waters, berthing at a confined quay, movement down a canal and unloading beneath gantry cranes. Rather less obvious are the minimum dimensions needed for a collection of functions. The weatherdeck layout of a warship is an important example. Figure 15.6 shows the minimum length required for a typical frigate upon which may be imposed working from the stern: the need to land a helicopter, hangar it, fit a missile system with adequate arcs of fire controlled by a radar guidance system with similar arcs, machinery uptakes and downtakes, a second missile system, mast acting as an aerial, bridge, gun and forecastle. It is not infrequently the case that the length of the ship is governed by such considerations resulting in both mass and volume to spare-although not often very much. If there is excessive mass and volume resulting then ingenious ways of reducing length are considered; for example, retractable hangars, hangars between split funnels, guided weapons *efflux* over the stern. There is however not usually much pressure to reduce length because length is generally beneficial to the ship's hydrodynamic performance.

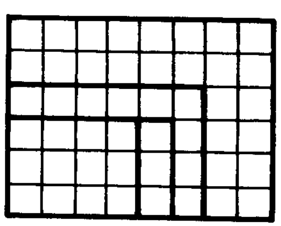


Fig. 15.7 *Container stacks*

Container ships are similarly constrained by the linear dimensions of their payload. The additions of standard container dimensions do give designers of such ships choices in beam and depth close to the steps shown in Fig. 15.7 imposed by their internal stowages.

PARAMETRIC STUDIES

The stage at which the dimensions and their ratios will be considered depends much upon the nature of the investigation and the requirements of the owners. Whether or not they are considered during concept and pre-feasibility study, it is certain that they must be thought about before the ship is declared to be a viable proposition. There are large numbers of parameters of assistance to naval architects upon which we have already dwelt throughout these books. Faced by them all, a sense of proportion is not easy to acquire. However, there are a few which are very dominant and rather more which are especially helpful in directing the choices before the detailed calculations are embarked upon.

So far, we have derived from economic or military arguments the payload, the appropriate volume and mass of the ship and constraints upon specific linear dimensions. Now, the dimensions must be derived to achieve the best compromise among many conflicting features-speed, endurance, strength, stability, seakeeping, manoeuvrability, cost, production, architecture, protection, habitability, signature suppression, survivability, logistics, reliability, etc. In making such choices, it is necessary to have constantly in mind the effect of changes to the parameters, qualitatively and quantitatively and it is necessary to remind ourselves of some work discussed throughout these chapters.

Length The longer the ship the better, in general, is the longitudinal seakeeping and the smaller the power required for a given speed. Against this, there is an increase in longitudinal structure and there may be more difficulty in achieving high manoeuvrability. Usually, with fuel prices relatively dominant, the most important consideration is the minimization of the propulsive power needed and long narrow ships find favour. There is a limit brought about by the awkwardness of building narrow compartments at the ends and values of ® much in excess of 8.0 are not very common for that reason. Nor should greater

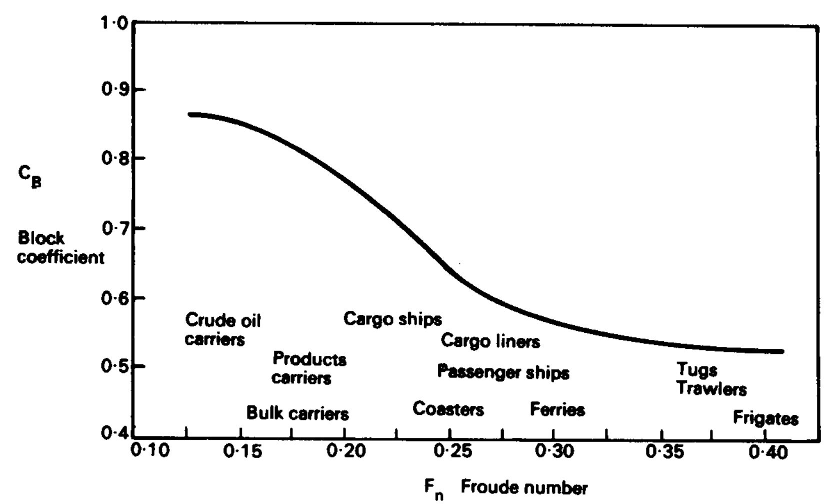


Fig. 15.8 Optimum block coefficient

length be allowed to cause a reduction in beam that would create stability difficulties or a reduction in draught that might cause slamming.

Prismatic and block coefficients These coefficients give an idea of the fineness of the ends relative to the middle part of the ship or to the product BT; for fine forms as with warships, prismatic coefficient is preferred as an aid while bluff merchant ship forms are judged usually on the basis of block coefficient. Prismatic coefficient has an important effect upon residuary resistance and optimum values have already been shown in Chapter II, Fig. 11.18. Low prismatic coefficients mean fine ends which can give problems in confining within the ship's lines bulky fittings, like diesel generators, sonar arrays or weapon systems. Optimum block coefficients as a function of Fn have been proposed for various types of merchant ships and these are shown in Fig. 15.8.

Dimensional ratios B/T, T/D Beam-to-draught ratio is of major importance to initial transverse stability and natural period ofrol1. Figures of around 2- 2! are common in weight dominated designs and about 3!-4 are usual for warships and passenger ships. There is a slight increase in resistance as B/T increases. Draught/depth ratio T/D is extremely important to large angle stability since it determines the point of deck edge immersion. It also determines freeboard and is therefore a measure of deck wetness and it indicates the reserve of buoyancy for survivability. Frigates tend to values of T/D around 0.5. Common values of various ratios of large numbers of ships are given in Fig. 15.9 and Table 15.4.

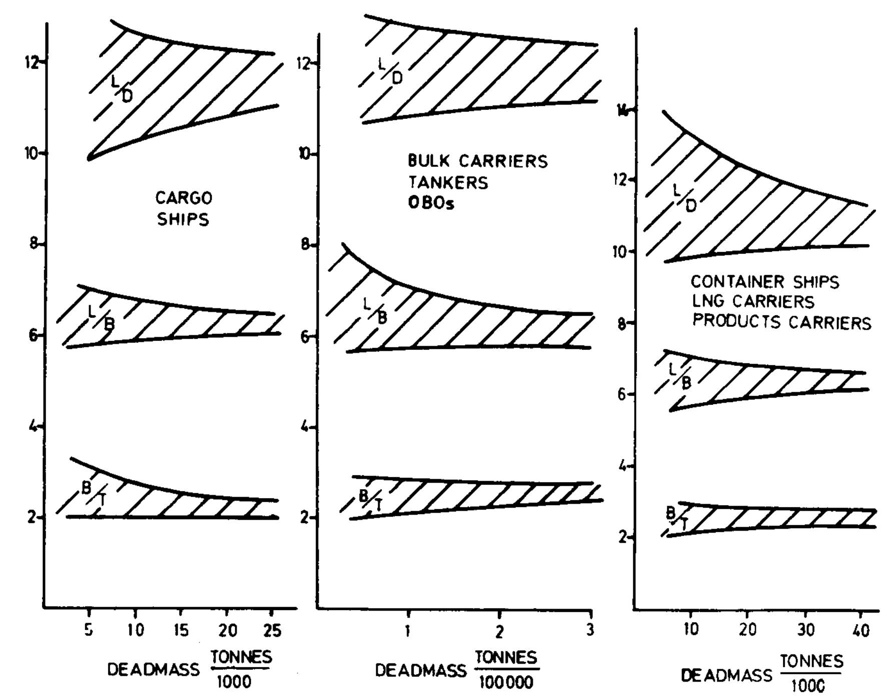


Fig. 15.9 General ranges of principal dimensions

Table 15.4
Typical warship type dimension ratios

Warship type	'V 10^3m^3	$LI'V^{1/3}$	LIB	LID	BID	BIT	F'V [Usl yI(g. 'V$^{1/3}$)]
WW2 battleship	40-60	7	7	14	1.8	2.5-3	0.8
WW2 destroyer	2-3	8-9	10	16	1.8	3-3.5	1.5
Minehunter	0.5	5-6.5	5-6	8	1.4	3.2-4	0.8
Corvette	1-2	7-8	7-8	11	1.5	3.5	1.3
Frigate	3-5	7-8.5	8-9.5	13	1.5	2.8-3.5	1.2
Cruiser	7-10	7-8.5	8-10	12	1.4	2.5-3.2	1.1
AIC Carrier	13-90	6-7.5	6-8	9	1.3	3.3-4.1	0.8

Us = Ship speed
F'V = Froude Displacement Number

Ship form The effect of form on resistance has been discussed at length in Chapters 10 and 11. There are some additional features that have to be borne in mind. Flare of the ship's side at the waterline of some 15 degrees can be very beneficial in keeping -K-M-constant with increasing displacement so that adequate stability can be maintained with variable payload. Knuckles higher up the ship's side avoid excessive flare and are often useful in throwing water clear as the ship pitches, so that spray is not whipped up by the wind. Large flat bottom areas, especially in the first 20 per cent of the length could well invite excessive slamming. Vee sections forward are beneficial to seakeeping. High sheer at the forecastle often helps to keep a dry fore-deck.

Waterplane coefficients A high waterplane coefficient C_w has a beneficial effect on seakeeping, although secondary to the effects of length. A fine angle of entrance at the bow gives a good start to the hydrodynamic flow over the ship and can be beneficial to resistance and to noise reduction.

A good relationship for stability purposes between C_w and C_p for small warships is

$$C_w = 0.44 + 0.52C_p.$$

Longitudinal centres and bulbs The effects of LCB position are relatively smar' on resistance but the addition of a bulb forward can be important to resistance especially at high Fn for cargo ships with a low block coefficient; however bulbs must be considered carefully during tank tests and generalizations are misleading. The centre of lateral resistance below water is important to manoeuvrability and it is useful in the early stages to have some flexibility available in the provision of deadwood aft or forefoot cutaway so that the stability coefficients can be massaged after tank tests.

Propeller While optimum design points for propellers have been discussed in Chapter II, some more general observations are worth keeping in mind. Slow revving large diameter propellers tend to be favoured wherever possible because they enable higher QPC to be achieved, they are less noisy and they should cause less excitation to the hull. There are limits however. They must be adequately immersed and enjoy satisfactory clearance from the ship's hull and from a dock bottom. More important, there will be a speed of propeller rotation below which an additional reduction gear train will be required, causing a step change in gearbox costs. While there may be some special reason for adopting controllable pitch propellers, ducts or shrouds; vertical axis propellers and other devices, it must be remembered that a well-designed fixed-pitch helical propeller is difficult to better for efficiency.

FEASIBILITY STUDIES

We now come to the very nub of the ship design process. How can it best be represented in the mind of the designer? What sort of order can be imposed upon a process which is so highly interactive? A change to anyone parameter will affect many factors and, in turn, require changes to other parameters; it is virtually impossible to change one dimension or parameter without significant effects upon many dependent variables. As we have already discussed, the process is iterative and convergence upon the final solution can only be achieved by going over the elements repetitively until they all match.

There are many ways of representing this convergent process but one of the most evocative is the design spiral (Fig. 15.10). This conveys both the interactive and the iterative nature of the whole ship design. It is presented here because it is at the feasibility stage that most of the factors come into play for the first time. Generation of the need for the ship is represented at the centre of the spiral from economic or military argument. From that, the spiral begins first with the identification of the characteristics of the payload or cargo and the complement. The prime parameter is chosen and the concept phase of the design process may proceed as already described. Some sectors of the diagram may be left for later turns of the spiral.

The order in which factors are considered and, indeed, which ones are omitted is a matter of judgement but broadly follows that shown. The number of circuits made until satisfactory convergence has been achieved will also vary with each ship design but they will be least when the most appropriate prime parameter has been chosen. If a less appropriate prime parameter has been chosen, convergence will still Occur but it will take longer. Feasibility study will certainly take more than one circuit and the stage at which the ship may be considered viable is also a matter of judgement. It is often useful to calculate the effects of varying one specific parameter on the parameters and performances represented in other sectors of the spiral, simply to discover how influential it is, e.g. to see how small changes in each dimension affect stability, strength, seakeeping, survivability, cost and propulsion. This variational analysis gives a designer a 'feel' for the design, assisting his judgement as he proceeds.

One especial difficulty in warship design is in the assessment of the complement. Despite many rules, assessment of the necessary complement has a high opinionative element affected by the standards of good ship husbandry, cross training, command philosophy, contract hours of personnel and calls on the crew in emergencies. Moreover, taking as it does 25 per cent of the volume of the

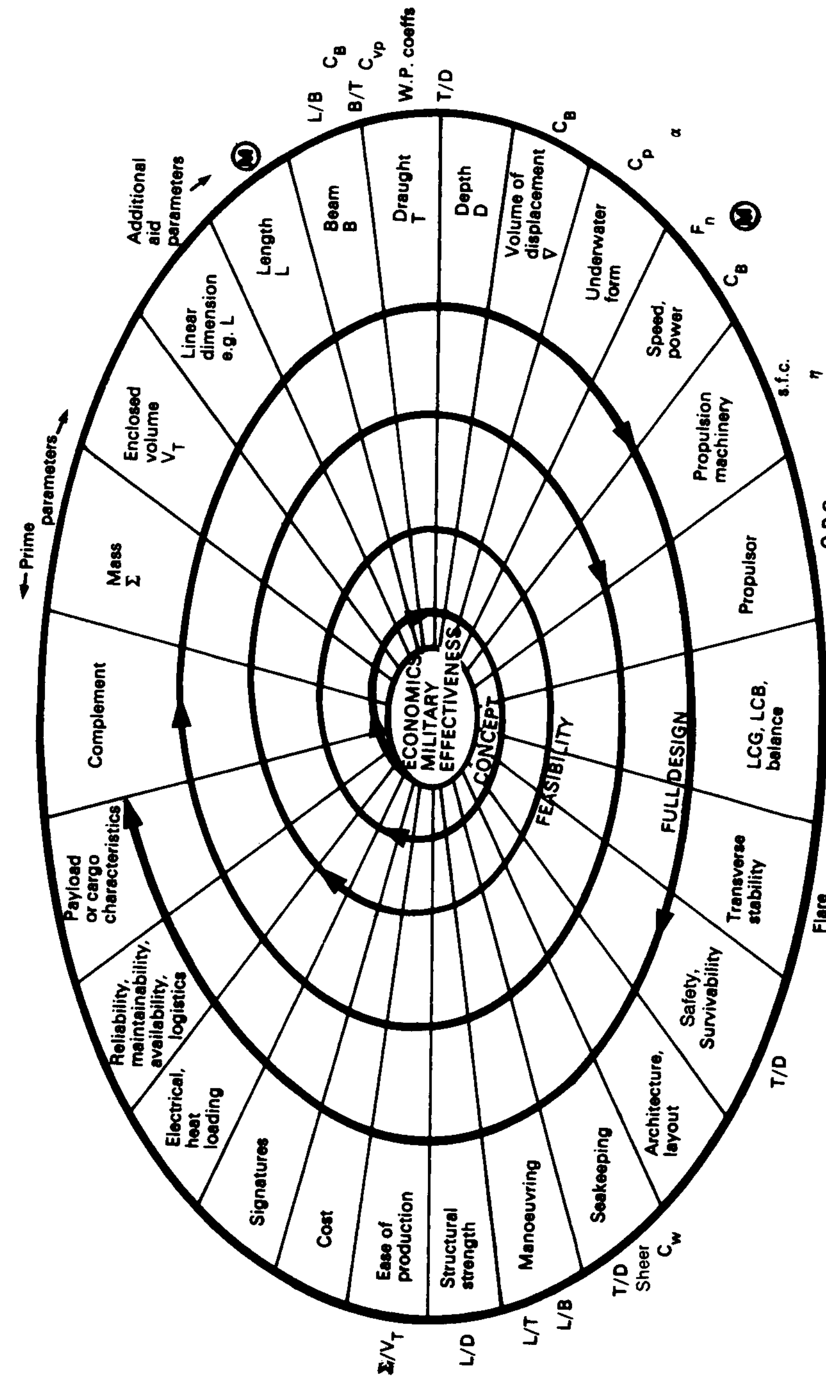


Fig. 15.10 The design spiral

ship, it has a profound effect on ship size and cost. Merchant ships also may find a small crew economical and a bigger investment in automation necessary. Wise designers will therefore set up very early in feasibility, a dialogue with those concerned to establish firmly what accommodation should be provided.

At the full design stage changes will still occur but by this stage the changes will be more in the nature of adjustments and the interactions will be much reduced. At the design production stage the nature of the activity changes and the spiral can be discarded. Design then becomes an important exercise in communication between the detailed creative activity in the shipyard design office and those who order the materials and build the ship.

Figure 15.10 also shows around the periphery some of those aid parameters which have been discussed in the previous section. It is not a complete list and students will wish to augment this *aide-memo ire* for themselves.

The object of the feasibility stage is to ensure that the elements of the ship design form a matching set. In considering such matters as survivability and strength it is necessary also to decide upon the standards to which the ship design is to conform. These will depend upon the scenario of operations prescribed and the constraints, like cost, which are imposed. In warship design, the days when nothing was too good for the fighting man have long passed and the decisions on what standards of habitability, protection and weapons can be afforded are difficult and painful. Thus, at the feasibility stage the basic compromise among all of the conflicting demands upon a ship is struck.

At this stage also items which require development or which involve any technical risk are identified, programmes are drawn up and preparations are made for the major commitment of full design development.

FULL DESIGN

It would be foolhardy to assume that nothing is going to change after the feasibility phase. As details are worked out the need for adjustments becomes apparent and the consequences of quite trivial matters can sometimes be extensive. Furthermore, some of the assumptions implicit in the feasibility study will not develop as expected, especially those with high technical risk, and the owner or the Naval Staff may well have a change of mind as market forces or military threat change. The full design phase retains its dynamic character but is, nevertheless, the stabilizing period for the design.

During this time, the compartments which have hitherto been spaces on a layout are considered individually, piping systems and other services are designed and inserted into the ship, electrical distribution evolves, data highways and the flow of information, people, material and command are all developed. The number of designers required increases by an order. As these matters are developed, conflict is inevitable and adjustments to the general arrangement layout to alleviate the problems are made continually. The more complex the ship, the more difficult and prolonged is the process. Tank tests during this time will have provided the final 10 per cent accuracy on the powering required and will have demonstrated the need for adjustments to underwater form, propeller dimensions and other hydrodynamic matters.

Although most, if not all, of the procedures used in developing the design lire now, or could be, computerized, it is instructive to describe some of the munuul methods used previously. This can give a clearer understanding of the principles involved. Generally, the computer has merely enabled them to be carried through in more detail, more accurately and in a shorter time scale. That is to say the computer has not changed what the designer is aiming to achieve but, rather, the means of achievement.

It has already been shown that definitions are important and must be precise and comprehensive. There must be no room for ambiguity. Consistency is vital if databases of information are to be used intelligently. Thus weight and cost data must be broken down in a disciplined and well-defined way. These divisions should be consistent with the systems and capabilities used to define the ship. Information on equipments must precisely define their form fit and function.

As the design progresses decisions will be made and, as a result, elements of the proposed ship will emerge. It is important that such decisions, and their results, are recorded in such a way that they can be readily retrieved. Put another way there must be an audit trail which can be followed later to establish what happened, why and who was responsible. Design records or logs can contain the minutes of all important meetings and conference decisions. In the case of warships a *Ship's Cover* was always produced and good ones are invaluable in following the thought processes that led to designs being configured in a particular way. Clearly their value depends very much upon the skill of the compiler.

From the beginning of the design process the naval architect is aware of ho\\ one element of the design interacts with another, often requiring a compromise between conflicting desires. This can apply at all levels of detail. Thus:

- a change in upper deck layout which involves moving the funnels wil impact upon the positioning of the machinery low in the ship and th, uptakes, downtakes and removal routes;
- the level of manoeuvrability demanded may necessitate fitting twin shafts o some form of transverse thruster;
- changing the helicopter to be carried will impact upon the flight deck ~ and layout, the hangar, support services, the strength of the flight deck an possibly the supporting transverse bulkheads.

One way of illustrating these interactions is the *design influence diagran* (see Fig. 15.13).

Many people are involved in the detailed development of the design. Ead must be aware of the decisions taken by others that affect what they themselve are doing. For some of these decisions the use of what may be termed *maste general arrangements* is useful. Thus one master can show all doors anI hatches, indicating in which direction they open. Another can show the siz of stiffeners on decks and bulkheads, and for the latter, which side of th bulkhead they are placed. These will avoid duplication, or the possibility tha the designers of the compartments on both sides of a bulkhead will assume tha the stiffeners and doors do not take up space within their compartment.

Other masters can be used for the routing of each pipework, cable or trunk system. Trunks can be sized to indicate the affect on the deckhead height available.

Very often a compromise must be struck between various conflicting requirements. This is one way in which the naval architect's skill, or 'art', is brought into play.

There is no absolute 'right design' or even 'optimum design'. There are a range of *optimization* techniques available but, powerful as these may be, they are limited to specific design aspects. Even those of widest scope do not embrace all design considerations. The designer will find them very useful but the final overall design balance is a personal one.

There are available to the designer many aids and processes that range in usefulness from valuable to handy upon which to draw. Let us look at some of these.

Scheduling In all, some 20,000 drawings may be required to describe a complex warship. Work instructions and machine computer tapes are derived from them. This great bulk of work must be arranged in logical order according to a schedule which accords with the production sequences. While less complex, the merchant ship design has similarly to be controlled.

There are many aids available to assist scheduling. Bar charts and bring-up systems are typical. One of the most powerful is the network which shows each event connected to those which must precede it. Figure 15.11 shows a fragment of a network schedule. There are paths through it which determine the total length of time for the whole process, called critical paths. It is these that excite the attention of the good manager. Network scheduling can quite readily be programmed for a computer and daily comparisons of the schedule intent with achievement are common as a control tool. Time, material, cost, labour, bought-in items, parts lists, tools, all normally form part of the scheduling events and processes and very comprehensive management control systems for shipyards are now available.

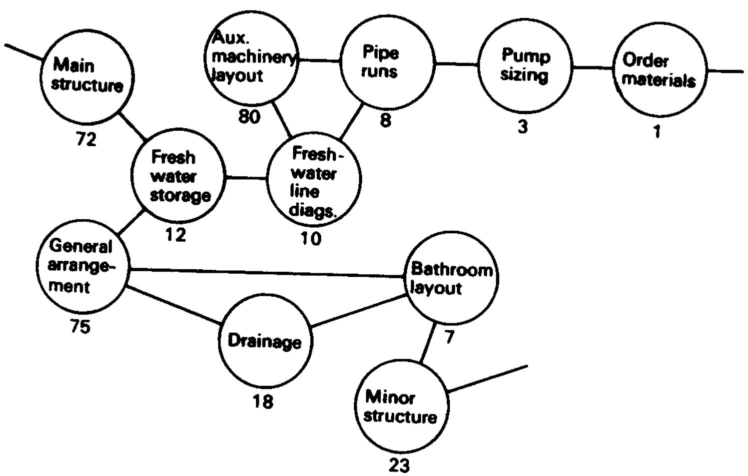


Fig. 15.11 Fragment of a drawing network schedule

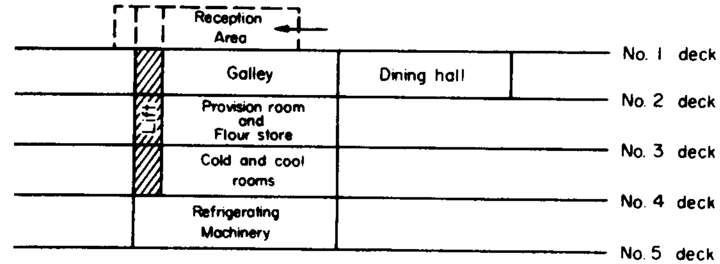


Fig. 15.12 Catering arrangements in destroyer

Flow Diagrams Simple flow diagrams to assist local arrangement in a ship are exemplified by Fig. 15.12. This ideal positioning of the commissariat arrangements in a destroyer is derived from common-sense consideration of the flow of goods in replenishment and food preparation. Similar arrangements may be derived for messdecks and bathrooms based upon considerations of comfort, noise, heat, safety, escape, convenience and the flow of people to toilets, action stations, canteen, etc. Where there are particularly difficult areas, land architects have used devices like trip frequency matrices or diagrams on which lines representing each flow expose bottleneck areas. Flow of information for weapons control is vital to a warship, even though it is made easier by the introduction of data buses. Flow of fluid and electrical power are pursued by the processes of Chapter 14.

