THE STEAM-ENGINE

AND

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BY

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PREFACE TO THE THIRD EDITION

IN this edition the book has been thoroughly revised and to some extent re-written.

The chapter on Steam Turbines is new, and so is the greater part of the chapter on Gas and Oil Engines.

In dealing with the properties of steam a departure has been made from the earlier treatment, by accepting the characteristic equation of Callendar along with the steam tables derived from it by Mollier. The older data, based chiefly on experiments by Regnault, which were for a long time followed by engineers, involved serious inconsistencies and errors. Callendar's method of treatment escapes inconsistencies, and gives figures which are in harmony with the most trustworthy and most recent results of experiment.

The theoretical work of Callendar on this subject, notwithstanding its importance, is but little known to engineers. In Germany it has been applied by Mollier to the calculation of steam tables which, with his kind permission, are here reproduced with additions making them more suitable for use by English engineers.

Mollier has also enriched technical thermodynamics by the invention of novel graphic methods of representing the properties of steam and of solving steam-engine problems. He has allowed the author to reproduce his new steam charts, as well as the tables, and thereby to place in the hands of English students material of great interest and practical value.

The Fahrenheit scale of temperature has been definitely abandoned: the Centigrade scale is used throughout. To adhere to Fahrenheit degrees and the quantities dependent on them is to maintain a wholly unnecessary and exceedingly inconvenient barrier not only between applied science and the science of the physical laboratory but also between the engineering of England and that of other countries.

The steam tables, as printed here, give pressures and volumes in both metric and British units. Students should familiarize themselves with the metric units; in many cases the calculations required in practice are most easily made by employing them throughout.

The author desires to express his cordial thanks to Professor Callendar and Dr Mollier for assistance given in this part of the work. He has also to acknowledge help, in respect of other features of this edition, from Mr Dugald Clerk, Professor Dalby, Professor B. Hopkinson, Dr E. Hopkinson, Mr Charles Parsons, Mr J. B. Peace, Engineer Commander Roome, R.N., and other friends, as well as various engineering firms who have kindly supplied illustrations and particulars of their manufactures.

The book, with these alterations, retains the same general character as before. It aims primarily at a reasoned rather than a descriptive treatment, and it deals not only with the thermo-dynamics of heat engines but with their principal mechanical aspects, as actual machines.

J. A. EWING.

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CHAPTER I.

1. Heat-Engines in general. In the scientific treatment of the steam-engine we have in the first place, and mainly, to regard it as a heat-engine—that is, a machine in which heat is employed to do mechanical work. Other aspects of the steam-engine will present themselves when we come to examine the action of the mechanism in detail, but the foremost place must be given to thermodynamic considerations. From the thermo-dynamic point of view the function of a heat-engine is to get as much work as possible from a given supply of heat, or (to go a step further back) from the combustion of a given quantity of fuel. Hence a large part of our subject is the discussion of what is called the *efficiency* of the engine, which is the ratio of the work done to the heat supplied. We have to consider on what conditions efficiency depends, how its value is limited in theory and how nearly the limiting value may be attained in practice. We have to describe means of testing the efficiency of engines, and the results which such tests have given in actual cases. Much of what has to be said in regard to efficiency is applicable to all heat-engines, whatever be the character of the substance which is made use of as the means of doing work within the engine. In all practical heat-engines work is done through the expansion by heat of a fluid which exerts pressure and overcomes resistance as it expands. Thus in steam-engines the working substance is water and water-vapour, and work is done by the pressure which the substance exerts while its volume is undergoing change. In air-engines the working substance is

atmospheric air; in gas-engines and oil-engines it is a mixture of air with combustible gas or vaporised oil and with the products of combustion. These last are important instances of what are sometimes called internal combustion engines, in which the heat is developed by combustion occurring within the working substance itself instead of reaching the substance from an external source. We may have heat-engines in which the working substance is not a fluid, and examples might even be named in which a substance that is alternately heated and cooled is made to do work not in consequence of changes in its volume or in its form, but in consequence of some other effect of heat such, for instance, as the loss and gain of magnetic quality. A complete list of typical heat-engines would include a mention of guns, in which the heat that is generated by the combustion of an explosive does work in giving energy of motion to a projectile. We have, however, to do only with those types of heat-engine whose object is to change the potential energy of fuel into a manageable mechanical form, so that they may serve as prime movers to other mechanism. Of such engines the steam-engine is by far the most important.

As a preliminary to the study of the modern engine it will be useful to review, if only very briefly, some of the stages through which it has passed in its development. In any such historical sketch the largest share of attention necessarily falls to the work of Watt, whose inventions were as remarkable for their scientific interest as for their industrial importance. But it should be borne in mind that a process of evolution had been going on before the time of Watt which prepared the steam-engine for the immense improvements it received at his hands. The labours of Watt stand in a natural sequence to those of Newcomen, and Newcomen's to those of Papin and Savery. Savery's engine, again, was the reduction to practical form of a contrivance which had long before been known as a scientific toy.

2. Hero of Alexandria. The earliest notices of heat-engines are found in the *Pneumatics* of Hero of Alexandria, which dates from the second century before Christ. One of the contrivances mentioned there is the æolipile, a steam reaction-turbine consisting of a spherical vessel pivoted on a central axis and supplied with steam through one of the pivots. The steam

escapes by bent pipes facing tangentially in opposite directions, at opposite ends of a diameter perpendicular to the axis. The globe revolves by reaction from the escaping steam, just as a Barker's mill is driven by escaping water. Another apparatus described by Hero (fig. 1)[1] is interesting as the prototype of a class of engines which long afterwards became practically important. A hollow altar containing air is heated by a fire kindled on it; the air in expanding drives some of the water contained in a spherical vessel beneath the altar into a bucket, which descends and opens the temple doors above by pulling round a pair of vertical posts to which the doors are fixed. When the fire is extinguished the air cools, the water leaves the bucket, and the doors close. In another device a jet of water driven out by expanding air is turned to account as a fountain. Several other philosophical toys or pieces of conjuring apparatus of the like kind are also described, but there is no suggestion that the methods they illustrate could be applied on a large scale or turned to any useful account.
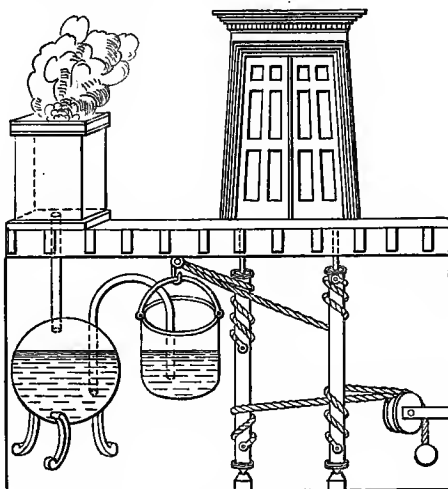
Fig. 1. Apparatus described by Hero.

3. Della Porta and De Caus. From the time of Hero to the 17th century there is no progress to record, though here

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[1] From Greenwood's translation of Hero's *Pneumatics*, edited by B. Woodcroft, 1851.

and there we find evidence that appliances like those described by Hero were used for trivial purposes, such as organ-blowing and the turning of spits. The next distinct step was the publication in 1606 of a treatise on pneumatics by Giovanni Battista Della Porta, in which he shows an apparatus similar to Hero's fountain, but with steam instead of air as the displacing fluid. Steam generated in a separate vessel passed into a closed chamber containing water, and drove the water out through a pipe which opened near the bottom of the vessel. He also points out that the condensation of steam in the closed chamber may be used to produce a vacuum and suck up water from a lower level. In fact, his suggestions anticipate very fully the principle which a century later was applied by Savery in the earliest commercially successful steam-engine. In 1615 Salomon De Caus gives a plan of forcing up water by a steam-fountain which differs from Porta's only in having one vessel serve both as boiler and as displacement-chamber, the hot water being itself raised.

4. Branca's Steam Turbine. Another line of invention was taken by Giovanni Branca (1629), who designed an engine shaped like a water wheel, to be driven by the impact of a jet of steam on its vanes, and, in its turn, to drive other mechanism for various useful purposes. But Branca's suggestion was unproductive, and we find the course of invention revert to the line followed by Porta and De Caus.

5. Marquis of Worcester. The next contributor is one whose place is not easily assigned. To Edward Somerset, second marquis of Worcester, appears to be due the credit of proposing, if not of making, the first useful steam-engine. Its object was to raise water, and it worked probably like Porta's model, but with a pair of displacement-chambers, from each of which alternately water was forced by steam from an independent boiler, or perhaps by applying heat to the chamber itself, while the other vessel was allowed to refill. The only description of the engine is found in Art. 68 of Worcester's *Century of Inventions* (1663). There are no drawings, and the notice is so obscure that it is difficult to say whether there were any distinctly novel features except the double action. The inventor's account leaves much to the imagination. It is entitled "A Fire Water-work," and runs thus :—

"An admirable and most forcible way to drive up water by fire, not by drawing or sucking it upwards, for that must be as the Philosopher calleth it, *Intra sphaeram activitatis*, which is but at such a distance. But this way hath no Bounder, if the Vessels be strong enough; for I have taken a piece of a whole Cannon, whereof the end was burst, and filled it three-quarters full of water, stopping and scruing up the broken end; as also the Touch-hole; and making a constant fire under it, within 24 hours it burst and made a great crack. So that having a way to make my Vessels, so that they are strengthened by the force within them, and the one to fill after the other, I have seen the water run like a constant Fountaine-stream forty foot high; one Vessel of water rarified by fire driveth up forty of cold water. And a man that tends the work is but to turn two Cocks, that one Vessel of water being consumed, another begins to force and re-fill with cold water, and so successively, the fire being tended and kept constant, which the self-same Person may likewise abundantly perform in the interim between the necessity of turning the said Cocks."

Later articles in the *Century of Inventions* contain notices of a device which under the name of a "Water-commanding Engine" received protection by Act of Parliament and was experimented on by Worcester on a large scale at Vauxhall. But there is nothing to show distinctly that the Water-commanding Engine was a heat-engine at all, and the meagre accounts that have been given of it rather point to the conclusion that it was a form of "Perpetual Motion." In any case the experiments led to no practical result.

6. Savery. The steam-engine became commercially success-ful in the hands of Thomas Savery, who in 1698 obtained a patent for a water-raising engine, shown in fig. 2. Steam is admitted to one of the oval vessels A, displacing water, which it drives up through the check-valve B. When the vessel A is emptied of water, the supply of steam is stopped, and the steam already there is condensed by allowing a jet of cold water from a cistern above to stream over the outer surface of the vessel. This produces a vacuum and causes water to be sucked up through the pipe C and the valve D. Meanwhile, steam has been displacing water from the other vessel, and is ready to be condensed there. The valves B and D open only upwards. The supplementary boiler and furnace E are for feeding water to the main boiler; E is filled while cold and a fire is lighted under it; it then acts like the vessel of De Caus in forcing a supply of feed-water into the main boiler F. The gauge-cocks G, G for testing the level of the water in the boiler are an interesting feature of detail. Another form of Savery's engine had only one displacement-chamber and worked intermittently. In the use of artificial means to condense the

steam, and in the application of the vacuum so formed to raise
water by suction from a level lower than that of the engine, the
action used by Savery was probably an advance on that proposed
or used by Worcester; in any case Savery's was the first engine to
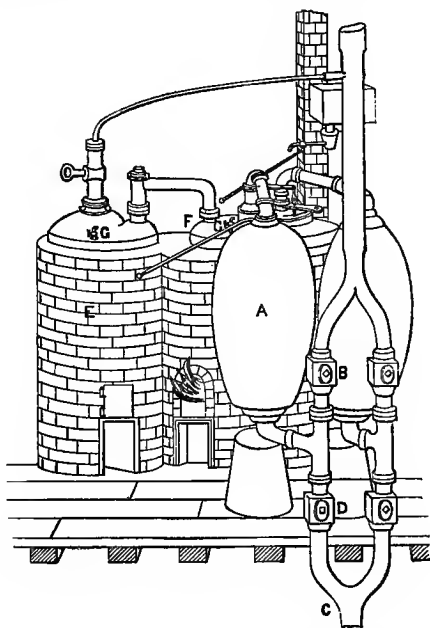
Fig. 2.   Savery's Pumping Engine, 1698.

take a really practical shape.   It found considerable employment
in pumping mines and in raising water to supply houses and
towns, and even to drive water-wheels.   A serious difficulty which
prevented its general use in mines was the fact that the height
through which it would lift water was limited by the pressure the
boiler and vessels could bear.   Pressures as high as 8 or 10
atmospheres were employed—and that, too, without a safety-valve.
But Savery found it no easy matter to deal with high-pressure
steam; he complains that it melted his common solder, and forced
him, as Desaguliers tells us, "to be at the pains and charge to have
all his joints soldered with spelter."   Apart from this drawback
the waste of fuel was enormous, from the condensation of steam
which took place on the surface of the water and on the sides of
the displacement-chamber at each stroke; the consumption of
coal was, in proportion to the work done, some twenty times

greater than it is in a good modern steam-engine. In a tract called *The Miner's Friend*, Savery alludes thus to the alternate heating and cooling of the water-vessel: " On the outside of the vessel you may see how the water goes out as well as if the vessel were transparent, for so far as the steam continues within the vessel so far is the vessel dry without, and so very hot as scarce to endure the least touch of the hand. But as far as the water is, the said vessel will be cold and wet where any water has fallen on it ; which cold and moisture vanishes as fast as the steam in its descent takes place of the water." Before Savery's engine was entirely displaced by its successor, Newcomen's, it was improved by Desaguliers, who applied to it the safety-valve (invented by Papin), and substituted condensation by a jet of cold water within the vessel for the surface condensation used by Savery.

To Savery is ascribed the first use of the familiar term " horse-power " as a measure of the performance of an engine.

7. Gunpowder Engines. Some twenty years before the date of Savery's patent, proposals had been made by several inventors to raise water by means of the explosive power of gunpowder. One scheme was to explode the powder in a closed vessel furnished with valves which opened outwards and allowed a great part of the air and burnt gases to escape when the explosion took place. As the gas that remained became cool a partial vacuum was formed in the vessel, and this was used to draw up water from a lower level. It does not appear that these schemes were ever put in practice except experimentally. The most interesting of the gunpowder engines was that of Huygens (1680), who for the first time introduced the piston and cylinder as constituent parts of a heat-engine. In Huygens' engine the piston was set at the top of a vertical cylinder and a charge of powder was exploded below it. This expelled part of the gaseous contents through valves which opened outwards, and then the cooling of the remainder caused the piston to descend under atmospheric pressure. The piston in descending did work by raising a weight through the medium of a cord and pulley.

8. Papin. In 1690 Denis Papin, who ten years before had invented the safety-valve as an adjunct to his "digester," suggested that the condensation of steam should be employed to make a vacuum under a piston which had been previously raised by the

expansion of the steam. Papin had been associated with Huygens in his experiments on the production of a vacuum under a piston by means of gunpowder, and had described Huygens' machine to the Royal Society. Noticing that after the explosion enough gas remained in the cylinder to fill about one-fifth of its volume, after cooling, he cast about for some means of obtaining a better vacuum. "By another way, therefore, I endeavoured to attain the same end, and since it is a property of water that a small quantity of it, converted into steam by heat, has an elastic force like that of air, but when cold supervenes, is again resolved into water so that no trace of the said elastic force remains, I saw that machines might be constructed wherein water, by means of no very intense heat and at small cost, might produce that perfect vacuum which had failed to be obtained by the use of gunpowder." He goes on to describe what was unquestionably the earliest cylinder and piston steam-engine, and his plan of
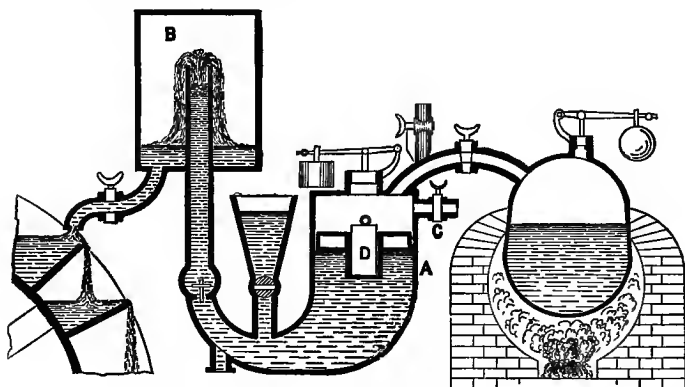
Fig. 3. Papin's modification of Savery's Engine, 1705.

using steam was that which afterwards took practical shape in the atmospheric engine of Newcomen. But his scheme was made unworkable by the fact that he proposed to use but one vessel as both boiler and cylinder. A small quantity of water was placed at the bottom of a cylinder and heat was applied. When the piston had risen the fire was removed, the steam was allowed to cool, and the piston did work in its down-stroke under the pressure of the atmosphere.

