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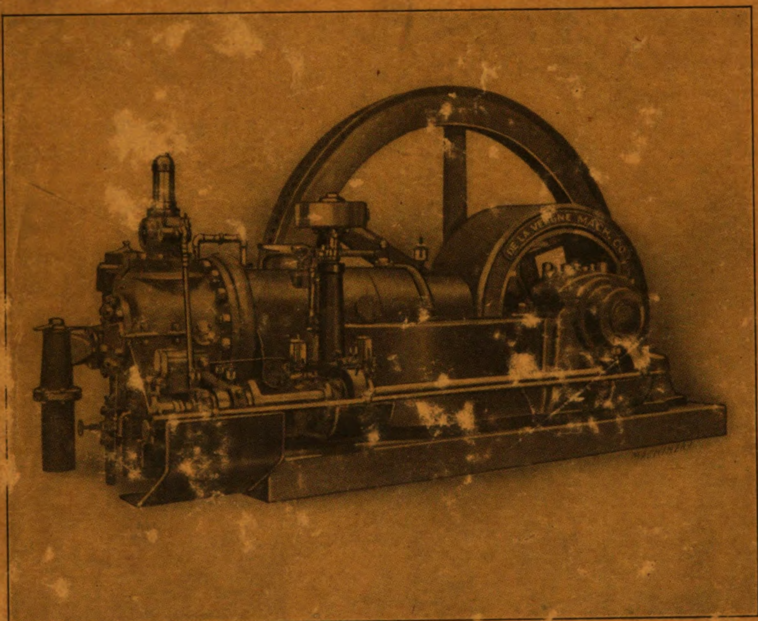
FORMULAS AND DATA
FOR
GAS ENGINE DESIGN

SECOND EDITION



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NUMBER 65

FORMULAS AND CONSTANTS FOR GAS ENGINE DESIGN

SECOND EDITION

CONTENTS

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INTRODUCTION*

The past fifty years have been remarkable for mechanical development in many lines, and one of the most important improvements of that period was made in the internal combustion engine. This type of prime mover, which comprises gas, gasoline, oil, alcohol and other engines of various descriptions in which the fuel is burned in the work cylinder, dates from the introduction of the crude Lenoir engine in 1860. This was the first gas motor brought before the public, although by no means the first actually built. Since that date there has been a steady improvement in design and growth in size, until to-day there are in use gas engines of four thousand horsepower, or more, driving dynamos with the regularity and smoothness which characterize the best steam engines.

Although essentially simple, the simplicity of the gas engine is more apparent than real. Possessed of but few parts, there are nevertheless many defects likely to affect the gas engine that never trouble the steam engine. Poor ignition, back firing, carbon deposits, poor compression and short circuits are a few of the troubles that have played havoc with its reputation as a reliable motive power. The simplicity of its mechanism is in part responsible for these troubles. Not sufficient care was taken in the design and construction and the conditions of operation often were notoriously bad. But the advent of the motor boat and automobile has been followed by a great improvement in design and construction. The number of users has greatly increased, and their requirements for uninterrupted service have made imperative the elimination of the ordinary faults inherent in the earlier designs.

The steam engine is rapidly passing for small stationary plants, and it is practically out of the automobile field. The competition between it and the gas engine as the motive power of the automobile resulted in a complete victory for the latter with its simpler construction, compactness, great power, superior economy and other characteristics, which have won for it the leading place, and the trend is toward the use of gas engines as the motive power for all purposes. The modern gas producer and the improved gas engine are for manufacturing plants practically as simple to operate as the steam engine, and the cost of fuel and labor is less. Cheap fuel can be used; there is no boiler to explode, no tubes to leak, no trouble with water supply and little or no waste of fuel after the engine is stopped.

In the evolution of the gas engine, little attention was given to the weight of the motor per horsepower until it had reached a stage which made it reliable and adapted it to wide fields of usefulness. The chief question which first concerned the designers and manufacturers was how much power it was possible to obtain from a single engine working under economical loads, and how reliable it could be made under con-

* MACHINERY, April, 1910.

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tinuous operation. The question of reducing the weight of gas engines arose when builders of boats and airships began to experiment with this form of motive power. The elimination of superfluous metal in the gas engine is following the lines of development of the steam engine. A hundred years ago a steam engine which produced ten horsepower weighed some ten to twenty tons, including engine, boiler and all equipment. Similarly, the early gas engines were heavy and unnecessarily clumsy. Even those first used in automobiles were so large, heavy and clumsy that when mounted on trucks they left apparently little room for passengers or freight, but the systematic cutting down of weight has progressed rapidly.

CHAPTER I

SUGGESTIONS IN THE DESIGN OF COMBUSTION ENGINES*

Motors of any type—gasoline, electric or steam—are compromises from beginning to end; we cannot design any one part to obtain the theoretically best results from it without interference with some other feature. With the gasoline engine there are certain advantages in a short stroke, but in the short-stroke motor we also meet with many objections. This same condition applies to the number of cylinders and their mounting (horizontal, vertical or oblique), to the casting *en bloc* or individually, and to the amount of compression. In many cases the best results can only be arrived at by actual "cut-and-try" experiments, and it is, therefore, not possible to give any definite figures except for purposes of illustration. The suggestions given are based upon an extensive experience with automobile and marine engines.

Any steam engine will deliver more than its rated horsepower, owing to the fact that it is built to run economically by cutting off the steam as it enters the cylinder during the early part of the stroke. With a cut-off of one-quarter or one-fifth, the expansive qualities of the steam are utilized, and the engine is rated at its economical output. If, however, more power is required temporarily, the point of cut-off is shifted ahead until the required power is obtained, up to the maximum attainable, when steam enters during the full length of the stroke. With gasoline engines, however, this conservative rating has never been observed. The A. L. A. M. rating, for example, gives the highest capacity of the engine under the best conditions; under unfavorable conditions, such as bad carburetor adjustment, poor compression, weak spark, and innumerable other troubles, the power output may be an indefinite amount less.

* MACHINERY, April, 1910.

Lack of Economy in Present Automobile Engines

It seems to have always been the aim of the automobile engine designer to obtain all the power possible from a given bore engine, regardless of gasoline economy or other considerations. Hence the old-time efficient suction valves have been superseded by mechanically operated valves, offering no resistance to the incoming charge; and, in addition, the valve is left open long after the piston reaches the end of the outward stroke, thus taking advantage of the momentum of the gases in the pipes to crowd more than atmospheric pressure into the cylinder. The result of all this, of course, is increased power, the same as in the case of the steam engine taking steam the full length of the stroke; but who would buy a steam engine working in such an inefficient manner?

An engine working under the above conditions will exhaust at about 60 pounds pressure. This 60 pounds multiplied by 125 per cent of the piston displacement is a complete loss, and amounts to nearly 50 per cent of the original power of the charge. Compound engines have been built to utilize this exhaust pressure, but there is no excuse for this complication when the same results may be accomplished by proper design of a single-expansion engine, by simply carrying the expansion down as far as may be found desirable. When expansion is carried beyond certain limits in a steam engine, trouble is encountered from condensation, but as there is little or no moisture in the explosive mixture, we need fear no trouble from that source with the gasoline engine. With a compound engine of the gasoline type, a great deal of the unused power of the gases is lost by cooling when passing through the ports, a large quantity is used to fill the ports, and the extra piston adds a large amount of extra friction, so that the only benefit obtained is practically the muffling action.

There should be no side pockets (for valves) in the cylinder heads, for the inside surface should be trimmed down to the minimum possible. The less internal surface we have exposed to the heat, the smaller the number of heat units that will be absorbed by the metal. Whatever is taken into the metal has to be disposed of, either by air or water, which adds to the difficulties of air cooling in the one case, and to the size of the cooling apparatus and the amount of water carried in the other. In both cases there is a loss in efficiency, for every heat unit radiated is one heat unit less power delivered at the shaft.

If we could obtain a material for the cylinders that was an absolute non-conductor of heat, we would not have cause to worry over any system of cooling. If the metal would not conduct heat, it would not absorb any. But suppose we go to the other extreme and obtain a metal that is a perfect heat conductor; then as soon as the first explosion takes place the outside of the cylinder would be just as hot as the inside surface; it would instantly become red hot. How would we keep the latter cylinder cool?

According to the above reasoning, it appears most desirable to have a very poor conductor of heat for the cylinder, for while we cannot obtain the absolute non-conductor, the nearer we reach that point the less heat will be absorbed by the metal at each explosion, and the greater proportionate effect the incoming charge will have toward cooling the internal surface. If the heat of the explosion were constantly applied, this beneficial result could, of course, not be expected.

Cylinder and Valves

The head of the cylinder should be as nearly hemispherical in form as possible and polished inside. With our best endeavors, however, it will be a very much flattened hemisphere unless the length of the stroke is carried far beyond the limits of good practice, for the cubic contents of a hemisphere would be too great for the piston displacement, and hence the compression would be too low.

With the pocketless head the placing of the valves has always been more or less of a problem. The most popular method at present appears to be the placing of the valves obliquely, and to mount the cam-shaft directly over the center of the line of cylinder tops. Another method is that which the H. H. Franklin Mfg. Co. has adopted of mounting the valves concentrically; that is, the inlet valve is inside of the exhaust valve.

Ratio between Bore and Stroke

As to the problem of the ratio between the bore and the stroke, there is a wide divergence of opinion among designers. The writer is very strongly in favor of the long stroke. Abroad, the long-stroke engine has become very popular on automobiles, and where racers have been limited by the bore, the long-stroke types have carried off the honors.

The reasons for a long stroke are several: 1. With a given cubic content of compression space (and upon that the amount of power obtainable depends), the shape is nearer that of a hemisphere, and consequently possesses the least absorbing surface possible for the quantity of mixture. 2. There is less piston diameter for a given charge, which gives less leakage area. 3. There are less reversals of motion of the reciprocating parts, thus reducing the number of hammer blows, giving higher mechanical efficiency and longer life to the engine.

It may be argued that the engine will have to run slower and thus deliver less power. This difference of speed, however, is made up by the additional amount of the charge due to the additional length of the compression space, which, of course, is made proportional to the length. As a higher piston speed is practicable, an increase of power is thus obtained, although not commensurate to the additional weight; but the advantages obtained outweigh in value the objectionable weight.

Increase of Efficiency of Present Engines

The lack of efficiency of present engines has been mentioned. We will now take up the problem of how to increase it. For illustration, suppose that the compression space of motors as at present constructed is 25 per cent of the combined piston sweep and clearance, and let the bore and stroke ratio be 1 to 1. Design an engine with double this stroke, without increasing the bore; let the compression space remain the same, and design the cam so that the incoming charge will be shut off at one-half the stroke. When the piston returns and compresses this charge it will have the same maximum compression as the first-mentioned engine. Now the explosion takes place, and the pressure at one-half the outward stroke falls to the exhausting pressure of the former engine. Both engines have received the same amount of charge and have thus far delivered the same amount of power. The first one exhausts at this point, but the last one has as much additional distance to travel, commencing with a charge of about 60 pounds pressure multiplied by four times the compression space. Owing to the latter fact the pressure will drop very slowly during the remainder of the stroke in comparison with the early portion, and there is no compression pressure to subtract from it, as the compression did not begin until after passing this point. Consequently the last half of the stroke is a pure gain, and is obtained at a high efficiency, for we have no losses in compression to subtract, nor the loss due to large leakage and heat losses under the high pressure and intense heat at the beginning of the stroke. Taking these facts into consideration we will probably receive, in addition, a third of the power of the first half of the stroke, thus obtaining a gain of 25 per cent in efficiency. The output of the motor, however, is only two-thirds of the maximum obtainable under the old system, and the efficiency and power may be still further increased by raising the compression, for the high point of compression is reached so much later in the stroke that the point of self-ignition will be just that much nearer the dead center. Another advantage of this plan is the fact that the average temperature is lower, and it is thus better suited for air cooling. The dimensions given are used only for convenient illustration. They will, however, probably be found, after experiment, to be not very far from the correct ones.

For a company that intends to take up the manufacture of such an engine the best plan will probably be to make up an adjustable engine—one in which the compression space and the length of the stroke can be varied at will, independently of one another. The cam should be designed so as to give a variable cut-off. This latter feature should also be made an element of the standard engine, as it can then be used in place of a throttle valve, and also for providing a larger charge when at high altitudes, so that the same compression may be obtained then as at lower levels. With the present type of engine there is a large loss in power capacity in high altitudes, as has often been demonstrated with automobiles when taken to elevated sections of the country.

CHAPTER II

FORMULAS AND CONSTANTS FOR COMBUSTION ENGINE DESIGN*

In the following chapter are given a number of formulas abstracted from an article by Mr. Sanford A. Moss, published in MACHINERY, February, 1906, and from an article by Mr. G. W. Rice, published in the *Sibley Journal of Engineering*, June, 1906. This latter article was abstracted in MACHINERY, October, 1906. The formulas are rational whenever possible, but in some cases, of course, they are necessarily empirical. An attempt has, however, been made to place them on a more rational basis than usual. The constants and coefficients given are for the most part taken from an investigation of current practice in gas engine design made at the Cornell University a few years ago. The data on which the formulas are based was obtained from a great number of builders and represent the average of a great variety of commercial engines. An effort has been made to arrange the formulas in the most convenient form for the use of the designer.