Virtual reality, mock ups and models Very congested or complicated spaces (e.g. command centres) where there are important man-machine interfaces used to be modelled full scale, usually in wood. Nowadays *virtual reality* techniques are likely to be used (see computers in Chapter 2) using the ship definition contained within the CAD system. Whether virtual reality, mock up or model is used the aim is to bring together operators, equipment designers and human factors experts to achieve a layout of maximum efficiency.

The next most important area of the ship to be modelled virtually or physically, is that of the main propulsion machinery spaces. Physical models, ifused, are likely to be at a scale of one-tenth or one-fifth full scale. At that scali, it is possible to gain access to nearly everything. It is used firstly for the desi~ of the pipe runs, ventilation trunking, placing of ancillary equipment to check access and removal routes and sometimes later for production purposes. Clear plastic and coloured pipes and fittings make comprehension easy. Smaller scale wooden models are often made for the anchor arrangements and, in copper, for assessing electromagnetic radiation efficiency.

It is becoming more common now to develop full-scale the total weapons system ashore to assess its compatibility and, even, for production of cable runs and other services. This shore test facility is a major investment of many millions of pounds and needs to be judged on the basis of the gains in operating efficiency and amortization of cost to the ships under production.

During production of a ship, the intentions expressed by the various layouts can be brought together in chalk in the bare compartments to produce 2D lineouts. Where justified, wires can be run through a space to create a 3D lineout. Sometimes, the desired layout is left until these lineouts are available. They are invaluable in making last-minute checks that all of the drawings-often produced by many separated authorities-are indeed compatible. Crossing pipes and ventilation trunking, unacceptably low deckheads, interferences and plain forgetfulness are uncovered and corrected. Again virtual reality techniques can be used in place of physical line outs.

Work Study In principle, a work study practitioner considers a certain process, perhaps design, fabrication, installation or maintenance, and asks:

What is achieved?	*Why* is it necessary?
Where is it done?	*Why* there?
When is it done?	*Why* then?
By whom is it done?	*Why* by that person?
How is it done?	*Why* in that way?

In asking 'Why?' the practitioner also seeks alternatives and then assesses the advantages and disadvantages of the various solutions to decide which is the best. This critical examination of the facts often reveals that jobs are unnecessary or are carried out in a certain way by certain people for purely historical reasons. The end product should be a better way of achieving the desired aim. Better may mean that the process is cheaper, quicker, requires less people or is in some other way superior depending upon the terms of reference under which the study was conducted. Work study does not need a highly specialized practitioner; all engineers and technologists should use the tool when it is likely to be profitable. It is often an expensive tool and it should itself be the subject of cost-effectiveness enquiry before being employed.

Value Engineering This is also a questioning process but must be quantified to be really useful. It is readily applied to making economies in the production process. An element of the design is critically examined with a view to achieving the same function more cheaply.

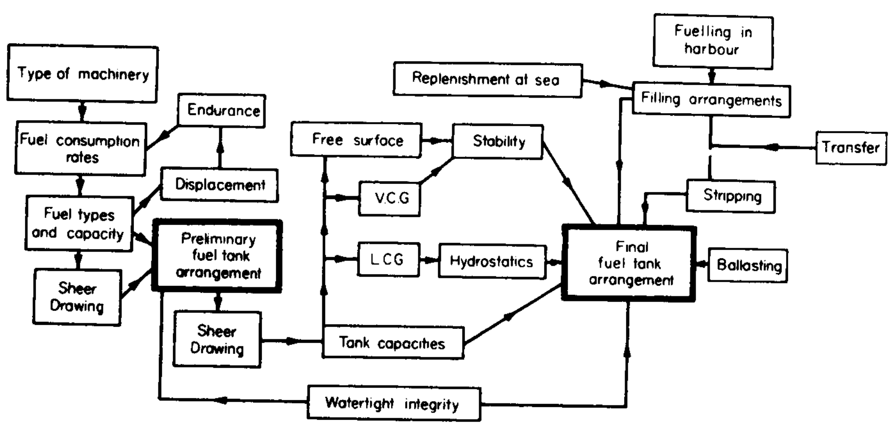


Fig. 15.13 Design influence diagram for fuel tanks

COMPUTER-AIDED DESIGN (CAD)

Initially it was the computer's ability to carry out repetitive calculations rapidly and accurately that made them attractive for much of the traditional design work such as calculation of displacement from a table of offsets and then on (o stability. Since then they have had a much more fundamental impact upon the work of the naval architect. In outline terms the following comments indicate what can be achieved.

- A ship's three-dimensional form can be defined accurately, which is the starting point for design and manufacture. Books are available on techniques that can represent and manipulate fair shapes such as the hull by patches, splines, polynomials, elements, etc., within the constraints placed on it by the designer.
- From the hull form can be derived the hydrostatic and stability characteristics, which affect the static behaviour of the ship in still water.
- Computational fluid dynamics techniques can be used to study flow past the hull and its appendages, and the wave system generated.
- Defining the seaway leads to a determination of the ship responses in terms of overall motions and structural behaviour. Finite element analysis methods can be used for global and local calculations.
- Similarly dynamic responses can be determined in response to control surface movements to establish the directional stability and manoeuvrability.
- Layouts can be studied and the computer can generate automatically the areas and volumes of compartments and tanks. The layouts can provide the basis for a computer-generated 'walk through' of the design showing how spaces will look (colour, texture as well as spatial layout can be shown) to assist both the designer and prospective owner. Accessibility and lines of view can be assessed.
- As decisions are made on structure, equipments, systems and fittings, weight distributions can be kept up-dated and, with them, the stability, strength and other ship design properties.
- Power calculations and propulsor designs can be produced.

As discussed elsewhere, design is an iterative process. The computer enables those iterations to be made more rapidly and accurately, and in greater de. than was previously possible. It remains important, however, that assessments at each iteration are at levels consistent with the firmness of the design. Some calculations have only been possible as the power of computers has increased; others have become economic as costs have dropped; applications are now available on desk-top computers which once needed a large main frame.

At first individual calculations were computerized. These were progressively adapted to form suites of integrated programs where output from one provided an automatic input to others, with many programs interacting with each other. These integrated suites of programs are known as *computer-aided design (CAD) systems.* Some calculations are still carried out separately from the main suites of programs. For instance, CFD calculations of flow around a hull are usually studied separately although using inputs from programs defining the hull shape.

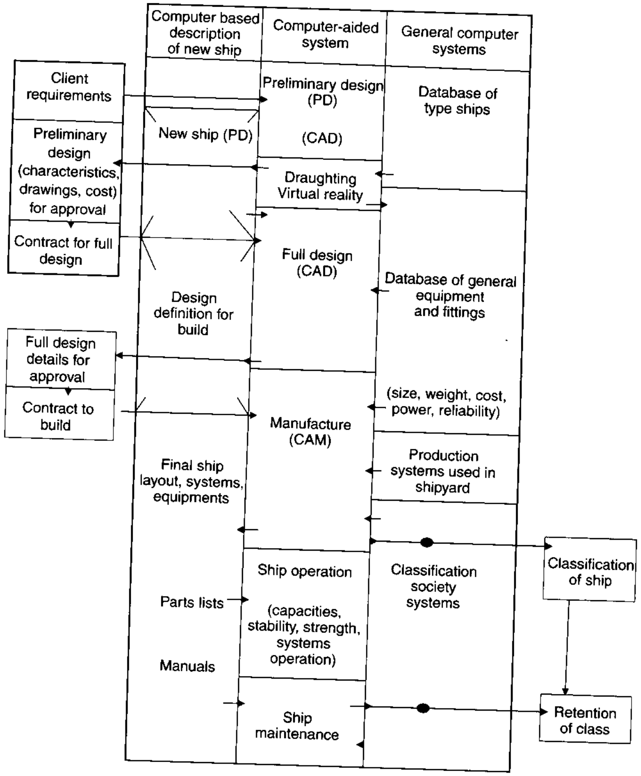


Fig. 15.14 *Schematic of computer-aided processes*

There are now CAD systems for a wide range of applications-large displacement vessels, small craft, surface warships, submarines, container ships, multi-hull vessels and so on. Some deal with the preliminary design phases and others relate to the full design process. And the usefulness of the systems is not restricted to the design process. The design outputs can feed directly into the shipyard and be integrated with their programs for *computer-aided manufacture* (CAM). Thus the form is defined digitally and there is no need to supply tables of offsets. The structural information and lists of equipment, fittings, cable runs and piping provide a data base for feeding directly into material ordering systems. Shape information provides the input for computer controlled cutting machines, associated with nesting programs for maximum economy of material. These CAM applications lead to greater accuracy of fit making fabrication easier and resulting in less built-in stress. They can considerably reduce the manpower and time for production.

In the same way, design and manufacturing information feeds into maintenance manuals and routines and can be used as the basis for recording the state of structure and systems during service. Such information assists surveyors to keep up with the deterioration of the structure during the life of the ship. Similarly the information provides a base for enabling ship staff to keep

track of stability and strength during loading, unloading and operation 0r the ship. In fact there is virtually no activity in the design office, shipyard or Oil board ship that cannot be controlled, or at least influenced, by computers. Whether such systems are economic and appropriate must always be considered. Because an activity can be computerized it does not follow that it should.

The pace of change in the field of CAD/CAM has been, and continues to be, rapid. Thus it is not profitable, in the context of this book, to discuss current systems with their relative merits. The reader should refer to the technical press for information on developments. As an example the Royal Institution of Naval Architects produces, in its journal *The Naval Architect,* a regular review of CAD/CAM developments.

Design for the life intended

There are many aspects of the intended life of the ship which were discussed earlier in this chapter that strongly influence the style and standards of the ship. The first considerations are for efficient use and economical production but so also must the in-life support and the intentions over modernization be considered. While this book is not intended to be instructive on the important subject of production, some reference is needed in so far as it affects design. Let us then look briefly at design for use, design for production, design for availability, design for support and design for modernization.

DESIGN FOR USE

It is important that systems and equipments should be easy to use. The term 'user friendly' is often used in everyday life and is a descriptive one. In ships, easily understood and used equipments mean less specialized training for operators. Equally important they are more likely to be used effectively when operators are under the stress of an emergency or in action.

Where a human being is involved a designer must take account of the way people function physically and mentally. In the general sense naval architects have always taken account of human factors in their designs, e.g. the pulWf-person could exert on a rope, the sizing of furniture, the provision of minimum ventilation standards. However, the efficient blending of human and material in design can only be achieved by a more positive approach and a more formal application of human factor principles. This involves calling upon the professional expertise of the physiologist and psychologist as well as the engineer and scientist.

In some areas of design this has been widely recognized in recent years, e.g. in the design of control consoles. Even in this relatively simple example several distinct problems need to be tackled, viz.:

(a) Ergonomics. The size and layout of the console must be such that all dials can be read; all controls must be readily reached, levers requiring fine adjustment must be operable with minimum force; and so on.

(h) Design 'philosophy'. The design oflayout must help the operator in understanding the results of actions. Thus for a machinery control console a control panel which represents diagrammatically actions within the machinery (e.g. flow of steam) is useful. So also are groupings of dials indicating physical state with controls which change those states; positions of controls for valves indicating whether the valve is open or shut; dangerous conditions immediately obvious by red or flashing lights, aural alarms, etc.

(c) Training. The need to train for routine tasks is obvious. The training for emergency conditions is also vital because of their importance when they occur and in the fact that, hopefully, they Occur only infrequently. Realism is essential.

All too often, a designer of an equipment or system assumes that the user will think and act as the designer would. The different levels of experience and ability of the designer and user are too often reflected in the operating instructions and handbooks issued so making it difficult for the user. Apart from this, however, a proper analysis of what a person does in certain circumstances, and why it is done, often shows that popularly held concepts are invalid. A common finding is that people tend to see what is expected and ignore what is not expected. This may be by a biased selection from the total data available or even by imagining data that is not there. Radar operators may generate bogus tracks or ignore obvious contacts. Thus, for a control console, it is necessary to determine the information to be presented to a controller, what form it should take (e.g. digital or analogue), what should be the result of any actions (e.g. should they directly modify the machinery state) and to establish how an operator will know when the control system itself, including its feedback, is at fault rather than the system being controlled.

In most systems, once the totality of actions has been deduced, a choice exists as to which the human should do and which should be left to the machine. In deciding, the relative strengths and weaknesses of both must be considered. Thus a machine is good at repetitive calculation whereas a human may make errors, particularly when tired or under stress. The human is better at pattern recognition than a machine. Imagine programming a machine to pick out a relative on a photograph which a human can do quickly. In some cases the division between the human and the machine is fixed but increasingly the interaction between the two is varied to suit the need. Thus in controlling a ship it may be desirable to give the helmsman more of the total task when conditions are quiet (e.g. on an ocean passage) to encourage alertness and to allocate more to the machine when the operator would otherwise be overloaded (e.g. operating in a congested seaway in bad weather).

It was the advent of the computer that provided the human factor practitioner with the greatest problems and greatest challenge. It offers now scope for the development of systems with artificial intelligence, with the human and the machine effectively having a dialogue and supporting each other. This may be for training purposes using part-task simulators or in the operational role.

Even without artificial intelligence as such, the growing capability of **the** computer for a given size and cost opens up prospects of decision or training 'aids'. These will often appear similar to 'games' giving the human a chance to study 'what happens if' so facilitating selection of the action that leads to the desired outcome. Many of these potential applications involve probabilities. Few outcomes of high-level actions are certain, depending upon circumstances of the environment and what actions others mayor may not carry out. Most individuals find probabilities hard to conceive in other than relatively simple cases. In multi-variant problems the best method of presenting data for human decisions is by no means clear, although in general terms it has been shown that graphical rather than alpha-numeric displays are preferable.

DESIGN FOR PRODUCTION

A good designer will have a feel for features which are likely to prove difficult to build, and which will, therefore, be expensive of time and money. Without that feel it will not be possible to arrive at sound compromises so necessary in ship design. Ideally designers will have had some experience of production. As an example, plating curved in only one dimension is much cheaper to produce than plating curved in two dimensions, but for the hull there may be a resistance or radar signature penalty. For many years now warships have dispensed with sheer and camber on decks other than weather decks. In steel structures the more welding that can be done by machine, the cheaper and more reliable it will be. The longer the straight runs of welding the less time will be absorbed in setting up the machines. Using stiffeners in one direction only, avoiding intersection of welds and intercostal members will help, as will the use of steels which do not need special pre-heating procedures and the use of standard sections. Using swedged or corrugated bulkheads can avoid the need to weld stiffeners to plating. Confined areas must be large enough to provide access for fabrication and application of protective coatings. Grouping system components to reduce the length of cable and piping runs will reduce cost and weight. Some design features will be universally advantageous. Others will depend upon the production methods used, and equipment available, in the building yard. In general, the more pre-outfitting that can be done before the main sections of the ship are joined together, the better. Designers need to b~ continual dialogue with the production team to provide break points which are compatible with the most economic production process.

DESIGN FOR AVAILABILITY

Availability has been seen earlier to be one of the main constituents of effectiveness or value. In turn, it is dependent on reliability and maintainability. These terms possess a range of meanings, the vernacular no less important than the mathematical, presently described. Reliability engineering developed slowly after it was first proposed by Sir Alfred Pugsley in 1930 but it received great impetus from the space programme 30 years later.

Let us consider the time taken for a large population of similar devices tested in identical conditions to fail. The number of failures in each interval of time

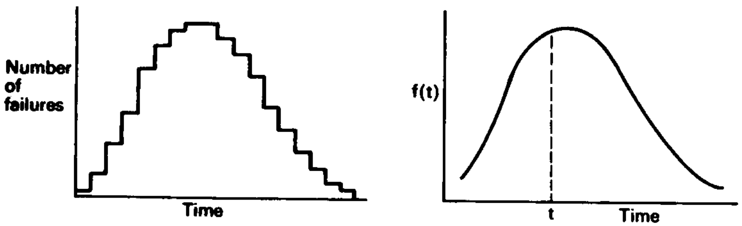


Fig. 15.15

may be plotted as a histogram. With sufficiently small intervals of time the number of failures divided by the total population may be represented by a smooth curve $f(t)$ called the probability density function.

If we now regard the behaviour of any one of the items as represented by the behaviour of the total sample, it is clear that the probability of failure up to time t is

$$F(t) = \int_0^t f(t)\,\mathrm{d}t$$

Because the total area of the curve must be unity, the probability of not failing is

$$R(t) = 1 - F(t) = 1 - \int_0^t f(t)\,\mathrm{d}t$$

This is the reliability. It is defined formally as the probability of a device performing adequately for the period of time intended under the operating conditions encountered. Note the dependence on specifying the environment. The integral curve $F(t)$ is the cumulative failure distribution and $R(t)$ is the reliability distribution function. There is one other important parameter $z(t)$, the instantaneous hazard rate or failure rate which can be shown to be

$$z(t) = \frac{f(t)}{R(t)}$$

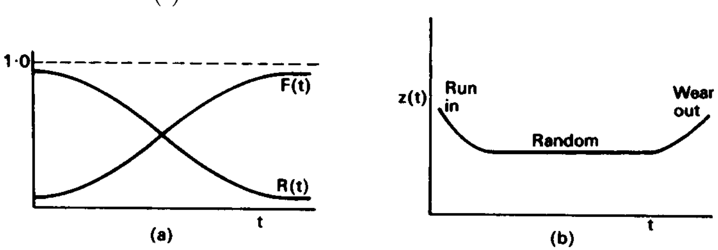


Fig. 15.16 (a) Reliability functions, (b) Bathtub curve

Testing or experience shows that the reliability functions may follow a specific statistical distribution. Wearout, for example, is often found to follow a Gaussian law, the probability density function being

$$f(t) = \frac{1}{\sigma\sqrt{2\pi}}\exp - \frac{1}{2}\left(\frac{t-\mu}{\sigma}\right)^2$$

where μ and σ are the mean and standard deviations.

A most important distribution related often to complex equipments subject to random failure is the exponential for which

$$f(t) = \frac{1}{\theta}\exp\left(-\frac{t}{\theta}\right) = \lambda\exp(-\lambda t)$$

$$R(t) = \exp\left(-\frac{t}{\theta}\right) = \exp(-\lambda t)$$

$z(t) = \frac{1}{\theta} = \lambda$, a constant, the failure rate. θ is its inverse known as the mean time between failures (MTBF). It is this which accounts for the main portion of the bathtub curve of Fig. 15.16(*b*). This type of curve typifies many systems in which, after an early period of running in, it enjoys a very long period of roughly constant failure rate until, as it ages, wearout makes the hazard rate rise again.

The exponential distribution is actually a special case of the Poisson distribution which may, in order to assess the number of spares to be carried, be used to predict the number of failures k in a stated time

$$p(k) = \sum_{i=0}^{i=k}\frac{(\lambda t)^i}{i!}\exp(-\lambda t)$$

The mathematical convenience of the exponential form makes it a temptation to apply to every case. It is important therefore to be assured of its relevance. Where it is applicable, it is salutary to examine the probability of failure; for an MTBF of 3000 hours for example the reliability is

Mission time (hours)	3000	750	120	24	4
Reliability	0.368	0.779	0.961	0.994	0.999

while the probability of there being failures in 3000 hours from the Poisson function is

Number of failures k	0	1	2	3	4
Probability	0.37	0.26	0.08	0.019	0.004

The overall reliability of a group of components which go to make up an equipment can be computed by combining reliabilities by laws very similar to those governing the combination of resistors in parallel or series. The components may not be physically in series but if their dependence on each other is sequential they will be functionally so and the overall reliability is the product of each, $\Pi R(t)$

$$R_s(t) = \prod_{i=1}^{n} R_i(t)$$

If, for example, there were 10 components each with a reliability of 0.99 placed in series, the overall reliability would be $0.99^{10} = 0.905$. If in parallel, equally loaded, the components' unreliability is similarly combined, so that

$$R_s(t) = 1 - \prod_{i=1}^{n} F_i(t) = 1 - \prod_{i=1}^{n}(1 - R_i(t))$$

In the above case, units in parallel would then have a reliability of $1-(0.01)^{10}$ which is almost unity.

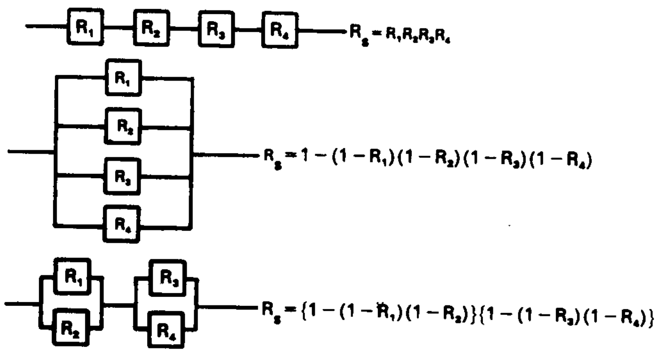
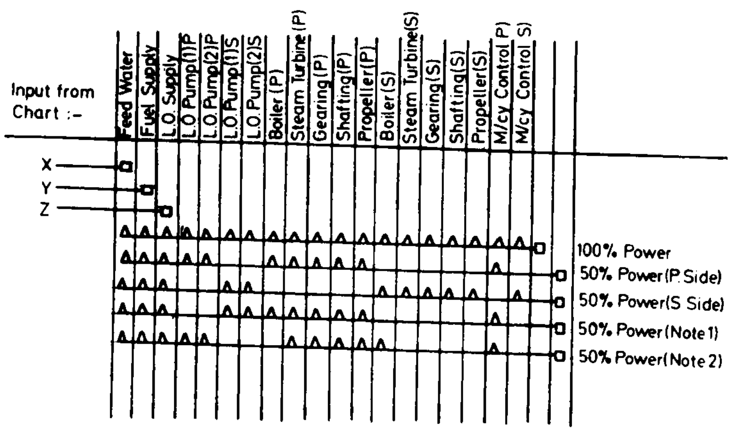


Fig. 15.17 *Some system reliahiliry models*

This gives the clue to reliability modelling. Systems of components whose individual reliabilities operating in similar environments are known from data bases can be built up from knowledge of their functional arrangements. Functional dependency diagrams like Fig. 15.18 assist. The reliability modelling then enables the overall reliability to be calculated and improved, if necessary by the incorporation of redundant equipments in parallel where there are weak points.

Reliability modelling is important not only to the assessment of risk but to adequate quality assurance and the rational scaling of spare parts. Maintainability in addition to its pragmatic meaning of ease of maintenance, is susceptible to mathematical treatment in which the variate is the time to repair for a population of faults.



Notes (1) Shows S. L.O. Pump can serve P side equipments
(2) Shows S. boiler can supply steam to P. turbine
(3) Brackets signify either L.O. Pump can fulfill need ie one of the pair is standby

Fig. 15.18 Part of speed characteristic dependency diagram

Availability at its simplest is the ratio of the time when a system or equipment is 'up', i.e. able to work to the total time. Redundancy in a system obviously raises availability and hence overall effectiveness.