After hearing of Savery's engine in 1705 Papin turned his attention to improving it, and devised a modified form, shown

in fig. 3, in which the displacement-chamber A was a cylinder, with a floating diaphragm or piston on the top of the water to keep the water and steam from direct contact with one another. The water was delivered into a closed air-vessel B, from which it issued in a continuous stream against the vanes of a water-wheel. After the steam had done its work in the displacement-chamber it was allowed to escape by the stop-cock C instead of being condensed. This second engine of Papin's was in fact a non-condensing single-acting steam-pump, with steam-cylinder and pump-cylinder in one. A curious feature of it was the heater D, a mass of hot metal placed in the diaphragm for the purpose of keeping the steam dry. Among the many inventions of Papin was a boiler with an internal fire-box,—the earliest example of a construction that is now almost universal[1].

9. Newcomen's " Atmospheric " Engine. While Papin was thus going back from his first notion of a piston-engine to Savery's cruder type, a new inventor had appeared who made the piston-engine a practical success by separating the boiler from the cylinder and by using (as Savery had done) artificial means to condense the steam. This was Newcomen, who in 1705, in conjunction with Savery and with Cawley, gave the steam-engine the form shown in fig. 4. The piston was connected by a chain with one end of an overhead beam. Steam admitted from the boiler to the cylinder allowed the piston to be raised by a heavy counterpoise hanging from the beam near the other end. Then the steam-valve was shut and a jet of cold water entered the cylinder and condensed the steam. The piston was consequently forced down by the pressure of the atmosphere and did work on the pump through the medium of a long rod which hung from the other end of the beam. The next entry of steam expelled the condensed water from the cylinder through an escape valve. The piston was kept tight by a layer of water on its upper surface. Condensation was at first effected by cooling the outside of the cylinder, but an accidental leakage of the packing water past the piston showed the advantage of condensing by a jet of injection water, and this plan took the place of surface condensation. The engine used steam which had a pressure little if at all greater

[1] For an account of Papin's inventions, see his *Life, and Correspondence with Leibnitz and Huygens*, by Dr E. Gerland, Berlin, 1881. See also Muirhead's *Life of Watt*.

than that of the atmosphere; sometimes indeed it was worked with the manhole-lid off the boiler. The function of the steam
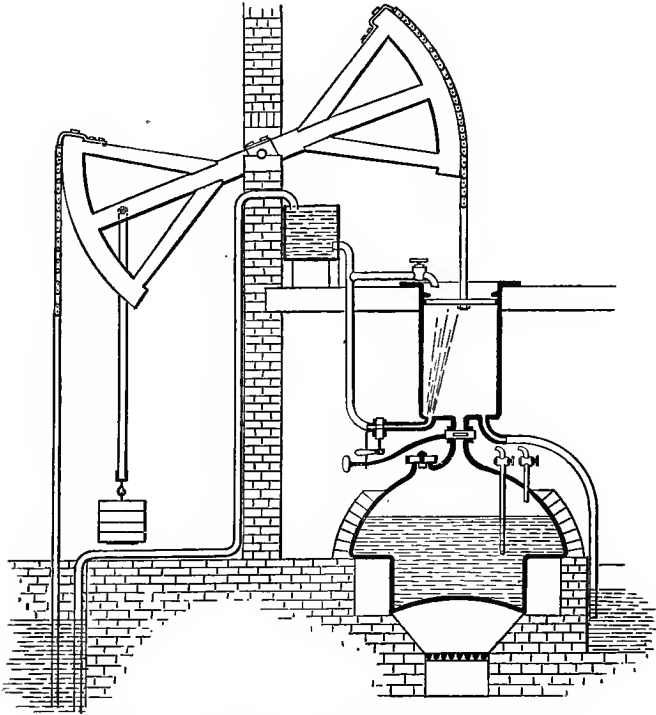
FIG. 4. Newcomen's Atmospheric Engine, 1705.

was merely to allow the piston to be raised, by making the pressure on the under side equal or nearly equal to the pressure on the top, and then to produce a vacuum by being condensed. Newcomen's engine was essentially the cylinder and piston of Papin combined with the separate boiler of Savery.

About 1711 Newcomen's engine began to be introduced for pumping mines. It is doubtful whether the engine was originally automatic in its action or depended on the periodical turning of taps by an attendant. An old print of an engine erected by Newcomen in 1712 near Dudley Castle shows a species of automatic gear. The common story is that in 1713 a boy named Humphrey Potter, whose duty it was to open and shut the valves of an engine he attended, made the engine self-acting by causing the beam itself to open and close the valves by means of cords

and catches. This rude device was simplified in 1718 by Henry Beighton, who suspended from the beam a rod called the plug-tree, which worked the valves by means of tappets. By 1725 the engine was in common use in collieries, and it held its place without material change for about three-quarters of a century in all. Near the close of its career the atmospheric engine was much improved in its mechanical details by Smeaton, who built many large engines of this type about the year 1770, just after the great step which was to make Newcomen's engine obsolete had been taken by James Watt.

Like Savery's engine, Newcomen's was put to no other use than to pump water—in some instances for the purpose of turning water-wheels to drive other machinery. Compared with Savery's it had the great advantage that the intensity of pressure in the pump was not in any way limited by the pressure of the steam, but could be made as great as might be desired by reducing the area of the pump plunger. It shared with Savery's, in a scarcely less degree, the defect already pointed out, that steam was wasted by the alternate heating and cooling of the vessel into which it was led. Even contemporary writers complain of its "vast consumption of fuel," which appears to have been scarcely smaller than that of the engine of Savery.

10. James Watt. In 1763 James Watt, an instrument maker in Glasgow, while engaged by the University in repairing a model of Newcomen's engine, was struck with the waste of steam to which the alternate chilling and heating of the cylinder gave rise. He saw that the remedy, in his own words, would lie in keeping the cylinder as hot as the steam that entered it. With this view he added to the engine a new organ—namely, the *condenser*—a vessel separate from the cylinder, into which the steam should be allowed to escape from the cylinder, to be condensed there by the application of cold water either outside or as a jet. To preserve the vacuum in his condenser he added a pump, called the air-pump, whose function was to pump from it the condensed steam and water of condensation, as well as the air which would otherwise accumulate by leakage inwards or by being brought in with the steam or with the injection water. Then as the cylinder was itself no longer used as the chamber in which the steam was condensed he was able to keep it continuously hot by

clothing it with non-conducting bodies, and in particular by the use of a *steam-jacket,* or layer of hot steam between the cylinder and an external casing. Further, and still with the same object, he covered in the top of the cylinder, taking the piston-rod out through a steam-tight stuffing-box, and allowed steam instead of air to press upon the piston's upper surface. The idea of using a separate condenser had no sooner occurred to Watt than he put

it to the test by constructing the apparatus shown in fig. 5. There A is the cylinder, B a condenser (of the type now distinguished as a sur-face-condenser) and C is the air-pump. The cylinder was filled with steam above the piston, and a vacuum was formed in the surface-condenser B. On opening the stop-cock D the steam rushed over from the cylinder and was condensed, while the piston rose and lifted a weight. A fuller account of this experiment will be found in Watt's narrative, below.
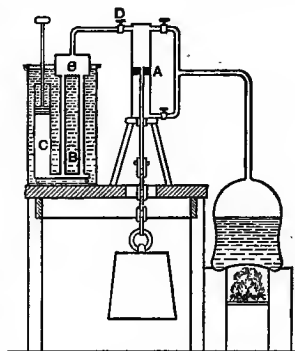
Fig. 5.
Watt's Experimental Apparatus.

After several trials Watt patented his improvements in 1769; they are described in his specification in the following words, which, apart from their immense historical interest, deserve careful study as a statement of principles which to this day guide the scientific development of the steam-engine :—

" My method of lessening the consumption of steam, and consequently fuel, in fire-engines, consists of the following principles:—

" *First,* That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire-engines, and which I call the steam-vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

" *Secondly,* In engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam-vessels or cylinders, although occasionally communicating with them ; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water or other cold bodies.

" *Thirdly,* Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the

steam-vessels or condensers by means of pumps, wrought by the engines themselves, or otherwise.

"*Fourthly*, I intend in many cases to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in common fire-engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the open air after it has done its office.......

"*Sixthly*, I intend in some cases to apply a degree of cold not capable of reducing the steam to water, but of contracting it considerably, so that the engines shall be worked by the alternate expansion and contraction of the steam.

"*Lastly*, Instead of using water to render the pistons and other parts of the engine air and steam-tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver and other metals in their fluid state."

The fifth claim was for a rotary engine, and need not be quoted here.

The "common fire-engine" alluded to was the steam-engine, or, as it was more generally called, the "atmospheric" engine, of Newcomen. Enormously important as Watt's first patent was, it resulted for a time in the production of nothing more than a greatly improved engine of the Newcomen type, much less wasteful of fuel, able to make faster strokes, but still only suitable for pumping, still single-acting, with steam admitted during the whole stroke, the piston still pulling the beam by a chain working on a circular arc. The condenser was generally kept cool by the injection of cold water, but Watt has left a model of a surface-condenser made up of small tubes, in every essential respect like the condensers now used in marine engines. He also used, as we have seen, a surface-condenser in the experimental apparatus by which the practicability of condensation in a separate vessel was first demonstrated.

11. Watt's pumping-engine of 1769. Fig. 6 is an example of the Watt pumping-engine of this period. It should be noticed that, although the top of the cylinder is closed and steam has access to the upper side of the piston, this is done only to keep the cylinder and piston warm. The engine is still single-acting; the steam on the upper side merely plays the part which was played in Newcomen's engine by the atmosphere; and it is the lower end of the cylinder alone that is ever put in communication with the condenser. There are three valves,—the "steam" valve *a*, the "equilibrium" valve *b*, and the "exhaust"

valve *c*. At the beginning of the down-stroke *c* is opened to
produce a vacuum below the piston and *a* is opened to admit
steam above it. At the end of the down-stroke *a* and *c* are shut
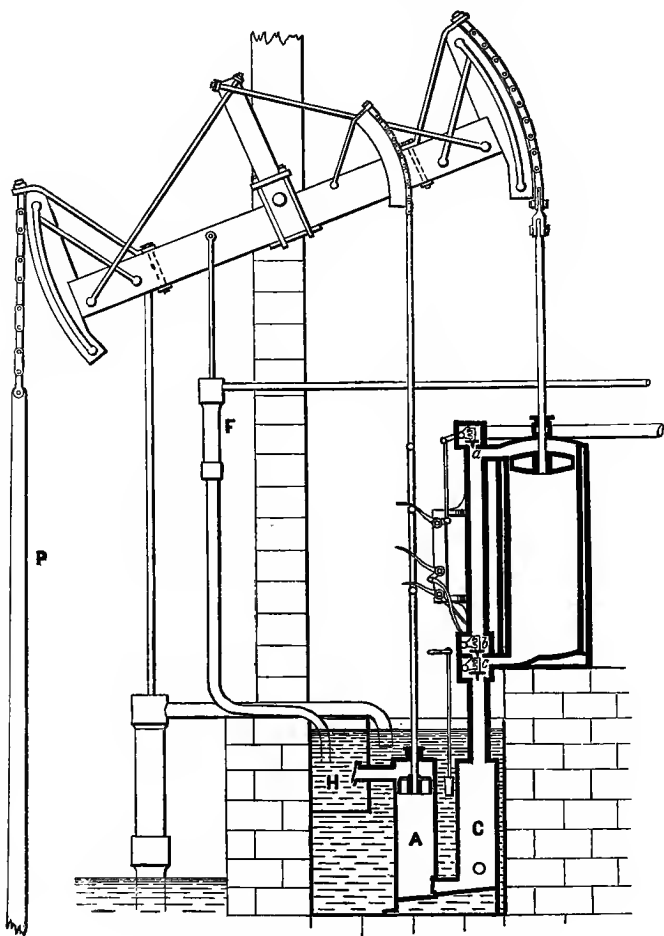
Fɪɢ. 6.   Watt's Single-acting Engine, 1769.

and *b* is opened. This puts the two sides of the piston in
equilibrium, and allows the piston to be pulled up by the pump-
rod P, which is heavy enough to serve as a counterpoise.   C is the
condenser, and A the air-pump, which discharges into the hot well
H, whence the supply of the feed-pump F is drawn.

12. Watt's narrative of his invention. In a note appended to the article "Steam-Engine" in Robison's *System of Mechanical Philosophy* (1822) Watt has given the following account of the experiments and reflexions which led up to his first patent. This narrative is of so particular interest that no apology need be made for reproducing it in full.

" My attention was first directed in the year 1759 to the subject of steam-engines, by the late Dr Robison himself, then a student in the University of Glasgow, and nearly of my own age. He at that time threw out an idea of applying the power of the steam-engine to the moving of wheel-carriages, and to other purposes, but the scheme was not matured, and was soon abandoned on his going abroad.

"About the year 1761 or 1762, I tried some experiments on the force of steam, in a Papin's digester, and formed a species of steam-engine by fixing upon it a syringe one-third of an inch diameter, with a solid piston, and furnished also with a cock to admit the steam from the digester, or shut it off at pleasure, as well as to open a communication from the inside of the syringe to the open air, by which the steam contained in the syringe might escape. When the communication between the digester and syringe was opened, the steam entered the syringe, and by its action upon the piston raised a considerable weight (15 lbs.) with which it was loaded.

" When this was raised as high as was thought proper, the communication with the digester was shut, and that with the atmosphere opened; the steam then made its escape, and the weight descended. The operations were repeated, and though in this experiment the cock was turned by hand, it was easy to see how it could be done by the machine itself, and to make it work with perfect regularity. But I soon relinquished the idea of constructing an engine upon this principle, from being sensible it would be liable to some of the objections against Savery's engine, viz. the danger of bursting the boiler, and the difficulty of making the joints tight, and also that a great part of the power of the steam would be lost, because no vacuum was formed to assist the descent of the piston. [I, however, described this engine in the fourth article of the specification of my patent of 1769 ; and again in the specification of another patent in the year 1784, together with a mode of applying it to the moving of wheel-carriages.]

" The attention necessary to the avocations of business prevented me from then prosecuting the subject farther; but in the winter of 1763-4, having occasion to repair a model of Newcomen's engine belonging to the Natural Philosophy class of the University of Glasgow, my mind was again directed to it. At that period, my knowledge was derived principally from Desaguliers, and partly from Belidor. I set about repairing it as a mere mechanician, and when that was done and it was set to work, I was surprised to find that its boiler could not supply it with steam, though apparently quite large enough (the cylinder of the model being two inches in diameter and six inches stroke, and the boiler about nine inches diameter). By blowing the fire it was made to take a few strokes, but required an enormous quantity of injection water, though it was very lightly loaded by the column of water in the pump. It soon occurred that this was caused by the little cylinder exposing a greater surface to condense the steam than the cylinders of larger engines did in proportion to their respective contents. It was found that by shortening the column of water in the pump, the boiler could supply the cylinder with steam, and

that the engine would work regularly with a moderate quantity of injection. It now appeared that the cylinder of the model being of brass, would conduct heat much better than the cast-iron cylinders of larger engines (generally covered on the inside with a stony crust), and that considerable advantage could be gained by making the cylinders of some substance that would receive and give out heat slowly: of these, wood seemed to be the most likely, provided it should prove sufficiently durable.

"A small engine was therefore constructed with a cylinder six inches diameter, and twelve inches stroke, made of wood, soaked in linseed oil, and baked to dryness. With this engine many experiments were made; but it was soon found that the wooden cylinder was not likely to prove durable, and that the steam condensed in filling it still exceeded the proportion of that required for large engines according to the statements of Desaguliers. It was also found, that all attempts to produce a better exhaustion by throwing in more injection, caused a disproportionate waste of steam. On reflection, the cause of this seemed to be the boiling of water in vacuo at low heats, a discovery lately made by Dr Cullen, and some other philosophers (below 100°, as I was then informed), and, consequently, at greater heats, the water in the cylinder would produce a steam which would, in part, resist the pressure of the atmosphere.

"By experiments which I then tried upon the heats at which water boils under several pressures greater than that of the atmosphere, it appeared, that when the heats proceeded in an arithmetical, the elasticities proceeded in some geometrical ratio; and by laying down a curve from my data, I ascertained the particular one near enough for my purpose. It also appeared, that any approach to a vacuum could only be obtained by throwing in large quantities of injection, which would cool the cylinder so much as to require quantities of steam to heat it again, out of proportion to the power gained by the more perfect vacuum; and that the old engineers had acted wisely in contenting themselves with loading the engine with only six or seven pounds on each square inch of the area of the piston.

"It being evident that there was a great error in Dr Desaguliers' calculations of Mr Beighton's experiments on the bulk of steam, a Florence flask, capable of containing about a pound of water, had about one ounce of distilled water put into it; a glass tube was fitted into its mouth, and the joining made tight by lapping that part of the tube with packthread covered with glazier's putty. When the flask was set upright, the tube reached down near to the surface of the water, and in that position the whole was placed in a tin reflecting oven before a fire, until the water was wholly evaporated, which happened in about an hour, and might have been done sooner had I not wished the heat not much to exceed that of boiling water. As the air in the flask was heavier than the steam, the latter ascended to the top, and expelled the air through the tube.