The cases to which the rules mainly apply are that of a single-acting trunk piston, stationary gas engine between 5 and 100 horsepower, and that of the light-weight type engine used in automobile practice. Some formulas for heavy marine practice, however, are also given when this practice differs from ordinary stationary and automobile gas engine practice. In the following, will be given, first, rules and formulas which are especially applicable to stationary gas engine practice, and second, formulas more especially applicable to automobile and marine engines.

I. FORMULAS FOR STATIONARY GAS ENGINE PRACTICE

In the following formulas all pressures and stresses are in pounds per square inch, and all dimensions in inches. The maximum pressure during normal operation varies from 250 to 350 pounds, the usual value being 300 pounds per square inch. The stresses in the various parts which are of most importance are the continuously repeated stresses due to constant repetition of the normal pressure, and not the occasional higher stresses due to a high value of the maximum pressure produced by excessive explosions now and then. Hence the normal value of the maximum pressure should be used in the formulas rather than the occasional extreme value sometimes occurring.

Thickness of Cylinder Walls

The thickness of the cylinder walls depends upon the stress s which can be safely allowed for continuous service. Considering the cylinder as an indefinitely long pipe with uniform fluid pressure, and adding a

* MACHINERY, February and October, 1906.

constant for reboring, crooked cores, etc., the thickness necessary for a stress s is

$$t = \left(\frac{1}{2s}\right) pd + \frac{1}{4}$$

in which formula

t = thickness of cylinder walls,

p = maximum pressure during normal operation,

d = diameter of cylinder.

Owing to the stiffening effect of the jacket, unstressed portions of walls, and cylinder ends, a rather high value of the apparent stress may be used in this formula. A safe value is 2450. Then

$$t = 0.000204\, pd + \frac{1}{4}$$

If p has the usual value, 300, this reduces to

$$t = \frac{d}{16} + \frac{1}{4}$$

Thickness of Jacket Walls and Water Jacket

The thicknesses of the jacket walls, T, and water jacket, j, are determined almost wholly by considerations of molding and casting, and depend directly on the thickness of the cylinder walls t.

Safe values are

$$T = 0.6\, t, \text{ and } j = 1\tfrac{1}{4}\, t$$

Cylinder Head Studs

If the number of cylinder head studs is q, satisfactory results will be obtained by use of the empirical expression

$$q = \frac{2\,d}{3} + 2.$$

The nearest whole number must, of course, be used.

If the initial load on the studs, caused by screwing them up, is not greater than the load due to explosion, the latter gives the maximum stress s at the root of the threads of the studs. (The nuts, of course, may be carelessly screwed up tighter, causing unknown stresses.) It may be shown (since the area at the root of a thread is about 0.7 of the outside area) that the outside diameter o necessary for a stress s is

$$o = \frac{d}{\sqrt{0.7\, s}} \sqrt{\frac{p}{q}}$$

A safe value for the stress is 7800. Then

$$o = 0.0135\, d \sqrt{\frac{p}{q}}$$

If p has the usual value, 300, and q is 8, or thereabouts, this reduces to $o = 1/12\, d$.

Length of Piston

Let u be the ratio of the length of the connecting-rod (distance between centers) to the radius of the crank. A usual valve for this is about 5. Let b be the average bearing pressure on the projected area of piston (Ld square inches) during the explosion stroke. The piston must be long enough to give a safe value to b in order to avoid undue wear. It can be shown that the average total load on the projected area of the piston (due to connecting-rod thrust only) is $pd^2 \times \frac{0.22\,\pi}{4\,u}$. Hence the length of piston L, necessary for a bearing pressure b is

$$L = \left(\frac{0.22\,\pi}{4u}\right)\frac{pd}{b}$$

A safe value for b is 7 pounds per square inch. Then

$$L = \frac{0.025\,p\,d}{u}$$

If p has the usual value of 300, and u the usual value 5, this reduces to

$$L = 1\tfrac{1}{2}\,d$$

The weight of the reciprocating parts produces an additional pressure on the projected area of the piston, which is usually slight compared with the rod thrust. In cases where it becomes appreciable it should be taken account of, however, by adding the bearing pressure produced by the weight to that produced by the rod thrust.

Thickness of Rear Wall of Piston

The thickness Z of the rear wall of the piston depends upon the stress s which can be safely allowed. By considering the wall as a circular plate, fixed at the circumference, and without ribs, it may be shown that the thickness necessary for a stress s is

$$Z = \left(\frac{0.41}{\sqrt{s}}\right) d\sqrt{p}$$

Owing to the fact that ribs are usually added to help support the wall, a high value of the apparent stress may be used in this formula. A safe value is 5320. Then

$$Z = 0.00562\ d\sqrt{p}$$

If p has the usual value, 300, this reduces to

$$Z = \frac{d}{10}$$

Length and Diameter of Wrist-pin

Let s be the stress in the wrist-pin due to continuous repetition of the pressure p, and b the bearing pressure on the projected area of the wrist-pin at the instant of maximum pressure, when the tendency to squeeze out the oil is greatest. The length l_2 and diameter d_2 of the wrist-pin must be such as to give s and b safe values. By taking the wrist-pin as a beam uniformly loaded, and supported at points l_2 inches apart, it may be shown that the diameter and length necessary for a stress s and a bearing pressure b are

$$d_2 = d\sqrt[4]{\frac{\pi}{4sb}}\sqrt{p}, \text{ and } l_2 = d_2\sqrt{\frac{\pi s}{4b}}$$

Safe values for s and b are 10,500 and 2800, respectively. Then

$$d_2 = 0.0128\, d\sqrt{p} \text{ and } l_2 = 1\tfrac{3}{4}\, d_2$$

If p has the usual value of 300, this reduces to

$$d_2 = 0.22\, d, \text{ and } l_2 = 1\tfrac{3}{4}\, d_2$$

Area of Mid-section of Connecting-rod

Let k be the factor of safety of the connecting-rod, or ratio of the breaking load to the actual maximum working load. Then the area must be such that k has a safe value. Let c be the distance from center to center of rod, and r the radius of gyration of the mid-section.

If the mid-section is round, having a diameter D, $r^2 = \dfrac{D^2}{16}$

If the mid-section is rectangular, having a height H, then $r^2 = \dfrac{H^2}{12}$.

It can be shown (by using Ritter's formula for long columns; end coefficient unity for ends free but guided; elastic limit of material 35,000; modulus of elasticity, 29,000,000; neglecting obliquity and inertia of rod which nearly neutralize each other) that the area a necessary to give a factor of safety k is

$$a = \frac{k}{44{,}560}\, p\, d^2\left(1 + \frac{0.00012\, c^2}{r^2}\right)$$

A safe value for k is 3.9. Then

$$a = 0.0000875\, p\, d^2\left(1 + \frac{0.00012\, c^2}{r^2}\right)$$

If p has the usual value 300, if the mid-section of the rod is circular and of diameter D, and if the proportions are such that $1 + \dfrac{0.00012\, c^2}{r^2}$ has a value of about 1.6 as is usually the case, this expression reduces to

$$D = 0.23\, d$$

Diameter of Crank-pin

The diameter of d_3 of the crank-pin depends upon the stress s which can be safely allowed. Let l_3 be the length of crank-pin journal, l_1 the length of the main bearing journal, and m one-half of the distance from center to center of the main bearings. A center-crank engine is assumed.

$M = m - (\frac{3}{8} l_s + \frac{1}{4} l_1)$ is a quantity needed in our formulas. This is the arm of the effective bending moment on the crank-pin for the stress caused by the reaction on the main bearing due to the explosion. This bending moment is the only one which need be taken into account. It can be shown that all other effects, such as inertia, centrifugal force, obliquity of rod, effect of counter balances, weight of flywheels, belt-pull, etc., all practically neutralize each other.

Then the usual relation between stress and bending moment gives as the diameter necessary for a stress s

$$d_s = \sqrt[3]{\frac{4}{s} M p d^2}$$

A safe value for s is 10,600. Then

$$d_s = \sqrt[3]{0.00038 M p d^2}$$

If p has the usual value of 300, and if the proportions are such that M is about 0.6 d, as is usually the case, this expression reduces to

$$d_s = 0.41 d$$

Length of Crank-pin Journal

Let d_s be the diameter of the crank-pin, and let b be the average bearing pressure on the projected area of the crank-pin due to the average value of load during a complete cycle. The length of the crank-pin must be such that b has a safe value, in order to avoid heating. It can be shown that the average value of the total load on the crank-pin, taken regardless of directions, is about 14½ per cent of the maximum load due to the explosion. Hence the length l_s of crank-pin necessary for a bearing pressure b is

$$l_s = \left(\frac{0.145 \pi}{4 b}\right) \frac{p d^2}{d_s}$$

A safe value for b is 213. Then

$$l_s = \frac{0.000535 p d^2}{d_s}$$

If p has the average value 300, and if d_s is 0.41 d as previously given, this expression reduces to

$$l_s = 0.95 d_s$$

Dimensions of Crank Throws

Let x be the thickness (in the direction of the shaft axis) of the throws of a center-crank engine, y the breadth (perpendicular to the shaft axis) and d_s the diameter of the crank-pin. Then the following are safe values:

$$x = \frac{5}{8} d_s, \quad y = 2\frac{1}{8} x.$$

Diameter of Crank-shaft at the Main Bearings

The diameter of the crank-shaft at the main bearings depends on the stress s which can be safely allowed at the inner edges of the main bearing journals. Let l_1 be the length of the main bearing journal, and l the length of the stroke. A center-crank engine is assumed.

$M_1 = 0.325\, l_1 + 0.09\, l$ is a quantity needed in our formulas. This is the arm of the effective bending moment on the crank-shaft at the inner edge of the main bearing, for the stress caused by the reaction on the main bearing due to the explosion. It can be shown that this gives a moment equal to the combined bending and twisting moment, taking flywheel, weight, belt-pull, etc., into account. Then the usual relation between stress and bending moment gives as the diameter necessary for a stress s

$$d_1 = \sqrt[3]{\frac{4}{s} p\, d^2 M_1}$$

A safe value for s is 9500. Then

$$d_1 = \sqrt[3]{0.000422\ p\, d^2 M_1}$$

If p has the usual value 300, and if the proportions are such that M_1 is about $0.4d$, as is usually the case, this expression reduces to $d_1 = \frac{3}{8} d$.

Length of Main Bearing Journals

A single cylinder with two main bearings is assumed. Let d_1 be the diameter of the crank-shaft at the main bearing, and let b be the average bearing pressure on the projected area of the main bearing due to the average value of the load during a complete cycle. The length of the main bearing must be such that b has a safe value, in order to avoid heating. It can be shown that the average value of the total load on the main bearings, taken regardless of directions, and taking into account belt-pull, flywheel weight, etc., is about one-third of the maximum load due to the explosion. Hence the length of each main bearing necessary for a bearing pressure b, is

$$l_1 = \left(\frac{\pi}{24\, b}\right) \frac{p\, d^2}{d_1}$$

A safe value for b is 123. Then

$$l_1 = \frac{0.001068\ p\, d^2}{d_1}$$

If p has the average value 300, and if d_1 is $\frac{3}{8}\, d$, as previously found, this expression reduces to

$$l_1 = 2\tfrac{1}{4}\, d_1$$

Outside Diameter of Flywheel

The stress in the rim of a cast-iron flywheel of the usual type depends directly upon the velocity of the rim V, in feet per minute. Hence the flywheel diameter should be such as to give V a safe value. If N is the number of revolutions per minute, the diameter D necessary to give a velocity V, is

$$D=\left(\frac{12\,V}{\pi}\right)\frac{1}{N}$$

A safe value for V is 3220 feet per minute. Then

$$D=\frac{12{,}300}{N}$$

Weight of Flywheels

Let W be the total weight of all flywheels, in pounds, for the case of a single cylinder, hit-and-miss engine. Let f be the speed fluctuation coefficient. This is the ratio of the difference between the maximum and minimum revolutions per minute, to the average revolutions per minute, N. The flywheels must be such as to give a safe value to f. Let H be the rated brake horsepower, and D the outside diameter of the wheels. The greatest fluctuation is at light loads, and the least working load is taken to occur when the engine misses three times between each fire. Then it can be shown (on the basis that maximum indicated horsepower is 1.4 times rated brake horsepower H; that the radius of gyration of an average flywheel is 0.83 of the outside radius; and that the ratio of the energy added to the wheel and causing the maximum acceleration in the case considered, to the net indicated energy developed per cycle if exploding every time, is 1.197) that the flywheel weight required to give a fluctuation coefficient f is

$$W=\frac{272{,}300{,}000{,}000}{f}\times\frac{H}{D^2\,N^3}$$

A safe value for f for ordinary engines is 0.054 (5.4 per cent). Then

$$W=5{,}000{,}000{,}000{,}000\,\frac{H}{D^2\,N^3}$$

If we insert the average value of the rim diameter found above $\left(D=\frac{12{,}300}{N}\right)$ this expression reduces to

$$W=\frac{33{,}000\,H}{N}$$

II. FORMULAS FOR LIGHT-WEIGHT TYPE GAS ENGINES

In the following are given formulas applying particularly to light-weight gas engines, especially such as are used in automobile and marine practice. As these formulas are deduced from actual practice as exemplified in approved designs of such machines, they are of special value to designers. All pressures and stresses are in pounds per square inch, and all dimensions in inches. In order to get the maximum explosion pressure which is necessary for finding the stresses in the engine parts, the assumption is made that the compression pressure is one-fourth of the maximum explosion pressure. This assumption is very nearly correct and is used throughout the remainder of this chapter.