DESIGN FOR SUPPORT

This is not a major influence on merchant ship design but for the dense warships it is extremely important. Reliability and maintainability studies, already discussed, identify those equipments which may require removal during the operational life of the ship. Gas turbine change units, for example, are planned to be replaced at a specified life and occasionally need replacement at unexpected moments. The requirement is to change them within 48 hours and removal routes have to be designed into the ship (often up the air downtakes). Diesels usually need portable plates through which they may be unshipped. Access around equipment for maintenance has to be planned carefully, including room to withdraw rotors, and maintenance envelopes are usually shown dotted on detailed layout drawings. Propeller shaft withdrawal and rudder removal in dry dock have to be thought about. Room has to be left in the layouts for spares close to where they may be needed, especially if they are heavy or bulky.

Electronic equipment is often subject to a repair-by-replacement policy and standardization here plays an especially important role. Standard boards or cards of electronic elements assist in easing a vast logistic problem. Some weapons elements must also be changed during weapons update periods, often by ship's staff, so that some parts of the weapons system keep pace with advances by a potential enemy. Such equipment usually needs to be kept in a controlled, dry environment so that ancillary equipment has to be reliable and easily maintainable.

Impending failure can be predicted by various health monitoring devices based, for example, on spectroscopic examination of oil or vibration measurement. This monitoring equipment is often designed into the control consoles Wi' the machines.

DESIGN FOR MODERNIZATION

A warship may last up to 30 years, during which time its weapons become obsolete. Taking a warship to pieces to replace whole weapons systems is time-consuming and expensive, especially when the system depends on elements scattered throughout the ship. Although this has long been recognized, only recently have advances been made which can substantially isolate a weapon system from the ship platform itself. This type of isolation has appeared under the general heading of modularity. One specific example is known as the MEKO system (Germany). While the principle is simple, namely to provide a self-contained module housing a weapon system, practice has been bedevilled by the need for that system to draw upon information from the gyro compass, the surveillance radar, the command system (AIm and many other parts of the ship. This problem has now been solved by the introduction of data highways or data buses which are able to carry packaged information electronically around the ship to be drawn off where required. It is likely therefore that, although modularized ships do tend to be somewhat larger and, initially, more expensive, they will become the normal way of designing large warships. Other services for the modules like chilled water and electrical supplies may still be drawn from a central source to avoid the expense of duplication.

Zoning of warships also aims, as far as is practicable, to keep functions of a ship within zones that can be sealed off. They form not only fire, smoke and watertight subdivision but sections of the ship which are independent of others to reduce vulnerability to action damage and to ease modernization. It is not, of course, possible to make them totally independent because some systems, like electrical power, are almost certainly to be distributed ship-wide. Nevertheless, it is worthwhile where it can be done economically.

THE SAFETY CASE

Design merely to meet regulations is no longer acceptable. Nor can it be assumed that a ship and its individual parts are either safe or unsafe. There are grades of acceptability which must be individually considered as compromises among the conflicting demands of profitability, facility of operation, social acceptability, personal danger and potential litigation. Such compromises are arrived at through formal procedures which are grouped together under the description of the Safety Case and may be implemented within a ship or equipment safety management system. While some aspects are conducive to a numerical approach, decisions on the standards to be adopted are often matters of judgement.

There are two elements to perceived risk; the likelihood of occurrence and the consequences of that occurrence. Judgement is based on the compound of these two elements. Thus, a low likelihood of occurrence but a catastrophic outcome would often be regarded as unacceptable, while a likely event resulting in a trivial outcome would be judged on the basis of its nuisance or economic impact. Of course, behaviour of operators and the environment within which they must work are inextricably mixed with the event.

Formal steps in the development of a safety case are clear:

(i) identification of all potential hazards and the likelihood of their occurrence in defined circumstances and environment;
(ii) identification of the consequences of each event;
(iii) establishment of a system for controlling the occurrence and its consequence, including escape and rescue, if relevant;
(iv) issue of codes of practice and communications which constitute a safety management system;
(v) institution of auditing procedures for the safety management system.

There are many techniqes available which help in the assessment of safety, for example reliability engineering, failure mode and effect analysis, fault tree diagrams, dependency diagrams. These may often expose the need for clear

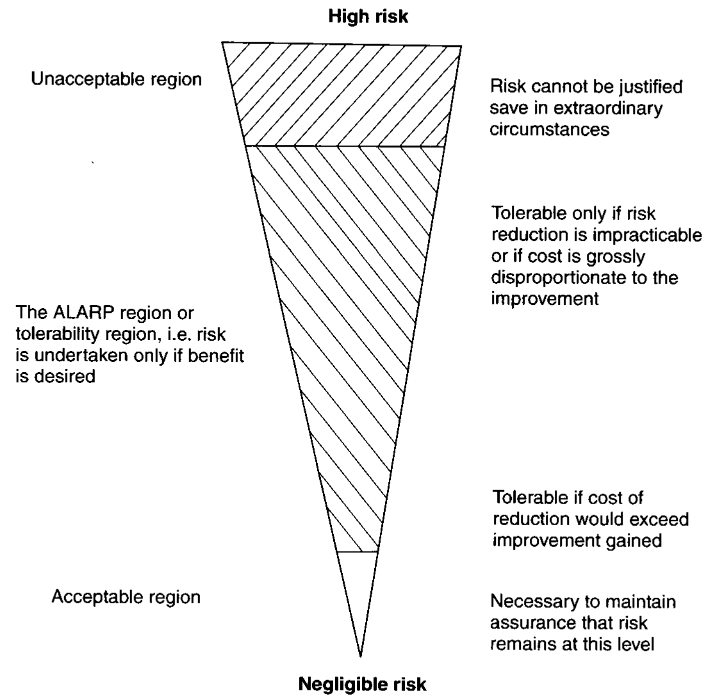


Fig. 15.19 *ALARP guidance*

information for operators or formal training, as well as equipment duplication or interlocking which physically makes dangerous operating procedures difficult.

Among the aids to reaching a decision on the acceptability of a system is the ALARP principle. This is an acronym for 'as low as is reasonably practicable'. Words which may help in the decision process are shown in Fig. 15.19. It is of little comfort to a designer to know that, despite their earnest endeavours, events subsequent to a catastrophic event will question their judgements as to what was an acceptable compromise among all the conflicting demands. That, however, is no reason for avoiding the problem.

The whole panoply of safety assessment has to be reserved for behaviour of the ship subject to major hazards such as flooding, fire and explosion, loss of steering, fracture of principal structure and capsize in a seaway. Nevertheless, the general approach to safety assessment is applicable to all equipments and systems in a ship, while inspiring the ship's company in contributing to the safety consciousness of all the crew.

Conclusion

This chapter has outlined the process of ship design. There is no substitute for the experience which application to a real ship design project can bring. Illumination and understanding derive from attempting the design of a ship in accordance with the philosophy and processes with which this book has been so sketchily dealing. It is not an arid application of engineering rote; it contains a great deal of art and artfulness, judgement and management. Beware of the facile claims of breakthroughs in ship design! Quoting from a similar engineering endeavour:

> The design required a series of difficult decisions, interrelated and far from the deductive, linear progression often imagined by people without experience in engineering.
>
> The sequence of decisions here is typical of the trade-offs between simplicity of design, availability of material and practicability of construction that make engineering an art rather than simply-or merely-applied science.'

16 Particular ship types

Passenger ships

At one time, passenger ships were typified by passage across the Atlantic. Deposed by air traffic, the long sea voyages now belong to the cruise ships of the tourist trade and little else. Almost all other passenger traffic by sea is confined now to ferries on relatively short journeys. Even the definition of a passenger ship is no longer related exclusively to those carrying more than twelve people who are not members of the crew. A maze of international regulation has grown up through IMO to present exemptions and exclusions reminiscent of the old tonnage rules. Passenger ships are certificated by the United Kingdom in accordance with the following classes:

Class I. Ships engaged on voyages (not being short international voyages) any of which are long international voyages

Class II. Ships engaged on voyages (not being long international voyages) any of which are short international voyages

Class II(A). Ships engaged on voyages of any kind other than international voyages

Class III. Ships engaged only on voyages in the course of which they are at no time more than 70 miles by sea from their point of departure and not more than 18 miles from the coast of the United Kingdom, and which are at sea only in fine weather and during restricted periods

Class IV. Ships engaged only on voyages in partially smooth waters, or voyages in smooth and partially smooth waters

Class V. Ships engaged only on voyages in smooth waters

Class VI. Ships engaged on voyages with not more than 250 passengers on board, to sea, or in smooth or partially smooth waters, in all cases in fine weather and during restricted periods, in the course of which the ships are at no time more than 15 miles, exclusive of any smooth waters, from their point of departure nor more than 3 miles from land

Class VI(A). Ships carrying not more than 50 passengers for a distance of not more than 6 miles on voyages to or from isolated communities, islands or coast of Scotland and which do not proceed for a distance of more than 3 miles from land.

The Merchant Shipping Regulations give the legal definition of many of the terms. Design of passenger ships is then dominated for each class by strict regulations concerning:

watertight subdivision
fire boundaries

freeboard
life-saving appliances
transport of dangerous goods

While basic regulations exist for each of these features, exemptions are aimed at trading off one against the other in order to accommodate different architecture. A lesser standard of watertight subdivision for example might be permitted in some classes provided more life-saving appliances were carried. These trade-offs are nowhere more complicated than in the design of RoRo ships.

In the early part of the 20th century, passengers needed to be conveyed as rapidly and as comfortably as possible between continents. Great trans-atlantic liners served this need in varying degrees of luxury and, it must be said with varying degrees of safety, at least until the 1930 Merchant Shipping Act recognized the frailties apparent from such disasters as the *Titanic.* That legislation led to ships able to survive flooding of two or more main compartments and to the provision of more acceptable firefighting facilities and life-saving equipment.

After the Second World War, there emerged slowly a market for ships which did not follow one specific line. A public demand for cruising to many different places in pursuit of holidays and cultural interests led to fleets of cruise ships. For a while, economies of scale drove the size of these ships up and up. At present the largest ships on order have a gross registered tonnage of 136,000. These ships provide a holiday experience in their own right and for some people the ports visited are of secondary importance. However, such large ships are restricted as to the ports they can visit and often embarkation and disembarkation times are long. For these reasons the medium sized ship remains popular. Nevertheless ships carrying several thousand passengers with a passenger to crew ratio of 3:1 are common. Some ships are now being designed to carry the number of people that can be carried by a large aircraft. Thus the aircraft that deposed the ocean liner are now an integral part of the cruise business. More recent ships tend to have higher speeds to enable greater distances to be covered in a given time. Some are relatively small to enable them to visit small islands away from the main tourist spots.

Safety arrangements and evacuation procedures for huge numbers need special attention. Architecture to appeal to clients has led to the introduction of massive public rooms and to atriums that create spectacular compartments through a dozen decks. These demand some close attention to firefighting and movement of large crowds. Rapid evacuation of passengers and crew from a considerable height above the sea has led to escape chutes and self-inflatable life-rafts. Disabled people require special attention, not only in an emergency but for normal movement around the ship. Swimming pools, gymnasiums, open areas for use in competition and sport are all normal requirements. These are just some of the problems that are added to the proper attention to regulatory demands in the design of such fascinating ships. While safety provision to required standards remains paramount, there arises conflict among safety and comfort; zoning of ventilation, sills to watertight doors, fire and

Table 16.1
Comparison of large RoRo and Cruise Ship

	RoRo	Cruise Ship
Length BP, m	146	224
Beam, m	26	31.5
Draught, m	6	7.75
Gross Tonnage	27000	70370
Deadmass, Tonnes	5350	7000
Displacement, Tonnes	14000	35800
Propulsion power, MW	18	28
Side Thruster, MW	1.8	9
Speed, knots	20.5	22.3
Passengers	2120	2634
Cabins	217	1024
Crew	141	920
GRT/Passenger	12.7	26.7

smoke barriers, freedom of movement for passengers and crew all lead to compromises that will have to be debated at the design stage. Nevertheless, as Table 16.1 shows, the cruise ship is relatively capacious compared with the RoRo ferry.

At the smaller end of the market, the regulations for vessels in Classes III to VI have become more stringent in the UK following the *Marchioness* disaster on the Thames. Most new designs in the future will achieve a one-compartment subdivision. Reliance on rescue from shore up to 60 miles away does demand an abundance of personal life saving appliances and a time to sink after an accident which is prolonged. Even for operation close inshore, designers should never forget that the sea is hungry, and sometimes very cold.

Ferries and RoRo ships

The concept of an integrated transport system that gave rise to container ships is applicable equally to Rollon-Rolloff ships and to ferries. It is often econ_ ical and quicker to transport people and goods from shore to shore in their own vehicles. Trains, lorries, cars, caravans and coaches may have to be embarked, conveyed and disembarked in a safe and efficient manner, while their passengers and travellers on foot require to be looked after in a manner compatible with the distance and fare.

A ship, of course rises and falls with the tide so that the facilities which permit embarkation need vertical adjustment and, indeed, restraint. Trains are long and inflexible so that long hinged bridges on to the ship will allow slow movement to matching rails in the ship. As it is embarked to one side, the ship will heel and a rapid heel compensating system must transfer fluid to the opposite side. Wheeled vehicles can be dealt with more slowly but heel is quite noticeable as a heavy lorry is embarked. Fore-and-aft transfer of weight to ensure that there remains adequate freeboard at the bow must also be available.

Stability of the ferry must, of course, be under continuous review during loading, often with the help of a portable computer.

Matching of ship and shore depends on local trading conditions, varying from a simple ramp let down on to a beach to a road system that mates with the ship, sometimes even at two levels to speed up operations. Indeed, capital investment in such facilities is a significant contributor to that economic evaluation which must precede the definition of ship and shore as an integrated system before design begins. Aspects of the ferry will be dominated by the specific requirements for its class of operations; thus, Class III or IV ferries may well be excused from supplying extensive lifesaving or evacuation facilities. Domestic facilities too will depend on length of voyage and local sea conditions.

The most common RoRo ferry has evolved during the last fifty years. It is a sad fact that regulation of safety lags much behind innovation and must rely so often on the lessons of failure. A string of disasters starting in the 1950s with the loss of the *Princess Victoria* and working their way through *Herald of Free Enterprise* and *Estonia* in the 1980s among many others, finally persuaded the authorities that regulation to improve safety must be introduced. IMO brought out STAB 90 in 1990, beginning the process of defining minimum standards for all RoRo passenger ships, for adoption early in the 21st century. Such standards are now regarded as adequate although all nations, even those who signed the Convention, have not been able to enforce them and tragedies are still occurring.

The basic problem is very simple. By their very nature, RoRo ferries need large open spaces. If, for any reason, these spaces become flooded, the integrity of the ship is threatened and foundering can be very rapid indeed. The car decks can become flooded if the doors are breached or if a side collision allows the sea to enter. The regulations now require the ship to sustain collision damage which penetrates to a depth of one fifth of the beam of the ship. Two watertight doors never to be opened unless the ferry is secured alongside must also be fitted forward and aft. Often, the inner door doubles as the ramp to the shore.

Meeting the survival requirement should the car deck become flooded is also relatively simple, although it may reduce capacity slightly. A belt of buoyancy is provided by watertight compartments each side of the cardecks above and below the normal waterline. The inner barrier provides additional protection against a collision and the contained compartments give buoyancy as the ship heels reducing the tendency to capsize. This tendency was very apparent in the *Herald of Free Enterprise* disaster. The estimation of time to founder is an important feature of the safety case and regulations require the ship to be evacuated within 30 minutes of the order to abandon ship. This requirement has a significant effect upon lifeboat arrangements, leading to a lower lifeboat embarkation deck and nested lifeboats. Figure 16.1 shows a typical RoRo cross-section. Machinery uptakes may sometimes be split to use the watertight side spaces and two funnels are not uncommon.

Many ferries are now fitted with side thrusters to assist them in coming alongside while others have adopted vertical axis propellers which serve a similar purpose. Diesel-electric propulsion using pods at the propeller have

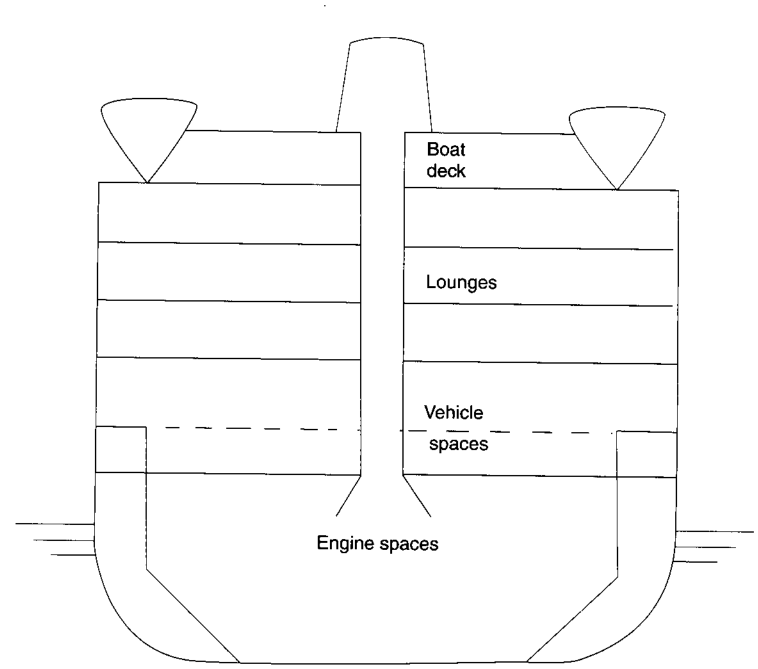


Fig. 16.1 *Typical RoRo cross-section*

also been fitted. Economic analysis has introduced a trend towards higher speeds that require gas turbine propulsion while other designs have used surface effect vehicles as RoRo ships.

Freight RoRo vehicles are basically similar to passenger ferries but are also trending towards higher speeds for economic reasons. Their cargo is somewhat more predictable and less variable in dimensions and can lead to their exclusive use on lines. Transportation of cars for trade is typical.

Aircraft carriers

The characteristics of an aircraft carrier are profoundly affected by the type aircraft that it is required to operate, which may be fixed wing, deflected jet, vertical take off or helicopter. Unless the types and numbers of aircraft are known with some precision, the aircraft carrier will be larger and more expensive than it need be; there is a high price to pay for flexibility.

Fixed wing carriers are complicated ships, often of 2000 compartments and carrying 4000 crew. As well as all the domestic, navigational and machinery requirements associated with all surface ships, the aircraft carrier must operate, direct and maintain perhaps fifty complex aircraft. Fixed wing aircraft are catapulted by one of several catapults up to 100 m long at the fore end of the flight deck while the ship is steaming head to wind. Because they normally require a length for landing not available to them on a ship, the aircraft are retarded on landing by an arresting gear; a hook on the aircraft is directed on to a wire stretched transversely across the flight deck which is connected to a damping mechanism below. For both physiological and practical reasons, accelerations and decelerations higher than 5 or 6 g cannot be achieved and this gives minimum possible lengths for catapults and for arrester wire pull-out. To give flying speed of 120 knots to a 30 tonnef aircraft, for example, a catapult would need about 30 m of constant 6 g acceleration and 30 MW power, some of which is contributed by the aircraft. Angle of descent of the aircraft and clearance over the stern, spacing and pull-out of arrester wires, centring gear and length of catapult and bridle catching gear thus all contribute in dictating minimum flight deck length.

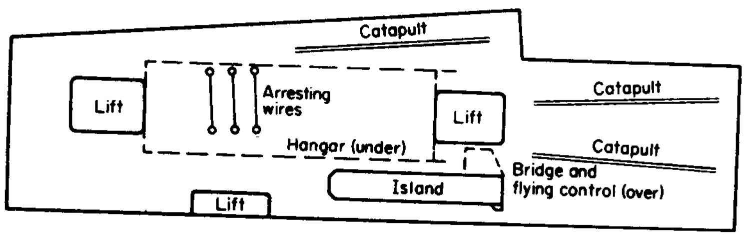


Fig. 16.2 Flight deck

Also accommodated by the flight deck are aircraft deck parks, fuelling positions, weapons areas, servicing positions, helicopter operating areas, landing aids and six or more aircraft and weapons lifts. The island itself, traditionally on the starboard side, houses the bridge, flying control, action information centre, flight deck personnel and the long range radars and communications equipment for detecting and controlling aircraft many hundreds of miles away. Layout of the flight deck, in fact, determines the length of the ship (and thence displacement) fairly closely and the naval architect usually struggles to keep island size and flight deck length to a minimum. With an aircraft specified, however, scope is limited.

The hangar below decks needs to be as wide and as long as possible and at least two decks high, even to accommodate aircraft with folded wings. Not only does this cause difficult access and layout problems for the rest of the ship, but it gives rise to some formidable structural problems, particularly if the hangar is immediately below the flight deck when the wide span grillages must support 30 tonnef aircraft landing on at high vertical deceleration.

At least thirty different piping systems are required, including flight deck fire main, fuelling, defuelling, air, hydraulics and liquid oxygen. Widespread maintenance facilities are required near the hangar for both aircraft and weapons. Underwater, a side protection system is fitted against mine and torpedo attack and armour is disposed around the vitals. The flight deck itself normally constitutes armour, although in some older carriers, this was relatively thin and the hangar was left open below it, armour being provided at hangar deck level.

Boats are stowed in pockets in the ship's side in order to keep the flight deck clear, together with items of ship's equipment, fairleads, capstans, etc. Accom-

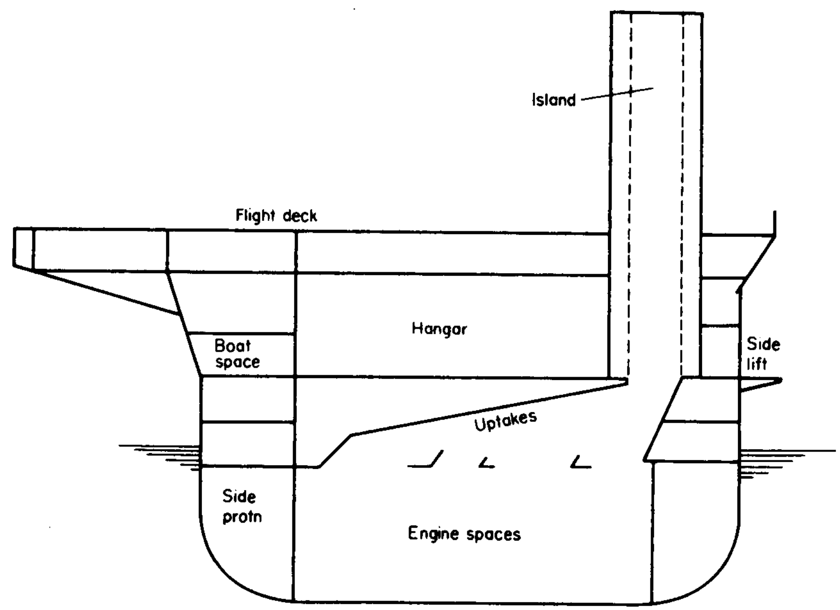


Fig. 16.3 Aircraft carrier section

modation arrangements in the ship are conventional except that additional spaces for aircrew such as briefing rooms, flying clothing cloakrooms and air intelligence spaces are needed.

Such is the power needed-over 150MW for large carriers-that steam machinery plant is usual; a nuclear reactor avoids the need for frequent refuelling. A fleet train is needed for regular provisioning, of course, but an oiler is vulnerable.

These complicated ships are expensive to build and run and are now confined to quite few countries. However, ships designed to carry helicopters and 'vertical short take-off and landing, VSTOL' aircraft are now common. Deflected jets, pioneered in the Harrier, enable such aircraft to take off and land vert~ cally, to hover and even to fly backwards. The vertical evolutions do use a lot ~ fuel which could be put to better use in extending the range. As a result, ramps have been devised of a shape which permit such aircraft to take off under their own power with a very short run. The profile of the ramp is critical in imposing a nose-up attitude without allowing the forces which the ramp imparts to the undercarriage to cause it damage. With the need for long catapults and arresting gear obviated such ships can be smaller and cheaper. The structural design is a particularly important aspect of aircraft carrier design. Because they are intrinsically asymmetric and require major discontinuities to accommodate lift wells, there is a tendency to torsional vibration with nodes at the structural weaknesses. A dynamic analysis to determine the modes is now possible.