"When the water was all evaporated, the oven and flask were removed from the fire, and a blast of cold air was directed against one side of the flask, to collect the condensed steam in one place. When all was cold, the tube was removed, the flask and its contents were weighed with care; and the flask being made hot, it was dried by blowing into it by bellows, and when weighed again, was found to have lost rather more than four grains, estimated at 4½ grains.

"When the flask was filled with water, it was found to contain about 17¼ ounces avoirdupois of that fluid, which gave about 1800 for the expansion of water converted into steam of the heat of boiling water.

"This experiment was repeated with nearly the same result; and in order to ascertain whether the flask had been wholly filled with steam, a similar quantity of water was for the third time evaporated; and, while the flask was still cold, it was

placed inverted, with its mouth (contracted by the tube) immersed in a vessel of water, which it sucked in as it cooled, until in the temperature of the atmosphere it was filled to within half-an-ounce measure of water. [In the contrivance of this experiment I was assisted by Dr Black. In Dr Robison's edition of Dr Black's lectures, Vol. I. page 147, the latter hints at some experiments upon this subject as made by him; but I have no knowledge of any except those which I made myself.]

"In repetitions of this experiment at a later date, I simplified the apparatus by omitting the tube, and laying the flask upon its side in the oven, partly closing its mouth by a cork having a notch on one side, and otherwise proceeding as has been mentioned. I do not consider these experiments as extremely accurate, the only scale-beam of a proper size which I had then at my command not being very sensible, and the bulk of the steam being liable to be influenced by the heat to which it is exposed, which, in the way described, is not easily regulated or ascertained; but, from my experience in actual practice, I esteem the expansion to be rather more than I have computed.

"A boiler was constructed which showed, by inspection, the quantity of water evaporated in any given time, and thereby ascertained the quantity of steam used in every stroke by the engine, which I found to be several times the full of the cylinder. Astonished at the quantity of water required for the injection, and the great heat it had acquired from the small quantity of water in the form of steam which had been used in filling the cylinder, and thinking I had made some mistake, the following experiment was tried :—A glass tube was bent at right angles, one end was inserted horizontally into the spout of a tea-kettle, and the other part was immersed perpendicularly in well-water contained in a cylindric glass vessel, and steam was made to pass through it until it ceased to be condensed, and the water in the glass vessel was become nearly boiling hot. The water in the glass vessel was then found to have gained an addition of about one-sixth part from the condensed steam. Consequently, water converted into steam can heat about six times its own weight of well-water to 212°, or till it can condense no more steam. Being struck with this remarkable fact, and not understanding the reason of it, I mentioned it to my friend Dr Black, who then explained to me his doctrine of latent heat, which he had taught for some time before this period (summer 1764), but having myself been occupied with the pursuits of business, if I had heard it I had not attended to it, when I thus stumbled upon one of the material facts by which that beautiful theory is supported.

"On reflecting further, I perceived, that in order to make the best use of steam, it was necessary, first, that the cylinder should be maintained always as hot as the steam which entered it; and secondly, that when the steam was condensed, the water of which it was composed, and the injection itself, should be cooled down to 100°, or lower, where that was possible. The means of accomplishing these points did not immediately present themselves; but early in 1765 it occurred to me, that if a communication were opened between a cylinder containing steam, and another vessel which was exhausted of air and other fluids, the steam, as an elastic fluid, would immediately rush into the empty vessel, and continue so to do until it had established an equilibrium; and if that vessel were kept very cool by an injection or otherwise, more steam would continue to enter until the whole was condensed. But both the vessels being exhausted, or nearly so, how was the injection water, the air which would enter with it, and the condensed steam, to be got out?

"This I proposed, in my own mind, to perform in two ways. One was by adapting to the second vessel a pipe reaching downwards more than 34 feet, by

which the water would descend (a column of that length overbalancing the atmosphere), and by extracting the air by means of a pump.

"The second method was by employing a pump, or pumps, to extract both the air and the water, which would be applicable in all places, and essential in those cases where there was no well or pit.

"This latter method was the one I then preferred, and is the only one I afterwards continued to use. In Newcomen's engine, the piston is kept tight by water, which could not be applicable in this new method; as, if any of it entered into a partially exhausted and hot cylinder, it would boil and prevent the production of a vacuum, and would also cool the cylinder, by its evaporation during the descent of the piston.

"I proposed to remedy this defect by employing wax, tallow, or other grease, to lubricate and keep the piston tight. It next occurred to me, that the mouth of the cylinder being open, the air which entered to act on the piston would cool the cylinder, and condense some steam on again filling it, I therefore proposed to put an air-tight cover upon the cylinder, with a hole and stuffing-box for the piston-rod to slide through and to admit steam above the piston to act upon it instead of the atmosphere. [N.B. The piston-rod sliding through a stuffing-box was new in steam-engines; it was not necessary in Newcomen's engine, as the mouth of the cylinder was open, and the piston stem was square and very clumsy. The fitting the piston-rod to the piston by a cone was an after improvement of mine (about 1774).] There still remained another source of the destruction of steam, the cooling of the cylinder by the external air, which would produce an internal condensation whenever steam entered it, and which would be repeated every stroke; this I proposed to remedy by an external cylinder containing steam, surrounded by another of wood, or of some other substance which would conduct heat slowly.

"When once the idea of the separate condensation was started, all these improvements followed as corollaries in quick succession, so that in the course of one or two days, the invention was thus far complete in my mind, and I immediately set about an experiment to verify it practically. I took a large brass syringe, 1¾ inches diameter, and 10 inches long, made a cover and bottom to it of tin-plate, with a pipe to convey steam to both ends of the cylinder from the boiler; another pipe to convey steam from the upper end to the condenser (for, to save apparatus, I inverted the cylinder). I drilled a hole longitudinally through the axis of the stem of the piston, and fixed a valve at its lower end, to permit the water which was produced by the condensed steam on first filling the cylinder, to issue. The condenser used upon this occasion consisted of two pipes of thin tin-plate, ten or twelve inches long, and about one-sixth inch diameter, standing perpendicular, and communicating at top with a short horizontal pipe of large diameter, having an aperture on its upper side which was shut by a valve opening upwards. These pipes were joined at bottom to another perpendicular pipe of about an inch diameter, which served for the air and water-pump; and both the condensing pipes and the air-pump were placed in a small cistern filled with cold water. [N.B. This construction of the condenser was employed from knowing that heat penetrated thin plates of metal very quickly, and considering that if no injection was thrown into an exhausted vessel, there would be only the water of which the steam had been composed, and the air which entered with the steam, or through the leaks, to extract.]

"The steam-pipe was adjusted to a small boiler. When steam was produced, it was admitted into the cylinder, and soon issued through the perforation of the rod, and at the valve of the condenser. When it was judged that the air was expelled, the steam-cock was shut, and the air-pump piston-rod was drawn up, which leaving

the small pipes of the condenser in a state of vacuum, the steam entered them and was condensed. The piston of the cylinder immediately rose and lifted a weight of about 18 lbs., which was hung to the lower end of the piston-rod. The exhaustion cock was shut, the steam was readmitted into the cylinder, and the operation was repeated, the quantity of steam consumed, and the weights it could raise were observed, and, excepting the non-application of the steam-case and external covering, the invention was complete, in so far as regarded the savings of steam and fuel.

"A large model, with an outer cylinder and wooden case, was immediately constructed, and the experiments made with it served to verify the expectations I had formed, and to place the advantage of the invention beyond the reach of doubt. It was found convenient afterwards to change the pipe-condenser for an empty vessel, generally of a cylindrical form, into which an injection played, and in consequence of there being more water and air to extract, to enlarge the air-pump.

"The change was made, because, in order to procure a surface sufficiently extensive to condense the steam of a large engine, the pipe-condenser would require to be very voluminous, and because the bad water with which engines are frequently supplied would crust over the thin plates, and prevent their conveying the heat sufficiently quick. The cylinders were also placed with their mouths upwards, and furnished with a working-beam and other apparatus as was usual in the ancient engines; the inversion of the cylinder, or rather of the piston-rod, in the model, being only an expedient to try more easily the new invention, and being subject to many objections in large engines."

13. Development of Watt's Engine: the rotative type. In a second patent (1781) Watt describes the "sun-and-planet" wheels and other methods of making the engine give continuous revolving motion to a shaft provided with a fly-wheel. He had intended to use the crank and connecting-rod, for this purpose (a mechanical device familiar even at that time from its use in the common foot-lathe), and had even made a model of it, but the application of the crank to the steam-engine had meanwhile been patented by one Pickard, and Watt, rather than make terms with Pickard, made use of his sun-and-planet motion until the patent for the application of the crank expired. The reciprocating motion of earlier forms had served only for pumping, but by this invention Watt opened up for the steam-engine a thousand other channels of usefulness. The engine was still single-acting; the connecting rod was attached to the far end of the beam, and that carried a counterpoise which served to raise the piston when steam was admitted below it.

14. Further improvements by Watt. In 1782 Watt patented two further improvements of the first importance, both of which he had invented some years before. One was the use of double action, that is to say, the application of steam and vacuum to each side of the piston alternately. The other, which had

been invented as early as 1769, was the use of steam expansively, in other words, the plan (now used in all engines that aim at
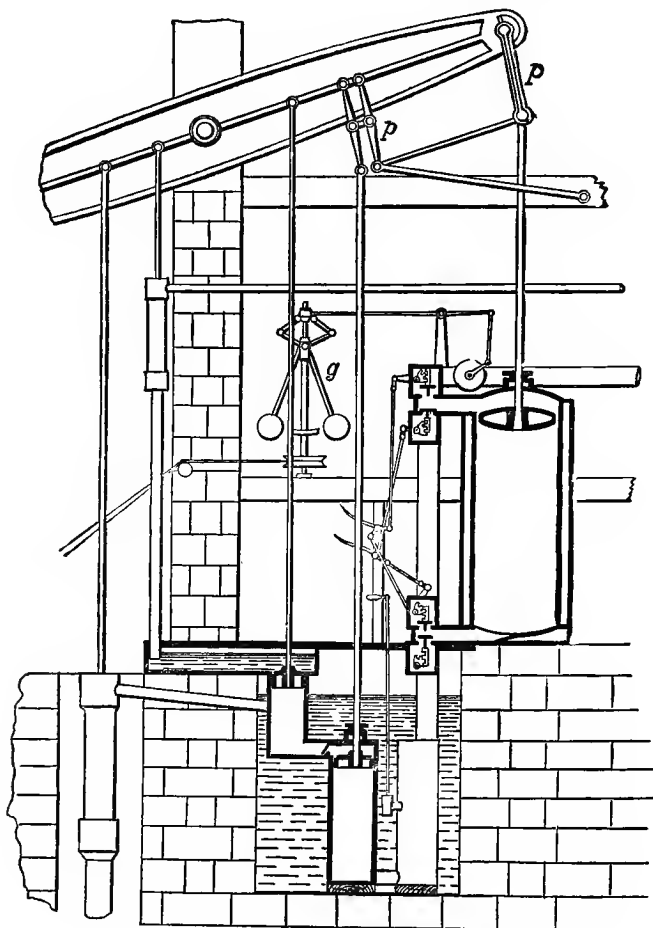
Fig. 7.   Watt's Double-acting Engine, 1782.

economy of fuel) of stopping the admission of steam when the piston had made only a part of its stroke, and allowing the rest of the stroke to be performed by the expansion of the steam already in the cylinder.   To let the piston push as well as pull the end of the beam Watt devised his so-called parallel motion, an arrangement of links connecting the piston-rod head with the beam in such a way as to guide the rod to move in a very nearly

straight line. He further added the throttle-valve, for regulating the rate of admission of steam, and the centrifugal governor, a double conical pendulum, which controlled the speed by acting on the throttle-valve. The stage of development reached at this time is illustrated by the engine of fig. 7 (from Stuart's *History of the Steam-engine*), which shows the parallel motion *pp*, the governor *g*, the throttle-valve *t*, and a pair of steam and exhaust valves at each end of the cylinder.

Among other inventions of Watt were the "indicator," by which diagrams showing the relation of the steam-pressure in the cylinder to the movement of the piston are automatically drawn; a steam tilt-hammer; and also a steam locomotive for ordinary roads,—but this invention was not prosecuted. As an inventor Watt was skilfully seconded by his assistant Murdoch, to whose ingenuity, he says, are due "many improvements"— amongst them, the introduction of the slide-valve as a means of controlling the admission and release of steam.

In partnership with Matthew Boulton, Watt carried on in Birmingham the manufacture and sale of his engines with the utmost success, and held the field against all rivals in spite of severe assaults on the validity of his patents. A special Act of Parliament was obtained which extended the patent monopoly for a term of twenty-five years from 1775. Notwithstanding Watt's knowledge of the advantage to be gained by using steam expansively he continued to employ only low pressures—seldom more than 7 lbs. per square inch over that of the atmosphere. His boilers were fed, as Newcomen's had been, through an open pipe which rose high enough to let the column of water in it balance the pressure of the steam. Following Savery, he adopted the term "horse-power" as a mode of rating engines and gave it a particular meaning, by defining one horse-power as the rate at which work is done when 33,000 lbs. are raised one foot in one minute. This estimate was based on trials of the work done by horses; it is excessive as a statement of what an average horse can do in working continuously for any long time, but Watt purposely made it excessive in order that his customers might have no reason to complain on this score.

15. **Non-condensing Steam-Engines.** In the fourth claim in Watt's first patent, the second sentence describes a

non-condensing engine, which would have required steam of
a considerably higher pressure than served in the condensing
engine. His narrative also shows that he had made experiments
in this direction before devising the separate condenser. This,
however, was a line of invention which Watt did not follow up,
perhaps because so early as 1725 a non-condensing engine had
been described by Leupold in his *Theatrum Machinarum*.
Leupold's proposed engine (for the main features of which he
professes himself indebted to Papin) is shown in fig. 8, which
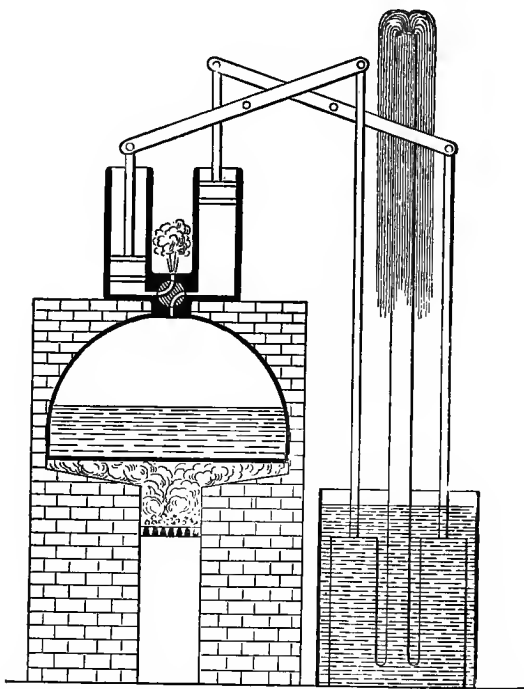
FIG. 8. Non-condensing Engine described by Leupold (1725).

makes its action sufficiently clear. Watt's aversion to high-
pressure steam was strong, and its influence on steam-engine
practice long survived the expiry of his patents. So much indeed
was this the case that the terms "high-pressure" and "non-
condensing" were for many years synonymous, in contradis-
tinction to the "low-pressure" or condensing engines of Watt.
This nomenclature no longer holds good; in modern practice

many condensing engines use as high pressures as non-condensing engines, and by doing so are able to take advantage of Watt's great invention of expansive working to a degree which was impossible in his own practice.

16. Use of comparatively high-pressure steam. The introduction of the non-condensing and, at that time, relatively high-pressure engine was effected in England by Trevithick and in America by Oliver Evans about 1800. Both Evans and Trevithick applied their engines to propel carriages on roads, and both used for boiler a cylindrical vessel with a cylindrical flue inside—the construction now known as the Cornish boiler. In partnership with Bull, who had been a workman in the employment of Boulton and Watt, Trevithick had previously made direct-acting pumping-engines, with an inverted cylinder set over and in line with the pump-rod, thus dispensing with the beam that had been a feature in all earlier forms. But in these " Bull " engines, as they are called, a condenser was used, or, rather, the steam was condensed by a jet of cold water in the exhaust-pipe, and Boulton and Watt successfully opposed them as infringing the patent for condensation in a separate vessel. To Trevithick belongs the distinguished honour of being the first to use a steam-carriage on a railway; in 1804 he built a locomotive in the modern sense, to run on what had formerly been a horse-tramway in Wales; and it is noteworthy that the exhaust steam was discharged into the funnel to force the furnace draught, a device which, 25 years later, in the hands of George Stephenson, went far to make the locomotive what it is to-day. In this connexion it may be added that as early as 1769 a steam-carriage for roads had been built in France by Cugnot, who used a pair of single-acting high-pressure cylinders to turn a driving axle step by step by means of pawls and ratchet-wheels. To the initiative of Evans may be ascribed the early general use of high-pressure steam in the United States, a feature which for many years distinguished American from English practice.