Ratio of Length to Diameter

While in stationary gas engines running at slow speed, the stroke is about 1.5 times the bore for thermodynamic reasons, practice differs in high-speed petrol engines.

Let l = cylinder length in inches,
D = diameter of cylinder in inches.

Values of l and D were plotted from actual designs, giving 1.07 as mean value of A in formula $l = A\,D$.

The designer's formula is, hence,

$$l = 1.07\,D.$$

Let D = the cylinder bore,
l = length of stroke,
R. P. M. = revolutions per minute,
C = clearance as a fraction of piston displacement.

The equation for the maximum horsepower is a rational formula, the constant in it being based on current practice.

$$\text{H. P. per cylinder} = \frac{D^2 \times l \times \text{R. P. M.} \times (0.48 + 0.1\,C)}{14{,}000}.$$

Thickness of Cylinder Wall

The thickness of the cylinder wall depends on the stress which can safely be allowed for continuous service. On account of the desire for lightness and the stiffening action of the jacket wall, this stress is taken as high as possible; in fact, instead of allowing the usual constant for reboring, it was found on plotting the data from engines in actual practice that this constant had a negative value of ⅛ inch.

Let t = thickness of cylinder wall,
s = allowable stress per square inch,
p = maximum explosion pressure,
D = cylinder diameter.

The designer's formulas are then:

$$t = \frac{p\,D}{5300} - \tfrac{1}{8}\text{ inch.}$$ (Light automobile practice).

$$t = \frac{p\,D}{3700} - \tfrac{1}{8}\text{ inch.}$$ (Medium weight practice).

$$t = \frac{p\,D}{3200} - \tfrac{1}{8}\text{ inch.}$$ (Heavy marine practice).

$$t = \frac{D}{16}$$ (Rough rule, not considering pressure).

Thickness of Integral Cast Cylinder Heads

The common form of head is that of a flattened ellipse. Liberal fillets should be used where the head joins the cylinder wall, and the head may be gradually reduced in thickness when approaching the center. Close to the cylinder wall $t = 0.005\, D \sqrt{p}$; at the center $t = p\, D \div 1.5\, s$.

Thickness of Jacket Wall

The jacket wall is made as thin as it can be cast in the foundry; in some cases it is deposited electrolytically of copper; in other cases the cylinder is cast without a jacket, turned up inside and out and a thin metal jacket of copper or brass applied. This latter practice came to the front a great deal a few years ago. In cylinders made in this manner the cylinder wall has a constant thickness, which is something which cannot be said of the ordinary type. It is also of a very light construction.

Length of Piston

The normal pressure between piston and cylinder wall for any point in the piston stroke is equal to the pressure on the piston head divided by the ratio of connecting-rod to crank length. By assuming an average clearance and different ratios of connecting-rod to crank, it was found that the average pressure on the piston head when the connecting-rod and crank were at right angles, giving the maximum normal pressure on the piston, was 0.23 times the maximum pressure. The designer's formulas are:

$$l = 0.0167\, p \frac{D}{c}$$

$$l = 1.125\, D$$

in which

p = maximum pressure on piston in pounds per square inch,
c = ratio of the connecting-rod to the crank,
l = length of the piston.

Thickness of Rear Wall of Piston

Let t = thickness of unribbed rear wall of piston,
p = maximum pressure in pounds per square inch,
D = diameter of cylinder.

The designer's formula is

$$t = 0.0034 \sqrt{p} \times D.$$

By plotting between piston head thickness and cylinder diameter, we get the rough design formula: Allow 1-16-inch thickness per inch of cylinder diameter.

Dimensions of Piston Rings

In the consideration of piston ring dimensions, the first proportion with which we are interested is the diameter to which the outside of the cast-iron ring is finished. This must be a diameter slightly greater than the bore of the cylinder so as to furnish a sufficient packing action to the piston. This diameter, d, is the same for eccentric turned rings as for non-eccentric ones, and by plotting between ring diameter and cylinder diameter it was found that the ring was turned to 1.03 times the cylinder diameter. Due to the heat of the burning gases expanding the piston head, that end of the piston which is in contact with the hot gases must be made slightly smaller down to the first ring than the rest of the piston. This allowance is usually taken as 0.001 inch per inch diameter of cylinder.

For plain rings of constant thickness the width, w, should be 0.07 of the cylinder diameter and the thickness, t, of the ring 0.5 of the width. The number of rings used by different builders varies widely, the common practice being three at the head end of the piston and one, known as an oil ring, at the open end.

The designer's formulas are:

$$d = 1.03\,D; \quad w = 0.07\,D; \quad t = 0.5\,w.$$

Design of Wrist-pin

The average pressure on the piston-pin will be the same as on the crank-pin, neglecting inertia effects.

p = maximum pressure in the cylinder,
d = diameter of wrist-pin,
l = length of wrist-pin,
D = cylinder diameter.

The designer's formulas are:

$$d\,l = 0.000445\,pD^2$$
$$l = 2\tfrac{1}{4}\,d$$
$$d = 0.225\,D$$

Crank-pin Design in Engines with Main Bearing Each Side of Crank-pin

Below are given data on the ultimate strength of 14 crank-shafts having an average ultimate strength of 95,000 pounds per square inch.

Autocar	85,000	Pierce	105,000	Columbia	90,000
Moline	90,000	Haynes	90,000	Covert	80,000
Packard	100,000	Lozier	100,000	Acme	90,000
St. Louis	70,000	S. and M.	125,000	Thomas	105,000
Nameless	85,000	Welch	115,000		

The designer's formulas for this type of crank-shafts are:

$$d = \frac{D}{43.2}\sqrt{p} = \text{diameter of pin,}$$

$$l = 1\,1/3\ d = \text{length of pin.}$$

The latest practice employs steel of special composition giving greater hardness and a very high tensile strength.

When special alloy steels are used, these formulas may have to be modified to suit the high tensile strength of the steel employed.

Crank-pin Design in Engine not having Main Bearing Each Side of Crank-pin

Assume for this type of engine $d = 2$ inches on the average.

$$d = \frac{D}{36.5} \sqrt{p} + 0.9 \text{ inch},$$

$$l = 3.75\, d - 3.75 \text{ inches}.$$

A general average of all cases shows that the diameter of the crank-pin equals $\frac{D}{2.8}$. Again the general average shows that the projected area of the crank-pin is 1/5 of the piston area.

Design of Main Bearings

d = diameter of main bearing.

The length of main bearing per cylinder in four-cylinder engines with five main bearings is 2.82 d.

The length of main bearing per cylinder in four-cylinder engines with three main bearings is 1.54 d.

The length of main bearings per cylinder in two-cycle engines is 4.45 d. (This applies to one- and two-cylinder engines only.)

The relative lengths of these bearings, among themselves, varies with the cylinder arrangement—whether they are cast in pairs, separately, etc. In all cases, the bearing at the flywheel or power end of the shaft is made longer than any of the others because the weight of the wheel rests almost directly on it and, therefore, the average total pressure is much greater than on the others.

The designer's formulas are—for length of journal—given above.

$$\text{Diameter} = 7.24 \sqrt[3]{\frac{\text{H. P. per cylinder}}{\text{R. P. M.}}}.$$

Crank Throws or Webs

Let d = diameter of main bearing,
d_1 = diameter of crank-pin,
h = depth of crank throws,
b = thickness of crank throws,
b_1 = thickness of crank throws on flywheel side,
b_2 = thickness of long crank throws,

The designer's formulas are:

$$d^3 = b\, h^2$$
$$h = 2.6\, b$$
$$b_1 = 1.25\, b$$
$$h = 1.33\, d_1$$
$$b_2 = 1.25\, b_1$$

Inertia Effects of Reciprocating Parts

Let F = inertia effects in pounds per square inch of piston area,
W = weight of (piston + 2/3 connecting-rod),

N = R. P. M.,
r = one half stroke, in feet,
c = ratio of connecting-rod to crank,
D = cylinder diameter, inches,
w = weight of reciprocating parts per square inch of piston area.

$$F = \frac{W \times N^2 \times r \times 0.00034}{0.7854\, D^2} \times \left(1 + \frac{1}{c}\right)$$

Now by plotting we find that the weight of reciprocating parts is 0.55 pounds per square inch of piston, and the value of c is 4. We may then rewrite the above equation as follows:

$$\begin{aligned} F &= 1.25\,(w \times N^2 \times r \times 0.00034) \\ &= 1.25\,(0.55 \times N^2 \times r \times 0.00034) \\ &= 0.0002435\,(N^2 \times r) \end{aligned}$$

This gives us a simple equation for inertia effects of a given engine at a given speed.

Stress in Connecting-rod Bolts

The stress in the bolts of the connecting-rod is almost entirely due to the inertia pressures at the end of the stroke. This stress may be found from the preceding formula by plotting the maximum inertia pressures at the engine's rated speed with the reduced bolt area. That is the area at the bottom of the threads. The average ratio of thread area to bolt area is 0.65 for the sizes commonly used in automobile engine construction.

Flywheel Design

In the design of a flywheel for an automobile engine we have a proposition entirely different from the design of a flywheel for any type of stationary engine. In the automobile the function of the flywheel is not to keep the engine speed constant, but to furnish a storage reservoir of energy sufficient to start the car under any working conditions or to keep the engine turning over when running at very low speed and under heavy load. Current practice does not help us as much as it might in this particular, for the weights of flywheels used for the same powered engines vary widely among the different builders. The weight depends, first upon the diameter, and this depends, to a large extent upon where the wheel has to be put; second, upon the weight of the loaded car, relative to the power of the engine. It also depends upon the gearing ratio of the car and other things relative to the car design.

By plotting between engine stroke and flywheel diameter, we find that the diameter varies from 4.9 to 2.9 times the engine stroke, the average value of flywheel diameter being 3.5 times the engine stroke.

Engine Weight

Instead of comparing the engine weight with the horsepower, as is usually done, let us compare it with the cubic inches of piston displacement. By plotting between the weight of the complete engine and cylinder volume in cubic inches, we find:

$$W = 1.125\,V + 100.$$

On plotting between engine weight without flywheel and cubic inches of piston displacement, we find:

$$W = 1.125\ V.$$

This indicates that irrespective of the power of the engine, the builders have always used a flywheel of about 100 pounds weight.

By plotting between engine weight and horsepower, we find the average value to be 17.6 pounds per horsepower.

Diameter and Lift of Exhaust Valves

Let D = cylinder diameter,
L = length of stroke,
N = R. P. M.,
S = allowable speed of gas in feet per minute = 3520,
d = diameter of exhaust valve,
h = lift of exhaust valve.

In high-speed engines the ring area open to gas passage, seems to be the all important item, and not the diameter of the valve itself. The tendency being to keep the valves large in diameter, and to make the lift as small as possible, 7/16 inch was the highest lift noted on about 80 engines, with cylinder sizes up to 7 × 9 inches, while the theoretical lift would be ¼ of the diameter of the valve. About 5/16 inch is a popular lift in this country, while the French use much lower lifts. These low valve lifts are used in order to get a quick closing valve and to prevent hammering of the cams on the valve push rods.

The designer's formula is:

$$D^2LN = 84{,}500\ dh.$$

Valve Thickness

For the thickness of the exhaust and inlet valves the formula of Reuleaux may be used:

$$t = r\sqrt{\frac{p}{s}}$$

in which

t = thickness,
r = radius of supporting circle,
p = maximum pressure in cylinder,
s = fiber stress.

As given by another designer this is modified to read:

$$t = 0.45\ d\sqrt{\frac{p}{s}}$$

The maximum normal pressure of the valve on its seat is given by several authorities as 900 pounds per square inch and when a conical seated valve is used the angle is usually taken between 45 and 70 degrees, which makes the effective lift of the valve equal to the real lift times the sine of the valve angle which may be approximated at 0.75. The diameter of the valve stem is taken as 1/5 valve diameter.

Inlet Valve Design

Most that has been said relative to the exhaust valve may be applied to the inlet valve. The valves themselves are very often made interchangeable, but they are usually given different lifts, that of the inlet valve being smaller. The designer's formula is:

$$D^2LN = dh \times 107{,}000.$$

Speed of Exhaust Gases through Pipe

Let D = cylinder diameter in inches,
L = length of stroke,
N = R. P. M.,
S = allowable speed of gas in feet per minute = 6550,
a = area of exhaust pipe (nominal).