Detection of submarine contacts by sonobuoys and the prosecution of attacks by homing torpedo can be done effectively by helicopters many miles from the parent ship. Escort destroyers and frigates are able to carry one or perhaps two helicopters of medium capability but there is a need for some ships to accommodate larger numbers. Helicopter carriers with a dozen or so aircraft can be adapted into relatively simple ships capable of arming, operating and maintaining the aircraft as well as accommodating the necessary personnel. This last task is not negligible; it is not unusual for as many as 25 people to be required for each helicopter and such ships are space demanding. The ships may also act as a garage for the maintenance of the escorts' aircraft.

One interesting way of getting helicopters or VSTOL aircraft to sea quickly without a purpose-built ship is by using standard containers. Using a normal container ship the containers may be stacked like children's building bricks to make a hangar while the containers themselves can accommodate spares, personnel, servicing facilities, communications, weapons and operations spaces. Purpose-built parts are required for the hangar roof and for bridging the gaps between containers. The total kit to achieve such a fitment is quite large and a 250 m container ship may be able to house only eight or ten aircraft, with two spots for take-off and landing. Nevertheless if there has been proper preparation in advance it can represent a very useful way of augmenting aviation capability in a national emergency by taking ships up from trade. The ships themselves will of Course need a certain amount of modification by removing vertical obstructions to safe flying. Figure 16.4 shows a possible arrangement.

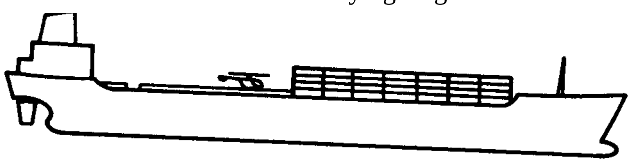


Fig. 16.4

Bulk cargo carriers

Cargo which may be carried in bulk includes oil, ore, chemicals, vegetable oils, molasses, liquefied gas, coal, grain and forest products. Even cars and containers present a homogeneous cargo whose effects upon the ship are reasonably predictable.

Economic arguments indicate that most oil should be carried worldwide in very large crude carriers, VLCCs and some 500 such vessels ply the oceans. They may carry as much as 300,000 tonnes of crude oil in holds arranged perhaps six longitudinally and three or four abreast. The block coefficient is often around 0.8. Machinery aft is usually diesel driving a single shaft and a boiler produces steam for domestic use, for heating the bunker oil and for steam cleaning cargo tanks. There is also an inert gas system for cargo tanks to prevent the build up of an explosive mixture above the cargo. There is a superstructure aft with bridge and accommodation and a central walkway along the upper deck to protect personnel from the effects of a very small freeboard. Loss by structural failure is now fairly rare even though cracking is

not and should be dealt with in good time. Most loss is attributable to collision or grounding giving rise to severe environmental pollution. In 1990, the USA introduced an Oil Pollution Act which required all oil tankers using their waters to have double skins and IMO followed in 1993. Unhappily, evidence suggests that this would not have helped to reduce pollution in many of the major environmental disasters that the world has suffered and designers have sought more effective measures.

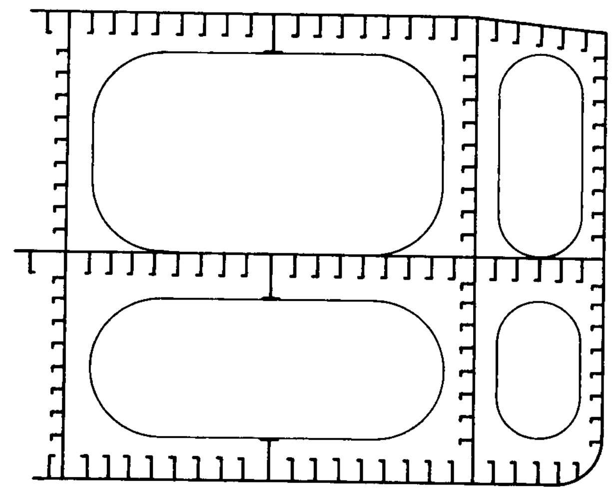


Fig. 16.5 Modern mid-deck tanker midship section

One way of containing at least some of the cargo after a grounding is to subdivide the ship by additional oiltight decks. Spaces below would be maintained by regulated air vents at atmospheric pressure so that, should the bottom be pierced, there is a pressure differential forcing the cargo inwards. It is then allowed to weir into empty tanks. There are several variations on this theme, which could work, at least partially until there is structural disintegration, eveif' in the dynamic fluid conditions that prevail. Of course, an increase in cost and a reduction in payload is inevitable.

Losses of ore carriers caused great anxieties during the 1980s. Many of them disappeared without trace, presumably by structural failure. There is evidence to suggest that the scarphing of the midships structure towards the ends into a reduced section modulus was not, in the 1970 designs, carried out with sufficient care and that local stress concentrations caused cracking which propagated fast. Some 10 to 15 per cent of the length from the stern has been shown to suffer high stresses in a seaway. Damage to structure by the huge grabs used to unload cargoes exacerbates the problem while some cargoes generate highly corrosive fluids that further damage the structure with time. These matters have led to strong pressures for more high quality steels in critical parts of the ship. More intensive survey is now adopted and cooperation within lACS hus led to

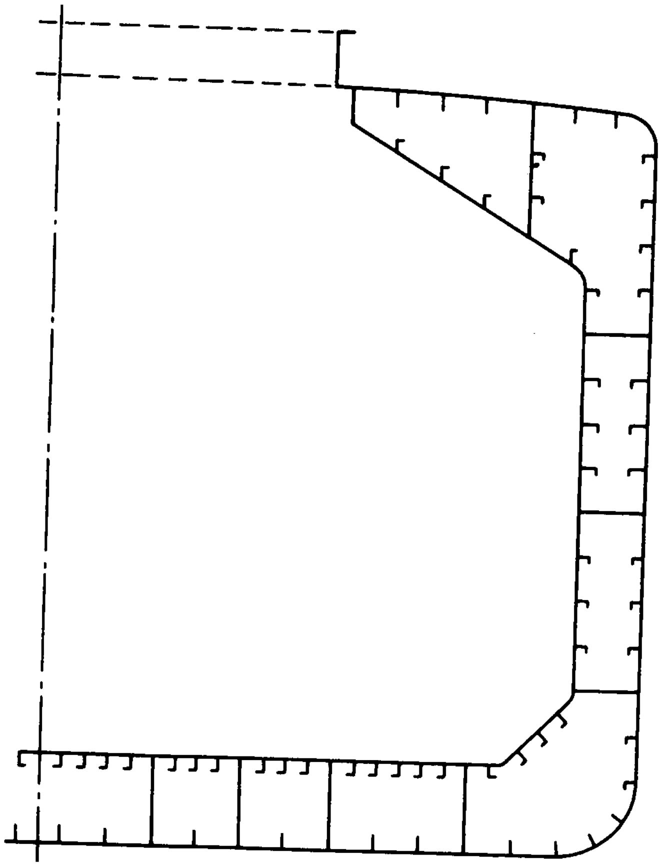


Fig. 16.6 *Traditional bulk carrier midship section*

new regulations which provide good structural management, including hull condition monitoring.

Slamming can be troublesome in bulk carriers and masters need to be advised on how to minimize these effects; it tends to be a highly tuned condition so that small changes in course or speed are usually effective in reducing its incidence. Of particular importance to the bulk carrier is the sequence of loading and unloading. It is relatively easy to cause structural damage by large shear forces between full and empty tanks as well as unacceptable still water hull flexure. Masters must be supplied with examples of these effects, often programmable readily on a microcomputer, to guide decision making. Designers may also wish to present their classification societies with a direct structural analysis rather than simply following their rules..This is entirely acceptable and can lead to improved structure.

Grain carriers, often affected by the need to sail the Great Lakes of North America are common. Grain carriers introduce special problems of stability due to the free surface effects of the grain. While shifting boards reduce the

effects, large holds make these difficult to fit; covering the surface with bagged grain damps the free surface but does not remove the problem. Masters are required to take the ullages (i.e. height above the free surface) of the grain and to apply these to diagrams supplied by the Authorities (see Chapter 14) for determining revised stability data. Expected angles of heel of the ship should not exceed 12 degrees due to possible shifting of the grain (see also Chapter 5).

A type of bulk carrier of increasing importance is the liquefied natural gas carrier. Natural gas, predominantly methane, is given off in vast quantities in a few areas of the world, particularly at oilfields. For use in other countries, it needs to be transported economically. The gas is first liquefied by compressing it and cooling it to temperatures around minus 100°C in which condition it is pumped into transit tanks. Such tanks are isolated from the ship's structure by very thick insulation and the ship is fitted with double bottom and side protection. During its passage across the sea, certain of the gas boils off naturally either to waste or it may, economically, be used to help drive the ship, perhaps by gas turbine. To minimize this loss and avoid carrying refrigeration machinery, the ship needs to be fast and reliable and the characteristics of the ship are, again, determined by the economics of operation.

Material for the liquefield gas tanks must not be brittle at these very low temperatures-which render even rubber brittle. Suitable materials are certain aluminium alloys and some nickel steel alloys. It is very important, of course, to prevent the mild steel of the ship from being reduced in temperature by leakage of the liquid, whereby it may become brittle (see Chapter 5 on brittle fracture).

Every cargo brings its particular problems but none more than special liquid cargoes such as molasses, sulphuric acid and sulphur, including thermal stresses, explosive vapours and corrosive materials.

Submarines

Submarines are vehicles designed to operate principally at considerable depth. Most applications to date have been to warships. Commercial applications have been oceanographic research vessels and small vehicles for laying piP" and servicing well-heads on the sea bed. Submarines are uneconomic for general commercial work.

Naval submarines were originally of limited capability. They were very good at attacking shipping with torpedoes, their invisibility enabling them to approach merchant ship targets unobserved. Nowadays, modern cruise missiles enable them to attack accurately land targets well inland. Other missiles mean they can engage surface ships or aircraft. The submarine is an ideal vehicle for landing small groups of special forces personnel on defended beaches. Thus the submarine has become more of a general purpose vessel than one with a single. albeit vital, mission.

Because the vessel has to operate on the surface and submerged all the usual naval architecture problems have to be studied for both conditions. Some have to be studied during the transition phase.

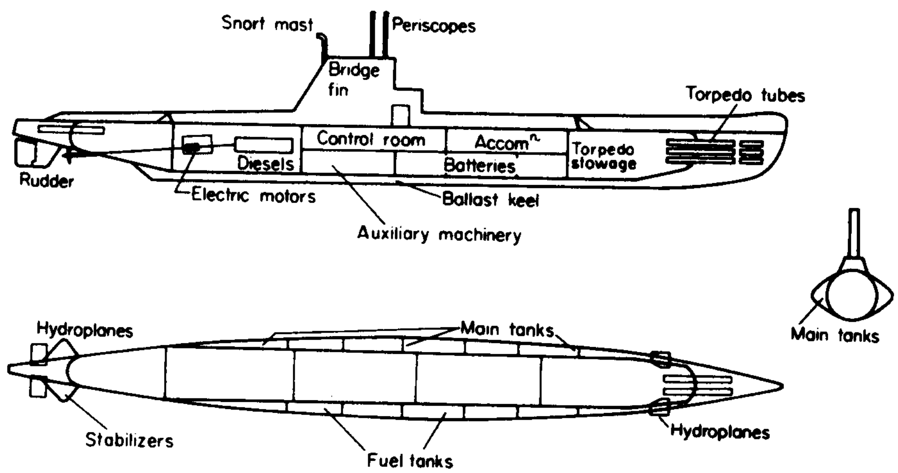


Fig. 16.7 *Diagrammatic arrangement of conventional submarine*

The general design of a conventional submarine is illustrated in Fig. 16.7. Major differences compared with a surface ship are:

(a) the shape, which is conditioned by the need to have efficient propulsion submerged;

(b) the enclosure of the main portion of the vessel in a pressure hull which is usually circular in cross-section to enable it to withstand high hydrostatic pressure at deep diving depths. The circular section means greater draught generally than a surface ship of the same displacement. It also requires that a docking keel be provided, unless special cradles are available, and a top casing for men to move around on in harbour;

(c) the hydroplanes, for controlling depth and trim angle; usually two sets are provided, one aft and forward or on the bridge fin;

(d) tanks, usually external to the pressure hull, which can be flooded to cause the vessel to submerge;

(e) a dual propulsion system. The submerged propulsion system is usually electric drive supplied by batteries and surface propulsion is usually by diesels. The batteries need frequent recharging, which means that a conventional submarine has to operate on the surface or at periscope depth for considerable periods. These disadvantages are overcome in nuclear submarines or in vessels with air independent propulsion;

(f) periscopes and sensor masts to enable the vessel to operate close to the surface;

(g) a special air intake, the snort mast, to enable air to be taken in when operating at periscope depth;

(h) special means of controlling the atmosphere inside the submarine. Apart from the normal conditioning equipment, carbon dioxide absorbers and oxygen generators are provided.

Comments on some aspects of submarine design are:

(a) Hydrostatics. Although the hydroplanes can take care of small out-ol'-balance forces and moments, the vessel when submerged must have a buoyancy almost exactly equal to its weight and B must be vertically above G. For reasons of safety, in practice the buoyancy is usually maintained slightly in excess of the weight. The capacity of tanks within the pressure hull for adjusting weight and longitudinal moment is limited, so that initial design calculations must be accurate and weight and moment control are more critical than for a surface ship. In the latter case, errors involve change of draught and trim from the design condition. If a submarine is too heavy it will sink and if too light it will not submerge. If Band G are not in the same vertical line, very large trim angles will result as B cannot move due to movements of 'wedges' of buoyancy. The term trim here is used in the conventional sense. In submarines, fore and aft angles are usually termed pitch and the term trim is used to denote correct balance between buoyancy and weight. This critical balance between weight and displacement means that if weights are ejected, e.g. a torpedo, then a carefully metered quantity of water must be taken on board immediately to compensate.

(b) Stability. The stability of the submarine for heel and depth when submerged were discussed in Chapter 4. In the submerged state, longitudinal and transverse -BGare the same. On the surface, the usual calculations can be applied but as the submarine dives the waterplane reduces considerably as the bridge fin passes through the surface. In this condition, B may still be relatively low and a critical stability condition can result.

Submarines are subject to special tests, the *trimming and inclining experiments,* to prove that the hydrostatic and stability characteristics are satisfactory in all conditions. The correct standard condition is then achieved by adjustments to the ballast keel.

(c) Strength. The pressure hull must be able to withstand the crushing pressures at deep diving depth. The hull is substantially axisymmetric to minimize bending stresses in the highly loaded circumferential hoop stress direction. In the absence of significant longitudinal bending longitudinal stiffening would b~ inefficient and the structure is treated as a ring stiffened cylinder.

An approximation to the effect of depth of operation on the weight of hull can be obtained by considering the simple case of a circular cylinder which is a good shape for withstanding external pressure. A sphere is better, but this form is used only for certain research vehicles. If stiffening is ignored and hoop stress is used as the design criterion then, for a given material, the permissible stress will be constant. Ignoring the ends of the cylinder, the thickness of hull plating, and hence hull weight, required will be proportional to pressure multiplied by the diameter. For a given diameter, the buoyancy is constant so the ratio of structural weight to buoyancy increases linearly with increasing pressure, i.e. depth. There will be a depth when there will be nothing available for payload. For a given depth of submergence, the ratio remains constant. If the diameter is allowed to increase, then at a given depth the hull weight increases as the diameter whereas the buoyancy increases as the square of the diameter. The proportion of the buoyancy devoted to structure is inversely proportional to the diameter. Other weights, e.g. machinery, crew, etc., are usually relatively less for a larger ship which should have a higher deadweight/displacement ratio.

Design is usually carried out assuming axial symmetry of structure and loads. This idealization enables approximate and analytical solutions to be applied with some accuracy. Subsequently detailed analyses can be made of non axi-symmetric features such as openings and internal structure. The dome ends at either end of the pressure hull are important features subject usually to finite element analysis and model testing.

Initial analysis of a pressure hull with heavy transverse bulkheads is as a uniformly loaded ring stiffened cylinder, the longest compartment being taken as the most critical. The maximum mean plating hoop stress Occurs circumferentially mid way between frames. The maximum longitudinal stress occurs on the inside of the plating in way of the frame, important because it is an area of heavy welding.

Buckling of the hull is possible and the following are assessed:

(a) Inter-frame collapse, i.e. collapse of the short cylinder of plating between frames under radial compression. Such a failure is likely to occur in a large number of nodes.

(b) Inter-bulkhead collapse, i.e. collapse of the pressure hull plating with the frames between bulkheads. This is sensitive particularly to the degree of out-of-circularity in construction.

(c) Frame tripping.

The design is developed so that any buckling is likely to be in the inter-frame mode and keeping risk of collapse at 1.5 times the maximum working pressure acceptably small. The effects of frames, shape imperfections and cold working residual stresses are allowed for empirically. Small departures from circularity can lead to a marked loss of strength. In one case, the pressure causing yield at 0.25 per cent shape imperfection on radius was only half that required for perfect circularity.

(d) Dynamic stability. This has already been discussed in Chapter 13 in some detail. The limited diving depth available for reasons of strength reduces the time available for corrective action should the vessel suddenly take on a bow down attitude. For example, at 20 knots and assuming that the vessel is already at 50 m depth with a collapse depth of 200 m, a 30 degree angle means that the vessel reaches her collapse depth in about 30 sec. If the depth of water available is less than collapse depth, as it would be in many coastal areas, then the time available is even less.

(e) Powering. For a given displacement, a submarine has a greater wetted surface area than a surface ship. This means a greater frictional resistance, which, for comparable conditions, means that the submarine must operate at depths where the wavemaking resistance is substantially reduced. In practice, this means operating at depths of the order of half the ship length or more.

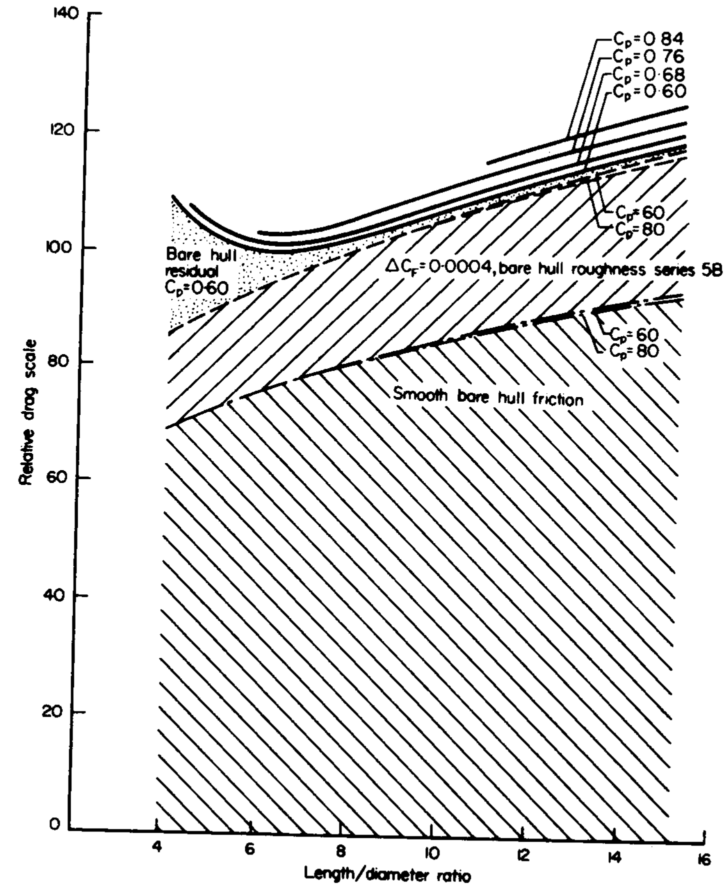


Fig. 16.8 *Variation in total resistance with length/diameter ratio and prismatic coefficient*

The importance of length/diameter ratio and prismatic coefficient on submerged resistance is shown in Fig. 16.8. The USS *Albacore* provided much useful data on submerged performance at high speed. It had a length/bearw ratio of about 7.4 compared with the ratio of about 10 in Second World War submarines. It also had a single screw which gave high propulsive efficiency.

Sources of power in a conventional submarine have for many years been the lead-acid battery driving an electric motor for underwater propulsion and the diesel engine for charging the batteries or propelling the boat on the surface. Air is drawn down the conning tower or the snort mast. Other methods have been tried, notably the steam submarines of the 1930s and the Walter turbines using oil and high-test peroxide in the 1950s. Various alternatives to lead acid are being introduced successfully to make much larger powers available. Even larger powers might derive from the research into hot batteries which naturally bring other problems or from fuel cells which benefit from the exothermic reaction of oxygen and hydrogen brought together to make water.

Over the years batteries have been improved to provide greater endurance underwater. However, the combat efficiency of a submarine is dependent on its ability to remain submerged and undetected, so much effort has recently been devoted to developing air independent propulsion (AIP) systems to provide some of the benefits of nuclear propulsion without the great expense (see Chapter 14). Solutions such as closed-cycle diesel engines, fuel cells and Stirling engines are being considered. The systems still require a source of oxygen such as high-test peroxide or liquid oxygen. Some have considered an artificial gill to extract dissolved oxygen from seawater. Fuel sources for fuel cell application include sulphur free diesel fuel, methanol and hydrogen.

Providing a submarine with a propulsion system to enable it to remain submerged for long periods necessitates making provision for better control of the atmosphere for the crew. The internal atmosphere can contain many pollutants some deriving from the new materials carried and others becoming important because they build up to dangerous levels over an extended period. A much more comprehensive system of atmosphere monitoring and control is needed than that in earlier conventional submarines.

COMMERCIAL SUBMARINES

SO far commercial applications of the submarine concept have been limited to relatively small vessels although some have dived very deep. Many have been unmanned, remotely operated vehicles. Most of these applications have been associated with deep ocean research or exploitation of the mineral wealth of the oceans. Small submersibles are also used to rescue the crews of disabled sub-

Table 16.2
A range of submersibles

Name	Diving Depth (m)	Length (m)	Width (m)	Height (m)	Displacement (Tonnef)	Crew
Deep diving manned submersibles						
Sea Cliff	6000	7.9	3.6	3.6	24.0	3
Aluminaut	4600	15.4	3.0	5.0	67.6	6
Alvin	4000	7.6	2.4	3.9	16.7	3
Sea Turtle	3000	7.9	3.6	3.6	24.0	3
Cyana	3000	5.7	3.0	2.1	8.5	3
Deep Quest	2440	12.2	5.8	4.0	52.0	4
Shinkai 2000	2000	9.3	3.0	2.9	25.0	3
Unmanned submersibles						
CURV III	6700	3.05	2.13	2.13	4.90	Tethered
AUSS	6000	5.2	1.27	1.27	1.27	Free
Argo	6000	4.8	1.0	1.18	1.59	Tethered
Angus	4000	4.27	1.83	1.52	2.45	Tethered
SAR	4000	4.57	1.22	1.22	3.63	Tethered
SBT	1400	4.0	2.8	3.0	8.00	Tethered

marines or for investigations of shipwrecks (see Table 16.2). Another .I'pltl&. tion has been for the leisure industry where submersibles take people down tu view the colourful world below the sea. These naturally tend to operate inurel"l where water clarity is high and the fish life is abundant. Submersibles cal'ryill~ 40 passengers are in service in Florida and the Caribbean.