17. Compound Engines. Hornblower and Woolf. Amongst the contemporaries of Watt one name deserves special mention. In 1781 Jonathan Hornblower constructed and patented what would now be called a compound engine, with two cylinders of different sizes. Steam was first admitted into the smaller

cylinder, and then passed over into the larger, doing work against a piston in each. In Hornblower's engine the two cylinders were placed side by side, and both pistons acted on the same end of a beam overhead. This was an instance of the use of steam expansively, and as such was earlier than the patent, though not earlier than the invention, of expansive working by Watt. Hornblower was crushed by the Birmingham firm for infringing their patent in the use of a separate condenser and air-pump.

Soon after the expiry of Watt's master patent in 1800 the compound engine was revived by Woolf, with whose name it is often associated. Using steam of fairly high pressure, and cutting off the supply before the end of the stroke in the small cylinder, Woolf expanded the steam to six or even nine times its original volume. Mechanically the double-cylinder compound engine has this advantage over an engine in which the same amount of expansion is performed in a single cylinder, that the thrust or pull exerted by the two pistons in the compound engine varies less throughout the action than that which is exerted by the piston of the single-cylinder engine. This advantage may have been clear to Hornblower and Woolf, and to other early users of compound expansion. But another and a more important merit of the system lies in a fact of which neither they nor for many years their followers in the use of compound engines were aware—the fact that by dividing the whole range of expansion into two parts the cylinders in which these are separately performed are subject to a reduced range of fluctuation in their temperature. This, as we shall have occasion to point out more particularly in a later chapter, limits to a great extent a source of waste which is present in all steam-engines, namely, the waste which results from the heating and cooling of the metal by its alternate contact with hot and cooler steam. The system of compound expansion is now used in nearly all large engines that pretend to economy. Its introduction forms the only great improvement which the steam-engine has undergone since the time of Watt; and we are now able to recognize it as a very important step in the direction set forth in his " first principle," that the cylinder should be kept as hot as the steam that enters it.

18. The Cornish Pumping Engine. Woolf introduced the compound engine somewhat widely about 1814, as a pumping

engine in the mines of Cornwall.   But it met a strong competitor
there in the high-pressure single-cylinder condensing engine,
which was at that time developing, in the hands of Trevithick
and others, into a machine of great efficiency, and which had
an evident advantage over Woolf's in the simplicity of its con-
struction.   Woolf's engine fell into comparative disuse, and the
single-cylinder type took a form which, under the name of the
Cornish pumping engine, was for many years famous for its great
economy of fuel.   In this engine the cylinder was set under one
end of a beam, from the other end of which hung a heavy rod
which operated a pump at the foot of the shaft.   Steam was
admitted above the piston for a short portion of the stroke, there-
by raising the pump-rod, and was allowed to expand for the
remainder.   Then an equilibrium valve, connecting the spaces
above and below the piston, as in fig. 6, was opened, and the
pump-rod descended, doing work in the pump and raising the
engine piston.   The large mass which had to be started and
stopped at each stroke served by its inertia to counterbalance the
inequalities of steam pressure which were due to expansive
working, for the pump rods and other reciprocating parts stored
up energy of motion in the early part of the stroke, when the
steam pressure was greatest, and gave out energy in the later part,
when expansion had greatly lowered the pressure.   The frequency
of the stroke was controlled by a device called a cataract, con-
sisting of a small plunger pump, in which the plunger, raised at
each stroke by the engine, was allowed to descend more or less
slowly by the escape of fluid below it through an adjustable
orifice, and in its descent liberated catches which held the steam
and exhaust valves from opening.   A similar device controlled the
equilibrium valve.   The cataract could be set to give a pause at
the end of the piston's down-stroke, so that the pump cylinder
might have time to become completely filled.

The Cornish engine is interesting as the earliest form which
achieved an efficiency at all comparable with that of good modern
engines.   For many years monthly reports were published of the
"duty" of these engines, the "duty" being the number of foot-
pounds of work done per bushel or (in some cases) per cwt. of
coal.   The performance of the engines became a matter of almost
sporting interest to mining engineers, and no pains were spared
to "beat the record."   The average duty of engines in the

Cornwall district rose from about 18 millions of foot-pounds per cwt. of coal in 1813 to 68 millions in 1844, after which less effort seems to have been made to maintain a high efficiency[1]. In individual cases much higher results were reported, as in the Fowey Consols engine, which in 1835 was stated to have a duty of 125 millions. This (to use a more modern mode of reckoning) is equivalent to the consumption of only a little more than 1¾ lb. of coal per hour per horse-power—a result surpassed by very few engines in even the best recent practice. It is difficult to credit figures which, even in exceptional instances, place the Cornish engine of that period on a level with the most efficient modern engines—in which compound expansion and higher pressure combine to make a much more perfect thermodynamic machine ; and apart from this there is room to question the accuracy of the Cornish reports. They played, however, a useful part in the process of steam-engine development by directing attention to the question of efficiency, and by demonstrating the advantage to be gained from high-pressure and expansive working, at a time when the theory of the steam-engine had not yet taken shape.

It may be added that the success of the Cornish type was no doubt largely responsible for the tendency which down to a very recent period engine builders have shown to interpose a beam between the steam-cylinder and the pump or crank on which work is being done. For a long time the beam appears, in one form or another, as an almost inevitable part of a steam-engine. The lesson to be learnt from Bull's early direct-acting engine was apparently, in general, overlooked.

19. Revival of the Compound Engine. The final revival of the compound engine did not occur until about the middle of the nineteenth century, and then several agencies combined to bring it about. In 1845 M'Naught introduced a plan of improving beam engines of the original Watt type, by adding a small high-pressure cylinder with a piston acting on the beam between the centre and the fly-wheel end. Steam of higher pressure than had formerly been used, after doing work in the new cylinder, passed into the old or low-pressure cylinder, where it was further expanded. Many engines whose power was proving insufficient for

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[1] *Min. Proc. Inst. C.E.*, vol. xxiii., 1863.

the extended machinery they had to drive were "M'Naughted" in this way, and after conversion were found not only to exert more power but to show a marked economy of fuel. The compound form was selected by William Pole for the pumping engines of Lambeth and other waterworks about 1850; in 1854 John Elder began to use it in marine engines; in 1857 E. A. Cowper added an intermediate receiver or reservoir for steam between the high- and low-pressure cylinders, which made it unnecessary for the low-pressure piston to be just beginning when the other piston was just ending its stroke. To keep the steam hot in the receiver he fitted it with a steam jacket. As the mechanical construction of engines and boilers improved and facilities therefore increased for the use of high-pressure steam, compound expansion came into more general use, its advantage becoming more conspicuous with every increase in boiler pressure. Now-a-days there are few large land engines and scarcely any marine engines that are not compound. In marine practice especially, where economy of fuel is a much more important factor in determining the design than it is on land, the principle of compound expansion has been greatly extended by the general introduction of triple and quadruple expansion engines, in which the steam is made to expand successively in three or in four cylinders. Even in locomotives for railways, where other considerations are of more moment than the saving of coal, compound expansion has found some application, though its use there is comparatively rare.

The growth of compound expansion has been referred to at some length, because it forms the most definite improvement which the piston and cylinder type of steam-engine has undergone since the time of Watt. For the rest, the progress of the steam-engine has consisted in its adaptation to particular uses, in the invention of features of mechanical detail, in the recognition and application of thermo-dynamical principles, in better structural design and in improved methods of manufacture by which it has profited in common with all other machines. These have made possible the use of steam with eight or ten times the pressure of that employed by Watt, and have allowed the mean speed of movement of the piston to be greatly increased, with consequent gains both in the amount of power obtainable from an engine of given size and in the efficiency of the action.

20. Application to locomotives. The adaptation of the steam-engine to railways, begun by Trevithick, became a success in the hands of George Stephenson, whose engine the " Rocket," when tried along with others on a part of the Liverpool and Manchester railway at Rainhill in October 1829, not only distanced its competitors but settled once and for all the question whether horse traction or steam traction was to be used on railways. The principal features of the " Rocket " were an improved steam-blast for urging the combustion of coal and a boiler (suggested by Henry Booth, the secretary of the railway) in which a large heating surface was given by the use of many small tubes through which the hot gases passed. Further, the cylinders, instead of being vertical as in earlier locomotives, were set at a slope, which was afterwards altered to a position still more nearly horizontal. To these features there was added later the " link motion," a contrivance which enabled the engine to be quickly reversed and the amount of expansion to be readily varied. In the hands of George Stephenson and his son Robert the locomotive took a form which has been in all essentials maintained by the far heavier locomotives of modern practice.

21. Application to steamboats. The first practical steamboat was the tug " Charlotte Dundas," built by William Symington, and tried in the Forth and Clyde Canal in 1802. A Watt double-acting condensing engine, placed horizontally, acted directly by a connecting-rod on the crank of a shaft at the stern, which carried a revolving paddle-wheel. The trial was successful, but steam towing was abandoned for fear of injuring the banks of the canal. Ten years later Henry Bell built the " Comet," with side paddle-wheels, which ran as a passenger steamer on the Clyde; but an earlier inventor to follow up Symington's success was the American Robert Fulton, who, after unsuccessful experiments on the Seine, fitted a steamer on the Hudson in 1807 with engines made to his designs by Boulton and Watt, and brought steam navigation for the first time to commercial success.

The American river boats soon began to use high-pressure steam, but English engineers looked askance on a practice which led to frequent explosions. They were moreover slow to realise that high pressure is a necessary condition of economical working. In 1835 it was usual for the pressure in marine boilers to be no

more than 4 or 5 pounds per square inch above the pressure of the atmosphere, and for many years later pressures of 20 or 25 pounds were common. With the introduction of compound working and with the substitution of cylindrical boilers for the weak box-boiler originally used on board ship the pressure rose considerably. In 1872 Sir F. J. Bramwell, describing the typical marine practice of that time[1], gave a list of engines—all compound—in which the pressure ranged from 45 to 60 pounds. The consumption of coal in these engines was generally from 2 to 2½ pounds per hour per indicated horse-power, and the mean piston speed was about 350 feet per minute. Nine years later Mr F. C. Marshall gave a similar list[2], in which the mean pressure was 77 pounds, the mean piston speed about 460 feet per minute, and the consumption of coal a trifle under 2 pounds per hour per indicated horse-power. These engines were also of the type in which steam is successively expanded in two cylinders. The triple expansion type of engine was introduced by A. C. Kirk in 1874 but did not come into general use until after 1881. It became the normal type of marine engine, and was associated with a marked advance in boiler pressure and a considerable gain in economy of fuel. Reviewing the progress of marine engineering in the decade from 1881 to 1891 Mr Blechynden[3] gave a list of triple engines with boiler pressures of about 160 pounds and piston speeds of about 500 or 600 feet per minute. These engines consumed on the average about 1½ pounds of coal per indicated horse-power-hour[4]. In later practice pressures of 200 pounds and over are not unusual and the piston speed is sometimes as high as 900 or 1000 feet per minute. In many cases quadruple expansion is employed, that is to say the steam expands successively in four stages through four separate cylinders.

The progressive rise in steam-pressure and in piston speed has not only increased the efficiency of engines but has greatly reduced their bulk for a given power. The rate at which work is done per square inch of piston area is equal to the mean piston speed multiplied by the mean effective pressure, and increased pressure of admission implies increased mean pressure throughout the stroke.

[1] *Proc. Inst. Mech. Eng.*, 1872.   [2] *Proc. Inst. Mech. Eng.*, 1881.

[3] *Proc. Inst. Mech. Eng.*, 1891.

[4] On this subject see further a paper by Sir A. J. Durston on the progress of Marine Engineering, read at the International Congress of Naval Architects, 1897. *Engineering*, July 9–16, 1897.

The remarkable efficiency now reached by the marine engine is in part due to its great size; a big engine being, *cæteris paribus,* rather more efficient than a small one. The rapid growth in size is a marked feature of recent progress. Ten thousand indicated horse-power as the performance of a single set of engines is by no means unusual, and several of the vessels engaged in the Transatlantic passenger service, and also a few armoured cruisers, have engines driving twin screws in which the two sets together develop over 30,000 horse-power. These figures are eclipsed in some ships which are driven by turbine engines. The cruisers of the "Invincible" class (1907—9) have turbine engines of 41,000 horse-power, and the turbines of the great Cunarders "Lusitania" and "Mauretania" develop over 70,000 horse-power in propelling them at a speed which exceeds 25 knots. It may be questioned whether so great a concentration of power within a ship would have been practicable had it not been for the possibilities which the introduction of the steam turbine has opened up.

22. **The Steam Turbine.** The introduction of the steam turbine as a practical engine dates from 1884 when the Hon. Charles Parsons took out his first patent for what became known as the Parsons Compound Turbine. For some years it was made in small sizes only and the steam was discharged into the atmosphere without condensation. Under these conditions its efficiency was comparatively low. One of the most important advantages which the steam turbine possesses over the reciprocating engine is its ability to utilize the energy of low-pressure steam by expansion down to the best vacuum obtainable in a condenser, and it was only when this characteristic was turned to account that the steam turbine became a serious rival to the reciprocating engine in respect of economy of steam. In 1891 Parsons adapted his turbine for use with a condenser, and it then began to be used on a fairly large scale as a power generator in electric supply stations. Its efficiency at that date was found, in tests by the present writer, to be comparable with that of a good reciprocating compound engine of the same capacity, but the figures then obtained were much improved on later in turbines of larger size and modified design. Before long it was recognized that the steam turbine, in addition to possessing conspicuous advantages in respect of constructive simplicity, compactness, and freedom from vibration,

was a highly economical generator of power on the largest scale, surpassing in this respect the best steam-engines of the older type. The invention of the steam turbine has consequently effected a revolution in steam-engine practice. The largest demands for power occur in central stations from which electric energy is distributed for traction or other uses, and in fast passenger vessels and war ships. In all these cases the steam turbine now takes the leading place. It is to the genius of Parsons that we owe not only the leading idea of the modern steam turbine, as well as the working out of mechanical details which have been essential to its success, but also its adaptation both to dynamo driving and to the propulsion of ships.

The first application of the turbine to marine propulsion was made by Parsons in the "Turbinia" in 1897, a little experimental vessel of 100 tons which was fitted with turbines of 2,100 horse-power driving three propeller shafts. The "Turbinia" attained what was then a record speed for a ship of any size. This was soon followed by further trials of turbine propulsion in the destroyers "Viper" and "Cobra," and in 1901 the first passenger vessel to be driven by steam turbines—the "King Edward"—was built on the Clyde. Her success led to the adoption of turbines first in various cross-channel packets and later in ocean-going passenger steamers of the largest size. In war ships the use of steam turbines has a special advantage in enabling the machinery to be placed at a low level beneath the protective deck, in addition to the general advantages of reduced bulk and weight, reduced vibration, reduced liability to break-down and reduced consumption of fuel and oil, which also apply to other vessels. The successful trials of the turbine driven cruiser "Amethyst" in 1904 demonstrated the superiority of steam turbines so conclusively that they are now used in all new ships of the British Navy from battleships to torpedo-boats. The tendency is for the turbine to displace reciprocating engines generally in large and fast ships. In slow-going cargo boats it is at a disadvantage on account of the difficulty of getting sufficiently high speed at the turbine blades without making the turbines unduly bulky and the leakage losses unduly large.

In the steam turbine, of whatever form, just as in the water turbine, the force directly operative to do useful work is derived from the kinetic energy of the working fluid, either by the impulse

of a jet or jets impinging on and sliding over moveable blades, or by the reaction on orifices or guides from which the jets issue. The pressure of the steam, instead of being exerted on a piston, is employed in the first instance to set the fluid itself in motion. There is a conversion of pressure-energy into velocity-energy as a preliminary step towards obtaining the effective work of the machine.

When this is done in a single operation the velocity acquired by the steam is immensely greater than the velocities with which water turbines have to deal, in consequence of the much smaller density of steam as a working fluid. Early attempts to design a steam turbine fell short of practical success mainly because of the difficulty of arranging for a sufficiently high velocity in the moving parts to utilize a good proportion of the kinetic energy of the steam. There was the further difficulty of getting the energy of the steam into a suitable kinetic form without excessive waste. The problem of getting the steam to form a jet in which the particles should have a common direction, without undue dispersion, when the steam expands through an orifice from a region of high to a region of low pressure, was solved by Dr Gustaf de Laval, who in 1889 introduced a form of steam turbine in which this was accomplished and in which also the velocity of the moving blades was so high as to make it practicable to recover a fair proportion of the kinetic energy of the jet. The novel features of De Laval's turbine were the form of diverging nozzle which served to produce the jet and the mechanical devices by which an exceptionally high speed was attained in the wheel carrying the vanes or blades on which the jet impinged. His turbine, which will be described in a later chapter, consists essentially of a wheel carrying a single ring of blades on which the steam jet or jets act simply by impulse. It has met with considerable success especially in comparatively small sizes and its efficiency in them is fairly good, but it is not adapted for developing large amounts of power, and it has not been applied to the propulsion of ships.