The designer's formula is:

$$a = \frac{D^2LN}{50{,}000}.$$

Speed of Gases through Inlet Pipe

The designer's formula is:

$$a = \frac{D^2LN}{80{,}000}.$$

S = 10,000 feet per minute.

Two-cycle Port Design

The design of ports for two-cycle engines depends upon two important factors. First, the height of the port determines the valve timing of the engine, and this timing must be arranged to give the proper results when the engine is running at slow speed. Next, the ports must be extended around the cylinder until a sufficient area is obtained to give the required engine speed. The two points then to consider are valve timing, and limiting gas velocities. This valve timing is very nearly constant for all the engines, the average values being 88 degrees for the inlet ports and 110 degrees for the exhaust ports. The velocity of the gases through the ports was found by assuming full port opening from the time the port began to open to the time it closed. The exhaust gas velocity was found to be quite constant at about 7500 feet per minute. The inlet gas velocity varied with the crank-case pressure, but as this pressure is either about 4 or 8 pounds, we find two values for inlet gas velocity. The gas velocity corresponding to 4 pounds is 12,000 feet per minute, while that corresponding to 8 pounds is 24,000 feet per minute.

Compression Pressure and Clearance*

In designing the cylinders of a gasoline motor, one of the first things to determine is the volume of the clearance space. In order to find

* Herbert C. Snow, MACHINERY, July, 1910.

this, the compression which the motor is to have must be known. Theoretically the compression of an engine depends upon the clearance, and from theory we can compute the compression of any engine of which we know the clearance volume. In practice we never get a full cylinder of explosive mixture, and the percentage which we do get depends upon the engine speed, the amount the engine is cooled, the temperature of entering charge, and the make of carburetor.

In automobile motors the compression varies all the way from 60 pounds to 100 pounds per square inch gage. The formula most generally used for finding the clearance space is:

$$\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n \text{ or } p_1 = p_2\left(\frac{v_2}{v_1}\right)^n$$

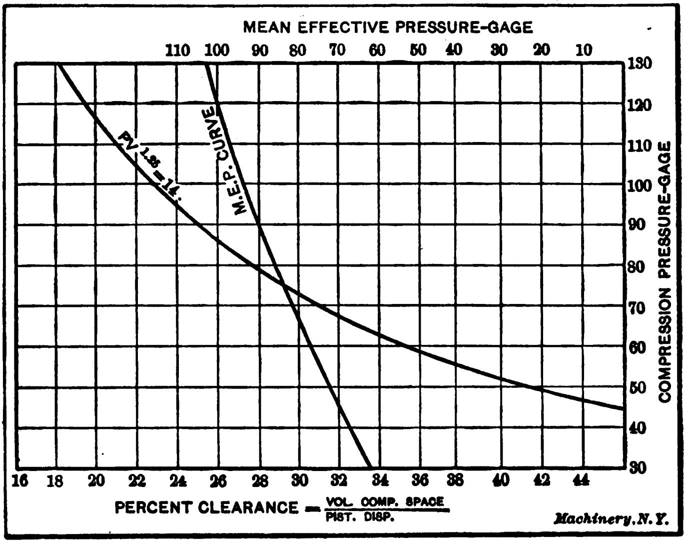


Fig. 1. Chart for Determining Percentage of Clearance Space in Gas Engine Cylinders

where p_1 = compression pressure in pounds, absolute,

p_2 = initial pressure,

v_1 = volume of clearance space,

v_2 = total volume of cylinder (piston displacement plus clearance space).

The value of n given varies from 1.21 to 1.41. This value of n varies with the engine speed, valve timing, valve diameter, valve lift, size and shape of the inlet pipes, and temperature of jacket walls. It will be seen from this that the compression cannot be calculated exactly, since it depends on so many variable quantities. It has been found by making tests of a number of gasoline motors that a value of $n = 1.25$, when the initial pressure is 14 pounds, gives very close results.

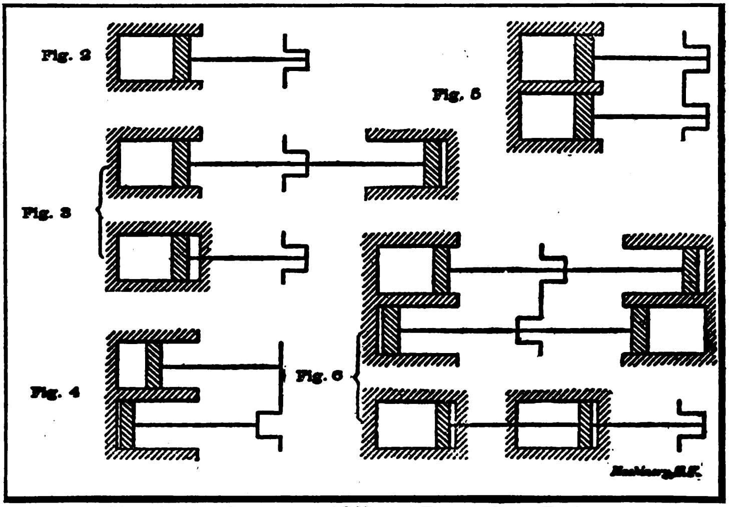
Using this value the accompanying diagram, Fig. 1, was plotted. To find the percentage of clearance for a given compression, find the compression in pounds gage in the column at the right, and follow across to the compression curve, thence to the bottom of the diagram where the percentage of clearance is read.

To find the mean effective pressure for a given compression, take the compression in the column to the right, as before, and follow across to the M. E. P. curve, thence to the top where the mean effective pressure is read. These curves give values which come very close to the average for automobile motors.

Formula for Gas Engine Flywheels*

The following formula for the calculation of flywheels for gas engines is applied by Mr. R. E. Mathot to all classes of engines. If, in the formula,

P = the weight of the rim (without arms or hub) in tons,
D = diameter of the center of gravity of the rim in meters,



Figs. 2 to 6. Diagrams of Different Types of Gas Engines

a = the amount of allowable variation,
n = the number of revolutions per minute,
N = the number of brake horsepower,
K = coefficient varying with the type of engine,

$$\text{then, } P = K \frac{N}{D^2an^3}$$

If D is transformed into feet, the formula will read:

$$P = K \frac{10.75\,N}{D^2an^3}.$$

* MACHINERY, August, 1907.

The coefficient K, which varies with the type of engine, is determined as follows:

$K = 44{,}000$ for Otto-cycle engines, single-cylinder, single-acting. (Fig. 2.)

$K = 28{,}000$ for Otto-cycle engines, two opposite cylinders, single-acting, or one cylinder double-acting. (Fig. 3.)

$K = 25{,}000$ for two cylinders single-acting, with cranks set at 90 degrees. (Fig. 4.)

$K = 21{,}000$ for two cylinders, single-acting. (Fig. 5.)

$K = 7000$ for four twin opposite cylinders, or for two tandem cylinders, double-acting. (Fig. 6.)

The factor a, the allowable amount of variation in a single revolution of the fly-wheel, is as follows:

For ordinary industrial purposes	1/25 to 1/30
For electric lighting by continuous current	1/50 to 1/60
For spinning mills and similar machinery	1/120 to 1/130
For alternating current generators in parallel	1/150

The total weight of the fly-wheel may be considered as equal to $P \times 1.4$.

CHAPTER III

CRANK-SHAFTS FOR INTERNAL COMBUSTION ENGINES*

The crank-shaft may rightly be termed the "backbone of the engine," and the designer should use his best judgment and skill in designing this most important member. In the following, the formulas are given for crank-shafts for determining the diameter, and the breadth and thickness of cheeks, of single-acting, single-cylinder, four-cycle engines of medium speed with overhanging flywheels and two bearings, of the ordinary type up to 75 horsepower. The formulas have been actually applied to engines up to 60 horsepower, and as far as known, no crank-shaft has ever failed. The data on which the formulas are based have been obtained from a complete line of engines built by one of the largest manufacturers, and from a few sizes from some other concerns.

Diameter of Crank-shaft

The strength of a shaft as regards twisting, is proportional to the cube of the diameter, while the torque exerted upon the shaft by the force of the explosion is proportional to the square of the bore of the cylinder and to the throw of the crank, which is half the stroke. Expressing this as a formula we may write:

$$\frac{B^2 \times \frac{1}{2} S}{D^3} = c \qquad (1)$$

in which B = bore of cylinder in inches,
S = stroke of piston in inches,
D = diameter of crank-shaft in inches,
c = constant determined by good practice.

The usual value of constant c for existing successful engines varies between 10 and 19, the average value being about 15, which, in the following formulas is assumed as good average practice.

The stroke S may be expressed as a function of the bore B. Assume, for example,

$$S = a B$$

Substituting in Equation (1) we have:

$$\frac{B^2 \times \frac{1}{2} a B}{D^3} = 15$$

Solving for D we have:

$$D = B \sqrt[3]{\frac{a}{30}} \qquad (2)$$

* MACHINERY, April, 1910.

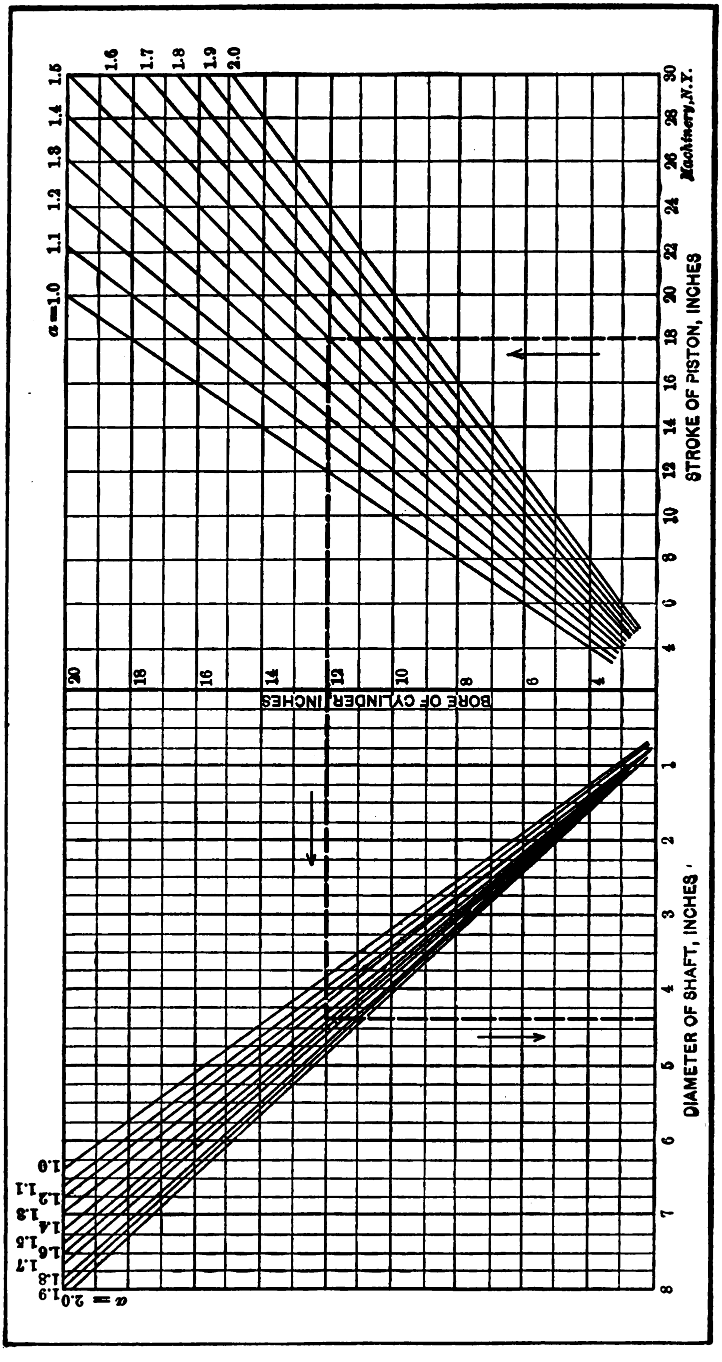


Fig. 7. Diagram for Obtaining the Diameters of Crank-shafts for Gas Engines

The average engine has a stroke of about 1½ times the bore, but this factor varies from 1¼ to 2. For $a = 1\frac{1}{4}$, $D = 0.347\ B$, and for $a = 2$, $D = 0.405\ B$.

Fig. 7 is a diagram for determining diameters of crank-shafts for gas engines, according to the formula given. To use the diagram, find the intersection of the bore- and stroke-lines at the right-hand side, and determine the corresponding value of a on the diagonal lines. Then follow the horizontal line from the intersection to the left-hand side of the chart to the same value of a on the diagonal lines, and then follow the vertical line down to the proper diameter of shaft. The heavy dotted line in the diagram shows the procedure for a 12-inch bore, 18-inch stroke engine, the diameter of crank-shaft obtained being 4⅜ inches.