In the above type of operations the submersible may be the only way of tackling a problem, e.g. the servicing of an oil wellhead in situ. In the leisure application. very special economic considerations apply. The carriage of bulk cargoes by submarine is unlikely to become commonplace because of the extra costs of building and operating submarines. Because the pressure hull must be cylindrical for strength efficiency, draughts for a given internal capacity are likely to be much greater than the corresponding surface ship. This complicates docking and restricts the harbours and routes such vessels can use. Special trans-shipment arrangements might be necessary. Submarine building costs are likely to be several times that of the corresponding surface ship, reducing as overall size increases. Safety would present special problems as the vessel would have very little time to respond to an emergency before exceeding its collapse depth or hitting the seabed.

To operate submerged in the ballast condition, it must be possible to introduce ballast water equal in weight to the cargo carried. This leads to a desire for a high density cargo. It would not be economic to cut large openings in the pressure hull so the cargo would ideally be capable of being loaded and discharged rapidly through relatively small openings.

Some have argued that surface units will be so vulnerable in a future hightech war that only submarines could be used with any reasonable chance of reaching the desired destination. So far such pessimism has not been borne out in the major conflicts of recent years. Also the cessation of the Cold War makes the type of scenario envisaged in such thinking less likely. Certainly the peacetime penalties associated with the construction and operation of these vessels are too great to make it likely that any country would embark upon any significant build programme.

A more likely scenario, although one not yet accepted, is the use of commercial submarines to obtain the capability to operate under ice. This might be to exploit minerals on the ocean floor or to obtain access to areas normally cut o~ by extensive ice fields. Such vessels would need to be nuclear powered or use some other form of air independent powering.

Container ships

There has been a revolution in the transportation of goods throughout the world. Goods may now be collected at their point of origin by lorry or train and taken to the port for sea transit. There they are deposited within a system of gantries or cranes which take them to the ship or to temporary storage until they can be embarked. The containers are standard worldwide and are called TEUs. Millions of the TEUs are available for hire in the knowledge that they will fit into ships specially built to receive them and take them to their destinations, where the reverse process takes place.

The ships themselves adopt various systems to hold the containers. The ships are like hollow shoeboxes, stowing containers below a single deck which has very large hatches. More of them may be carried above the deck in stacks. There has evolved also an open container ship constructed like a double-skinned U without a deck. The containers may be locked together by fittings, such as twist locks and are lashed so that they are unable to move even in severe weather. Alternatively, ships may be fitted with long vertical stanchions throughout so that a container can be housed between four of them. Fig. 16.9 shows a typical cross-section of an open container ship.

A huge range of ships now exists. Small coasters may carry a few containers on deck while at the other end of the scale,' ships 350 metres long carrying 10,000 containers are not uncommon.

In large ships there may be 20 containers athwart ships in stacks 20 high, of which a quarter at least may be above deck. Closed ships may also have stacks five or six high above the hatches and lashed in various ways to avoid movement in bad weather.

Regulations have, of course, evolved to make these ships relatively safe and losses of complete ships are rare. Containers, however, are regularly lost overboard and present a significant hazard to small ships, especially yachts, and a drain on the insurance market. Forces on the lashings in rough weather are high and the efficiency of the devices depend much on the vigilance of the crew (twist locks alone are insufficient).

Anxieties over safety have been expressed by Vossnack who points out that the statutory freeboard is extremely low, thereby keeping the underdeck tonnage (and harbour dues) low. As shown in Fig. 16.9 this gives an angle of deck edge immersion and uncontrollable flooding often around thirty degrees. Moreover, minimum crew numbers based on gross tonnage raise doubts about their adequacy in foul weather.

Further development of the carriage of standard packaged cargo such as motor vehicles may be expected. There have been studies to provide a faster service across the oceans. Demands on the design for high speed ships could be considerable in terms of machinery and air intakes for gas turbines which would reduce the number of packages that could be carried. Structure to deal with the enhanced impact loads at the bow and on the bottom would also have to be more substantial. Of course, the final decision would rest upon the predicted economy of trading off an amount of cargo against its earlier delivery.

Frigates and destroyers

These vessels cover a range of displacement from about 2500 tonnef to 6000 tonnef while having the same general roles. Lengths will be in the range 100 to 150m, with length to beam ratio of about 8:1. The larger ships can fulfil more of these roles and operate effectively in more severe sea conditions. The two titles are imprecise, the word 'destroyer' originating as 'torpedo boat destroyer' a hundred years ago.

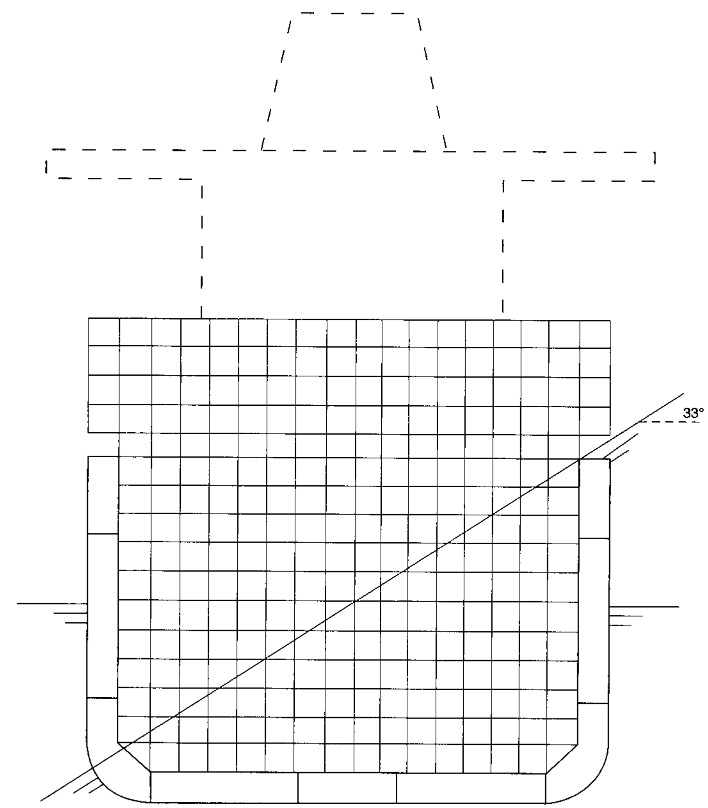


Fig. 16.9 Open container ship cross-section

The roles covered by these ship types include:

- Anti-air Warfare (AAW)
- Anti-submarine Warfare (ASW)
- Anti-surface Warfare (ASuW)
- Task force protection
- Self defence
- Shore bombardment

Because they cover a wide range of functions these vessels are often described as 'maids of all work'. All carry a gun for shore bombardment and for use against pirates or drug runners. Some are intended to provide one primary capability with limited secondary abilities in other roles. Compared to general purpose ships these 'specialist' vessels have the advantages that:

- individual ships can be smaller;
- each ship is less expensive and more hulls can be afforded on a given budget enabling a military presence to be exerted in more areas at anyone time;

- the smaller ships tend to have lower signatures;
- the loss of one ship is less damaging to the total military capability of the task force although the loss may be critical to the particular mission being undertaken.

The advantages of the general purpose ship are:

- if both AAW and ASW functions, say, are needed then one ship suffices which is less expensive than two specialist hulls;
- the larger ship provides a more stable platform for the weapon systems and those systems are less likely to be degraded by adverse sea conditions;
- there is more scope for providing some duplication of, or protection to, vital servIces;
- the manpower requirement for a given total military capability is less.

The following remarks relate mainly to the larger ships in the range and some features will not be provided in smaller hulls. A typical profile is shown in Figure 16.10.

For their size, these vessels have a high level of military capability. Indeed, some 60 per cent of the cost of a frigate is devoted to its fighting capability, compared with 25 for its mobility function and 15 for its float function. Because of this, careful attention must be paid to their vulnerability. Susceptibility to attack is reduced by giving them low signatures-radar cross-section, acoustic, infra-red, and magnetic. They cannot be made invisible but low signatures make their initial detection more difficult, makes it harder for incoming missiles to lock on to the target and makes it more likely that countermeasures will be effective.

The design must cater for some enemy weapons striking the ship. The most likely attack scenarios must be analysed and the layout and structure arranged

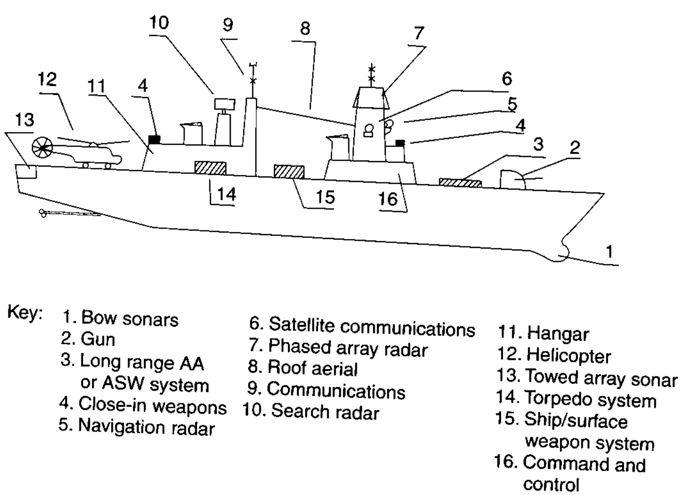


Fig. 16.10 Typical frigate profile

so as to limit the extent of damage and increase the chances of the ship bein~ able to maintain at least partial weapon capability for those weapons needed for the mission. Power supplies and other vital services must remain available. Typical analysis methods were touched upon in Chapter 5. Zoning can be used to reduce the spread of flooding and fire and to make vertical sections more autonomous as regards most essential services. Double skin bulkheads can help contain blast and reduce splinter penetration. Box girders under the upper deck can increase the residual longitudinal strength after damage and provide protected runs for vital services. Siting important compartments low down, or well away from the ship side, makes them less vulnerable. Providing separate machinery spaces, separated if possible, for each shaft reduces the risk of the ship being left helpless in the water.

One lesson that has been painfully learnt many times is that last minute changes to the design to reduce procurement cost or size is very likely to increase vulnerability to attack.

Vessels intended to act as escorts must be capable of relatively high speed, perhaps 28 to 30 knots. They may need to change their position in the task force screen or regain their position in the screen after prosecuting an anti-submarine attack. However, they are likely to spend a lot of time cruising at economical speeds, perhaps 14 knots. To provide reasonable economy at cruising speed combined with ability to go fast, these ships usually have a combined machinery plant. As described in Chapter 14, these typically involve gas turbines for the high speed in addition to, or in place of, diesels provided for cruising. A 6000 tonnef ship will have machinery giving some 40 MW. Block and waterplane coefficients will be about 0.50 and 0.75 respectively. To provide good manoeuvrability twin shafts and rudders are often fitted.

The weapon systems must be chosen so that the ship can act independently or as part of an integrated task force, providing defence in depth. In a task force, for instance, an aircraft carrier would provide the first line of defence from air attack by its fighters. Helicopters, from the carrier or the escorts, would provide long range anti-submarine defence. Progressively the long range, medium range and finally the close-in defensive weapon systems come into play. The escort's duty is to provide cover for the ships it is escorting and then for itself. At so~ point the command must decide whether, or not, to deploy decoys to seduce tne incoming attack. Computer-aided command systems help to ensure good, timely, decisions are made.

A wide range of communication frequencies are required and the ship will usually have a roof aerial and excite various structures such as masts. To meet the separation and height requirements for communications, the superstructure is often in two main sections. The space in between can be utilized for replenishment at sea and weapons.

High speed small craft

There is considerable scope for debate as to what is meant by both 'high speed' and 'small'. In this section the boundaries are drawn quite widely so as to embrace a number of interesting, and often technically challenging, hull configurations and propulsion systems. Each was introduced to overcome problems with earlier forms or to confer some new advantage. Thus catamarans, and other multi-hull configurations, avoid the problem of loss of stability at high speed suffered by round bilge monohulls. They also provide large upper deck spaces. Hydrofoil craft reduce resistance by lifting the main hull clear of the water. Air cushion vehicles give the possibility of having all the craft clear. Apart from reduced resistance this provides a degree of amphibiosity. The effect of waves on performance is minimized by the Small Waterplane Area Twin Hull (SWATH) concept. Some designs are tailored specifically to reduce wash so that they can operate at higher speeds in harbours or on waterways.

The choice of design must depend upon the particular requirements of the service for which it is intended. In some cases the result is a hybrid and the number of possible permutations is very large. Also, although most applications of these concepts have been initially to small craft some are now appearing in what may be termed medium size, especially for high speed ferry service. For simplicity, in the following sections the concepts are dealt with individually.

MONOHULLS

Most high speed small monohulls have until recently been hard chine forms. A notable exception were the German E boats of the Second World War. With more powerful small engines, round bilge forms have been pushed to higher speeds and have experienced high speed stability problems. For the hard chine forms, greater beam and reduced length give improved performance in calm water but lead to high vertical accelerations in a seaway. Their ride has been improved by using higher deadrise angles leading to a 'deep vee' form. This form was used, for instance, in the Atlantic Challenger *Gentry Eagle.*

Current practice is generally to favour round bilge for its lower power demands at cruising speed and for its seakindliness, but to move to hard chine at Froude numbers a little above 1.0 because of the stability problem. One advantage of the round bilge form in seakeeping is that it can be fitted with bilge keels much more readily than can chine forms.

MULTI-HULLED VESSELS

There are many applications: sailing catamarans, off-shore rigs, diving support vessels and ferries. The concept is not new. Two twin hulled paddle steamers of about 90 m length were built in the 1870s for cross channel service. One had two half hulls connected by cross girders and driven by paddle wheels placed in the parallel sided tunnel between the hulls. The other had two complete hulls. Both ships had a good reputation for reduced rolling, rolling only 5 degrees when other ships rolled 15 degrees.

The upper decks, spanning the two hulls, provide large areas for passenger facilities in ferries or for helicopter operations. In research vessels or mine countermeasure vessels they provide space for deployment of towed bodies of various kinds. General comparisons with monohulls are difficult because it depends whether such comparisons are made on the basis of equal length,

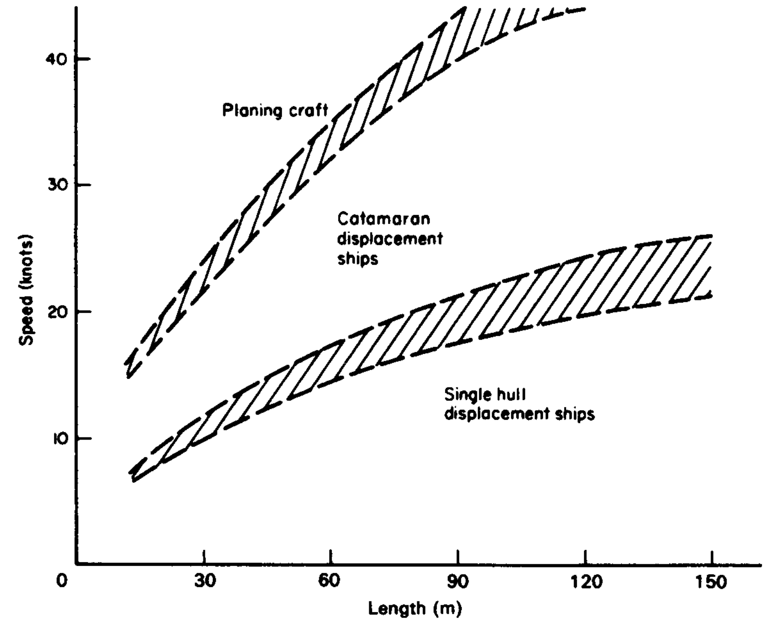


Fig. 16.11 *Areas of favourable operation*

displacement, or carrying capacity. What is needed are two design solutions each meeting the owner's requirements. The comparison in Fig. 16.11 is based on results for hulls of the same length and draught, the beam of the monohull being twice that of each catamaran hull. This is unfair to the mono hull. The greater wetted area of the catamaran leads to increased frictional resistance but their relatively fine hulls lead to reduced wave resistance at higher speeds. There will be interference effects between the two hulls. These will be less at high separation but this may make docking difficult and lead to excessive transverse stability. A reasonable separation of the hulls is about 1.25 times the beam of each. Generally the manoeuvrability of multi-hulls is good.

The increased transverse stability and relatively short length mean that *gooff'* seakeeping is not their strongest point. Improvements in this respect have been obtained in the wave piercing catamarans developed to reduce pitching, and in the SWATH designs where the waterplane area is very much reduced and longitudinal motions can be reduced by the use of fins or stabilizers if necessary.

Multi-hull designs suffer from a relatively high structural weight and to preserve payload some designs use aluminium to reduce structural weight. Wave impact on the cross structure must be minimized and high freeboard is needed together with careful shaping of the undersides. SWATH ships, because of their very small waterplane area are very sensitive to changes in load and its distribution. A system of water compensation is needed and this ballasting system can help mitigate heel due to damage leading to partial flooding of one hull.

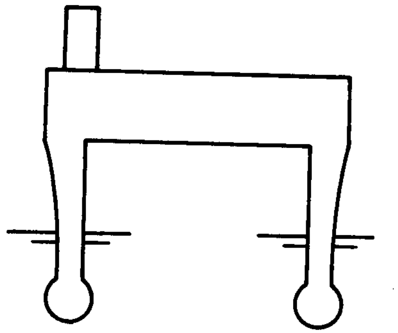


Fig. 16.12 SWATH concept

Propulsion of SWATH ships clearly invites a prime mover in each pod, or at least a propeller on each. For ships below about two thousand tonnef, the walls are insufficiently wide to permit the passage oflarge prime movers and designers have to conceive means of developing the necessary power in the 'tween decks and delivering it either as jet propulsion above water or to propellers on the ends of the submerged pods. Bevel gearing, conventional or superconducting electrical devices and hydraulic motors are all possibilities, although the driving motors themselves may not be readily removable for refitting.

In recent years, even for ships of significant size, such as frigates, considerable interest has developed in trimarans which have a long slender central hull with two narrow side hulls. The advantages claimed for this form are:

- reduced resistance and hence power for a given speed. (Said to be about 18 per cent less power for 28 knots in an escort sized vessel.) Greater fuel economy;
- improved seakeeping performance at high speed. Operational in higher sea states;
- large deck area, improved stability and reduced motions for helicopter operations;
- increased directional stability;
- better top weight growth margins.

Several studies have shown this configuration to have advantages for a wide range of applications from quite small ships up to aircraft carriers and cruise ships. To prove the validity of the concept the UK MOD decided to invest in a 97 m, 1100tonnef displacement, demonstrator vessel, which was launched in 2000. This is RV *Triton* with an overall beam of22.5 m, main hull beam 6m, side hull beam I m, and maximum draught 3.2 m. Powering is diesel electric and maximum speed 20 knots. *Triton* was built to DNV High Speed and Light Craft Rules.

SURFACE EFFECT VEHICLES

Vessels which benefit from an aerostatic force are called variously cushion craft, ground-effect machines, hovercraft, surface-effect ships and sidewalls. The

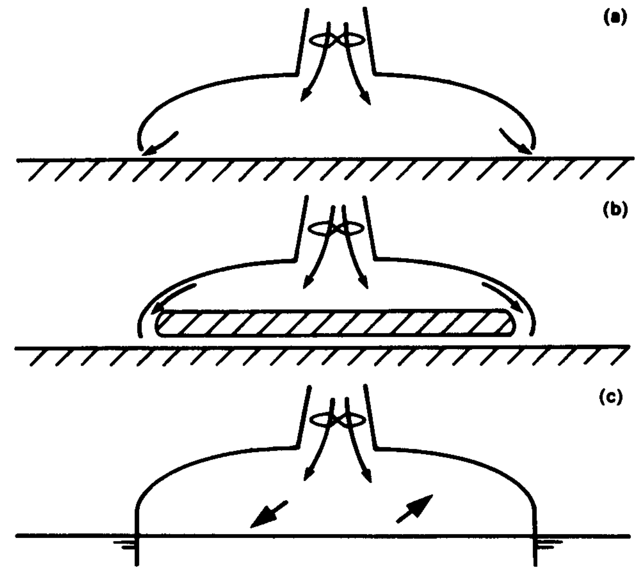


Fig. 16.13

aerostatic force is generated by a downward current of air creating an air cushion beneath the craft of which there are three general types:

(a) plenum chamber craft;
(b) peripheral jet craft;
(c) sidewall craft.

The plenum chamber craft is typified by the lawnmower of that design. Air is maintained in a plenum chamber and escapes around the periphery. Some rudimentary theory can be deduced to give an idea of the importance of the various parameters.

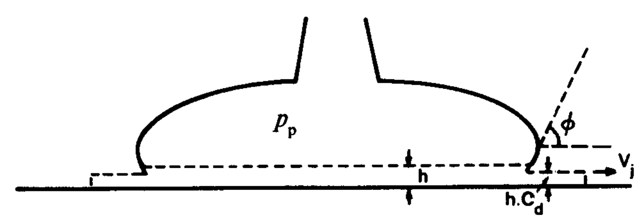


Fig. 16.14

The potential represented by the gauge pressure in the plenum p_p is converted, according to BernouilIi's law into the kinetic energy of discharge. Assuming that the discharge velocity V_j is large relative to the plenum air velocity,

$$p_p = \frac{1}{2}\rho V_j^2$$

The rate of mass flow of air escaping from the periphery of length l is

$$\dot{m} = lhC_d \rho V_j$$

C_d is a coefficient of discharge which is in practice dependent upon the angle ϕ. In a steady state the weight of the vehicle W is equal to the aerostatic force F,

$$W = F = p_p A = \frac{1}{2} \frac{\dot{m}^2 A}{\rho l^2 h^2 C_d^2}$$

A is the planform area. For a circular body which gives the largest ratio of planform area to periphery.

$$W = \frac{\dot{m}^2}{8\pi C_d^2 \rho h^2}$$

Thus the vehicle weight W can be supported by a fan whose necessary capacity rh diminishes with the clearance h over the surface. Moreover if h is decreased during operation the aerostatic force exceeds the weight so that the body is restored to the equilibrium position, i.e. there is vertical equilibrium. By a similar argument it is clear that there is also stable equilibrium if tilt about a horizontal axis occurs.

The peripheral or annular jet craft is more common because the air flow is more controllable. Rudimentary theory is rather less accurate. However, the value of a small value for h remains and the designer is faced with the problem of achieving a good lift using a small ground clearance yet needing a large ground clearance for the avoidance of obstacles and at sea, waves. This is overcome by making the lower part of the craft elastic using a heavy rubber skirt. Much research has been needed to produce skirts which are adequately robust. Truly amphibious craft result.

Because the hovercraft is above the water it has a low lateral resistance to disturbance by wind. If it is driven by air propellers they may have to be vectorable to control the positioning in wind and the stability in manoeuvre has to be the subject of study much like that of an aircraft. Large air rudders are consequently not unusual. Where a high degree of lateral stability is needed the two side walls of a rectangular hovercraft are extended into the water. The two ends of the craft remain sealed by rubber skirts to contain the air cushion. Such sidewall hovercraft, while no longer amphibious, nevertheless retain many of the advantages of the true hovercraft. Moreover, if the walls are now thickened, they provide a vertical buoyancy force so that the aerostatic force need not be so large. The designer must effect the compromise among these features which suits the particular needs.

As a craft hovers over the water, there is an indentation in the water which obeys Archimedes' Principle, i.e. its volume multiplied by water density is equal to the weight of the hovercraft. When the craft moves, the indentation moves with it causing transverse and divergent wave systems and a wave resistance just like a displacement ship. Sea friction is of course much reduced although there is some increase in the air resistance and an addition due to the dipping skirt.