Parsons, who successfully attacked the problem of designing a steam turbine at an earlier date than De Laval, proceeded in a different way. By dividing the whole range of expansion into many successive steps he prevented the steam from acquiring an inconveniently high velocity at any stage of the process. At each step in the Parsons steam turbine the steam suffers only a small

drop in pressure, and acquires only a moderate velocity; its kinetic energy is therefore easily extracted by means of blades moving at a moderate speed before it passes on to the next step. Moreover in each step the drop in pressure is too small to give rise to any difficulty in the formation of the jets. To form these the steam passes through fixed guide-blades which are distributed round the whole circumference of the revolving wheel and all the revolving blades are consequently in action at once. The steam streams from end to end of the turbine through an annular space between a revolving drum and the casing which surrounds it. Parallel rings of fixed guide-blades project inwards from the casing so as almost to touch the drum, and between these rings alternate rings of moving blades, fixed to the drum, project outwards so as almost to touch the casing. At each step in the expansion the steam streams through a ring of fixed guide-blades, forming a continuous ring of jets which impinge on the adjacent moving blades and give up to them the greater part of the jets' kinetic energy. The same process is repeated many times from ring to ring as the steam progresses from the high-pressure to the low-pressure end, and the size of the annulus containing the blades increases to accommodate the increased volume of the expanded steam. The force on the moving blades is exerted partly by impulse and partly by reaction, for the steam not only impinges on them with the velocity it has acquired in the preceding guide-blades but also is accelerated relatively to the moving blades themselves in the act of passing through them. The construction, which is of great simplicity, will be described later. It lends itself well to the generation of power on a large scale, especially in cases where a fairly high speed of rotation is wanted.

Since the invention of the Parsons turbine other forms have been designed, but only one besides that of De Laval differs widely enough to require mention here as a more or less independent type. This is the Curtis turbine, in which the kinetic energy of jets, formed as in the De Laval turbine, is extracted by their impinging not on a single ring of blades but on a series of rings, of which the alternate rings are fixed guide-blades whose function is simply to change the direction of the steam. The action, which is of the impulse type, will be described more fully later. Like the Parsons turbine the Curtis turbine has been applied on a large scale for the economical generation of electric power.

The steam turbine not only serves as a substitute for the reciprocating engine but may usefully supplement its action by taking steam which has already done duty under a piston, and extracting more work out of it by continued expansion through a range of pressures lower than those with which the piston and cylinder engine can properly deal. An important characteristic of steam turbines is their capacity to utilize the remaining energy of low-pressure steam, after the volume has become so great that further expansion in a cylinder would be impracticable. This has led to the successful use of what are called exhaust steam turbines as auxiliaries to other engines, with the result of adding materially to the output of power without increasing the expenditure of coal.

23. Development of the Theory of Heat-Engines. It is remarkable how little the infancy of the steam-engine has owed to scientific nursing. The early inventors had no theory of thermodynamics to guide them. Watt had the advantage, as he mentions in his narrative, of a knowledge of Black's doctrine of latent heat ; but there was no philosophy of the relation of work to heat until long after the inventions of Watt were complete. The theory of the steam-engine as a heat-engine may be said to date from 1824, when Sadi Carnot published his *Réflexions sur la Puissance Motrice du Feu.* He there showed that heat does work only by being let down from a higher to a lower temperature. But Carnot had no idea that any of the heat disappears in the process, and it was not until the doctrine of the conservation of energy was established in 1843 by the experiments of Joule, which determined the mechanical equivalent of heat, that the theory of heat-engines began a vigorous growth. Important data were furnished by Regnault's experiments on the properties of steam, the results of which were published in 1847. From 1849 onwards the science of thermodynamics was developed with extraordinary rapidity by Clausius, Rankine, and Thomson (Lord Kelvin), and was applied, especially by Rankine, to practical problems in the use of steam. The publication in 1859 of Rankine's *Manual of the Steam-Engine* formed an epoch in the philosophical treatment of the subject. While the thermodynamic theory was rigorous in itself, its direct application to steam-engine problems was limited to a greater degree than Rankine probably realized, owing to its being founded on certain simplifying assumptions

which are by no means fulfilled in real engines. In the ideal engine of the theory it was assumed that the cylinder and piston might be treated as behaving to the steam like non-conducting bodies,—that the transfer of heat between the steam and the metal was negligibly small. Rankine's calculations of steam-consumption, of work, and of thermodynamic efficiency involve this assumption, except in the case of steam-jacketed cylinders, where he estimates that the steam in its passage through the cylinder takes just enough heat from the jacket to prevent a small amount of condensation which would otherwise occur as the process of expansion goes on. If the transfer of heat from steam to metal could be overlooked, the steam which enters the cylinder would remain during admission as dry as it was before it entered, and the volume of steam consumed per stroke would correspond with the volume of the cylinder up to the point of cut-off. It is here that the actual behaviour of steam in the cylinder diverges most widely from the behaviour assumed in the ideal engine. When steam enters the cylinder it finds the metal chilled by the previous exhaust, and a portion of it is at once condensed. This has the effect of increasing, often very largely, the volume of boiler steam required per stroke. As expansion goes on the water that was condensed during admission begins to be re-evaporated from the sides of the cylinder, and this action is generally continued during the escape of the steam. In later chapters the effect which this exchange of heat between the metal of the cylinder and the working fluid produces on the economy of the engine will be discussed, and an account will be given of experimental means by which we may examine the amount of steam that is initially condensed and trace its subsequent re-evaporation. The influence which the walls of the cylinder exert is in fact immense, by the alternate give and take of heat between them and the steam. The exchanges of heat are so complex that there seems little prospect of submitting them to any comprehensive theoretical treatment, and we must rather look for help in the future development of engines to the scientific analysis of experiments made upon actual machines. Many such experiments have been made and their value is now fully realised, by no persons more than by the designers of the best modern engines. Questions relating to the influence on thermal economy of speed, of pressure, of ratio of expansion, of jacketing, of compound expansion, or of

superheating must in the main be settled by an appeal to experiment. The student must not, however, conclude that because the conditions under which an actual engine works are so complex as to make an exact theory of the action impracticable, no theory need be studied. The very complexity of conditions makes the study of theory more necessary, as a guide in judging what conditions are favourable to efficiency and what are unfavourable. Moreover the general theory of heat-engines gives the steam engineer a counsel of perfection, by assigning a limit of efficiency which engines may approach but cannot surpass. To interpret rightly the results of experiments requires a knowledge of the principles of thermodynamics and of the physical properties of steam, and the performance of an ideal engine such as that imagined by Rankine affords a valuable criterion with which to compare the results obtained in trials of actual engines.

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CHAPTER II.

24. Laws of Thermodynamics. The First Law. In the action of a heat-engine, heat is either taken in by the engine from a furnace or from some external source or is generated by the combustion of fuel within the engine itself. A portion of the heat thus supplied is spent in doing mechanical work and so ceases to exist as heat, being converted into another form of energy; and the remainder is rejected by the engine, still in the form of heat. The relation which holds between the heat supplied, the heat converted into mechanical energy, and the heat rejected depends on two general principles which are described as the two Laws of Thermodynamics. The first law states the fact that the amount of heat which disappears in the process (as heat) is proportional to the amount of mechanical work done in the engine; in other words, it states the principle of the Conservation of Energy in relation to the doing of mechanical work by the agency of heat. This may be expressed in the following terms:—*When mechanical energy is produced from heat a definite quantity of heat goes out of existence for every unit of work done; and conversely, when heat is produced by the expenditure of mechanical energy the same definite quantity of heat comes into existence for every unit of work spent.*

To put this statement into a numerical form we must have a unit for the measurement of quantities of heat as well as a unit for the measurement of mechanical work. For engineering purposes the foot-pound is the common unit of work in British and American usage, and the metre-kilogramme (or kilogrammetre) in Continental usage[1]. These convenient and familiar units are open

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[1] Since 1 metre = 3·28085 ft. and 1 kilogramme = 2·20462 lbs., one metre-kilogramme = 7·233 foot-pounds, when both are measured at the same place, so that gravity acts alike on the lb. and the kilogramme.

to the objection that they have slightly different values in different places on account of differences in the intensity of gravity; but these differences are scarcely large enough to be important from a practical point of view. In cases where greater precision of statement is required a particular locality or rather a particular latitude has to be specified, or recourse may be had to absolute units, such as the foot-poundal, the erg or the joule, which are independent of gravity[1].

Quantities of heat are expressed in terms of the *thermal unit*, which is the quantity of heat required to raise the temperature of unit quantity of water by 1 degree. The magnitude of this unit accordingly depends on what unit is used in reckoning the quantity of water, and on what scale the degree is taken by which the rise in temperature is measured. If we take one lb. of water, and a degree of the Fahrenheit scale, we have what is commonly called the British Thermal Unit. It is however of great advantage to avoid using the Fahrenheit scale, and students will do well to make their calculations in terms of centigrade degrees. The Pound-Degree Centigrade is accordingly to be preferred: it is greater than the British Thermal Unit or Pound-Degree Fahrenheit in the proportion of 9 to 5. If we take one gramme of water, and one degree on the Centigrade scale, we have a unit of heat called the gramme-degree or Calory which is in general use for scientific purposes. Engineers often use a unit one thousand times larger than this, namely the heat required to raise one kilogramme of water through one Centigrade degree.

To make the definition of the thermal unit precise we have to specify at what place in the scale of temperature the change through one degree is supposed to occur, for the specific heat of water is not quite constant. As was first shown by Regnault, it takes rather more heat to raise the temperature of a pound of water 1 degree if the temperature is high than if it is low. Later investigations have shown that when water is warmed from the

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[1] The erg is the absolute unit of work on the centimetre gramme second system, namely one centimetre-dyne, the dyne being the c.g.s. unit of force, which is the force required to give one gramme a velocity of one centimetre per second in one second. The joule is ten million (10⁷) ergs. At sea level in latitude 45°, where the acceleration due to gravity is 980·6 cm. per second per second, one kilogrammetre is equal to 9·806 joules. In the latitude of London it is 9·812 joules. For any latitude λ it may be calculated from the formula

$$9\cdot780\ (1+0\cdot0053\sin^2\lambda).$$

temperature of melting ice its specific heat at first decreases slightly as the temperature rises, reaches a minimum at about 35° C. and then increases continuously. In defining the thermal unit a standard temperature of 15° C. (59° Fah.) is usually taken, and this usage will be followed here. It may be added that for the purpose of exact definition it is necessary not only to specify the standard temperature, but also to say whether the 1 degree interval of temperature is to be taken on the scale of the mercury thermometer or on that of the air or other gas thermometer.

Our knowledge of the mechanical equivalent of heat is originally due to the experiments of Joule, which were begun in 1843 and continued for many years. Causing the potential energy of a raised weight to be spent in turning a paddle which generated heat by the agitation of the liquid in which it was immersed, and observing the increase in temperature which this brought about, Joule arrived at the figure 772 as the number of foot-pounds equivalent to one pound-degree Fahrenheit, and this was for long the commonly accepted value of the mechanical equivalent of heat. Later experiments by Joule himself gave a larger number; in 1878 an improved method of measurement, in which the mechanical stirring of water was still used, pointed to a value between 774 and 775. A comparison by Rowland[1] of the scale of the thermometer used by Joule with that of an air thermometer led to a further increase in this number, and very accurate experiments by Rowland himself, carried out by a generally similar method but dealing with larger quantities of work, gave the number 778, the standard temperature being about 60° Fah. and the interval of one degree being taken on the air thermometer. Subsequent determinations by Griffiths[2] and others confirmed the result that Joule's original value was too low and that a number not less than 778 should be accepted. In the experiments of Griffiths the water was heated by a measured expenditure of electrical energy, and this method was also adopted in determinations by Schuster and Gannon[3] and by Callendar and Barnes[4]. An important mechanical determination by Osborne Reynolds and W. M. Moorby[5], of the amount of work spent in raising the

[1] Rowland, *Proceedings of the American Academy*, 1879.
[2] Griffiths, *Phil. Trans.* 1893, vol. 184 A.
[3] Schuster and Gannon, *Phil. Trans.* 1895, vol. 186 A.
[4] Barnes, *Phil. Trans.* 1902, vol. 199 A.
[5] Reynolds and Moorby, *Phil. Trans.* 1898, vol. 190 A.

temperature of water from freezing to boiling point gives very approximately 778 foot-pounds as the mean value of the mechanical equivalent throughout that range : in other words, according to their measurements 180 times 778 is the number of foot-pounds of work required to raise the temperature of 1 lb. of water from 32° to 212° Fah. This mode of defining and measuring the mechanical equivalent has the advantage of escaping all ambiguity in regard to the thermometric scale used in specifying the unit of heat, but to compare it with the definition already given requires a knowledge of the relation of the mean specific heat of water between 0° and 100° C. to the specific heat at 15° C. According to Callendar and Barnes the mean specific heat from 0° to 100° C. is almost exactly equal to the specific heat at 15° C.

Taking the evidence together we may accept 778 foot-pounds as the most probable value of the mechanical equivalent of the British thermal unit or pound-degree Fahrenheit. This corresponds in round numbers to 1400 foot-pounds as the equivalent of the pound-degree Centigrade, and to 427 kilogrammetres as the equivalent of the kilogramme-degree Centigrade, or 427 metre grammes as the equivalent of the gramme-degree or Calory. These numbers will be used in any calculations that occur in this book[1] Since a definite number of mechanical units of work is equivalent to 1 thermal unit, we may, if we please, express

[1] In this statement the interval of one degree is to be understood as taken on the scale of the hydrogen thermometer, at 15° C. The student should refer to a careful summary of results in the article "Calorimetry" by Professor Callendar in the *Encyclopaedia Britannica*, 10th edition. He gives on review of the evidence 4·180 joules as the most probable value of the equivalent when the heat unit is the calory at 20° C., which is smaller than the calory at 15° in the ratio 0·9986 to 1. Hence for the equivalent of the calory at 15° we may take 4·186 joules, and consequently in latitude 45°, where the kilogrammetre is 9·806 joules, the calory is equivalent to $\frac{4\cdot186}{\cdot009806}$ or 426·9 metre grammes. Rowland's results taken by themselves give 4·188 joules as the equivalent of the calory at 15° C., which would correspond to 427·1 metre grammes in latitude 45°. Both these figures agree closely with the round numbers taken in the text. Griffiths' electrical measurements reduce to about 4·193 joules and Schuster and Gannon's to 4·190 when both are referred to 15° C. Reynolds and Moorby's measurements give 4·183 joules as the value of the mean calory between 0° and 100° C. The mean result of Barnes' electrical measurements throughout the same range is very nearly equal to the value found by him at 15° C., namely 4·183 as against 4·184. See Barnes, *loc. cit.*, also Griffiths, *Phil. Mag.* Nov. 1895, and his book on the Thermal Measurement of Energy (Camb. Univ. Press, 1901), where an interesting account is given of various methods by which the mechanical equivalent has been determined.

quantities of work in thermal units, or quantities of heat in foot-pounds or in kilogrammetres.

25.   The Second Law of Thermodynamics. *It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.*

This is the form in which the second law has been stated by Clausius[1].   Another statement of it, different in form but similar in effect, has been given by Lord Kelvin[2].   Its force may not be immediately obvious, but it will be shown below that this law sets a most important limit to the convertibility of heat into work. So far as the first law goes, there is nothing to prevent the whole heat taken in by an engine from changing into mechanical energy.   In consequence of the second law, however, as we shall presently see, no heat-engine converts, or can convert, more than a small fraction of the heat supplied to it into work ; a large part is necessarily rejected as heat.   The ratio

$$\frac{\text{Heat converted into work}}{\text{Heat taken in by the engine}}$$

is a fraction always much less than unity.   This fraction is called the *efficiency* of the engine considered as a heat-engine.

26.   The Working Substance in a Heat-Engine. In every heat-engine there is a *working substance* which alternately takes in and rejects heat.   In general it suffers changes of volume, and does work by overcoming resistance to these changes.   The working substance may be gaseous, liquid, or solid.   We can, for example, imagine a heat-engine in which the working substance is a long metallic rod, arranged to act as the pawl of a ratchet-wheel with closely pitched teeth.   Let the rod be heated so that it elongates sufficiently to drive the wheel forward through the space of one tooth.   Then let the rod be cooled (say by applying cold water), the ratchet-wheel being meanwhile held from returning by a separate click or detent.   The rod, on cooling, will retract so as to engage itself with the next succeeding tooth, which may then be driven forward by heating the rod again, and so on.   To

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[1] See Clausius, *Mechanical Theory of Heat*, translated by W. R. Browne.