Size of Bearings

For best results the mean pressure on the crank-pin should not exceed 400 pounds per square inch of projected area. In the following, 390 pounds per square inch has been assumed. The total pressure on the crank-pin bearing is the area of the piston multiplied by the mean effective pressure (M. E. P.).

Hence, if P_1 = unit pressure on pin (390 pounds per square inch of projected area),
B = bore of cylinder,
l = length of crank-pin bearing,
d = diameter of crank-pin bearing,

then:

$$P_1 = \frac{0.7854\ B^2 \times \text{M.E.P.}}{l\ d}$$

and if the mean effective pressure is assumed as 75 pounds per square inch, then,

$$l\ d = \frac{0.7854\ B^2 \times 75}{390} = 0.151\ B^2$$

As in these bearings, length and diameter are usually equal ($l = d$), we have:

$$d^2 = 0.151\ B^2, \text{ or } d = 0.39\ B \qquad (3)$$

When this formula gives d smaller than D, the value of d should be increased, because in practice the diameter of the crank-pin is never made smaller than that of the shaft. They are usually made of the same size, and it is even better practice to make the crank-pin at least 1⅛ times the diameter of the shaft.

The diameter of the shaft in the main bearings is usually about 1.1 times the diameter of the shaft, or

$$d_1 = 1.1\ D \qquad (4)$$

Making the shaft in this manner slightly increases the cost, but this style is used in the best designs.

The length of the main bearings varies from 1.75 to 2 times the diameter of the shaft, or

$$l_1 = (1.75 \text{ to } 2) \times D \qquad (5)$$

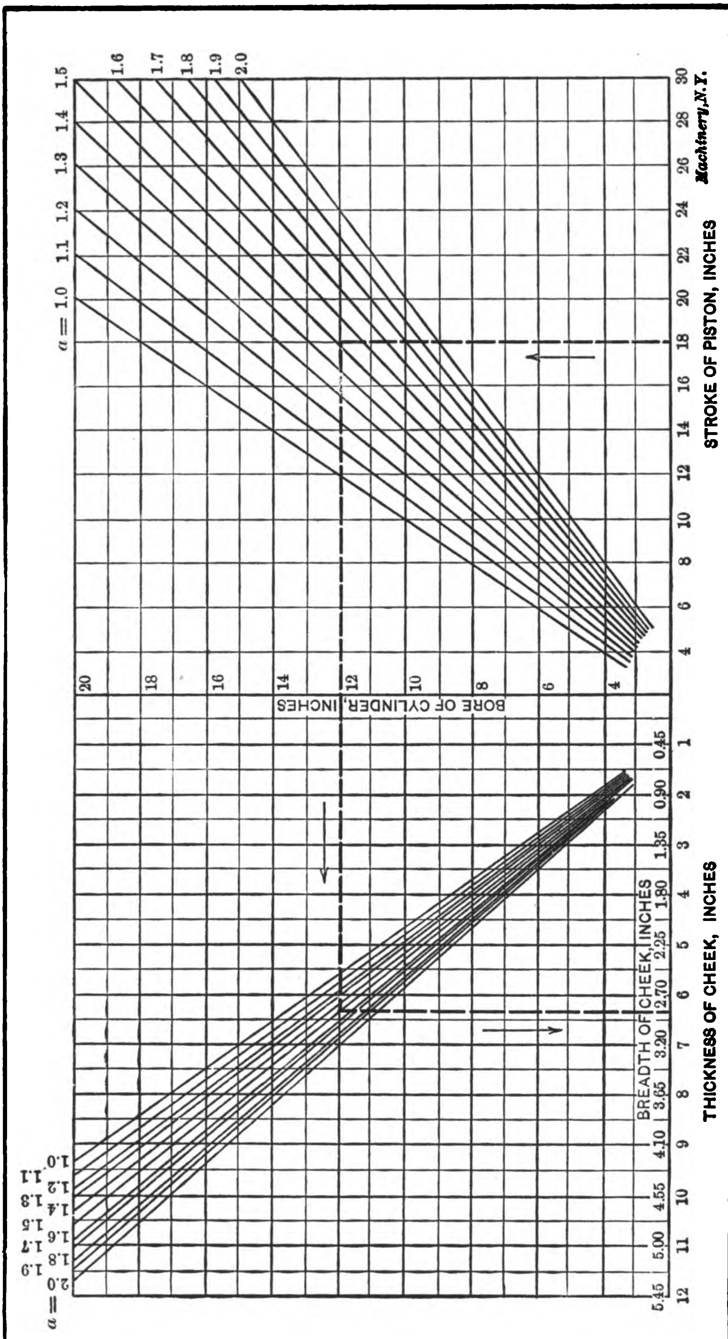


Fig. 8. Diagram for the Cheek Dimensions of Gas Engine Crank-shafts

Dimensions of Cheeks

Having now obtained the various diameters of the shaft and the lengths of the bearings, we will determine the formulas for the cheeks. The crank cheeks in horizontal and in the larger vertical engines are machined and rectangular in section. They act as a beam supported at one end. The strength of such a beam is directly proportional to its breadth b and to the square of its thickness t, and inversely proportional to the length (which, in turn, is proportional to the stroke S). The load acting at the end of the beam is proportional to the square of the bore B. Hence,

$$\frac{b\,t^2}{B^2\,S} = c.$$

The average value of the constant c is 0.045, which has been found by plotting the constant for existing engines and assuming an average value. The thickness t is usually made about 2.2 b, and as $S = a\,B$, as mentioned before, we have:

$$\frac{t^3}{2.2\,a\,B^3} = 0.045, \text{ or } t = \sqrt[3]{0.099\,a} \times B \qquad (6)$$

All dimensions have now been reduced to some term containing B, so that, having decided upon the bore and stroke of the proposed engine, it is an easy matter to proportion the crank-shaft.

In Fig. 8 is shown a diagram for determining the breadth and thickness of the cheeks for crank-shafts for gas engines. To use the diagram, find the intersection of the bore- and stroke-lines at the right, and determine the corresponding value of a on the diagonal lines. Then follow the horizontal line from the intersection to the left-hand part of the diagram, to the same diagonal line a. Then follow the vertical line down to the bottom of the diagram where the breadth and thickness of cheek are given. The heavy dotted line in the diagram shows the procedure for a 12-inch bore, 18-inch stroke engine. The breadth and thickness found are 2⅞ inches and 6⅜ inches, respectively.

The ends of the cheeks are made concentric with the pin or shaft opposite, that is, the pin end of the cheek is turned when revolving about the shaft center and *vice versa*. The length of the crank-shaft depends upon whether the driving pulley is to be bolted to the arms of the flywheel or keyed to the shaft. The latter method is preferable, although slightly more expensive. The shaft, however, should only be long enough to engage the pulley hub, thus keeping the driving pulley close to the flywheel. A longer shaft gives an inexperienced operator an opportunity to set the pulley out far enough to seriously strain the unsupported end of the shaft.

CHAPTER IV

THE CRANK-CASE PROBLEM*

Volumes of catalogues and trade literature have been published descriptive of the operation of what is known as the enclosed crank-case two-cycle motor, but very little real information has been given relating to the principles of design and the inherent difficulties to be met with in this class of engines.

Everyone knows that in this class of engines the crank-case is used as a pump to force the charge into the working cylinder; in fact, the crank-case is one end of the cylinder which has been enlarged enough to permit the crank-shaft and connecting-rod inside and give them room to work. This makes a double-acting cylinder just like a steam engine, except that one end is an air or gas compressor with the moving parts of the engine inside of it, and the other is an explosion engine. The same piston serves for both uses and, of course, sweeps through the same volume or "displacement" at each end, this volume in cubic inches being the area of the piston in square inches multiplied by the stroke in inches.

The engine end of the cylinder has means for admitting the mixture to it, and releasing the burned gases from it, these means being two ports, or series of ports, in the cylinder walls which are covered by the piston, except for a short time near the end of the stroke. The exhaust port is wider and opens first, at a point ten to fifteen per cent before the piston reaches the dead center. This gives a chance for the hot gases to escape and reduces the pressure in the cylinder before the piston uncovers the inlet port at five to eight per cent from the end of the stroke. This is necessary to prevent the burning gases from shooting down into the crank-case and firing the new charge before it is time. Before the exhaust opens, the pressure in the cylinder may be as much as thirty or forty pounds per square inch, while the pressure in the crank-case is seldom over six pounds. One of the difficulties in two-cycle design is to get these ports right, so that the pressure in the cylinder will be lower than the crank-case pressure by the time the inlet port is uncovered. This inlet port to the engine end is the discharge port for the compressing or pump end; and for all three of the events thus far mentioned, *viz.*, admission and discharge in the engine end and discharge in the pump end, the piston is the controlling valve. The fourth event, admission to the pump end, is also controlled by the piston in the type known as three-port, or valveless engine. In the two-port engine the admission to the pump is by a check valve.

In the operation of these pumps there are four main sources of loss in capacity or volume of charge: 1. Suction loss due to taking in

* MACHINERY, April, 1910.

charge at less than full atmospheric pressure. 2. Discharge loss due to failure of charge in crank-case to be fully delivered to the working end during the brief opening of the inlet port. 3. Exhaust loss due to entering charge partially mixing with exhaust gas, and going out the open exhaust port. 4. Leakage from the crank-case at bearings, etc.

Clearance space in the crank-case is often mentioned as a cause of reduced charge and is so, but in an indirect way, through its influence on suction and discharge. These latter are primarily due to fluid friction, or the resistance of the charge to passing through the inlet valve and inlet port. If there were no friction, the clearance would have no effect, but as a matter of fact it has a great effect in increasing both the suction loss and discharge loss, as will be shown.

The above losses are not all in the same direction; for instance, if an engine has a large clearance resulting in increased loss in (1) and (2) it will at the same time have a lower pressure and less leakage from the

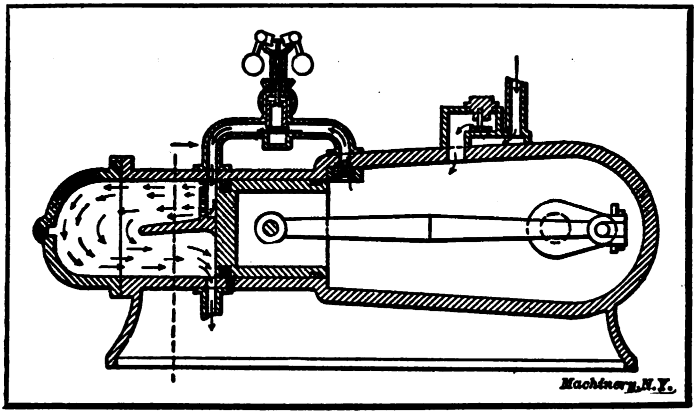


Fig. 9. Diagrammatic View of Two-cycle Engine of the Type having a Check Valve for the Admission to the Crank-case or Pump

crank-case (4), and likewise, because of the reduced volume actually delivered to the cylinder, less chance of blowing fuel out through the exhaust (3). Designers nowadays do not worry so much about (3) because the other results are such that it usually is not important, but he who attempts to make his pump *too efficient* must watch out for this loss. An interesting commentary on this point was made by Mr. L. H. Nash at a recent meeting of the American Society of Mechanical Engineers. Mr. Nash exhibited the accompanying illustration, Fig. 9, which is of one of the first two-cycle engines. Note the large space in the crank-case and the enormous baffle plate on the piston with its precautions for heading the charge straight for the cylinder head. In this case the designer evidently feared nothing so much as blowing out at the exhaust.

To return to the clearance, most designers work to keep this as low as possible. This is done by making the crank-case as close a fit as is safe, by using a short connecting-rod, by putting balance weights on the crank webs, and even by attaching false pieces in the pistons. The object of this is not merely to get a high crank-case pressure, but also to reduce the suction loss. If we designate the displacement or volume swept by the piston as 100, the waste space in the crank-case will vary in different designs from say 150 to 250 additional, say 200 as a fair average; then the volume of air in the crank-case when the piston is

TABLE I. SUCTION LOSS IN PER CENT OF DISPLACEMENT FOR VARIOUS CLEARANCES

Suction Pressure Below Atmospheric Pressure, Pounds per Square Inch	Clearance in Per Cent of Displacement			
	150	200	250	300
½	8.5	10.1	11.9	13.6
1	17.0	20.2	23.8	27.2
1½	25.5	30.3	35.7	40.8
2	34.0	40.4	47.6	54.4
2½	42.5	50.5	59.5	68.0

clear up will be 300 and when the piston is clear down, 200. The pressure at the end of the down stroke depends on the pressure at the beginning of the stroke and on the nature of the compression. Theoretically there are two possible ways of compressing, known as isothermal and adiabatic. Practically, it is impossible to compress exactly by either, the usual result being somewhere between them. For comparison we may use the isothermal for figuring pressures, as this favors allowance for leakage, etc. If then we had before compression a pressure of 14.7 pounds absolute, atmospheric pressure, the pressure after

TABLE II. DISCHARGE LOSS IN PER CENT OF DISPLACEMENT FOR VARIOUS CLEARANCES

Residual Pressure Above Atmospheric Pressure, Pounds per Square Inch	Clearance in Per Cent of Displacement			
	150	200	250	300
½	5.1	6.8	8.5	10.2
1	10.2	13.6	17.0	20.4
1½	15.3	20.4	25.5	30.6
2	20.4	27.2	34.0	40.8
2½	25.5	34.0	42.5	51.0

compressing from 300 volumes to 200 would be 22 pounds absolute or 7.3 pounds gage, but if the suction pressure is two pounds below atmospheric, the final pressure would be only 19 pounds absolute or 4.3 pounds gage. In other words, a loss of two pounds on the suction is a loss of 3 pounds in the maximum pressure, and with larger clearance would be still more. How much this affects the volume delivered to the cylinder cannot be answered absolutely, as it depends on the inlet port and the speed. When the piston uncovers the inlet port the charge in the crank-case at once begins to enter the cylinder and the pressure starts to drop. It does not drop entirely to atmospheric pressure, however, and probably seldom to less than 1 pound gage pressure on account of the very limited time that this port is open.