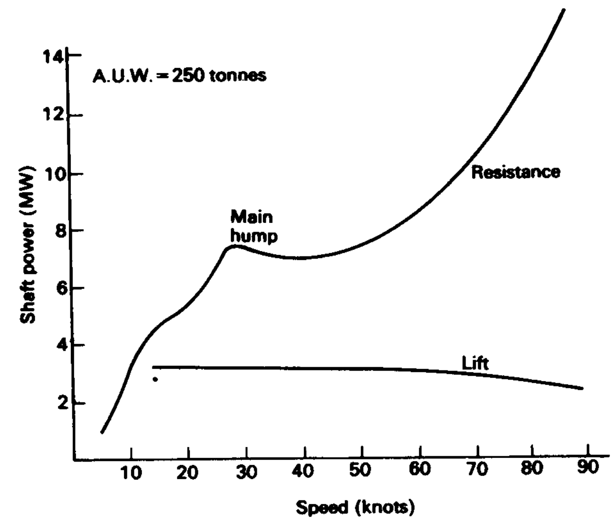


Fig. 16.15

As the craft increases speed, there comes a time when the indentation cannot properly keep up and the wave resistance enjoys a sudden reduction. The total resistance of the craft is characterized by Fig. 16.15.

High speed becomes readily possible with the sorts of power units that can be accommodated, representing one of the craft's major advantages. The resistance to motion of a hovercraft has in fact three components, each requiring study:

(a) aerodynamic resistance which varies as (velocity)2 and includes components for both the vehicle and the cushion itself;
(b) wave-making resistance which has a peak at low speeds and then falls away to a negligible value;
(c) momentum resistance which varies linearly with speed. This resistance arises from the fact that the air drawn into the craft leaves it at zero velocity relative to the craft and has therefore experienced an overall change of momentum which is proportional to the craft's velocity.

Another of the important advantages of the hovercraft over displacement craft is its relative invulnerability to underwater explosion, making it a good candidate for minehunting duties. Like the hydrofoil, its payload is relatively small and aluminium alloy aircraft standard construction is often advisable, especially in small craft. With their high power-to-weight ratio, gas turbines are often preferred for the propulsion units, both for the lift fans and the driving engines, although high speed diesels are not uncommon for craft operating at around 40 knots.

Seakeeping is generally poorer at the same sea state than for many other types of craft. Limiting sea states for various types of craft are shown roughly in Fig. 16.16.

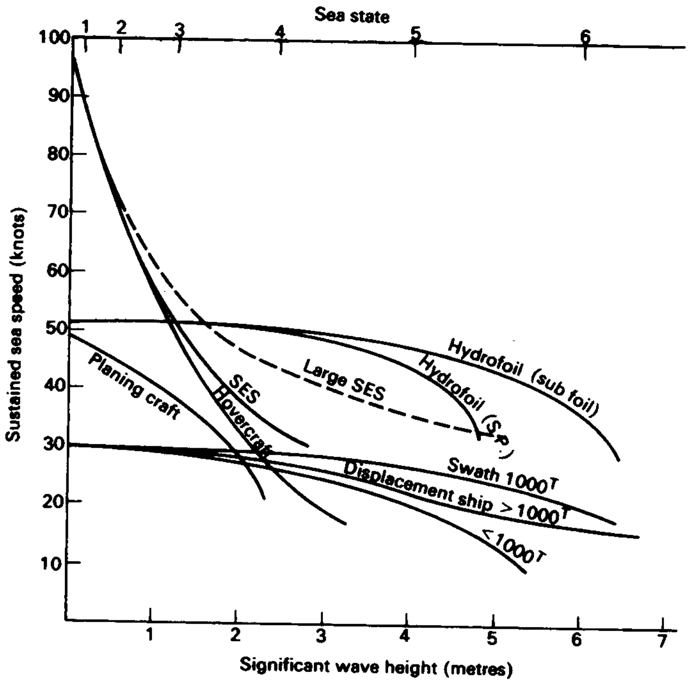


Fig. 16.16

HYDROFOIL CRAFT

A hydrofoil moving at speed through water can generate considerable lift, and if an efficient cross-section is chosen the associated drag will be relatively low. If hydrofoils are fitted below a conventional high speed craft, they generate

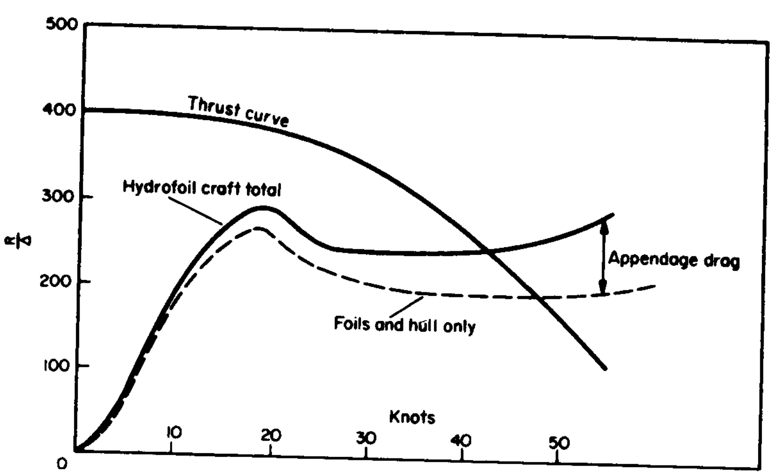


Fig. 16.17 Resistance curve for hydrofoil craft

increasing lift as the speed increases, the lift being proportional to the square of the velocity. If the craft has sufficient power available, there will come a time when the lift on the foils is sufficient to lift the hull completely clear of the water. Having lost the resistance of the main hull, the craft can accelerate until the resistance of the foils and air resistance absorb the power available. A typical curve of *R/* t1 against *V/* J*L* is shown in Fig. 16.17. The hump in the curve is associated with the very high wave resistance experienced just before the hull lifts clear of the water.

After the hull has lifted clear of the water, the lift required from the foils is constant. Thus, as the speed increases further, either the angle of incidence of the foil must reduce or the immersed area of the foil must decrease. This leads to two basic types of foil system, viz.:

(a) *Surface piercing foils* in which, as the craft rises higher, the area of foil immersed reduces as it passes through the water surface;
(b) *Completely submerged, incidence controlled foils* in which the foils remain always submerged and the lift generated is varied by controlling the angle of attack of the foils.

These two systems are illustrated in Fig. 16.18.

Longitudinal balance must also be maintained, and it is usual to have a large foil area just forward or just aft of the longitudinal centre of gravity with a small foil at the stern or bow respectively. Any ratio of areas is feasible provided the resultant hydrodynamic force acts in a line through the c.g. The planform geometries are also illustrated in Fig. 16.18.

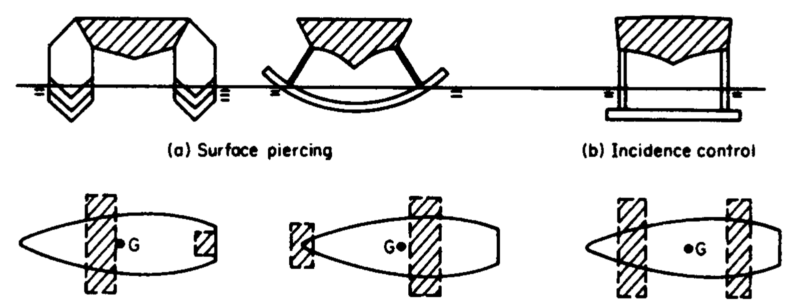


Fig. 16.18 Basic foil geometries

So far a calm water surface has been assumed. To understand what happens in waves, consider a surface piercing system as in Fig. 16.19. As the craft runs

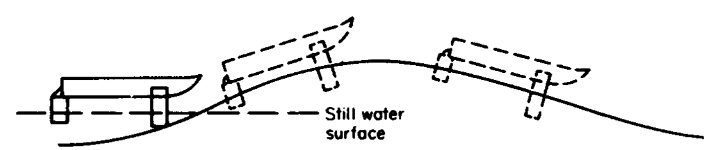


Fig. 16.19 Surface piercing system in waves

into the wave surface, the water level rises on the forward foil. The lift on the forward foil increases and this has the effect of raising the bow, keeping it clear of the wave surface. Having passed the crest of the wave, the process is reversed and the craft more or less 'contours' the waves. The more rapid the change of lift with draught on the foils the more faithfully will the craft follow the wave surface. By adjusting the rate of change, the movement can be lessened, giving a smoother ride but a greater possibility of the craft impacting the wave surface.

With the fully submerged foil system, the foil is unaware of the presence of the wave surface except through the action of the orbital motions of the wave particles. Thus, to a first order, this type of craft can pursue a level path which has attractions for small wave heights. In larger waves, the lift on the foils must be varied to cause the craft partially to contour the wave profile. In a small craft, the variation can be controlled manually but, in craft of any size, some form of automatic control is required which reacts to a signal from an altitude sensor at the bow.

It follows, that the same process which causes the craft to respond to variations in height of the water surface also provides the craft with a measure of trim stability. Roll stability will be present in a surface piercing system if the lift force which acts as the craft rolls, intersects the middle line plane of the craft above the vertical C.g. With a fully submerged foil system, roll stability is provided by means of flaps or ailerons which act differentially on the two sides of the craft in such a way as to provide a moment opposing the roll angle. This, again, is controlled by signals produced by a stable element in the craft.

Both types of hydrofoil have operated successfully for many years. Their role needs to be carefully tuned to their characteristics because, like most high performance craft they are not cheap either to buy or to run. High-speed passenger traffic in relatively calm water-up to sea state 4 or perhaps 5-has proved profitable while a 'presence' role in offshore surveillance may also be an important application. At high sea states, should the craft for any reason come off its foils, it is sometimes difficult to get up again and the craft is left wallowing in some discomfort. Impact with the water at speed should the craft come off its foils can be severe and the fore ends of these hydrofoil vessels need special strengthening and good subdivision. Aircraft standard construction is necessary in order to preserve a worthwhile payload. Aluminium alloy and fibre reinforced plastic are common.

Propulsion by water jet above water at top speeds is surprisingly efficient. The jet may be created by a diesel or gas turbine-driven high-performance pump. This avoids the need for bevel gearing for a drive down the struts or the highly angled shafting for a drive by a propeller in the water. Wind propulsion of a hydrofoil craft offers a fascinating challenge to any enthusiastic naval architect.

INFLATABLES

Inflatables have been in use for many years and, with a small payload, can achieve high speeds. The rigid inflatable is used by the Royal National Lifeboat

Institution. The rigid lower hull is shaped to make the craft more seakindly und the inflatable principle safeguards against sinking by swamping.

The RIB concept continues to develop rapidly and the craft are widely llsed by commercial firms, the military and other government departments. They arc available in lengths up to about 16m with speeds of up to 80 knots, although most operate in the range 30 to 40 knots. Petrol and diesel fuelled in-board or out-board propulsion units are common and some utilize waterjet propulsion. Single hull and catamaran versions are produced. The larger units come with wheel houses and in some cases are in competition with the fast planing craft.

The early RIB was a wooden hulled boat surrounded by an inflatable tube. The hull is now usually fabricated in GRP or polyethylene. A lot of research has gone into developing strong, durable materials for the collars. For further safety the collars are sub-divided. Besides being used in the leisure industry, the speed of RIBs makes them attractive to the military, the police and coastguard for life saving and for intercepting smugglers and gun runners. The offshore industry also makes wide use of them for diving work. Many are certified for SOLAS requirements as fast rescue craft.

COMPARISON OF TYPES

All the types of vessel discussed in this section have merits and demerits. A proper comparison can only be made by producing design studies of each to meet a given requirement and, as said earlier, the best solution may be a combination of more than one concept. Some requirements may point directly to one form, e.g. a landing craft capable of running up onto a hard surface may suggest an air cushion vehicle. This, however, will not be the usual situation.

Many of the craft in use today of these types are passenger carrying. The vast majority of operational SESs are used commercially for fast passenger transport and that, with speeds of over 40 knots commonplace, services can compete with air transport. Hydrofoils also enjoy considerable popularity for passenger carrying on short routes, e.g. the surface piercing Rodriguez designs and the Boeing Jetfoil with its fully submerged foil system. Catamarans are much used as high speed passenger ferries. Because of this common characteristic for carrying people, Fig. 16.20 presents an analysis of the BHP jpassenger plottflft against speed. This shows that hydrofoils generally need more power than catamarans and SESs and that SESs are very economical at high speed.

Offshore engineering

The spectacular recovery of gas, oil and minerals from the sea has presented naval architects with a fascinating range of problems. Drilling rigs, permanent platforms, buoyant terminals, submersible search vehicles, underwater habitats and many different servicing ships have all had to be designed to serve the industry.

In shallow waters drilling towers with their working platforms are often sunk on legs to the sea bottom, there to be secured by piles or mooring devices.

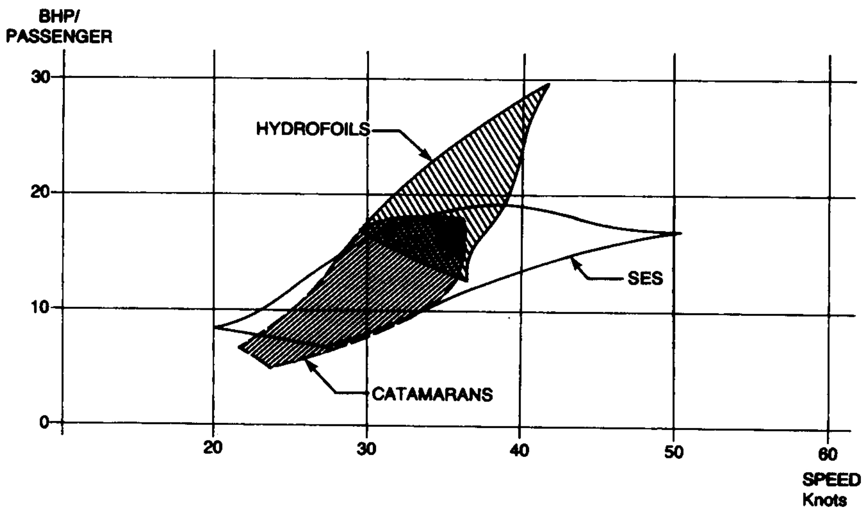


Fig. 16.20 BHP per passenger

Application of the simple hydrostatic and structural theories relevant to any floating body is straightforward.

Deeper waters and weather as fierce as the North Sea demand sterner measures. Two types of rig now predominate in such conditions, the one for exploration and the other for permanent mooring over the wellhead. The most favoured of the exploration rigs is now the semi-submersible. This comprises often two underwater cylinders each surmounted by two, three or four columns which support the drilling platform a long way above water level. Sometimes the cylinders are bent round to form a horizontal ring. Supporting a total payload of several thousand tonnes of pipes and mud they may readily exceed twenty thousand tonnes in displacement. The waterplane comprises four, six or eight separated shapes which importantly affect the hydrostatic behaviour of the rig including its static stability. The horizontal submerged cylinders are deep enough to be affected little by the surface waves and the inertia of the whole system is huge so that motion is not a problem. Structural strength, on the other hand, presents important difficulties due to both wind and wave dynamic loading and due to a wave crest above one cylinder with a trough above the other. Dynamic structural behaviour is in fact very important and the modal patterns of the structures under fluctuating excitation by the sea must be determined.

Such exploration platforms need to be mobile, often under tow by tugs but some of them are self propelled. More importantly they need to be kept still over the wellhead and in consequence, they employ vital positioning and control systems. These often comprise groups of propellers in tunnels or in retractable vectorable housings all controlled from computer systems fed by sensing devices. In addition, the rigs need to be hotels, to carry helicopters, drilling equipment, pipe racks, mud tanks, cement tanks, firefighting and life-saving equipment and diving facilities.

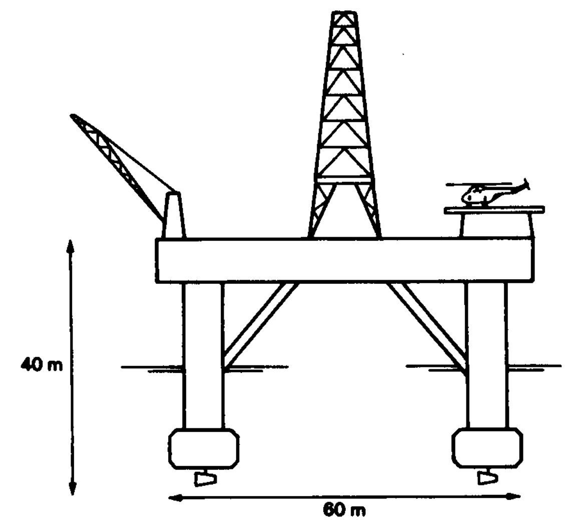


Fig. 16.21

Similar platforms with rather less elaborate arrangements can be used over the top of completed wells or, more usually groups of wells which have been capped to act as an oil terminal. Such a semi-submersible would be fixed in position by mooring cables and anchors spread out on the seabed in all directions. In preference, the oil industry today often employs the tension leg platform. This is a pontoon which would otherwise be floating freely if it were not held down by groups of hawsers fixed to the seabed. The buoyancy of the pontoon increases with the rise of tide so the pontoon has to be quite deep. The hawsers are sometimes spread to minimize lateral movement. Living quarters and the working platform are elevated high above the pontoon and the waves by lattice structures.

Vast underwater oil storage tanks are necessary where it is not possible to pipe the products ashore directly. Groups of capped wells feed the storage tank which disposes of its oil to a terminal at a sufficiently safe distance for btIIt tankers to moor and take it on board. These terminals are not just simple buoys. Hostile seas cannot be allowed to interfere with the taking up of the oil which is made as automatic as is possible. The terminal might comprise, for example, a large horizontal vee into which the tanker nestles, a tall tower carrying the oil hose surmounting it.

Oil and gas pipes on the sea bottom have to be inspected regularly. From the outside this may be done by a television camera mounted on a mobile saddle over the pipes or from a submersible which may be manned. Inside the pipes inspection is performed from vehicles called slugs which may record the condition of the welds and pipe material over very many miles, logging its position with great accuracy. On the surface many different types of ship are required to service and protect the rigs, fight fires and provide search and rescue. Compression chambers on board ship or sea bottom, or habitats for housing groups of people allow divers to remain under pressure for several days, avoiding the prolonged process of decompression after a single dive.

Recovery of minerals from the seabed has hardly begun on a major scale because it is not yet an economic venture. When it does become so, there will emerge a need for special types of vehicles of all descriptions, providing yet another wealth of interest for the naval architect.

Tugs

Tugs perform a variety of tasks and their design varies accordingly. They are needed to pull or push dumb barges or pull drones in inland waterways; they are needed to pull or push large ships in confined waters and docks, and they are needed to tow large ships on long ocean voyages. Concern for the impact on the environment of an incident involving spillage of oil from a tanker (or, indeed, any other hazardous cargo or normal bunker fuels) has led to the concept of the escort tug. Tugs are broadly classified as inland, coastal or ocean, the largest of the ocean tugs approaching 1000 tonnef in displacement. They are capable often of firefighting and salvage duties and may carry large capacity pumps for these purposes.

Essentially a tug is a means of applying an external force to the vessel it is assisting or controlling. That force may be applied in the direct or the indirect mode. In the former the major component of the pull is provided by the tug's propulsion system. In the indirect mode most of the pull is provided by the hydrodynamic lift due to the flow of water around the tug's hull, the tug's own thrusters being mainly employed in maintaining the tug's attitude in the water.

Apart from the requirements arising from the above, the main characteristics of tugs are:

(a) hull form and means of propulsion designed both for a given freerunning speed and a high thrust at zero speed (or bollard pull) or economical towing speed;
(b) upper deck layout to permit close access to ships with large overhang;
(c) a towing point above the longitudinal centre of lateral pressure, usually just aft of amidships on the centre line: the towing wire is often required to have a 180 degree clear sweep;
(d) good manoeuvrability;
(e) adequate stability when the towing wire is athwart ships and either veering from a self rendering winch or about to break.

Hull form is based on normal considerations of minimum resistance for the maximum free running speed which, for ocean tugs, is usually about 20 knots and for river tugs 12-16 knots. There are several restrictions to the selection of form; there is often a restriction on length, particularly for inland craft and frequently a need for minimum draught. Air drawing to propellers must be prevented, usually by adopting wide flat sections aft which give the propellers physical protection as well. A block coefficient of 0.55-0.65 is usual. The choice

of propulsion unit is offundamental importance because, like the trawler, there are two quite different conditions to meet, each at high efficiency-required free running speed and required bollard pull at zero speed or pull at towing speed.

Another way of classifying tugs is by the type and position of the propulsor units.

(a) *Conventional tugs.* These have a normal hull form and a traditional propulsion system of shafts and propellers. The propellers may be open or nozzled and of fixed or controllable pitch. These tugs may have steerable nozzles or vertical axis propellers. Some still employ paddle wheels. The main characteristics of these various propulsors are described in Chapter 10. Conventional tugs usually tow from the stern either with a tow hook or from a winch. They push with the bow.
(b) *Stern drive tugs.* These have a conventional hull form forward but the stern is cut away to provide room for twin azimuthing propellers. These propellers, which may be of fixed or controllable pitch, are in nozzles and can be turned independently through 360° providing very good manoeuvrability. Propeller drive is through two right angle drive gears and for this reason these vessels are sometimes called Z-drive tugs. They usually have their main winch forward and tow over the bow or push with the bow.
(c) *Tractor tugs.* These have an unconventional hull form. The propulsors are sited about one-third of the length from the bow under the hull, protected by a guard. A stabilizing skeg is fitted aft. Propulsion is by azimuthal units or vertical axis propellers. They usually tow over the stern or push with the stern.

In most operations involving tugs the assisted ship is moving at relatively low speed. In the escort tug concept the tug may have to secure to, run with and, in the event of an incident, control the assisted vessel at 10 knots or more. The success of such operations must depend upon the prevailing weather conditions and the proximity of land or underwater hazards, as well as the type and size of tug. Some authorities favour a free-running escort as not adding to the danger to ship and tug in the majority (event free) of operations. The tug would normally run ahead of the ship but has the problem of connecting up to it in the event of the ship experiencing difficulty. For that reason other authorities favour the tug being made fast to the escorted ship either on a slack or taut li~

The direct pull a tug can exert falls off with speed and indirect towing will be more effective at the higher speeds. One stern drive tug, displacement 614 tonnef, operational speed 14.5 knots and a static bollard pull of 53 tonnef, is capable of steering a 130,000 DWT tanker over the range 5.9 to 8.8 knots using the indirect method and below 5.9 knots using the direct method. This was on a course simulating an approach to Fawley on Southampton Water. With the tanker at 10 knots, engines stopped with rudder amidships, the tug brought her to rest in 15 minutes over an almost straightline distance of 1.25 miles.

Upper deck layout is dictated by the need to get close in to a variety of vessels and by the need to keep the towing point above the longitudinal centre of lateral pressure so that a lateral pull has a minimum effect on manoeuvrability. In a conventional tug (Fig. 16.22) the entire after half of the weather deck has only low obstructions and low bulwarks with tumble home and large freeing

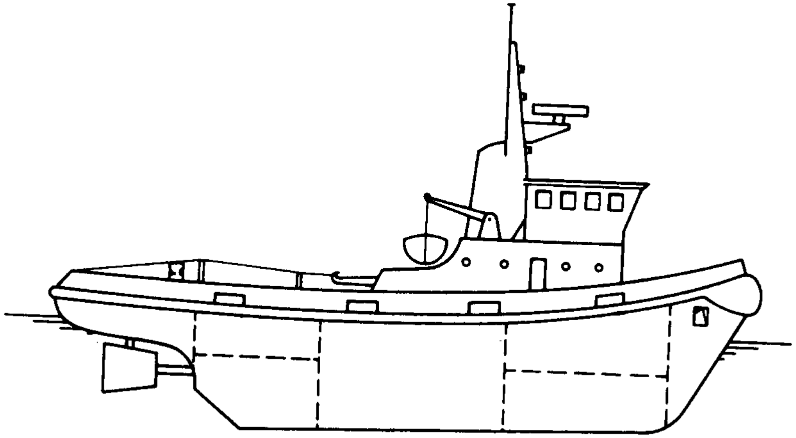


Fig. 16.22 *Conventional tug profile*

ports. Special towing hooks and slips are fitted. Superstructures are kept small and away from the sides where they might otherwise foul the attended ships. Hard wood fendering is fitted around the pushing areas and the structure inboard of these areas is reinforced.