[2] See Lord Kelvin's (Sir W. Thomson's) *Collected Papers*, vol. i., for his early investigations in thermodynamic theory.

make it evident that such an engine would do work, we have only to suppose that the ratchet-wheel carries round with it a drum by which a weight is wound up. The device forms a complete heat-engine, in which the working substance is a solid rod, which receives heat by being brought into contact with some source of heat at a comparatively high temperature, transforms a small part of this heat into work, and rejects the remainder to what we may call a receiver of heat, which is kept at a comparatively low temperature. The greater part of the heat may be said simply to pass through the engine, from the source to the receiver, *becoming degraded as regards temperature* in doing so. It will be seen presently that this is typical of the action of all heat-engines; when they are doing work they must take in heat at a comparatively high temperature and reject heat at a comparatively low temperature. They convert some heat into work only by letting down a much larger quantity of heat from a high to a relatively low temperature. The action is, to some extent, analogous to that of a water-wheel, which does work by letting down water from a high level to a lower level, change of level in the one case being the analogue of change of temperature in the other. But there is this important difference, that whereas in the action of the water-wheel none of the water disappears, in the action of the heat-engine an amount of heat disappears which is equivalent to the work done.

27. **Graphic Representation of Work done in the changes of volume of a Fluid**. In almost all actual heat-engines the working substance is a fluid. In some it is air, in some a mixture of several gases. In the steam-engine the working fluid is a mixture (in varying proportions) of water and water-vapour. In an engine of the usual cylinder and piston type work is done by changes of volume only; its amount depends solely on the relation of pressure to volume during the change, and not at all on the form of the vessels in which the change takes place. Let a diagram be drawn (fig. 9) in which the relation of the intensity of pressure to the volume of any supposed working fluid is graphically exhibited by the line ABC, where AM, CN are pressures and AP, CQ are volumes, then the work done by the substance in expanding from volume AP to volume CQ is the area of the figure $MABCN$. And similarly, if the substance be

compressed from volume CQ back to its original volume in such a manner that the line CDA represents the relation of pressure to volume during compression, a quantity of work is done *upon* the substance which is represented by the area $NCDAM$. Taking
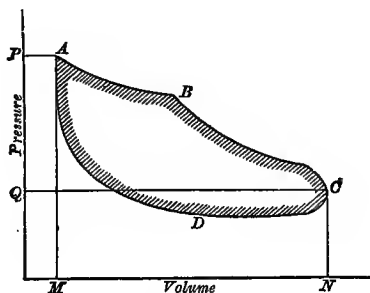
Fig. 9.

the two operations together, we find that the substance has done a net amount of work equal to the area of the shaded figure $ABCDA$, or $\int PdV$. This is an example and a generalization of the method of representing work which Watt introduced by his invention of the indicator; the figure $ABCDA$ may be called the *indicator diagram* of the supposed action.

28. Cycle of operations of the working substance. Modern forms of the indicator will be described in a later chapter. For the present it may suffice to say that the indicator draws automatically a diagram showing the relation of the pressure of the working fluid to the movement of the piston, or in other words to the volume of working fluid in the cylinder, and thus gives complete information as to the work done throughout the stroke. Generally in heat-engines the working substance returns periodically to the same state of temperature, pressure, volume, and physical condition. Each time this has occurred the substance is said to have passed through a complete cycle of operations. For example, in a condensing steam-engine, water taken from the hot-well is pumped into the boiler; it then passes into the cylinder as steam, passes thence into the condenser, and thence again as water into the hot-well; it completes the cycle by returning to the same condition as at first. In other less obvious cases, as in that of the non-condensing steam-engine, a little consideration will show that the cycle is completed, not indeed by the same portion of working

substance being returned to the boiler, but by an equal quantity of water being fed to it, while the steam which has been discharged into the atmosphere cools to the temperature of the feed-water.   In the theory of heat-engines it is of the first importance to consider as a whole the cycle of operations performed by the working substance (as was first done by Carnot in 1824).   If we stop short of the completion of a cycle matters are complicated by the fact that the substance is in a state different from its initial state, and may therefore have changed its stock of internal energy. After the cycle is completed, on the other hand, the internal energy of the substance is necessarily the same as at first, since the condition is in every respect the same.   Hence in regard to the cyclic process as a whole this equation must hold good,

$$\text{Heat taken in} = \text{work done} + \text{heat rejected.}$$

29. Internal Energy.   We have used here a phrase which requires some further explanation—the *internal energy* of a substance.   No means exist by which the whole stock of energy that a substance contains can be measured.   But we are concerned only with changes in that stock, changes which may arise from the substance taking or giving out heat, or doing work, or having work done upon it.   If a substance takes in heat without doing work its stock of internal energy increases by an amount equal to the heat taken in.   If it does work without taking in heat, it does the work at the expense of its stock of internal energy, and the stock is diminished by an amount equal to the work done.   In general, when heat is being taken in and the substance is at the same time doing work, we have

$$\text{Heat taken in} = \text{Work done} + \text{Increase of Internal Energy,}$$

which we may write $$dQ = dW + dE.$$

In a complete cycle there is, at the end, no change of E, and consequently $$Q_1 - Q_2 = W,$$

where $Q_1 - Q_2$ is the net amount of heat received in the cycle as a whole, namely the difference between the heat taken in and the heat rejected in the complete process.

30. Engine using a perfect gas as working substance. It is convenient to approach the theory of heat-engines by

considering, in the first instance, the action of an engine in which the working substance is what is called a perfect gas. Any one of the so-called permanent gases, or a mixture of them, such as air, is very nearly perfect. The word permanent, as applied to a gas, is to be understood only as meaning that the gas is liquefied with difficulty—by the use of extremely low temperature in conjunction, generally, with high pressure. So long as gases are under conditions of pressure and temperature widely different from those which produce liquefaction, they conform very approximately to certain simple laws—laws which may be regarded as *rigorously* applicable to ideal substances called *perfect* gases. After stating these laws we shall examine the efficiency of a heat-engine using a gas in a certain manner as working substance, and then show that the results so derived have a general application to all heat-engines whatsoever. In this procedure there is no sacrifice of generality, and a part of the process is of independent service in the discussion of actual air-engines.

The laws which have now to be stated are very nearly though not absolutely true for air, oxygen, nitrogen, hydrogen and carbonic oxide, except when at specially high pressures or specially low temperatures. Hydrogen probably comes nearest to the ideal of a perfect gas ; but no real gas is in this sense strictly perfect.

31. Laws of the permanent gases. Boyle's law. The laws which are very approximately true of the permanent gases, and may be regarded as strictly true of the ideal perfect gas, are the following :—

LAW 1 (Boyle). *The volume of a given mass of gas varies inversely as the pressure, provided the temperature be kept constant.*

Thus, if V be the volume of a given quantity of any gas, and P the pressure, then so long as the temperature is unchanged—

V varies inversely as P, or $PV = $ constant.

32. Charles's law. LAW 2 (Charles). *Under constant pressure equal volumes of different gases increase equally for the same increment of temperature. Also, if a gas be heated under constant pressure, equal increments of its volume correspond very nearly to equal intervals of temperature as determined by the scale of a mercury thermometer.*

If, for example, we take a vessel containing a quantity of

air and heat it from one temperature to another, taking care to
arrange the experiment so that the air may expand without any
change in its pressure, we shall find that a certain change of
volume takes place.   Let any other permanent gas then be substi-
tuted for the air in the vessel and let the experiment be repeated
by heating this other gas from the same initial to the same
final temperature as before, the pressure being still kept constant.
The volume will be found to have changed by sensibly the same
amount as was observed in the experiment with air.   And further,
if the experiment be varied by using a greater or smaller interval
of temperature, it will be found that the change of volume under-
gone by the air, or by any other gas that may be substituted for
it, is very approximately proportional to the magnitude of the
interval of temperature as measured on the scale of the
ordinary mercury thermometer.   This is equivalent to saying that
if we use a gas thermometer (in which air or any other gas is
allowed to expand without change of pressure) to measure tem-
peratures, defining equal intervals of temperature to be those
which correspond to equal expansions on the part of the gas, we
obtain a thermometric scale which is in substantial though not
perfect agreement with the usual mercurial scale, which defines
equal intervals of temperature to be those that correspond to
equal expansions of mercury in glass.

Thus if a gas be heated under constant pressure from a volume
V_0 at $0°$ C. to a temperature t its volume will become

$$V = V_0 (1 + \alpha t),$$

where α, which is called the coefficient of expansion, measures the
increase of volume per degree, expressed as a fraction of the
volume at $0°$.   Charles's Law states, in effect, that α is nearly the
same for all " permanent " or nearly " perfect " gases, and that its
value in any one of these gases is nearly constant whatever be the
range of temperature, when t is measured on the scale of a mercury
thermometer.

Experiment shows that the amount by which a gas expands
when its temperature is changed by one degree Centigrade, the
pressure being kept constant, is about $\frac{1}{273}$ of its volume at $0°$ C.
Thus if we take 273 cubic inches of air or any other permanent gas
at the temperature $0°$ C. and heat it to $1°$ C. its volume alters to
274 cubic inches.   If we heat it to $2°$ its volume becomes 275

cubic inches and so on. Similarly if the gas be cooled from $0°$ C. to $-1°$ C. its volume changes from the original 273 cubic inches to 272, and so on. In other words, α for a gas is approximately $\frac{1}{273}$.

Putting this in a tabular form, let the volume be

$$273 \text{ at } 0° \text{ C.}$$

It will become $\qquad$ $272 \text{ at } -1° \text{ C.}$

$$\vdots \qquad \vdots$$

and finally would be $\qquad$ $0 \text{ at } -273° \text{ C.,}$

if the same law could be held to apply at indefinitely low temperatures. Any actual gas would change its physical state before so low a temperature were reached, becoming first liquid and then solid, and the volume to which it would contract would consequently be not zero but the volume of the substance in the solid state.

33. Absolute temperature. The above result may be concisely expressed by saying that if temperature be reckoned, not from the ordinary zero but from a point $273°$ below the zero of the Centigrade scale, the volume of a given quantity of a gas, kept at constant pressure, is proportional to the temperature reckoned from that zero. Temperatures so reckoned are called absolute temperatures, and the point $-273°$ C. is called the absolute zero of temperature. Denoting any temperature according to the ordinary scale by t, and the corresponding absolute temperature by τ, we have

$$\tau = t + 273 \text{ on the Centigrade scale,}$$

and $\qquad \tau = t + 459\cdot4 \text{ on the Fahrenheit scale.}$

Charles's law shows that if temperatures be measured by thermometers in which the expanding substance is air, hydrogen, oxygen, or any other permanent gas, those intervals of temperature being called equal which correspond to equal amounts of expansion, then the indications of these thermometers always agree very closely with each other, and also agree, though less closely, with the indications of a mercury thermometer. It will be shown later that the theory of heat-engines affords a means of forming a truly absolute scale of temperature, in the sense that it is a scale which is independent of the properties, as to expansion, of any substance.

It will be further seen that this scale has the same absolute zero as we have arrived at here by considering the properties of the permanent gases, and also makes those intervals equal which are reckoned to be equal on the scale of the gas thermometer.

We are therefore justified in the use of the term absolute, as applied to temperatures measured by the expansion of an ideal gas[1].

34. Connection between Pressure, Volume, and Temperature in a gas.

By Boyle's law we have P varying as $\frac{1}{V}$ where the temperature is kept constant, V being the volume of a given quantity of any gas. By Charles's law we have P varying as τ when V is kept constant, τ being the absolute temperature. Combining the two laws, we have, for a given mass of any gas,

$$PV = R\tau \quad \text{...........................(1)}$$

where R is a constant depending on the specific density of the gas and on the units in which P and V are measured.

We may write

$$R = \frac{P_0 V_0}{273},$$

where V_0 is the volume and P_0 the pressure at 0° C. When the volume is reckoned per unit quantity of the gas we have a definite value of R for each gas, depending on the units employed. Taking

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[1] The position of the absolute zero is discussed in a paper by Professor Callendar (*Phil. Mag.* Jan. 1903), where it is shown that experiments by Regnault on the expansion of gases, as well as later experiments by Chappuis, agree in placing the absolute zero probably within a tenth of a degree of −273° C. This is deduced from observations with hydrogen, air, nitrogen, and carbonic acid, after applying in each case the correction which is required in consequence of the fact that the gas experimented on is not "perfect," the deviation from perfection being inferred from observations of how much the gas is cooled when it is "throttled" or forced through a porous plug or narrow orifice (see § 36, below). In the case of hydrogen the correction is very small, in the case of carbonic acid, which is a much less perfect gas, it is comparatively large. Callendar gives the following figures:

	Hydrogen	Air	Carbonic Acid
Observed coefficient of expansion per degree at 0° C., a	·0036613	·0036706	·003742
Zero inferred directly from coefficient of expansion, or $\frac{1}{a}$	273°·13	272°·44	267°·24
Correction to absolute zero, calculated from throttling experiments..................................	−·13	+·70	+5·8
Absolute zero deduced	273°·00	273°·14	273°·04

for example dry air, observations by Regnault show that one cubic metre at 0° C. and at the standard pressure of 760 mm. of mercury or 10333 kilogrammes per square metre weighs 1·293 kilogrammes. Hence V_0 is $\dfrac{1}{1·293}$ and R is $\dfrac{10333}{1·293 \times 273}$ or 29·27. Or using English units in which the volume is reckoned in cubic feet per lb. and the pressure in pounds per square foot, we have R for dry air $= 96·0$, the temperature being still taken in Centigrade degrees.

It is important to notice that R measures the external work done per degree when unit quantity of the gas is heated under constant pressure and consequently expands. For let the original temperature of the gas be τ_1 and its volume V_1, and let it be heated under constant pressure P till the temperature is τ_2 and the volume is V_2. Then we have $R\tau_1 = PV_1$ and $R\tau_2 = PV_2$, from which $R(\tau_2 - \tau_1) = P(V_2 - V_1)$, which is the external work done by the gas in expanding from V_1 to V_2. Let the interval of temperature be 1°, then R is equal to the external work.

Thus R is numerically expressed in units of work per unit of mass and per degree: in foot-pounds per lb., or in kilogrammetres per kilogramme. If we use the Centigrade degree in both cases, the ratio of the number which expresses R in foot-pounds per lb. to the number which expresses it in kilogrammetres per kilogramme is 3·28085, namely the number of feet in a metre. Values of R for various gases will be given later.

35. The Specific Heat of a gas. LAW 3 (Regnault). *In a gas the specific heat at constant pressure is independent of the pressure.*

By specific heat at constant pressure is meant the heat taken in by unit mass of the substance when its temperature rises 1°, while the pressure remains unchanged—the volume being allowed to change. The law states that this quantity is the same for any one gas, no matter what be the constant pressure at which the process of heating takes place. This, like the other laws, is approximately true for real gases, but is to be regarded as strictly true only of gases that are ideally " perfect."

Another important quantity in the theory of heat-engines is the specific heat at constant volume, that is, the heat taken in by unit mass of the substance when its temperature rises 1°, while the volume remains unchanged—the pressure being free to change.

We shall denote specific heat at constant pressure by K_p and specific heat at constant volume by K_v. These quantities may of course be stated either in heat units or in work units; in what immediately follows we take K_p and K_v in work units. An obvious difference between the heating of a gas at constant pressure and at constant volume is that when heated at constant volume the gas does no work, whereas heating at constant pressure involves expansion of the gas and consequently the gas does an amount of work equal to the product of the pressure and the increase of volume. Let unit mass of a gas be heated at constant pressure P from temperature τ_1 to temperature τ_2 (absolute). Let V_1 be the volume at τ_1 and V_2 the volume at τ_2. Heat is taken in, and external work is done by the expansion of the gas, namely—

Heat taken in $= K_p (\tau_2 - \tau_1)$.
Work done $= P (V_2 - V_1) = R (\tau_2 - \tau_1)$.

The difference between these quantities, or $(K_p - R)(\tau_2 - \tau_1)$, is the amount by which the stock of internal energy possessed by the gas has increased during the process. It will be shown immediately that this gain of internal energy is the same when the gas has its temperature changed in any other manner from τ_1 to τ_2 and is independent of the condition of the gas as to pressure.

According to Regnault's experiments K_p besides being independent of the pressure is nearly constant throughout a considerable range of temperature. But by later investigations it is known to increase slightly as the temperature rises, and at the very high temperatures which occur in gas-engine explosions the difference is important.

In a perfect gas K_v as well as K_p is independent of the pressure. It also increases slightly with the temperature, in such a manner that the difference between K_p and K_v remains constant at all temperatures.

36.   The Internal Energy of a gas.   LAW 4 (Joule).   *The Internal Energy of a given quantity of a gas depends only on the temperature.*

This is an inference from the fact, established by the experiments of Joule, that *When a gas expands without doing external work, and without taking in or giving out heat (and therefore with-*

out changing its stock of internal energy), its temperature does not change.

Joule connected a vessel containing compressed gas with another vessel which was empty, by means of a pipe with a closed stop-cock. Both vessels were immersed in a bath of water and were allowed to assume a uniform temperature. Then the stop-cock was opened, and the gas distributed itself between the two vessels, expanding without doing external work. After this the temperature of the water in the bath was found to have undergone no appreciable change. The temperature of the gas appeared unaltered, and no heat had been taken in or given out by it, and no work had been done by it.