With a clearance of 200 and a pressure in crank-case reduced to 1 pound at the point of inlet port closure, it will require 13.6 per cent of the return stroke before the charge remaining in the crank-case will expand to atmospheric pressure, and this reduces the charge delivered to the cylinder by 13.6 volumes in 100.

TABLE III. VOLUMETRIC EFFICIENCY OF CRANK-CASE FOR ONE POUND SUCTION AND ONE POUND RESIDUAL PRESSURE FOR VARIOUS CLEARANCES

	Clearance in Per Cent of Efficiency			
	150	200	250	300
Volumetric Efficiency	72.8	66.2	59.2	52.4

Going back to the suction pressure, if it is 2 pounds below atmospheric when the inlet valve or third port closes, it will require 41 per cent of the forward stroke to compress this charge up to atmospheric pressure resulting in a loss due to suction of 41 volumes. So that our net charge under these conditions will be $100 - (13.6 + 41) = 45.4$ volumes or 45.4 per cent of the piston displacement. This low capacity is frequently obtained in practice and the best practice seldom exceeds 60 to 65 per cent of the charge.

TABLE IV. CRANK-CASE PRESSURE (MAXIMUM) IN POUNDS GAGE PRESSURE FOR VARIOUS CLEARANCES

Suction Pressure Below Atmospheric Pressure, Pounds per Square Inch	Clearance in Per Cent of Efficiency			
	150	200	250	300
½	9.0	6.6	5.2	4.2
1	8.1	5.9	4.5	3.6
1½	7.3	5.1	3.8	2.9
2	6.5	4.4	3.1	2.2
2½	5.7	3.6	2.4	1.6

In Tables I and II the losses in capacity from both suction and discharge are tabulated for different percentages of clearance space in the crank-case. Clearances and capacities are stated in percentages of piston displacement. The tables are carried beyond the limits of good practice in order to show clearly the effect of excessive clearance or frictional pressure loss in suction or discharge. The "suction pressure" is the pressure in crank-case below atmospheric when the piston is up ready to compress and the "residual pressure" is the pressure above atmospheric when the piston has just closed the port into the cylinder and a new charge is about to be taken into the crank-case.

In Table III the net capacity is given under assumed conditions such as exist in good practice. These values are obtained by subtracting from 100 per cent the sum of the suction and clearance loss in Tables I and II. It will readily be seen that for some of the conditions of Tables I and II the capacity would be zero or less. A study of these tables may lead one to wonder not why his engine *does not run* sometimes, but why it *does run at all.*

In estimating the suction pressure if the clearance is known or *vice versa*, the highest pressure obtained in the crank-case is a valuable aid. This can be found with a ball check valve connected to the crank-case and to a low reading gage. Table IV shows what the pressure will be under stated conditions. It may be remarked that the usual value in good engines is from 4 to 6 pounds. A leaky bearing or stuck inlet valve will at once cut this down. It will be easily seen from the tables why it is that motors choke up and refuse to run at speeds above what they are designed for, on account of the loss of charge at both ends of the stroke, and also what are the best results which may be expected of motors of this type.

CHAPTER V

WATER REQUIRED TO COOL A GAS ENGINE*

The water pump for a gas engine is generally designed to carry off one-half the heat produced by combustion. At times one-quarter would be sufficient, but one-half is the amount that should be figured on. If the heat per minute generated by an engine is represented by q, then for a thermal efficiency of 12½ per cent, $q = 339.2 \times$ I. H. P., and $q \times 0.5 =$ the heat to be carried off by the water.

The constant 339.2 is obtained from the formula $q = \frac{\text{I. H. P.}}{E} \times \frac{33{,}000}{778}$.

$E =$ the thermal efficiency, which for gasoline is taken as 12½ per cent. Hence, $q = \frac{\text{I. H. P.}}{0.125} \times \frac{33{,}000}{778} = 339.2 \times \text{I. H. P.}$ Other constants may be obtained by substituting the thermal efficiency expected or known. The maximum efficiency of gasoline is about 19 per cent, and a number of very good engines have shown about 15 per cent efficiency, but for the general run, the safe figure to use is 12½ per cent.

Let

$t - t_1 =$ allowable rise in temperature,

$t =$ maximum temperature of water in degrees F.; (about 180 degrees should be the maximum temperature allowed),

$t_1 =$ normal temperature of water in degrees F.,

$W =$ the number of pounds of water required per minute, then

$$W = \frac{169.6 \times \text{I.H.P.}}{t - t_1}$$

$t - t_1 =$ the number of B. T. U. absorbed per pound of water.

As the pump is generally attached to the engine shaft, it will have the same number of revolutions as the engine. Let p equal pounds of water required per revolution, then

$$p = \frac{W}{\text{R. P. M.}}$$

As one gallon of water weighs 8.33 pounds,

$\frac{p}{8.33} =$ number of gallons required per revolution.

Let us take an example and assume that we wish to design a pump for a 20 I. H. P. gas engine which turns at 300 R. P. M.

$$q = 339.2 \times 20 = 6784 \text{ B. T. U.}$$

* MACHINERY, December, 1909.

$q \times 0.5 = 6784 \times 0.5 = 3392$ B. T. U., which is the amount of heat the water is to carry off.

$t = 180,\ t_1 = 60,\ 180 - 60 = 120.$

$$W = \frac{169.6 \times 20}{120} = 28.267 \text{ pounds.}$$

$$p = \frac{28.267}{300} = 0.0942 \text{ pound per revolution.}$$

$$\frac{0.0942}{8.33} = 0.0113 \text{ gallon per revolution.}$$

By referring to the accompanying table we see that a pump 1½ inch bore by 1½ inch stroke will answer. The number of gallons pumped

TABLE V. GIVING GALLONS PER STROKE FOR VARIOUS SIZES OF SINGLE-ACTING PUMPS

Diam.	Area	Stroke						
		1	1½	2	3	4	5	6
½	0.196	0.0008	0.001	0.002	0.0025	0.003	0.004	0.005
⅝	0.307	0.001	0.002	0.003	0.004	0.005	0.007	0.008
¾	0.442	0.002	0.003	0.004	0.006	0.008	0.010	0.012
⅞	0.601	0.0025	0.004	0.005	0.008	0.010	0.013	0.016
1	0.785	0.003	0.005	0.007	0.011	0.014	0.017	0.020
1¼	1.227	0.005	0.007	0.010	0.016	0.021	0.026	0.032
1½	1.767	0.007	0.011	0.015	0.022	0.030	0.038	0.045
1¾	2.405	0.010	0.015	0.021	0.031	0.041	0.051	0.062
2	3.142	0.014	0.021	0.027	0.041	0.054	0.068	0.082

per minute is equal to the number of R. P. M. of a single-acting pump multiplied by the number of gallons per revolution as given in the table.

The formula given is not theoretically correct, but answers all practical requirements.

If the thermal efficiency of a gas engine were 100 per cent, that is, if all the heat were converted into work, there would be no rejection of heat into the cylinder walls, and consequently no need for cooling water. Again, if the thermal efficiency were 50 per cent, one-half the heat would be rejected into the walls and exhaust while the other half was converted into work. The formula, therefore, is not theoretically correct, as it does not take into consideration the percentage of heat converted into energy and which thus disappears.

Using the same notation, the formula should properly be:

$$q = \left(\frac{\text{I. H. P.}}{E} \times \frac{33{,}000}{778}\right) - \left(\frac{33{,}000 \times \text{I. H. P.}}{778}\right)$$

$$= \frac{33{,}000 \times \text{I.H.P.} - 33{,}000 \times \text{I.H.P.} \times E}{778 \times E}$$

$$= \frac{660{,}000 - 660{,}000\,E}{778\,E}$$

If $E = 12\frac{1}{2}$ per cent, then:

$$q = \frac{660{,}000 - 82{,}500}{97.25} = 5938 \text{ B. T. U.}$$

The thermal efficiency of gas engines being rarely more than 20 per cent, the error in the given method, however, is not important and for practical purposes it is to be preferred because of its simplicity.

CHAPTER VI

COMMERCIAL RATINGS FOR INTERNAL COMBUSTION ENGINES*

Various empirical ratings have been proposed for automobile and other engines, based on the cylinder dimensions and the speed. These are founded either on tests of a large number of types, or on an analysis, such as follows, of the factors limiting the output of an engine running under stated conditions.

The Indicator Diagram

The power exerted by an engine during a working stroke is shown by such a diagram as that given by the indicator. The factors which

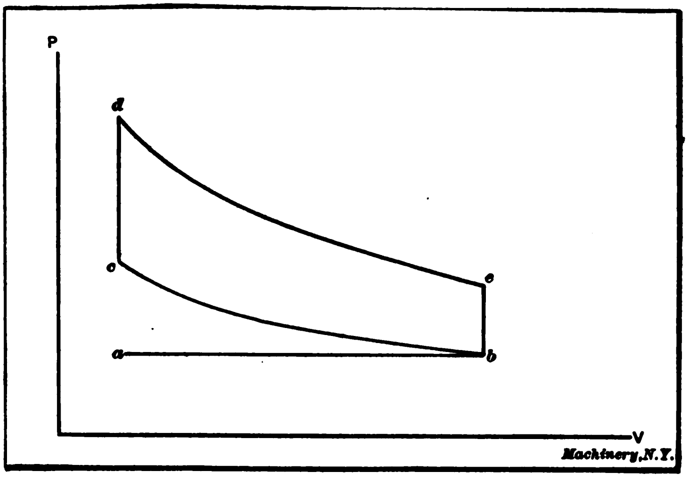


Fig. 10. Ideal Gas Engine Diagram

would influence this diagram in an internal-combustion engine may be listed as follows:

***a.* The diameter of the cylinder and the stroke of the piston.**

***b.* The type of the engine, whether 2-cycle or 4-cycle.**

***c.* The mechanical design, including the questions of port areas, jacketing, and gas velocities.**

***d.* The fuel used, the nature of the fuel limiting the temperature allowable during compression.**

***e.* The clearance, which is determined by reference to factors *b*, *c* and *d*.**

The ideal diagram of a gas engine is shown in Fig. 10. Gas enters

* MACHINERY, April, 1910.

the cylinder along *ab*, without friction, at constant pressure. It is compressed along *bc*, without gaining or losing heat, until the piston has reached the end of its stroke at *c*. Ignition then occurs. The spread of flame is instantaneous, so that the line *cd*, representing a rise of temperature and pressure, is straight and vertical. The piston immediately moves forward, the gas expanding behind it along *de*, again without gaining or losing heat. At *e*, the piston has reached the outer end of its stroke, the exhaust valve opens, and the gas passes out without friction. The pressure falls instantaneously to that at *b*, the remaining low-pressure gases leaving the cylinder (also without friction) during the return stroke of the piston along *ba*.