A dangerous condition arises when the towing wire is horizontal and athwartships tending to capsize the tug. A self-rendering winch or a wire of known breaking strain limits the amount of the pull the tug must be capable of withstanding without undue heel. $\overline{GM}$s of 0.6 m are not unusual. Integral tug/barge systems can give good economy by creating higher utilization of the propulsion section in association with several barges. The concept has been applied to combinations up to 35,000 tonne dead mass.

Fishing vessels

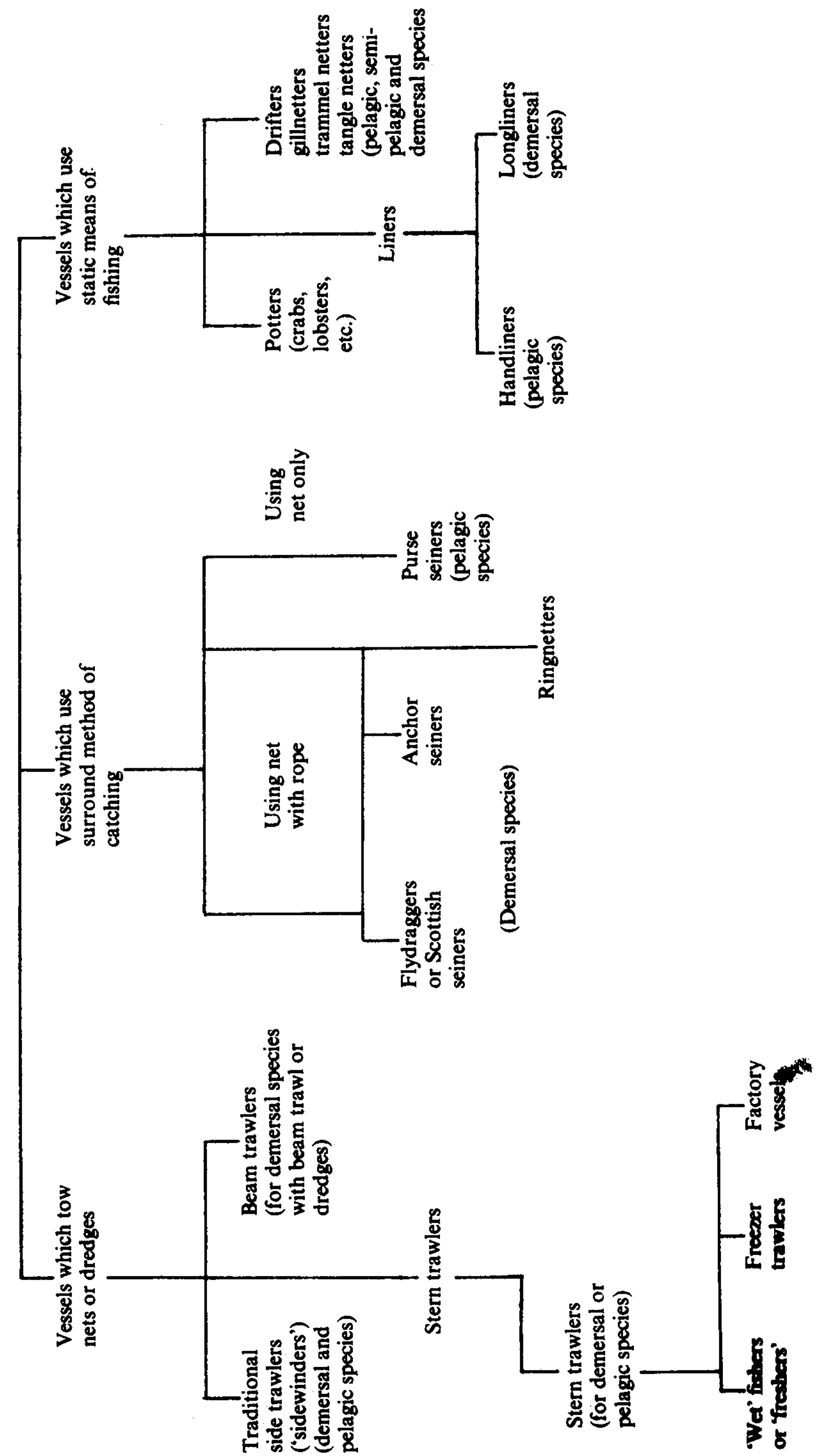
Fishing vessels have evolved over thousands of years to suit local conditions. Fish which live at the bottom of the sea like sole, hake and halibut and those which live near the bottom like cod, haddock and whiting are called demersal species. Those fish which live above the bottom levels, predominantly such as herring and mackerel, are called pelagic species. There are also three fundamental ways of catching fish:

(a) by towing trawls or dredges;
(b) by surrounding the shoals by nets, purse seines;
(c) by static means, lines, nets or pots.

These distinctions enable fishing vessels to be classified in accordance with Table 16.3.

The commonest type of fishing vessel is the trawler which catches both demersal and pelagic species. The trawl used for the bottom is long and stocking shaped and is dragged at a few knots by cables led to the forward gantry on

Table 16.3
Fishing vessel classification



the ship. When the trawl is brought up it releases its catch in the cod end down the fish hatch in the trawl deck. Operations are similar when trawling for pelagic species but the trawl itself has a wider mouth and is altogether larger.

Trawlers suffer the worst of weather and are the subject of special provision in the freeboard regulations. They must be equipped with machinery of the utmost reliability since failure at a critical moment could endanger the ship. Both diesel and diesel-electric propulsion are now common. Ice accretion in the upperworks is a danger in certain weather, and a minimum value of $\overline{GM}$ of about 0.75 m is usually required by the owner. Good range of stability is also important and broaching to is an especial hazard.

Despite great improvements in trawler design significant numbers of vessels are lost every year and many of them disappear without any very good explanation. It is probable that such losses are due to the coincidence of two or more circumstances like broaching to, open hatches, choked freeing ports, loss of power, critical stability conditions, etc.

To give adequate directional stability when trawling, experience has shown that considerable stern trim is needed, often as much as 5 degrees. Assistance in finding shoals of fish is given by sonar or echo sounding gear installed in the keel. No modern trawler is properly equipped without adequate radar, communication equipment and navigation aids. A typical stern fishing trawler is shown in Fig. 16.23.

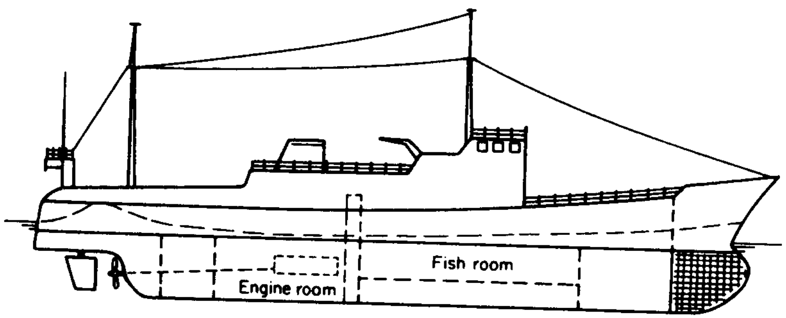


Fig. 16.23 *Stern trawler*

The trawler was the first type of ship for which a special analysis of resistance data was produced. A regression analysis of trawler forms for which tank tests have been made showed that a total resistance coefficient C_R = *RLj f:j,y2* is found to be a function of six geometrical parameters of the ship's form, *L/B*, *B/T*, C_m, C_p, longitudinal position of LCB and half angle of entrance of waterplane. From these, the power/speed curve can be produced to within an accuracy of a few per cent without the expense of tank tests.

Yachts

For many years, the design, construction and sailing of yachts has been a fascinating art about which whole books are regularly published. This is

because the science is too complex for precise solution-and indeed, few yuchtll-men would wish it otherwise. Some tenable theories have, in fact, been evolved to help in explaining certain of the performance characteristics of sailing boats.

A yacht, of course, obeys the fundamental theory described generally in this book for all surface ships. In addition, a yacht is subject to air forces acting on the sails and to water forces due to its peculiar underwater shape-forces which

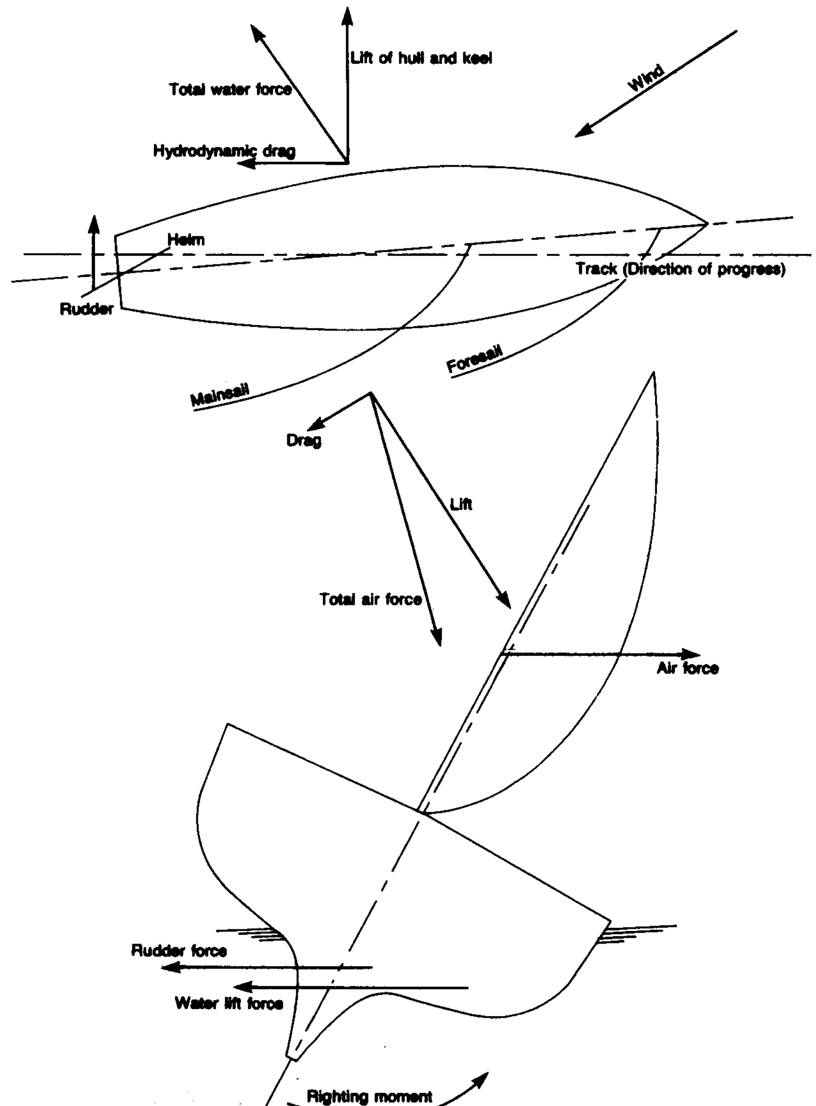


Fig. 16.24 Principal forces acting on a boat sailing into the wind

are negligible for ordinary surface ships. Sailing before the wind, a yacht is propelled by:

(a) the vector change of momentum of the wind, deflected by the sails;
(b) lift generated by the sails acting as aerofoils; because an aerofoil requires an angle of incidence, yachtsmen prefer sailing with wind on the quarter rather than dead astern, particularly when flying a spinnaker, to give them more thrust.

When sailing into the wind, the yacht is propelled only by a component of the lift due to the sails, acting as aerofoils. Lift and associated drag depend upon the set of the sails, their sizes, shapes, stretch and material, the angle of heel of the boat, the relative wind velocity and the presentation of the sails to the wind and to each other. Because the yacht does not quite point in the direction in which it is going and is also heeled, the hull too acts as an aerofoil, experiencing hydrodynamic lift and drag which exactly balance out the air forces when the boat is in steady motion.

The transverse couple produced by air and water forces is reacted to by the hydrostatic righting moment to keep the boat in stable equilibrium. Large angle stability and dynamical stability are clearly of great importance.

Longitudinally, the relative position of hydrodynamic lift, the centre of lateral resistance and the lateral component of air lift determine whether the rudder carries weather helm, as shown in Fig. 16.24, or lee helm. Lee helm is dangerous because, if the tiller is dropped or the rudder goes free by accident, the boat will not come up into the wind but veer away and increase heel. Ideally, to minimize rudder drag, a yacht should carry slight weather helm in all attitudes and it is towards this 'balance' that yacht designers aim.

The resistance of a yacht calculated or measured in the manner conventional for surface ships, is not a very helpful guide to the yacht designer. Minimum resistance is required at small angles of yaw and small angles of heel, and these are different from the conventional figures. Resistance in waves is also of considerable importance and varies, of course, with the response of the boat to a particular sea-because of augment of resistance due to pitching, a yacht may well sail faster in Force 4 conditions than it does in Force 5, or better in the Portland Reaches than off Rhode Island.

A yacht designer must therefore achieve minimum resistance yawed, heeled and in waves, good longitudinal balance in all conditions and satisfactory stability. The rig must not upset the longitudinal balance and must give maximum performance for sail area permitted, in all conditions and direction of wind, particularly close to the wind. While some theory helps, the process overall remains much an art.

Annex (Related to Chapter 11)

The Froude 'constant' notation (Froude, 1888)

This form of presentation is truly non-dimensional, although appearing a little strange at first to those used to the more common forms of non-dimensional presentation used in general engineering. It is not used these days but there is a lot of earlier data in this form. For that reason it is necessary to give students an introduction to the notation and to retain the Imperial units in which the data was derived.

As the characteristic unit of length, Froude used the cube root of the volume of displacement and denoted this by U. To define the ship geometry he used the following

$$(M) = \text{length constant} = \frac{\text{wetted length}}{U}$$

$$(B) = \text{breadth constant} = \frac{\text{wetted breadth}}{U}$$

$$(D) = \text{draught constant} = \frac{\text{draught at largest section}}{U}$$

$$(S) = \text{wetted surface constant} = \frac{\text{wetted surface area}}{U^2}$$

$$(A) = \text{section area constant} = \frac{\text{section area}}{U^2}$$

As performance parameters he used relationships between speed and displacement and speed and length

$$(K) = \frac{\text{speed of ship}}{\text{speed of wave of length } U/2} = V\left(\frac{4\pi}{gU}\right)^{\frac{1}{2}}$$

$$(L) = \frac{\text{speed of ship}}{\text{speed of wave of length } L/2} = V\left(\frac{4\pi}{gL}\right)^{\frac{1}{2}}$$

And as a resistance constant

$$(C) = \frac{1000 \times \text{resistance}}{\Delta (K)^2}$$

In verbal discussions, (M) is referred to as 'circular M', (B) as 'circular B', and so on.

The resistance used in calculating C can be the total, residuary or frictional resistance and these can be denoted by C_T, C_R and C_F respectively.

Standard expressions can be derived for each constant as in Table A.1. In this form they are dimensional.

For comparison of design data, the results from model tests were plotted in the form of C–K curves with the data corrected to represent a 16 ft model. Separate curves are drawn for each displacement value at which the model is run. (Fig. A.1.) Superimposed on the plots are curves showing the skin friction correction in passing from the 16 ft model to ships of various length.

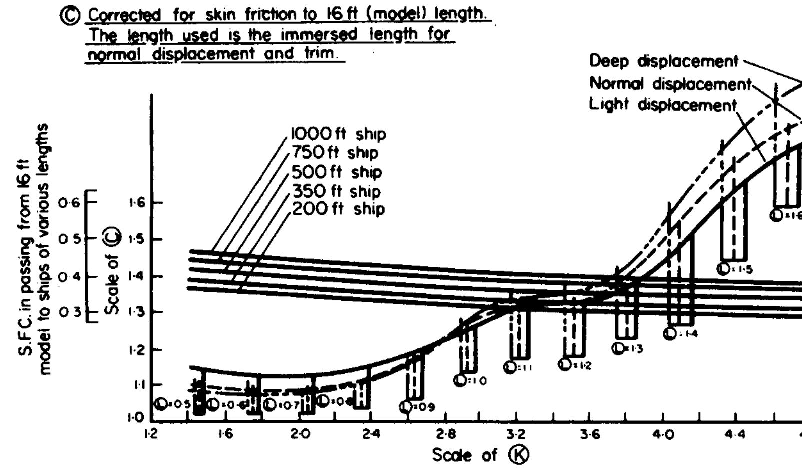
This plotting is included in the *Elements of Form Diagram* which is the standard method of summarizing the important information on ship's form and resistance. The principal dimensions and form coefficients are given in tabular form. The curve of areas, waterline and midship section for the design displacement are plotted non-dimensionally by dividing length dimensions by U and areas by U^2. To facilitate comparison of forms, the curves are drawn on a common base length and the ordinates of the curve of areas adjusted to keep the area under the curve unchanged.

The data from a large selection of models is assembled as plots of C against M for a range of K values. Generally, displacement and length are fairly well established by other design considerations so that for a given speed M and K are known. The best form of those already tested can be selected and the data for this form provides the first approximation to the resistance of the new design.

Table A.1
Froude 'constant' notation (sea water).

Parameter	Symbol	Expression: British units (tonf, ft, knots)	Expression: Metric units (tonne, m, knots)
Characteristic length	U	$3.271\,\Delta^{\frac{1}{3}}$	$0.992\,\Sigma^{\frac{1}{3}}$
Length	M	$0.3057\,L/\Delta^{\frac{1}{3}}$	$1.0083\,L/\Sigma^{\frac{1}{3}}$
Breadth	B	$0.3057\,B/\Delta^{\frac{1}{3}}$	$1.0083\,B/\Sigma^{\frac{1}{3}}$
Draught	D	$0.3057\,T/\Delta^{\frac{1}{3}}$	$1.0083\,T/\Sigma^{\frac{2}{3}}$
Wetted surface	S	$0.0935\,S/\Delta^{\frac{2}{3}}$	$1.0167\,S/\Sigma^{\frac{1}{3}}$
Section area	A	$0.0935\,A/\Delta^{\frac{2}{3}}$	$1.0167\,A/\Sigma^{\frac{2}{3}}$
Speed	K	$0.5834\,V/\Delta^{\frac{1}{6}}$	$0.5848\,V/\Sigma^{\frac{1}{6}}$
	L	$1.055\,V/\sqrt{L}$	$0.5824\,V/\sqrt{L}$
Resistance	C	$\dfrac{427.1\text{ e.h.p.}}{\Delta^{\frac{2}{3}}V^3}$	$\dfrac{579.7\,P_E}{\Sigma^{\frac{2}{3}}V^3}$

Note: The U.K. nautical mile, 6080 ft (1853.18 m) is greater than the international nautical mile which is 1852 m. P_E is the power in kW and Σ is the mass displacement.



Unit model	Light	Normal	Deep
Length constant			
Draught "			
Breadth "			
Skin "			
Largest sec: area			
" " coeff†			
Prismatic coeff†			

Ship Dimensions	Light	Normal	Deep
Length			
Beam (max)			
Draught (mean)			
Displacement			
Largest sec area			
Prismatic coeff†			
Skin			
L.C.B. Abaft F.P.			

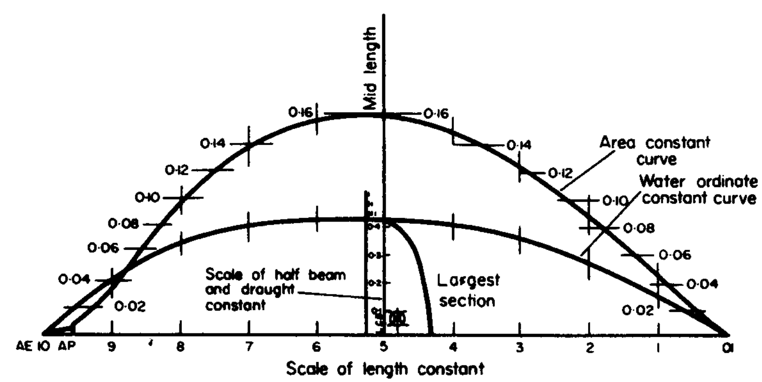


Fig. A.1 Elements of form diagram

Table A.2

O and f values. R. E. Froude's frictional data. Values at standard temperature = 15 °C = 59 °F (British Units), f related to S in ft², V in knots, R in lbf

Length (ft)	O	f	Length (ft)	O	f
5	0.15485	0.012585	40	0.10043	0.009791
6	0.1493	0.012345	45	0.09839	0.009691
7	0.1448	0.012128	50	0.09664	0.009607
8	0.1409	0.011932	60	0.0938	0.009475
9	0.13734	0.011751	70	0.09164	0.009382
10	0.13409	0.011579	80	0.08987	0.009309
11	0.1312	0.011425	90	0.0884	0.009252
12	0.12858	0.011282	100	0.08716	0.009207
13	0.1262	0.011151	120	0.08511	0.009135
14	0.12406	0.011033	140	0.08351	0.009085
15	0.1221	0.010925	160	0.08219	0.009046
16	0.12035	0.010829	180	0.08108	0.009016
17	0.11875	0.010742	200	0.08012	0.008992
18	0.11727	0.010661	250	0.07814	0.008943
19	0.1160	0.010596	300	0.07655	0.008902
20	0.1147	0.010524	350	0.07523	0.008867
21	0.1136	0.010468	400	0.07406	0.008832
22	0.11255	0.010413	450	0.07305	0.008802
23	0.11155	0.010361	500	0.07217	0.008776
24	0.1106	0.010311	550	0.07136	0.008750
25	0.10976	0.010269	600	0.07062	0.008726
26	0.1089	0.010224	700	0.06931	0.008680
27	0.1081	0.010182	800	0.06818	0.008639
28	0.1073	0.010139	900	0.06724	0.008608
29	0.1066	0.010103	1000	0.06636	0.008574
30	0.1059	0.010068	1100	0.06561	0.008548
35	0.10282	0.009908	1200	0.06493	0.008524

Fronde method

Using the 'constant' notation:

$$C_F = \frac{1000\ (\text{frictional resistance})}{\Delta K^2}$$

$$= \frac{1000}{\rho g U^3} \times \frac{fSV^{1.825}}{4\pi V^2/gU}$$

$$= O \cdot S\, L^{-0.175}$$

where

$$O = \frac{1000f}{4\pi\rho(gL/4\pi)^{0.0875}} = \text{'Circular } O\text{'}$$

Since for both model and ship

$$C_T = C_R + C_F$$

$$[C_T]_{\text{ship}} = [C_T]_{\text{model}} - [O_m - O_s]\, S\, L^{-0.175}$$

Table A.3

f values. R. E. Froude's skin friction constants (Metric units)

Length (m)	f	Length (m)	f	Length (m)	f
2	1.966	11	1.589	40	1.464
2.5	1.913	12	1.577	45	1.459
3	1.867	13	1.566	50	1.454
3.5	1.826	14	1.556	60	1.447
4	1.791	15	1.547	70	1.441
4.5	1.761	16	1.539	80	1.437
5	1.736	17	1.532	90	1.432
5.5	1.715	18	1.526	100	1.428
6	1.696	19	1.520	120	1.421
6.5	1.681	20	1.515	140	1.415
7	1.667	22	1.506	160	1.410
7.5	1.654	24	1.499	180	1.404
8	1.643	26	1.492	200	1.399
8.5	1.632	28	1.487	250	1.389
9	1.622	30	1.482	300	1.380
9.5	1.613	35	1.472	350	1.373
10	1.604				

f in metric units $= f$ (Imperial units) x 160.9.