Since the gas had neither gained nor lost heat, and had done no work, its internal energy was the same at the end as at the beginning of the experiment. The pressure and volume had changed, but the temperature had not. The conclusion follows that the internal energy of a given quantity of a gas depends only on its temperature, and not upon its pressure or volume; in other words, a change of pressure and volume not associated with a change of temperature does not alter the internal energy. Hence in any change of temperature the change of internal energy is independent of the relation of pressure to volume during the operation: it depends only on the amount by which the temperature has been changed.
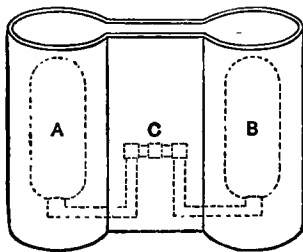
FIG. 10.

The apparatus used by Joule in this experiment is shown in the figure. The vessel *A* was filled with air compressed to more than 20 atmospheres, and *B* was exhausted. The water in the bath was stirred and the temperature noted before the stop-cock *C* was opened. After the gas had come to rest in the two vessels the water was again stirred, and was found to show the same

4—2

temperature as before, so far as tests made by a very sensitive thermometer could detect.

In another form of the apparatus Joule separated the bath into three portions, one portion round each of the vessels and one round the connecting pipe. When the stop-cock was opened the water surrounding A was cooled but this was compensated by a rise of temperature in the water surrounding B and C. The gas in A became colder in the act of expanding, but heat was given up in B and C as its eddying motion settled down, and when all was still there was neither gain nor loss of heat on the whole so far as could be detected in this form of experiment.

It is now, however, known that a very slight change of temperature does in fact take place when a gas expands without doing work. In later experiments by Joule and Thomson (Lord Kelvin) a more delicate method was adopted of detecting whether there is any change of internal energy when the pressure and volume change under conditions such that external work is not done. The gas was forced to pass through a porous plug by maintaining a constant high pressure on one side of the plug and a constant low pressure on the other. Care was taken to prevent any heat being gained or lost by conduction from outside. In this operation work was done upon the gas in forcing it up to the plug, and work was done by it when it passed the plug, by its displacing gas under the lower pressure on the side beyond the plug. If no change of temperature took place, and if the gas conformed to Boyle's Law, these two quantities of work would be exactly equal, and consequently no external work would be done on the whole. For let P_1 be the pressure and V_1 the volume before passing the plug, and P_2 the pressure and V_2 the volume after passing the plug, the volumes being in both cases stated per lb. of the gas. Then the work done upon the gas (per lb.) as it approaches the plug is $P_1 V_1$, and the work done by it as it leaves the plug is $P_2 V_2$. If the temperature is the same on both sides these quantities are equal in a gas for which PV is constant at any one temperature. Thus a 'perfect' gas which conforms both to Boyle's Law and to Joule's would in its passage of the plug have expanded without (on the whole) doing any work, and therefore without changing its internal energy, no heat being gained or lost. In such a gas no change of temperature should accordingly be found, as it passes the plug,

and if a change of temperature is observed it is due to the fact that real gases are not strictly 'perfect.'

In the experiments of Joule and Thomson small changes of temperature were in fact detected and measured in air and other real gases, on passing the porous plug. This Joule-Thomson effect, as it is called, is in general a cooling.

There is in general a small fall of temperature when the gas expands without doing work[1]. In other words, there is an appreciable deviation from Joule's Law, which like the other laws stated here is to be regarded as strictly true only in the case of the ideal or 'perfect' gas. Measurements of the small Joule-Thomson cooling effect supply a means of determining to what extent a gas deviates from the ideally perfect condition, and make it possible to frame an equation to express the relation of the three quantities, pressure, volume, and temperature, more accurately than by the simple form $PV = R\tau$. An equation connecting pressure, volume and temperature in any substance is called the characteristic equation of the substance[2].

To express the quantity of energy which becomes stored up in a gas when its temperature rises, or is extracted from the gas when its temperature falls, we may consider either the case of heating at constant volume, or at constant pressure, since the internal energy depends on the temperature and on nothing else.

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[1] See Lord Kelvin's *Collected Papers*, vol. ɪ., p. 333. The fall of temperature, small as it is, which occurs when a compressed gas is allowed to escape through a constricted orifice forms the basis of Linde's regenerative process of obtaining excessively low temperatures, which he has successfully applied to the liquefaction of air and to the separation of the oxygen of air from the nitrogen. For particulars see the author's *Mechanical Production of Cold* (Camb. Univ. Press, 1908).

In hydrogen at ordinary temperatures the cooling effect is negative, that is to say throttling causes a slight rise of temperature; but at low temperatures hydrogen behaves like other gases in showing a positive cooling effect. The rise of temperature results from a deviation from Boyle's Law which makes the product PV slightly greater on the high-pressure side.

[2] See the paper by Prof. Callendar already referred to (*Phil. Mag.* Jan. 1903). He finds that in any actual gas, in place of the equation $V = \dfrac{R\tau}{P}$ which is characteristic only of an ideally perfect gas, an equation of the form

$$V = \frac{R\tau}{P} + b - c$$

will serve well to express the facts, b being a small constant and c a term varying inversely as a certain power of the temperature. It will be shown in the Appendix how a characteristic equation in this form is applied by him to determine the properties of even so highly imperfect a gas as steam.

In the operation of heating any substance we have seen that

Heat taken in = work done + increase of internal energy.

Take the case of heating at constant volume, and suppose a unit quantity of gas (assumed perfect or sensibly perfect) to be so heated from absolute temperature τ_1 to absolute temperature τ_2. The heat taken in is

$$K_v (\tau_2 - \tau_1)$$

by definition of K_v, the specific heat at constant volume. No external work is done, and hence the whole of this heat goes to increase the stock of internal energy. But in whatever way the temperature be changed from τ_1 to τ_2 the change of internal energy is the same, for, as we have just seen, the internal energy depends only on the temperature. Hence this expression

$$K_v (\tau_2 - \tau_1)$$

measures the change of internal energy which the gas suffers whenever its temperature changes from τ_1 to τ_2 in any manner whatsoever, no matter how the volume and the pressure vary during the process.

37. Relation between the two Specific Heats. We are now in a position to establish a relation between the two specific heats of a gas, K_v and K_p. It was seen by § 33 that when a gas is heated from τ_1 to τ_2 in one particular way, namely, at constant pressure, the change of its internal energy, per unit mass, may be expressed as

$$(K_p - R)(\tau_2 - \tau_1).$$

This expression must agree with the one just found, namely $K_v(\tau_2 - \tau_1)$, and hence

$$K_v = K_p - R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2).$$

The ratio $\dfrac{K_p}{K_v}$ enters into many thermodynamic equations and is usually denoted by the letter γ. Using this symbol the above equation may be written

$$K_v = \frac{R}{\gamma - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3).$$

38. Values of the constants for Air and other gases. In the following tables data, founded chiefly on the experiments of Regnault, are collected for a number of gases. The first table gives the specific densities and volumes and values of R. The second gives the experimentally determined specific heat at con-

stant pressure, K_p, also the specific heat at constant volume, K_v, which is calculated from the equation $K_v = K_p - R$, and also γ as found by dividing K_p by K_v. Independent determinations of γ have been made, by inference from the observed velocity of sound and in other ways, and these agree fairly well with the values calculated from K_p and R. In the first table R is given in work units; in the second it is given in heat units, the numbers being divided by the mechanical equivalent of heat J. The centigrade degree is employed throughout. In the figures for density and volume the temperature is taken as 0° C. and the pressure as one atmosphere (760 mm. of mercury).

Values of the Density, Volume, and R

	Density kg. per cub. met.	Volume		R	
		Cub. metres per kg.	Cub. ft. per lb.	Kilogrammetres per kg.	ft.-pounds per lb.
Dry Air	1·293	0·773	12·39	29·27	96·0
Hydrogen	0·0899	11·126	178·2	421·1	1382
Oxygen	1·429	0·700	11·21	26·49	86·9
Nitrogen	1·251	0·799	12·80	30·25	99·2

Values of the Specific Heats, and γ

	K_p	R	K_v	$\gamma = \dfrac{K_p}{K_v}$
Dry Air	0·2375	0·0686	0·1689	1·406
Hydrogen	3·409	0·9862	2·423	1·407
Oxygen	0·2175	0·6203	0·1555	1·400
Nitrogen	0·2438	0·7085	0·1730	1·409

It should be noticed that the product of K_p into the density is very nearly the same for all the gases, and the same is true of the product of K_v into the density. In other words equal volumes of the various gases require very nearly equal quantities of heat to raise their temperatures equally, whether the heating is effected at constant pressure or at constant volume.

It is now considered probable that these values of the specific heats are too low. Callendar[1] has pointed out a source of error in Regnault's measurements which would have this tendency, and later determinations, made in other ways, agree in giving rather higher figures. In direct measurements of K_v by Joly[2] 0·172 was found for air and 2·402 for hydrogen. A measurement of K_p in air by Mr Swann[3], using electrical heating, gives a result about 2 per cent. greater than Regnault's, namely 0·242. Taking K_p for air to be 0·242, we should have 0·173 for K_v, since R is 0·0686, and this agrees well with Joly's measurement. These figures are for moderate temperatures, up to say 100°C. At high temperatures there is, as already stated, a very considerable increase.

39. Work done by an expanding fluid. We now return to the consideration of imaginary indicator diagrams, which exhibit the relation of the pressure to the volume of a fluid working substance during its expansion or during its compression, in order to study the form which the expansion or compression curve assumes in certain particular cases.

In most of the instances which present themselves in the theory of heat-engines such curves may be exactly or approximately represented by an equation of the form

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

where the index n has various numerical values but is a constant for any one curve. We proceed to find the values which n takes in two very important modes of expansion. Let AB, figure 11, be a curve of expansion, for any fluid, to which the general formula $PV^n =$ constant is applicable. The fluid is supposed to expand from A, where the pressure is P_1 and the volume V_1, to B, where the pressure is P_2 and the volume is V_2.
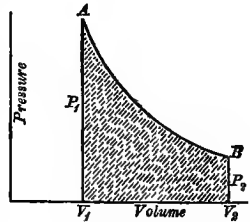
FIG. 11.

During this expansion it does an amount of work which is measured by the shaded area under the curve. That is to say, if W denote the work done during expansion,

$$W = \int_{V_1}^{V_2} P \, dV \quad \dots\dots\dots\dots\dots\dots(4),$$

[1] Callendar, *Appendix to First Report of British Association Committee on Gaseous Explosions*, 1908.

[2] J. Joly, *Proc. Roy. Soc.* vol. 41 (1886), and vol. 47 (1889).

[3] W. F. G. Swann, *Proc. Roy. Soc.* Dec. 1908.

the integral being taken between the limits V_2 and V_1. To integrate we have to remember that the pressure and volume at any point are such that

$$PV^n = P_1V_1{}^n = P_2V_2{}^n.$$

Hence substituting $\dfrac{P_1V_1{}^n}{V^n}$ for P in (4), this expression for the work done becomes

$$W = P_1V_1{}^n \int_{V_1}^{V_2} \frac{dV}{V^n},$$

which gives on integration

$$W = \frac{P_1V_1{}^n(V_2{}^{1-n} - V_1{}^{1-n})}{1-n} \dotfill (5).$$

This may also be written

$$W = \frac{P_1V_1(1 - r^{1-n})}{n-1} \dotfill (6),$$

where r is the *ratio of expansion*, that is to say, the ratio of the final volume V_2 to the initial volume V_1.

Since $P_1V_1{}^n = P_2V_2{}^n$, still another form in which the above result may be expressed is readily derived from equation (5), namely,

$$W = \frac{P_1V_1 - P_2V_2}{n-1} \dotfill (7).$$

If instead of expanding from A to B the fluid were compressed from B to A, the expression given above for W will measure the work spent upon the fluid instead of work done by it.

Further, if the working substance be a gas satisfying the characteristic equation $PV = R\tau$, this last result may be written in the form

$$W = \frac{R(\tau_1 - \tau_2)}{n-1} \dotfill (8),$$

since $P_1V_1 = R\tau_1$ and $P_2V_2 = R\tau_2$.

40. Adiabatic Expansion. We have next to consider particular modes in which any working substance may be expanded or compressed. One very important case is that which occurs when the fluid neither receives nor rejects heat as it expands, or as it is compressed. This mode of expansion or compression is called *adiabatic*, and a curve which exhibits the relation of P to V in such a process is called an adiabatic line. In an adiabatic process the substance is neither gaining nor

losing heat by conduction or radiation or internal chemical action, or by the establishment or subsidence of eddying motions within the substance itself. The subsidence of eddies heats a substance by the conversion of their kinetic energy into heat, just as if heat were communicated to it from outside. From this definition it follows that the work which a substance does when it is expanding adiabatically is all done at the expense of its stock of internal energy, and the work which is spent upon a substance when it is being compressed adiabatically all goes to increase its stock of internal energy. Adiabatic action would be realized if we had a substance expanding, or being compressed, without chemical change and without eddying motions, in a cylinder which (along with the piston) was a perfect non-conductor of heat, and was opaque to heat-rays.

In actual heat-engines the action is never strictly adiabatic on account of the fact that more or less heat passes by conduction between the working fluid and the inner surface of the cylinder. The more quickly the process of expansion or compression is performed the more nearly adiabatic it becomes, for there is then less time for this transfer of heat to take place.

Coming now to the particular case in which the working substance is a gas, since in adiabatic expansion or compression the work done is equal to the change of internal energy we may determine the law of adiabatic action in a gas as follows. Taking expression (8) for the work done, namely,

$$W = \frac{R(\tau_1 - \tau_2)}{n-1},$$

we have to find what value of n in the general formula $PV^n =$ constant will make the process adiabatic. We have seen (§ 36) that in any change of temperature from τ_1 to τ_2 a gas loses internal energy to the amount

$$K_v(\tau_1 - \tau_2),$$

which may be written $\dfrac{R(\tau_1 - \tau_2)}{\gamma - 1}$.

γ being (§ 37) the ratio of the two specific heats.

Hence, equating the work done with the loss of internal energy, the condition of adiabatic expansion is secured when

$$\frac{R(\tau_1 - \tau_2)}{n-1} = \frac{R(\tau_1 - \tau_2)}{\gamma - 1} \quad \ldots\ldots\ldots\ldots\ldots(9),$$

from which $n = \gamma.$

Expansion or compression in a gas will therefore be adiabatic when

$$P V^\gamma = \text{constant} \quad \dots\dots\dots\dots\dots\dots(10),$$

or in other words this is the equation of an adiabatic line for a gas[1].

41. Change of temperature in the adiabatic expansion of a gas. When a gas is expanding adiabatically its stock of internal energy is being reduced, and hence its temperature (to which the internal energy is proportional, by § 36) falls. Conversely, in adiabatic compression the temperature rises. The amount by which the temperature is changed may be found by combining the equations

$$P_1 V_1{}^\gamma = P_2 V_2{}^\gamma \ \text{ and } \ \frac{P_2 V_2}{P_1 V_1} = \frac{\tau_2}{\tau_1}.$$

Multiplying them together we have

$$\frac{\tau_2}{\tau_1} = \frac{P_2 V_2 P_1 V_1{}^\gamma}{P_1 V_1 P_2 V_2{}^\gamma},$$

whence

$$\frac{\tau_2}{\tau_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

or

$$\frac{\tau_2}{\tau_1} = \left(\frac{1}{r}\right)^{\gamma-1} \Bigg\} \quad \dots\dots\dots\dots\dots\dots(11),$$

where r is as before the ratio of expansion. This result of course applies to compression as well as to expansion along an adiabatic line.

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[1] It will be instructive to the student to obtain the same result in another way.
In the general equation

$$dQ = dW + dE,$$

dW is PdV and dE in a gas (by § 36) is $K_v d\tau$. In an adiabatic process $dQ = 0$. Hence in the adiabatic expansion of a gas

$$K_v d\tau + PdV = 0.$$

Substituting $\dfrac{R\tau}{V}$ for P, and dividing by τ,

$$K_v \frac{d\tau}{\tau} + R \frac{dV}{V} = 0,$$

which gives on integration

$$K_v \log_e \tau + R \log_e V = \text{constant}.$$

Writing $K_p - K_v$ for R, and dividing by K_v,

$$\log_e \tau + \gamma \log_e V - \log_e V = \text{constant}.$$

Further, since $\dfrac{PV}{\tau} = \text{constant}$,

$$\log_e P + \log_e V - \log_e \tau = \text{constant}.$$

Adding, we have

$$\log_e P + \gamma \log_e V = \text{constant},$$

or, as in the text, $\qquad P V^\gamma = \text{constant}.$

As to expansions which are not adiabatic, it follows, from the expressions given above for the external work done by an expanding gas and for the change of internal energy, that if n is less than γ the work done is greater than the loss of internal energy—that is to say, the gas is then taking in heat while it expands. On the other hand, if n is greater than γ the work done is less than the loss of internal energy; in other words, the gas is then rejecting heat by conduction to the walls of the containing vessel or in some other manner.