The actual indicator diagram of the 4-cycle type of engine is shown in Fig. 11. Gas enters as before during the out-stroke of the piston along *ab*, but port friction wire draws the charge so that the pressure

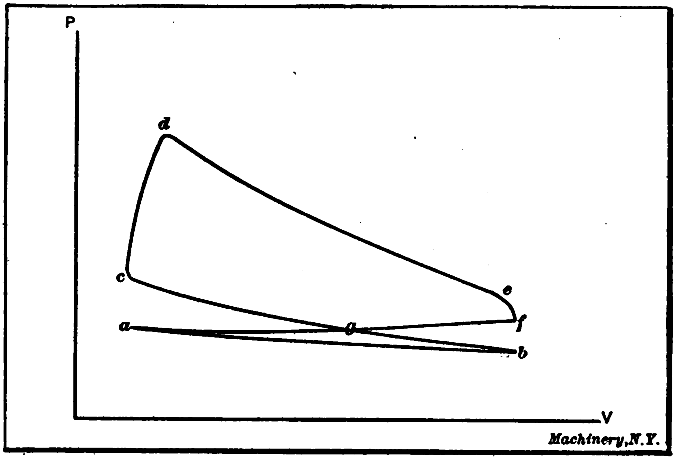


Fig. 11. Actual Diagram, Four-cycle Type Engine

gradually decreases. The pressure at *b* in a 4-cycle engine may be about 12 pounds absolute (2.7 pounds less than atmospheric pressure) and in a 2-cycle engine about 20 pounds absolute. Compression occurs along *bc*, but the curve does not rise as rapidly as the corresponding curve in Fig. 10. This is to be explained on the ground that heat is being given up to the cylinder walls, so that the temperature and pressure of the gas at any point are less than they would otherwise be. It has been found that the curve *bc* follows approximately the law

$$PV^n = \text{constant},$$

in which P and V are corresponding pressures and volumes and the value of n may be taken at 1.3. Then

$$P_b V_b^{1.3} = P_c V_c^{1.3}, \text{ for example.}$$

An important principle in gas engine design may here be suggested. The temperature due to the compression reached at *c* must not exceed that at which the gases will ignite of themselves. Suppose this temperature, for the fuel in question to be 600 degrees F. Then by a fundamental formula for gases,

$$\frac{P_c V_c}{T_c} = \frac{P_b V_b}{T_b} \text{ and } \frac{V_c}{V_b} = \frac{P_b T_c}{P_c T_b} \qquad (1)$$

in which the symbol T refers to the absolute temperature, equal to the Fahrenheit temperature plus 460.

We have, moreover, as already stated

$$P_c V_c^{1.3} = P_b V_b^{1.3} \text{ and } P_c = P_b \left(\frac{V_b}{V_c}\right)^{1.3} \qquad (2)$$

Combining Equations (1) and (2), we find

$$\frac{V_c}{V_b} = \left(\frac{T_c}{T_b}\right)^{-3.33}$$

The value of T_b depends upon the temperature of the gases when admitted to the cylinder, and the warming which they undergo by transfer of heat from the walls during the suction stroke. Suppose it to be 200 degrees F. We then find $T_c = 600 + 460 = 1060$, $T_b = 200 + 460 = 660$, and $\frac{V_c}{V_b} = \left(\frac{1,060}{660}\right)^{-3.33} = 0.206$, whence $\frac{V_c}{V_b - V_c} = 0.26$ the clearance expressed as a fraction of the stroke.

Ignition occurs at *c;* but as the spread of the flame is not instantaneous, the ignition path *cd* is swerved to the right by the movement of the piston. Further, the resulting rise of temperature attained at *d* is only a fraction of that which might be expected from a computation based on the composition of the gas. This fact is due to the increase of volume during ignition, the transfer of heat to the cylinder walls, and the slow burning, by reason of which some of the combustion occurs along the expansion path *de*. The temperature attained at *d* seldom exceeds 3000 degrees F. = 3460 degrees absolute, and the corresponding pressure, assuming no important change of volume to occur between *c* and *d*, is given by the formula,

$$\frac{P_d}{P_c} = \frac{T_d}{T_c}.$$

Along the expansion path *de* some heat is given up by the gas to the cylinder walls. Any "after-burning" due to incomplete ignition along *cd* will also affect the shape of *de*. In most cases, the curve in question may be approximately represented by the same sort of formula as that given for the compression curve.

$$PV^n = \text{a constant},$$

in which the value of n, for our present purpose, may be taken at 1.33. When the value of n is the same for the expansion curve as for the compression curve, we have the simple relation:

$$\frac{P_d}{P_c} = \frac{P_e}{P_b}.$$

The exhaust valve opens at *e*, and the gas is forced out of the cylinder against an appreciable resistance along *efa*. The indicated work is *cdefg* — *agb*.

The Designer's Diagram

We may now simplify the indicator diagram as in Fig. 12, eliminating the negative loop *agb* and the rounded corners of the diagram in Fig. 11, and treating the ignition line *cd* as straight and vertical. The indicated work of such a diagram would be represented by the enclosed area, or, we may say, by

$$mden - mcbn.$$

For an area like *mden*, subtended by a curve like *de*, we have the formula

$$\frac{P_d V_d - P_e V_e}{0.33}.$$

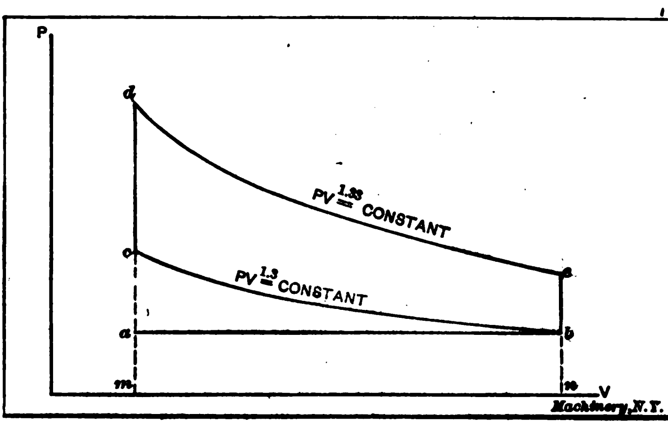


Fig. 12. Designer's Diagram

the 0.33 being the quantity $n - 1$, where n is the exponent of the equation representing the curve. Similarly,

$$mcbn = \frac{P_c V_c - P_b V_b}{0.3}, \text{ and}$$

$$cdeb = mden - mcbn = \frac{P_d V_d - P_e V_e}{0.33} - \frac{P_c V_c - P_b V_b}{0.3}.$$

Computation of Pressures and Work

For a 4-cycle engine, $P_b = 12 \times 144 = 1728$ pounds per square foot. To find P_c, we write

$$\frac{P_c V_c}{T_c} = \frac{P_b V_b}{T_b}, \text{ or } P_c = \frac{P_b V_b T_c}{V_c T_b},$$

and substituting values already given, we obtain:

$$P_c = \frac{1728 \times 1060}{0.206 \times 660} = 13{,}400 \text{ pounds per square foot, about.}$$

For P_d, we have, approximately,

$$\frac{P_d}{P_c} = \frac{T_d}{T_c};\ P_d = P_c \frac{T_d}{T_c} = 13{,}400 \frac{3000 + 460}{1060} = 44{,}000 \text{ pounds per sq. foot.}$$

For P_e, we write

$$P_e V_e^{1.33} = P_d V_d^{1.33};\ P_e = P_d \left(\frac{V_d}{V_e}\right)^{1.33} = 44{,}000 \times 0.206^{1.33} = 5880 \text{ pounds per}$$

square foot, about.

Let $V_b - V_c = V_e - V_d = D$, the displacement of the piston per stroke.

But $V_c = 0.206\ V_b$, whence $0.794\ V_b = D$; $V_b = V_e = 1.26\ D$; and $V_c = V_d = 0.26\ D$. Then,

$$cdeb = D\left(\frac{(44{,}000 \times 0.26) - (5380 \times 1.26)}{0.33} - \frac{(13{,}400 \times 0.26) - (1728 \times 1.26)}{0.3}\right) =$$

$$D\left(\frac{11{,}440 - 6780}{0.33} - \frac{3490 - 2180}{0.3}\right) = 9750\ D.$$

Under the assumed conditions, *viz.*, a pre-compression pressure of 12 pounds per square inch and a temperature of 200 degrees F., a compression curve $PV^{1.3} =$ a constant, a temperature after compression of 600 degrees F. and after ignition of 3000 degrees F., with an expansion curve $PV^{1.33} =$ a constant, with a piston displacement of D cubic feet, the work per active stroke, measured in foot-pounds, is 9750 D. This must be somewhat reduced to allow for the rounding of corners of the actual diagram, the negative loop *agb*, Fig. 2, etc., say to 9000 D, which then represents the indicated work to be expected from the engine.

Engine Capacity

Let S be the piston speed in feet per minute, s the piston speed in inches, A the area of the piston in square feet, a its area in square inches, and N the number of revolutions per minute. The displacement of the piston in cubic feet per stroke is then $AS \div 2N$. The number of active strokes per minute depends upon the form of engine. Thus, for 4-cycle engines, we have the following ratio of active strokes to total strokes:

Single-acting,	1-cylinder,	1 to 4;
	2-cylinder,	2 to 4;
	4-cylinder,	4 to 4;
	6-cylinder,	6 to 4;
Double-acting,	2-cylinder,	4 to 4;
	4-cylinder,	8 to 4.

Since there are $2N$ total strokes per minute, the number of active strokes in a single-acting engine, if 4-cylinder, is $2N$; and if 6-cylinder, $\frac{6}{4} \times 2N = 3N$. The indicated horsepowers of the two forms of engine are then $\frac{9000 \times 2ND}{33,000}$ and $\frac{9000 \times 3ND}{33,000}$: and since $D = AS \div 2N$, these horsepowers may also be written:

$$\frac{9000 \times 2ANS}{33,000 \times 2N} = 0.273AS, \text{ and } \frac{9000 \times 3ANS}{33,000 \times 2N} = 0.409AS.$$

Empirical Ratings

Since $S = \frac{s}{12}$ and $A = \frac{a}{144}$, we have for the 4-cylinder engine:

$$\text{I.H.P.} = 0.273 \times \frac{s}{12} \times \frac{a}{144} = 0.000158sa.$$

If d be the cylinder diameter in inches, $a = 0.7854d^2$, and

$$\text{I.H.P.} = 0.000158 \times 0.7854sd^2 = 0.000124sd^2.$$

If the piston speed be taken in feet as in standard nomenclature, then,

$$\text{I.H.P.} = 0.00149\,Sd^2 = \frac{Sd^2}{670}.$$

From this we may formulate the following rule: The indicated horsepower of a single-acting four-cylinder four-cycle engine is equal to the piston speed in feet per minute, multiplied by the square of the cylinder diameter in inches, and divided by 670. A 4¾-inch engine at 800 feet piston speed would then be rated at 28 H. P.

Ratings are sometimes based on the cylinder dimensions only, the number of revolutions per minute being regarded as standard. If L be the length of the stroke in inches,

$S = 2LN \div 12 = \frac{LN}{6}$, and, for a four-cylinder engine,

$$\text{I.H.P.} = \frac{LNd^2}{4020} = \frac{LNa}{4020 \times 0.7854} = \frac{LNa}{3155}$$

$$\text{I.H.P. per cylinder} = \frac{LNa}{3155 \times 4} = \frac{LNa}{12,620}$$

At 1000 R.P.M. the indicated horsepower per cylinder is then equal to the product of the length of stroke in inches, piston area in square inches, and number of revolutions per minute, divided by 12.62.

Another method of rating at 1000 R.P.M. may be derived as follows:

$$\text{I.H.P.} = \frac{LNd^2}{4020} = \frac{Ld^2}{4.02}, \text{ or}$$

$$\text{I.H.P.} = \frac{Ld^2}{16.08} \text{ per cylinder.}$$

If the diameter of the cylinder equals the stroke, we have:

$$\text{I.H.P.} = \frac{d^3}{16.08} \text{ per cylinder, or approximately } \frac{d^3}{16}.$$

For m cylinders this becomes:

$$\text{I.H.P.} = \frac{md^3}{16}.$$

Effective Horsepower

The effective horsepower may be assumed to be 85 per cent of the indicated horsepower. By inserting this value in the formulas for the indicated horsepower per cylinder we find the actual horsepower:

$$\text{H.P.} = \frac{Sd^2}{3150} = \frac{LNa}{14{,}900} = \frac{LNd^2}{19{,}000} = \frac{Nd^3}{19{,}000}.$$

The denominators are given in round numbers to facilitate calculations. They are, however, very close to the exact values.

Rules Based on Formulas Given

Under the stated conditions of pressure, temperature, etc., which are approximately correct for four-cycle gasoline automobile engines, the effective horsepower per cylinder may be estimated by any of the following rules:

1. The product of the piston speed in feet per minute times the square of the diameter in inches, divided by 3150.

2. The product of the stroke in inches, the area of the piston in square inches, and the number of revolutions per minute, divided by 14,900.

3. The product of the stroke in inches, the square of the diameter in inches, and the number of revolutions per minute, divided by 19,000.

4. The product of the cube of the diameter in inches times the number of revolutions per minute, divided by 19,000. This last rule holds good only when the stroke equals the diameter.

Rule 1 appears to be the simplest and most direct. The rule sometimes quoted, giving the horsepower of the whole engine as equal to the diameter squared multiplied by the number of cylinders and divided by 2.5, can be justified only on assumptions of a constant piston speed and fixed length of stroke for all engines. Rules in any one of the four forms given are, however, satisfactory and scientific as long as their application is limited to a specific fuel and specific type of engine. The constants will vary if the rules are extended to cover such cases as:

a. A double-acting four-cycle engine using producer gas.

b. The same engine using natural gas.

c. The same engine using blast furnace gas.

d. A single-acting four-cycle engine using kerosene.

e. A single-acting two-cycle engine using kerosene.

The following data will permit of establishing rules for the kinds of engines suggested, as well as for some other forms.