$RF = fSV^{1.825}$

RF = Frictional resistance, N

S = wetted surface, m²

V = speed, m/s

L = waterline length, m

f in metric units $= f$ in Imperial units x 47.87 when V is in knots.

where O_m and O_s are the 'circular O' values for model and ship respectively.

In other words, the total resistance of the ship expressed non-dimensionally can be obtained from that of the model by making a correction which is dependent on the skin friction. For this reason, the term $(O_m - O_s)\, S\, L^{-0.175}$ is known as the *skin friction correction*.

To assist in applying the above method O and f values are tabulated in Tables A.2 and A.3 where:

$$\text{Frictional resistance} = fSV^{1.825}$$

and

$$C_F = O \cdot S\, L^{-0.175}$$

The values are those agreed by the International Conference of Ship Tank Superintendents held in Paris in 1935 Values of $L^{-0.175}$ are tabulated in Table A.4.

If the model tests are carried out at other than 59 °F(15 °C), then the data have to be corrected for this by increasing or decreasing the C_F value by 2.4 per cent for every 10 °F the temperature is below or above this value. Thus, if the experiments are conducted at a temperature of t_1 °F

Table A.4
Value of $Ⓛ^{-0.175}$

Ⓛ	$Ⓛ^{-0.175}$	Ⓛ	$Ⓛ^{-0.175}$	Ⓛ	$Ⓛ^{-0.175}$
0.00	∞	1.00	1.0000	2.00	0.8858
0.05	1.6892	1.05	0.9915	2.05	0.8819
0.10	1.4962	1.10	0.9835	2.10	0.8782
0.15	1.3937	1.15	0.9758	2.15	0.8746
0.20	1.3253	1.20	0.9686	2.20	0.8711
0.25	1.2746	1.25	0.9617	2.25	0.8677
0.30	1.2345	1.30	0.9551	2.30	0.8644
0.35	1.2017	1.35	0.9488	2.35	0.8611
0.40	1.1739	1.40	0.9428	2.40	0.8580
0.45	1.1500	1.45	0.9370	2.45	0.8549
0.50	1.1290	1.50	0.9315	2.50	0.8518
0.55	1.1103	1.55	0.9262	2.55	0.8489
0.60	1.0935	1.60	0.9210	2.60	0.8460
0.65	1.0783	1.65	0.9161	2.65	0.8432
0.70	1.0644	1.70	0.9113	2.70	0.8404
0.75	1.0516	1.75	0.9067	2.75	0.8378
0.80	1.0398	1.80	0.9023	2.80	0.8351
0.85	1.0288	1.85	0.8979	2.85	0.8325
0.90	1.0186	1.90	0.8938	2.90	0.8300
0.95	1.0090	1.95	0.8897	2.95	0.8275
				3.00	0.8251

$$[Ⓒ_{F}]_{\text{model}} = [1 + 0.0024(59 - t_1)]O_{\text{m}}ⓈⓁ^{-0.175}$$

The wetted surface area used in determining Ⓢ is taken as the mean wetted girth of sections multiplied by the length on the waterline. It is therefore less than the true wetted surface area as the inclination of the surface to the middleline plane of the ship is ignored.

Some authorities calculate the appendage resistance. This can introduce an error and this is allowed for in the QPC factor which is deduced from ship trials. The National Physical Laboratory introduced a scaling factor β by which the model appendage resistance can be multiplied, although they recommended that the value of β should generally be taken as unity.

In the absence of firmer data for a ship, the value of Ⓢ can be obtained by applying the Haslar formula:

$$Ⓢ = 3.4 + \frac{Ⓜ}{2.06}$$

or the Taylor formula given in the main text.

Worked example

The Ⓒ-Ⓚ curve for a 16 ft model of a ship 570 ft long and 11,500 tonf displacement, corrected to standard temperature, is defined by the following table:

Ⓚ	2.2	2.3	2.4	2.5	2.6	2.7	2.8
Ⓒ	1.130	1.130	1.132	1.138	1.140	1.141	1.150

Ⓚ	2.9	3.0	3.1	3.2	3.3	3.4
Ⓒ	1.153	1.156	1.172	1.193	1.236	1.283

Deduce a plot of e.h.p. against speed for the clean condition assuming that Ⓢ is given by the Haslar formula with a multiplying factor of 1.03. Also, calculate the e.h.p. for the ship 6 months out of dock assuming that the skin frictional resistance increases by $\frac{1}{4}$ per cent per day out of dock.

Solution: From the formulae quoted in the text the following relationships can be deduced.

$$Ⓚ = 0.5834\frac{V}{\Delta^{\frac{1}{6}}}$$

Hence, in this case, $V = 8.144$ Ⓚ where V is in knots

$$Ⓛ = 1.055\frac{V}{\sqrt{L}} = \frac{1.055}{\sqrt{570}}V$$

$$U^3 = 11{,}500 \times 35 = 402{,}500\,\text{ft}^3$$

Hence $U = 73.83$ ft.

Table A.5
Froude analysis

Ⓚ	V (knots)	Ⓒ	SFC	Clean ship		Ⓛ	$Ⓛ^{-0.175}$	$\delta Ⓒ_{F}$	Dirty ship	
				Ⓒ	e.h.p.				Ⓒ	e.h.p.
2.2	17.9	1.130	0.371	0.759	5200	0.79	1.042	0.243	1.002	6770
2.3	18.7	1.130	0.368	0.762	5950	0.83	1.033	0.241	1.003	7810
2.4	19.5	1.132	0.366	0.766	6800	0.86	1.027	0.239	1.005	8890
2.5	20.4	1.138	0.363	0.775	7850	0.90	1.019	0.237	1.012	10,200
2.6	21.2	1.140	0.361	0.779	8800	0.94	1.011	0.236	1.015	11,500
2.7	22.0	1.141	0.357	0.784	10,040	0.97	1.005	0.234	1.018	13,000
2.8	22.9	1.150	0.354	0.796	11,400	1.01	0.998	0.232	1.028	14,700
2.9	23.7	1.153	0.351	0.802	12,800	1.05	0.992	0.231	1.033	16,500
3.0	24.4	1.156	0.349	0.807	14,050	1.08	0.987	0.230	1.037	18,000
3.1	25.2	1.172	0.347	0.825	15,750	1.11	0.982	0.229	1.054	20,150
3.2	26.1	1.193	0.345	0.848	18,000	1.15	0.976	0.227	1.075	22,800
3.3	26.9	1.235	0.343	0.892	20,800	1.19	0.970	0.226	1.118	26,000
3.4	27.7	1.283	0.341	0.942	24,000	1.22	0.966	0.225	1.167	29,600

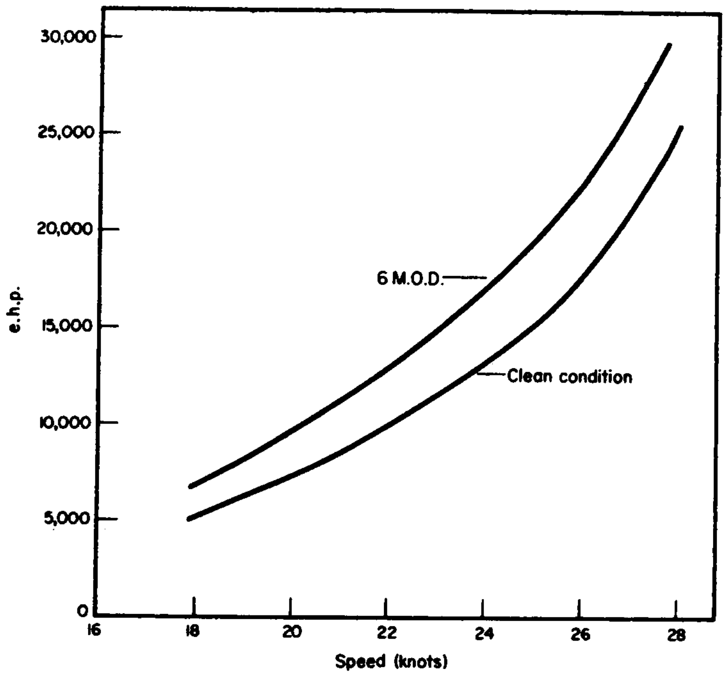


Fig. A.2 Plot of results

$$\text{Ⓜ} = \frac{L}{U} = \frac{570}{73.83} = 7.72$$

$$\text{Ⓢ} = 1.03\left[3.4 + \frac{\text{Ⓢ}}{2.06}\right] = 7.205$$

$$\text{Ⓒ}_F = O\ \text{Ⓢ}\ \text{Ⓛ}^{-0.175}$$

where, for $L = 570$, $O = 0.0711$

$$\therefore \quad \text{Ⓒ}_F = 0.0711 \times 7.205 \times \text{Ⓛ}^{-0.175}$$

$$= 0.512\ \text{Ⓛ}^{-0.175}$$

For the 16 ft model, $O = 0.1204$
and

$$\text{Ⓒ}_F = 0.867\text{Ⓛ}^{-0.175}$$

The reduction to be applied to the model Ⓒ to obtain the ship Ⓒ is 0.355 $\text{Ⓛ}^{-0.175}$

This is the skin friction correction, SFC.

$$\delta\text{Ⓒ}_F \text{ for the ship in the dirty condition} = 0.456\text{Ⓒ}_F$$

$$= 0.233\text{Ⓛ}^{-0.175}$$

$$\text{e.h.p.} = \frac{\Delta^{\frac{2}{3}}}{427.1}\text{Ⓒ}_T V^3 = 1.193\text{Ⓒ}_T V^3 \text{ in this case}$$

Table A.5 can now be constructed and the e.h.p./speed curves plotted as in Fig. A.2.

Bibliography

References are grouped by chapter, or group of chapters, with a final listing for general reading. Those references quoted will lead on to other useful references for the student to pursue.

The transactions, and conference proceedings, of the learned societies are among the main sources. Others are technical journals, research organizations and the universities. Much information on these sources can be gleaned from the Internet.

The following abbreviations have been used, with the letter Tstanding for *Transactions:*

RINA	The Royal Institution of Naval Architects (*INA* prior to 1960).
SNAME	The Society of Naval Architects and Marine Engineers.
DTMB	David Taylor Model Basin, USA.
IMO	International Maritime Organisation.
HMSO	Her Majesty's Stationery Office, now the Government Bookshop.

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Some useful web sites

Much useful data can be gleaned from the Internet. As an example, the RINA makes available all its technical papers which are issued for discussion. Other sites give details of the facilities at various research establishments. The LIoyds Register site gives information on the software they have available.

The following are some sites the student may find helpful. Many others relating top shipbuilders and equipment manufacturers, are regular contained in the advertisements in technical publications, such as *The Naval Architect* which is the journal of the RINA.

Learned societies

Royal Institution of Naval Architects	www.rina.org
Society of Naval Architects and Marine Engineers, USA	www.sname.org
Institute of Marine Engineers	www.imare.org.uk
The Nautical Institute	www.nautinst.org

International and government organizations

International Maritime Organisation	www.imo.org
Dept. of the Environment, Transport and the Regions	www.detr.gov.uk
UK Maritime & Coastguard Agency	www.mcagency.org.uk
US Coast Guard	www.uscg.mil
Defence Evaluation and Research Agency (UK)	www.dera.gov.uk
MARIN (Netherlands)	www.marin.nl
David Taylor Model Basin (USA)	www50.dt.navy

Classification Societies

International Association of Classification Societies	www.iacs.org.uk
American Bureau of Shipping	www.eagle.org
Bureau Veritas	www.veristar.com
China Classification Society	www.ccs.org.cn
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Answers to problems

Chapter 2

1. 0.73,0.80,0.97, 3.70m.
2. 8771 tonnef, 86.1 x 10^6 newtons, 103 m^3, 0.722.
3. 0.785, 0.333, 0.500.
4. 146.25 m^2, 145.69 m^2, 1.0039.
5. 2048 m^2.
6. 1.2 per cent low, 0.46 per cent low.
7. 9.33m^3, 0.705MN.
8. 5286 m^2, 1.65 m abaft 6 ord.
9. 2306 tonnef, 16.92m above 10 ord.
10. 403 tonnes.
11. 36,717m^3, 4.85m, 5.14m.
12. 243,500 m^4, 23,400 m^4.
13. 182.2 x $10^6 m^4$.
14. 39.42m, 48.17m.
15. 55.4 KN, 8.50 m.
16. 3.12m.
17. (a) 539.3 (Simpson 2),561.0 (trap).
    (b) 278.7 (Simpson 58-1), 290.0 (trap).
18. 0.41 per cent.
19. M = ih2(Yl + 6Y2 + 12Y3 + 18Y4 + ...) about lord.
21. (a) 2.09440, (b) 1.97714, (c) 2.00139, (d) 2.00000 correct.
22. (a) 201.06, (b) 197.33, (c) 199.75, (d) 200.59, (e) 200.51.
23. (a) 141.42, (b) 158.93, (c) 158.02, (d) 157.08 correct.
25. (a) 23.37, (b) 11.36.
26. (a) 4.37, (b) 7.23, (c) 9.3 per cent.

Chapter 3

1. (a) 146, (b) 544.
2. 4.94m.
3. 242m^3,0.12m^3/tonnef.
5. Approx 3 cm below keel.
6. 9 degrees.
7. 21 m.
8. 535MN, 4.1m below lWL (1331 Rule).
9. 39,650 tonnef, 4.34m below 1 WL.
10. 0.27 m below 2! WL.
11. 3.83 m, 5.65 m.
12. 5.94m. abaft amidships, 6.71 m.
13. 4.39 m.
14. 4.21 m forward, 4.70m aft.

15. 11.4m forward, 12.3m aft.
16. 1.34, 1.65 m.
17. 3.704m,2.744m.
18. 3.72MN, 4.06m, 5.06m.
19. 1.95 hours, 0.92m, 3.56m.
20. (a) 2388 tonnef, 3.36 m aft of amidships, (b) 208 tonnef, 3.58 m.
21. 3.77m fwd, 4.33m aft.
22. 3.22m,4A4m.
23. 9.22m forward, 9.28m aft.
24. $8847m^3$, 2.76m.
25. 115.8MN.
26. 12.86m before amidships, 2.74m, 2.66m.

Chapter 4

1. -.
2. -.
3. -.
4. 1.972m, 0.536m, 1.047m.
5. 3875 tonnefm, 1252 tonnefm, 15.6 degrees.
6. 2.05 m, 1.99 m.
7. 2.1 m, 15Am.
8. 2.9 cm.
9. 0.97T.
10. 1° 27'.
11. 4.388m.
12. -.
13. 7° 7'.
14. 90 degrees.
15. 2.39m,27.13m.
16. 1.25m.
17. 0.927 m.
18. lA4m.
19. 1°22'.
20. 0.21 m.
21. 0.28m.
22. 3.28 m.
23. 45 degrees.
24. -.
25. 0.37 m.
26. (a) 70Ao, (b) 84.9°, (c) 87.1°, (d) 95.50.
27. 74MJ.
28. 2.60, 1.035 : 1.
29. 0.67m.
30. OA7 m.
31. 2.53m, 2.82m, 0.09m, 4.57m, 75.SO.
32. 0.509 m, OA9 m.
33. -OA3 m.
34. -.

Chapter 5

1. 1.73 m port, 2.99 m starboard.
2. 13.25m,2.92m.
3. 288 tonnef, 4.70m, 5.83m; 6.23m, 5.81 m.
4. 20,740 tonnefm, 10°37',93.31 m (added weight), 91.58 m (lost buoyancy).
5. (i) 170 tonnef, (ii) 0.63 m, (iii) 0.59 m.
6. (a) 4.09m, (b) 3A2m, (c) 0.79m.
7. 7.06 m, 4.92 m forward; 4.33 m, 2.20 m aft.
8. 1.70 m.
9. 9.70m, +0.98m.
10. 40 tonnef, 94 tonnef, 22cm.
II. 10* degrees.
12. 1.8 degrees.
13. 53 tonnef, I! degrees.
14. 4.26m.
15. 4.63 min.
16. Starboard hit 40 per cent, 22! per cent; Port hit 40 per cent, 10 per cent.

Chapter 6

1. 35.7m from fore end, 486 tonnef/m.
2. SF, 0.21L from end = W/lOA; BM amidships = WL/32.
3. Max. SF., 30.75 tonnef; max BM, 132.6 tonnef/m.
4. SF, 48 tonnef 30 m from ends; BM 960 tonnef/m amidships.
5. Max. SF, 690 tonnef; max. BM, 13,000 tonnef/m.
6. 16mm.
7. SF, 714 tonnef at 3/4; BM 24,500 tonnef mat 5/6.
8. 113.5 pascals.
9. 55.3 and 65.5 MPa.
10. (/0, 93.7 MPa, (/K, 56.7 MPa.
II. 35.
12. zo, $1.59 m^3$; ZK, $1.22 m^3$; (/0, 71 MPa; (/K, 93 MPa.
13. (/0,40 MPa; (/K 58 MPa.
14. $52cm^2$.
15. 57.5,42.2, 19.7 MPa.
16. 42.9,42.9, 33.2 MPa.
17. 6M/WL = 0.144 and 0.115.
18. $20cm^2$.
19. 19mm.
20. 131 MPa, 0.3 degrees.
21. 55 mm approx.

Chapter 7

I. 260 MPa.
2. $7700cm^3$, $14,800cm^3$.
3. (a) 1:2.92, (b) 1:2.20.
4. (a) $2664cm^4$, (b) 7Amm.
5. 18mm.
6. 4A5 tonnef.
7. 7Acm.

8. (a) 20mm, (b) 8mm.
9. (a) 0.112 MPa, (b) 0.035 MPa, (c) 5.7mm.
10. (a) 0.11, (b) 0.73, (c) 0.22, (d) 0.40 MPa.
11. (a) 2.0, (b) 1.17.
12. 90 MPa.
13. 41.7 MPa, in rib at deck edge.
14. M_{EA} = 420, M_{AE} = 840, M_{BA} = 27,300, M_{BF} = 8480, M_{BC} = 35,760, M_{FB} = 2120 Nm.
15. M_{AB} = 0.77, M_{BC} = 3.07, M_{CB} = 12.36, M_{CG} = 2.07, M_{CD} = 14.43, M_{GC} = 1.02 tonnef/m.
16. 0.03 MN.
17. Torque 15.9 tonnef/m; BM =237 tonnef/m.
18. 554mm, 11.56kW.

Chapter 8

1. 75.5m, 7.7MN, 0.19 N/mm^2.
2. 18.36, 14.31 fwd, 22.41 aft tonnef/m^2.
3. -2.38 m forward, 6.91 m aft.
4. 8.7m.
5. (a) 97m, (b) 1,480 tonnef, (c) 41,000 tonnef/m.
6. (a) lift at 144m (b) max. force on FP 7,950 tonnef, (c) min tip mt. 700,000 tonnef/m, (d) floats off.
7. Max. BM, 184,000 tonnef/m; max. SF, 439 tonnef.
8. (a) 16.2m, (b) 1500 tonnef, (c) 57,000 tonnef/m.

Chapter 9

1. —.
2. 12.5 m/s, 8.0 s, 28.5 m.
3. 4.85 m, 3.03 m, 6.19 m.
4. $p(x) = \dfrac{1}{1.925}\exp\left\{-\dfrac{(x-2.19)^2}{1.18}\right\}$,

   $p(x) = \dfrac{x}{5.38}\exp\left\{\dfrac{-x^2}{10.75}\right\}$.
5. 0.16 MN, 41 knots, 1° 35′.
6. 1.64 m, 2.40 m, 3.00 m.
7. 1.56, 1.56; 6.2, 3.1; 25, 6.2; 56, 9.4; 100, 12.5; 156, 15.6; 306, 21.8; 506, 28.1; 625, 31.2.
8. 24.3 knots, 16.96 m.
9. —.
10. 57.5 c.p.m.

Chapter 10

1. 781 N, 1.63 m/s, 4.05×10^5.
2. 24.25, 27.64, 15.96, 12.37, 10.45 knots.
3. 34.2 knots, 19.7 knots.
4. 36 knots.
5. 0.524, 0.582, 13.4 MW.

6. 10.29 MW, 26.8 MW.
7. 28.14 knots.
8. 10.00, 12.11, 14.07 knots.
9. 28.257, 28.381, 28.370, 28.418, 28.319 knots.
10. 28.381 knots; 0.789, −0.453, 0.098, 0.034, −0.008
11. 9.68 tonnef (94910 N).
12. 2.58 MW, 0.97, 0.14.

Chapter 11

1. 12.54 knots.
2. $L = 233$ m, $B = 21.7$ m, $T = 7.2$ m, $\Delta = 200$ MN, 22.9 MW at 30 knots.
3. —.
4. 14 knots, 3.16 MW.
5. 35 per cent, 32.4 per cent.
6. 7.78 MW, 9.42 MW, $L = 150$ m, $B = 24$ m, $T = 9$ m.
7. 28 knots.
8. 14.9 MW, 44 MW
9. 4.36 m, 17.9 MW, 0.56, 4.88 m, 229 r.p.m.
10. 2.77 m, 0.716, 3.60 m.
11. 21700 N/m^2, 769 r.p.m.
12. 550 r.p.m., 33.7 N, 103 Ncm
13. 0.60, 0.961.
14. 12.14 tonnef.
15. 95.4 newtons.
16. 448 r.p.m., 0.12, 0.110, 1.009.
17. 0.70, 2.45 m/sec.
18. 0.04, 0.0458, 0.66, 28.8 MW.
19. 27.8 knots, 26.5 knots.

Chapter 12

1. 3.39 m, 76.5 m, 10.93 m/s.
2. 8.69, 3.61, 3.47 secs.
3. $91\frac{1}{2}$ degrees, $120\frac{1}{2}$ degrees.
4. a:b:c as 1: $\pi/2 : 4/\pi$.
5. 10.07 m.
6. 13.16 tonnef.
7. 3.71°, 5.00°, 0.93°, 16.5 s.
8. Heave = 0.41 (wave height) at $\omega_E = 0.60$.
9. $a = 0.083$, $b = 0.0079$ 3.62°.
10. —.
11. 8 degrees approx., 33.6 kW.
12. 5.56.
13. $a = 0.12$, $b = 0.015$.
14. 2.03 m^2 s.
15. —.

Chapter 13

1. 0.222 MN m.
2. $3\frac{1}{2}$ degrees.

3. 1.96m.
4. 1.25MN, l.4m.
5. 0.67MN, 0.184MNm.
6. 1.19MN, -0.194MNm.
7.-.
8. 16 tonnef. m, 237 tonnef/m.
9. 2.540 m^2, 2.127 m, 3.403 m^4, 0.833 m^4.
10. ±29°20'.
I I. 55.2,47.3, 41.4m^2.
12.14.8MNm.
13. A stable, B unstable; 19.4 m, 38.5 m.
14. 40.2m, 70.7m.
15. 40 m, stable, 2.38 knots.

Chapter 14

II. 328kN/m^2, 196kN/m^2.
12. 112.7 MN/m^2 and 470 kW, 1.97 MN/m^2 and 8.2 kW.
13.1046 litres/min at 0.99 MPa; 23.2kW for smooth pipes, 1046 litres/min at 1.06MPa; 25.6 kW for rough pipes.
14.21 IOl/s, 9401/s.
15.1O,7001/s.
16. Approx. 7°C, about 25cc.
17. About 4.65 x 10^3 *ljkg* from 16.6/15.6 to 21/17.
18. Slope 0.49, off Coil 14/13, mix 30/23.6; I.54m^3/sec. (including fresh air) is one solution.
19. Approx. 1020m^2.
20. (a) 64 per cent, (b) 57 per cent approx.
21. 30000GT, 23232 NT, (a) the same, (b) 23307.

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