The value of P_b, Fig. 12, in two-cycle engines equals 18 to 21 pounds; in four-cycle engines, 12 to 14 pounds, absolute. The value of T_b will range from 140 to 300 degrees F.

The values of n (the exponent for the expansion curve) vary from 1.2 to 1.38, being lower in larger engines. The values of the exponent for the compression curve are usually taken slightly smaller, say from 0.03 to 0.05 less than n for the expansion curve. T_c should not exceed 450 degrees F. if the gas contains more than 10 per cent, by volume, of hydrogen. In no case should it exceed 600 degrees F.

The value of P_c will usually be kept within the following ranges:

In automobile engines, 45 to 100 pounds.

In ordinary gasoline engines, 60 to 85 pounds.

In kerosene engines, 30 to 85 pounds.

In natural gas engines, 75 to 130 pounds.

In producer gas engines, 100 to 160 pounds.

In blast furnace gas engines, 120 to 190 pounds.

The value of T_d will seldom or never exceed 3000 degrees F., and may, without serious error, be taken at this value for all the fuels mentioned. The percentages of deduction to be made for the rounding of the corners of the diagram, etc., and losses between cylinder and crankshaft will aggregate from 15 to 25 per cent in four-cycle, and from 30 to 40 per cent in two-cycle engines, being greatest in small engines.

Approximate Horsepower Formulas for Gasoline Engines*

In the September 3, 1909, issue of *Industritidningen Norden*, Mr. E. Hubendick reviews the various approximate formulas which have been adopted or proposed by a number of societies and individuals for the horsepower of gasoline engines. In these formulas

D = diameter of cylinder,

N = number of cylinders,

S = length of stroke,

n = number of revolutions per minute.

The French Automobile Club's formula is:

H. P. = $0.07\ D^2N$, when the diameter is given in centimeters.

H. P. = $0.45\ D^2N$, when the diameter is given in inches.

In this formula the mean pressure has been assumed to be 5.3 kilograms per square centimeter (75 pounds per square inch), and the piston speed 5 meters (16 feet 5 inches) per second.

The Royal Automobile Club's (British) formula is:

H. P. = $0.0625\ D^2N$, when the diameter is in centimeters.

H. P. = $0.405\ D^2N$, when the diameter is given in inches.

* MACHINERY, November, 1909.

Mr. Arnon's formula is:

$\text{H. P.} = 0.0061\ D^2N$, when the diameter is in centimeters.

$\text{H. P.} = 0.1\ D^2N$, when the diameter is given in inches.

Mr. Faroux's formula is:

$\text{H. P.} = 0.0074\ D^{2.4}\ S^{0.6}$, when the diameter is in centimeters.

$\text{H. P.} = 0.121\ D^{2.4}\ S^{0.6}$, when the diameter is given in inches.

Another French formula is as follows:

$\text{H. P.} = 0.02562\ D^{2.4}\ N$, when the diameter is in centimeters.

$\text{H. P.} = 0.24\ D^{2.4}\ N$, when the diameter is given in inches.

Mr. T. Thornycroft's formula is:

$$\text{H. P.} = \frac{D^2\ S^{0.75}\ N}{35{,}000},$$ when the diameter is given in centimeters.

$$\text{H. P.} = \frac{D^2\ S^{0.75}\ N}{2{,}700},$$ when the diameter is given in inches.

Prof. H. L. Callender's formula is:

$\text{H. P.} = 0.0875D\ (D - 2.5)\ N$, when the diameter is given in centimeters,

$\text{H. P.} = 0.565D\ (D - 1)\ N$, when the diameter is given in inches.

In this latter formula the mean pressure is assumed to vary in the same proportion as $\left(1 - \frac{2.5}{D}\right)$ if the diameter is given in centimeters, and $\left(1 - \frac{1}{D}\right)$ if the diameter is given in inches.

The Royal Automobile Club's (Swedish) formula is:

$$\text{H. P.} = \frac{D^2SnN}{250{,}000},$$ when the diameter is given in centimeters.

$$\text{H. P.} = \frac{D^2SnN}{15{,}240},$$ when the diameter is given in inches.

In this connection the formula of the Association of Licensed Automobile Manufacturers should be included:

$$\text{H. P.} = \frac{D^2N}{2.5},$$ when the diameter is given in inches.

$\text{H. P.} = 0.062\ D^2N$, when the diameter is given in centimeters.

Some of the formulas proposed, with fractional exponents, are more cumbersome to use than would be the exact horsepower formula, and are of very doubtful value for their purpose. It is difficult to understand why one should be given an approximate formula at all, unless the form of that formula be such that it greatly facilitates computations, as compared with the exact formula. In this respect the A. L. A. M. formula is one of the best of those given.

UNIV. OF CALIFORNIA

OUTLINE OF A COURSE IN SHOP AND DRAFTING-ROOM MATHEMATICS, MECHANICS, MACHINE DESIGN AND SHOP PRACTICE

Any intelligent man engaged in mechanical work can acquire a well-rounded mechanical education by using as a guide in his studies the outline of the course in mechanical subjects given below. The course is laid out so as to make it possible for a man of little or no education to go ahead, beginning wherever he finds that his needs begin. The course is made up of units so that it may be followed either from beginning to end; or the reader may choose any specific subject which may be of especial importance to him.

Preliminary Course in Arithmetic

JIG SHEETS 1A TO 5A:—Whole Numbers: Addition, Subtraction, Multiplication, Division, and Factoring.

JIG SHEETS 6A TO 15A:—Common Fractions and Decimal Fractions.

Shop Calculations

Reference Series No. 18. SHOP ARITHMETIC FOR THE MACHINIST.

Reference Series No. 52. ADVANCED SHOP ARITHMETIC FOR THE MACHINIST.

Reference Series No. 53. USE OF LOGARITHMIC TABLES.

Reference Series Nos. 54 and 55. SOLUTION OF TRIANGLES.

Data Sheet Series No. 16. MATHEMATICAL TABLES. A book for general reference.

Drafting-room Practice

Reference Series No. 2. DRAFTING-ROOM PRACTICE.

Reference Series No. 8. WORKING DRAWINGS AND DRAFTING-ROOM KINKS.

Reference Series No. 33. SYSTEMS AND PRACTICE OF THE DRAFTING-ROOM.

General Shop Practice

Reference Series No. 10. EXAMPLES OF MACHINE SHOP PRACTICE.

Reference Series No. 7. LATHE AND PLANER TOOLS.

Reference Series No. 25. DEEP HOLE DRILLING.

Reference Series No. 38. GRINDING AND GRINDING MACHINES.

Reference Series No. 48. FILES AND FILING.

Reference Series No. 32. SCREW THREAD CUTTING.

Data Sheet Series No. 1. SCREW THREADS. Tables relating to all the standard systems.

Data Sheet Series No. 2. SCREWS, BOLTS AND NUTS. Tables of standards.

Data Sheet Series Nos. 10 and 11. MACHINE TOOL OPERATION. Tables relating to the operation of lathes, screw machines, milling machines, etc.

Reference Series Nos. 50 and 51. PRINCIPLES AND PRACTICE OF ASSEMBLING MACHINE TOOLS.

Reference Series No. 57. METAL SPINNING.

Jigs and Fixtures

Reference Series Nos. 41, 42 and 43. JIGS AND FIXTURES.

Reference Series No. 3. DRILL JIGS.

Reference Series No. 4. MILLING FIXTURES.

Punch and Die Work

Reference Series No. 6. PUNCH AND DIE WORK.

Reference Series No. 13. BLANKING DIES.

Reference Series No. 26. MODERN PUNCH AND DIE CONSTRUCTION.

Tool Making

Reference Series No 64. GAGE MAKING AND LAPPING.

Reference Series No. 21. MEASURING TOOLS.

Reference Series No. 31. SCREW THREAD TOOLS AND GAGES.

Data Sheet Series No. 3. TAPS AND THREADING DIES.

Data Sheet Series No. 4. REAMERS, SOCKETS, DRILLS, AND MILLING CUTTERS.

Hardening and Tempering

Reference Series No. 46. HARDENING AND TEMPERING.

Reference Series No. 63. HEAT TREATMENT OF STEEL.

Blacksmith Shop Practice and Drop Forging

Reference Series No. 44. MACHINE BLACKSMITHING.

Reference Series No. 61. BLACKSMITH SHOP PRACTICE.

Reference Series No. 45. DROP FORGING.

Automobile Construction

Reference Series No. 59. MACHINES, TOOLS AND METHODS OF AUTOMOBILE MANUFACTURE.

Reference Series No. 60. CONSTRUCTION AND MANUFACTURE OF AUTOMOBILES.

Theoretical Mechanics

Reference Series No. 5. FIRST PRINCIPLES OF THEORETICAL MECHANICS.

Reference Series No. 19. USE OF FORMULAS IN MECHANICS.

Gearing

Reference Series No. 15. SPUR GEARING.

Reference Series No. 37. BEVEL GEARING.

Reference Series No. 1. WORM GEARING.

Reference Series No. 20. SPIRAL GEARING.

Data Sheet Series No. 5. SPUR GEARING. General reference book containing tables and formulas.

Data Sheet Series No. 6. BEVEL, SPIRAL AND WORM GEARING. General reference book containing tables and formulas.

General Machine Design

Reference Series No. 9. DESIGNING AND CUTTING CAMS.

Reference Series No. 11. BEARINGS.

Reference Series No. 56. BALL BEARINGS.

Reference Series No. 58. HELICAL AND ELLIPTIC SPRINGS.

Reference Series No. 17. STRENGTH OF CYLINDERS.

Reference Series No. 22. CALCULATIONS OF ELEMENTS OF MACHINE DESIGN.

Reference Series No. 24. EXAMPLES OF CALCULATING DESIGNS.

Reference Series No. 40. FLYWHEELS.

Data Sheet Series No. 7. SHAFTING, KEYS AND KEYWAYS.

Data Sheet Series No. 8. BEARINGS, COUPLINGS, CLUTCHES, CRANE CHAIN AND HOOKS.

Data Sheet Series No. 9. SPRINGS, SLIDES AND MACHINE DETAILS.

Data Sheet Series No. 19. BELT, ROPE AND CHAIN DRIVES.

Machine Tool Design

Reference Series No. 14. DETAILS OF MACHINE TOOL DESIGN.

Reference Series No. 16. MACHINE TOOL DRIVES.

Crane Design

Reference Series No. 23. THEORY OF CRANE DESIGN.

Reference Series No. 47. DESIGN OF ELECTRIC OVERHEAD CRANES.

Reference Series No. 49. GIRDERS FOR ELECTRIC OVERHEAD CRANES.

Steam and Gas Engine Design

Reference Series Nos. 67 to 72, inclusive. STEAM BOILERS, ENGINES, TURBINES AND ACCESSORIES.

Data Sheet Series No. 15. HEAT, STEAM, STEAM AND GAS ENGINES.

Data Sheet Series No. 13. BOILERS AND CHIMNEYS.

Reference Series No. 65. FORMULAS AND CONSTANTS FOR GAS ENGINE DESIGN.

Special Course in Locomotive Design

Reference Series No. 27. BOILERS, CYLINDERS, THROTTLE VALVE, PISTON AND PISTON ROD.

Reference Series No. 28. THEORY AND DESIGN OF STEPHENSON AND WALSCHAERTS VALVE MOTION.

Reference Series No. 29. SMOKEBOX, FRAMES AND DRIVING MACHINERY.

Reference Series No. 30. SPRINGS, TRUCKS, CAB AND TENDER.

Data Sheet Series No. 14. LOCOMOTIVE AND RAILWAY DATA.

Dynamos and Motors

Reference Series No. 34. CARE AND REPAIR OF DYNAMOS AND MOTORS.

Data Sheet Series No. 20. WIRING DIAGRAMS, HEATING AND VENTILATION, AND MISCELLANEOUS TABLES.

Reference Series Nos. 73 to 78, inclusive. PRINCIPLES AND APPLICATIONS OF ELECTRICITY.

Heating and Ventilation

Reference Series No. 39. FANS, VENTILATION AND HEATING.

Reference Series No. 66. HEATING AND VENTILATING SHOPS AND OFFICES.

Data Sheet Series No. 20. WIRING DIAGRAMS, HEATING AND VENTILATION, AND MISCELLANEOUS TABLES.

Iron and Steel

Reference Series No. 36. IRON AND STEEL.

Reference Series No. 62. TESTING THE HARDNESS AND DURABILITY OF METALS.

General Reference Books

Reference Series No. 35. TABLES AND FORMULAS FOR SHOP AND DRAFTING-ROOM.

Data Sheet Series No. 12. PIPE AND PIPE FITTINGS.

Data Sheet Series No. 17. MECHANICS AND STRENGTH OF MATERIALS.

Data Sheet Series No. 18. BEAM FORMULAS AND STRUCTURAL DESIGN.

Data Sheet Series No. 20. WIRING DIAGRAMS, HEATING AND VENTILATION AND MISCELLANEOUS TABLES.