By way of exemplifying an adiabatic process suppose a quantity of dry air to be contained in a cylinder at a temperature of 15° C. ($\tau = 288$) and to be suddenly compressed to half its original volume, the process being so rapid that no appreciable part of the heat developed by compression has time to pass from the air to the cylinder walls. Here $r = \frac{1}{2}$, and taking γ for air to be 1·4 the temperature immediately after compression, before the gas has time to cool, is

$$\tau_2 = \tau_1 \left(\frac{1}{r}\right)^{\gamma-1} = 288 \times 2^{0\cdot4} = 380$$

or 107° C. The work spent in compressing the air, namely,

$$\frac{R\,(\tau_2 - \tau_1)}{\gamma - 1}, \text{ is } \frac{96\cdot0 \times 92}{0\cdot4} = 22080 \text{ foot-pounds}$$

for each lb. of air in the cylinder. The internal energy of the gas becomes increased by this amount; but if the cylinder be a conductor of heat the whole of this will in time become dissipated by conduction to surrounding bodies and the internal energy will gradually return to its original value, as the temperature of the gas sinks to 15° C.

During compression the pressure rises (following the law $PV^\gamma = $ constant), and just at the end its value is greater than at the beginning in the ratio r^γ to 1, that is $2^{1\cdot4}$ or 2·64 to 1. If as before we then suppose the temperature to sink slowly by conduction to 15° C. while the volume does not change, the pressure will fall with the temperature until it reaches a value only twice that which it had before the air was compressed.

42. Isothermal Expansion. Another very important mode of expansion or compression is that called *isothermal*, in which the temperature of the working substance is kept constant during the process.

In the case of a gas the curve of isothermal expansion is a rectangular hyperbola, having the equation

$$PV = \text{constant} = R\tau \quad \ldots\ldots\ldots\ldots\ldots(12).$$

This is a particular case of the general formula $PV^n = \text{constant}$. But equation (6) or (7) above will not serve to find the work done, for when $n = 1$ both the numerator and the denominator in these expressions vanish. To find the work done in the isothermal expansion of a gas we have

$$W = \int_{V_1}^{V_2} P\,dV$$

and

$$P = \frac{P_1 V_1}{V},$$

from which

$$W = P_1 V_1 \int_{V_1}^{V_2} \frac{dV}{V}.$$

Integrating,

$$W = P_1 V_1 (\log_e V_2 - \log_e V_1)$$

or

$$W = P_1 V_1 \log_e \frac{V_2}{V_1} = P_1 V_1 \log_e r \quad \ldots\ldots(13).$$

Instead of $P_1 V_1$ we may write PV, since the product of P and V is constant through the process, and again, since $PV = R\tau$,

$$W = R\tau \log_e r \ldots\ldots\ldots\ldots\ldots\ldots\ldots(14).$$

There is no need here to use a suffix with τ since the temperature does not change. These expressions give either the work done by a gas during isothermal expansion or the work spent upon it during isothermal compression[1].

During isothermal expansion or compression a gas suffers no change of internal energy (by § 36, since τ is constant). Hence during isothermal expansion the gas must take in an amount of heat just equal to the work it does, and during isothermal compression it must reject an amount of heat just equal to the work spent upon it. The expression $R\tau \log_e r$ consequently measures, not only the work done by or upon the gas, but also the heat taken in during isothermal expansion or given out during isothermal
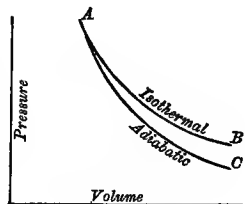
Fig. 12.

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[1] In calculations where this expression is involved it is convenient to remember that $\log_e$, the 'hyperbolic,' or 'natural,' or 'Napierian' logarithm of any number is 2·3026 times the common logarithm of the number.

compression. In the diagram, fig. 12, the line AB is an example of a curve of isothermal expansion for a perfect gas, called for brevity an isothermal line, while AC is an adiabatic line starting from the same point A.

The compression of air or any other gas in a real cylinder is approximately adiabatic when the process is very quickly performed, but approximately isothermal when it is performed so slowly that the heat has time to be dissipated by conduction while the process goes on.

43. Carnot's Cycle of operations. We shall now consider the action of an ideal engine in which the working substance, which we shall in the first place assume to be a perfect gas, is caused to pass through a cycle of changes each of which is either isothermal or adiabatic. The cycle to be described was first examined by Carnot, and is spoken of as Carnot's cycle of
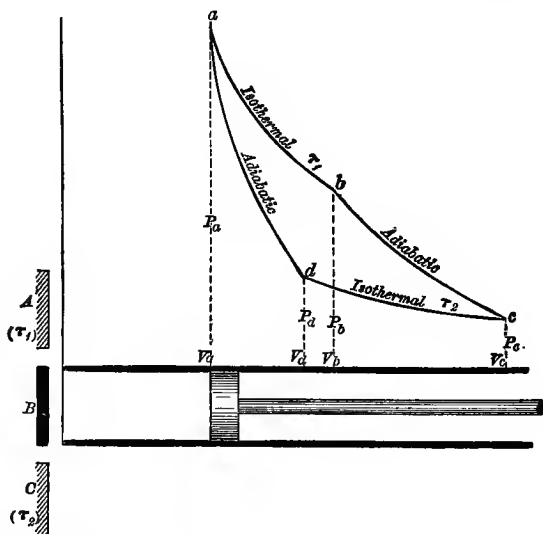
Fig. 13. Carnot's Cycle, with a gas for working substance.

operations. Imagine a cylinder and piston composed of a perfectly non-conducting material, except as regards the bottom of the cylinder, which is a conductor. Imagine also a hot body or indefinitely capacious source of heat A, kept always at a temperature τ_1, also a perfectly non-conducting cover B, and a cold body or indefinitely capacious receiver of heat C, kept always at

some temperature τ_2 which is lower than τ_1. It is supposed that A, B, or C can be applied at will to the bottom of the cylinder. Let the cylinder contain 1 lb. of a perfect gas, at temperature τ_1, volume V_a, and pressure P_a to begin with. The suffixes refer to the points on the indicator diagram, fig. 13.

(1) Apply A, and allow the piston to rise slowly through any convenient distance. The gas expands isothermally at τ_1, taking in heat from the hot source A and doing work. The pressure changes to P_b and the volume to V_b.

(2) Remove A and apply B. Allow the piston to go on rising. The gas expands adiabatically, doing work at the expense of its internal energy, and the temperature falls. Let this go on until the temperature is τ_2. The pressure is then P_c, and the volume V_c.

(3) Remove B and apply C. Force the piston down slowly. The gas is compressed isothermally at τ_2, since the smallest increase of temperature above τ_2 causes heat to pass into C. Work is spent upon the gas, and heat is rejected to the cold receiver C. Let this be continued until a certain point d (fig. 13) is reached, such that the fourth operation will complete the cycle.

(4) Remove C and apply B. Continue the compression, which is now adiabatic. The pressure and temperature rise, and if the point d has been properly chosen, when the pressure is restored to its original value P_a, the temperature will also have risen to its original value τ_1. [In other words, the third operation must be stopped when a point d is reached such that an adiabatic line drawn through d will pass through a.] This completes the cycle.

To find the proper place at which to stop the third operation, we have by equation (11), for the cooling during the adiabatic expansion of stage (2),

$$\frac{\tau_1}{\tau_2} = \left(\frac{V_c}{V_b}\right)^{\gamma-1}$$

and also, for the heating during the adiabatic compression of stage (4),

$$\frac{\tau_1}{\tau_2} = \left(\frac{V_d}{V_a}\right)^{\gamma-1}.$$

Hence
$$\frac{V_c}{V_b} = \frac{V_d}{V_a},$$

and therefore also
$$\frac{V_c}{V_d} = \frac{V_b}{V_a}.$$

That is to say, the ratio of isothermal compression in the third stage of the cycle is to be made equal to the ratio of isothermal expansion in the first stage, in order that an adiabatic line through d shall complete the cycle. For brevity we shall denote either of these last ratios (of isothermal expansion and compression) by r.

The following are the transfers of heat to and from the working gas, in the four successive stages of the cycle:—

(1) Heat taken in from $A = R\tau_1 \log_\epsilon r$ (by § 42).
(2) No heat taken in or rejected.
(3) Heat rejected to $C = R\tau_2 \log_\epsilon r$ (by § 42).
(4) No heat taken in or rejected.

Hence, the net amount of external work done by the gas, being the excess of the heat taken in above the heat rejected in a complete cycle, is

$$R(\tau_1 - \tau_2)\log_\epsilon r \, ;$$

this is the area enclosed by the four curves in fig. 13.

44. Efficiency in Carnot's Cycle. The *efficiency* of the process, namely, the fraction

$$\frac{\text{Heat converted into work}}{\text{Heat taken in}}$$

is $$\frac{R(\tau_1 - \tau_2)\log_\epsilon r}{R\tau_1 \log_\epsilon r} = \frac{\tau_1 - \tau_2}{\tau_1} \quad \ldots\ldots\ldots\ldots(15).$$

This is the fraction of the whole heat given to it which an engine following Carnot's cycle converts into work. The engine takes in an amount of heat, at the temperature of the source, proportional to τ_1; it rejects an amount of heat, at the temperature of the receiver, proportional to τ_2. It works within a range of temperature extending from τ_1 to τ_2, by letting down heat from τ_1 to τ_2 (§ 26), and in the process it converts into work a fraction of that heat, which fraction will be greater the lower the temperature τ_2 at which heat is rejected is below the temperature τ_1 at which heat is received.

45. Carnot's Cycle reversed. Next consider what will happen if we reverse Carnot's cycle, that is to say, if we force this imaginary engine to act so that the same indicator diagram as before is traced out, but in the direction opposite to that followed in § 43. Starting as before from the point a (fig. 13) and with the gas at τ_1, we shall require the following four operations:—

(1)   Apply B and allow the piston to rise.   The gas expands adiabatically, the curve traced is ad, and when d is reached the temperature has fallen to τ_2.

(2)   Remove B and apply C.   Allow the piston to go on rising.   The gas expands isothermally at τ_2, taking heat from C and the curve dc is traced.

(3)   Remove C and apply B.   Compress the gas.   The process is adiabatic.   The curve traced is cb, and when b is reached the temperature has risen to τ_1.

(4)   Remove B and apply A.   Continue the compression, which is now isothermal, at τ_1.   Heat is now rejected to A, and the cycle is completed by the curve ba.

In this process the engine is not doing work; on the contrary, a quantity of work is spent upon it equal to the area of the diagram, or $R\,(\tau_1 - \tau_2)\log_e r$, and this work is converted into heat. Heat is taken in from C in the first operation, to the amount $R\tau_2 \log_e r$.   Heat is rejected to A in the fourth operation, to the amount $R\tau_1 \log_e r$.   In the first and third operations there is no transfer of heat.

The action is now in every respect the reverse of what it was before.   The same work is now spent upon the engine as was formerly done by it.   The same amount of heat is now given to the hot body A as was formerly taken from it.   The same amount of heat is now taken from the cold body C as was formerly given to it.   This will be seen by the following scheme:—

Carnot's Cycle, Direct.

Work done by the gas $= R\,(\tau_1 - \tau_2)\log_e r$;

Heat taken from $A = R\tau_1 \log_e r$;

Heat rejected to $C = R\tau_2 \log_e r$.

Carnot's Cycle, Reversed.

Work spent upon the gas $= R\,(\tau_1 - \tau_2)\log_e r$;

Heat rejected to $A = R\tau_1 \log_e r$;

Heat taken from $C = R\tau_2 \log_e r$.

The heat rejected to the cold body is now equal to the sum of the heat taken in from the hot body and the work spent on the substance. This of course follows from the principle of the conservation of energy. But what is important to observe is that the reversal of the work has been accompanied by an exact reversal of each of the transfers of heat.

46.   Reversible engine. An engine in which this is possible is called, from the thermodynamic point of view, a *reversible* engine. In other words, a reversible heat-engine is one which, if forced to trace out its indicator diagram reversed in direction, so that the work which would be done by the engine when running direct is actually spent upon it, will reject to the source of heat the same quantity of heat as, when running direct, it would take from the source, and will take from the receiver of heat the same quantity as, when running direct, it would reject to the receiver. By "the source of heat" is meant the hot body which acts as source when the engine is running direct, and by "the receiver" is meant the cold body which then acts as receiver. An engine performing Carnot's Cycle of operations is one example of a reversible engine. The idea of thermodynamic reversibility in the sense here defined is of the greatest interest, for the reason that no heat-engine can be more efficient than a reversible engine when both work between the same limits of temperature; that is to say, when both engines take in heat at the same temperature and also reject heat at the same temperature. This theorem, due to Carnot, is of fundamental importance in the theory of heat-engines. It is deduced as follows from the laws of thermodynamics.

47.   Carnot's Principle. To prove that no other heat-engine can be more efficient than a reversible engine when both work between the same limits of temperature, imagine two engines R and S of which R is reversible, and let them work by taking in heat from a hot body A and by rejecting heat to a cold body C. Let Q_A be the quantity of heat which the reversible engine R takes in from A for each unit of work which it does, and let Q_C be the quantity which it rejects to C.

Now consider what consequences would follow if it were possible for S to be more efficient than R. It would take in less

heat from A and reject correspondingly less heat to C, in doing each unit of work. Denote the heat which it would take in from A by $Q_A - q$ and the heat which it would reject to C by $Q_C - q$.

Suppose that S working direct (that is to say, converting heat into work) be set to drive R as a reversed engine, so that R converts work into heat. For every unit of work done by the
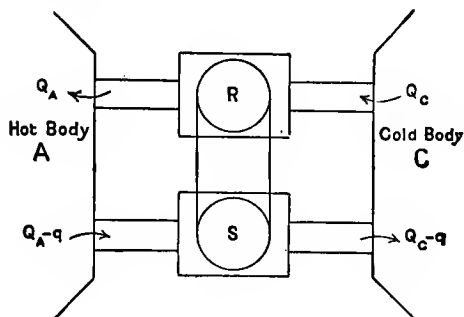
Fig. 14.

engine S on the reversible engine R the quantity $Q_A - q$ would be taken from A by the engine S, and the quantity Q_A would be restored to A by the reversed action of the engine R. This is because R being reversible restores to A when working reversed the same amount of heat as it would take from A when working direct. Hence the hot body would on the whole gain heat, by the amount q for every unit of work done by the one engine on the other. Again, S gives to C a quantity $Q_C - q$ while R takes from C a quantity Q_C and hence the cold body C would lose an amount of heat equal to q for every unit of work done by the one engine or the other. Thus the combined action of the two engines—one working direct, as a true heat-engine, and the other reversed, as what we might call a heat-pump—would result in a transfer of heat from the cold body C to the hot body A, and this process might evidently go on without limit. Moreover the two engines taken together form a purely self-acting system, for the whole power generated in one is spent on the other and is sufficient to drive the other; if we assume that there is no mechanical friction the double machine requires no help from without. Hence the supposition that the engine S could be more efficient than the reversible engine R has led to a result inconsistent with the second law of thermodynamics, for it has led us to construct, in

imagination, a self-acting machine capable of transferring heat, in any quantity, from a cold body to a hot body. The second law asserts that this is contrary to all experience, and we are therefore forced to the conclusion that no other engine S can be more efficient than a reversible engine R when both work between the same limits of temperature. In other words, when the source and receiver of heat are given, a reversible heat-engine is as efficient as any engine working between them can be.

Further, let both engines be reversible. Then the same argument shows that neither can be more efficient than the other. Hence all reversible heat-engines taking in and rejecting heat at the same two temperatures are equally efficient.

48. Reversibility the criterion of perfection in a heat-engine. These results imply that reversibility, in the thermodynamic sense, is the criterion of what may be called perfection in a heat-engine. A reversible engine is perfect in the sense that it cannot be improved on as regards efficiency: no other engine, taking in and rejecting heat at the same temperatures, will convert into work a greater fraction of the heat which it takes in. Moreover, if this criterion be satisfied, it is as regards efficiency a matter of complete indifference what is the nature of the working substance, or what, in other respects, is the mode of the engine's action.

49. Efficiency of a perfect heat-engine. Further, since all engines that are reversible are equally efficient, provided they work between the same temperatures, an expression for the efficiency of one will apply equally to all. Now, the engine whose efficiency was found in § 44, namely, an engine having a gas for working substance and performing Carnot's Cycle of operations, is one example of a reversible engine. Hence the expression which was obtained for its efficiency, namely,

$$\frac{\tau_1 - \tau_2}{\tau_1},$$

is the efficiency of any reversible heat-engine whatsoever taking in heat at τ_1 and rejecting heat at τ_2. And, as no engine can be more efficient than one that is reversible, this expression is the measure of *perfect efficiency*. We have thus arrived at the

immensely important conclusion that no heat-engine can convert into work a greater fraction of the heat which it receives than is expressed by the excess of the temperature of reception above that of rejection divided by the absolute temperature of reception.

50. Summary of the argument. Briefly recapitulated, the steps of the argument by which this result has been reached are as follows. After stating the experimental laws to which gases conform, and finding that they afforded a provisional means of defining temperature upon an absolute scale, we examined the action of a heat-engine in which the working substance took in heat when at the temperature of the source and rejected heat when at the temperature of the receiver, the change of temperature from one to the other of these limits being accomplished by adiabatic expansion and adiabatic compression. Taking a special case in which the engine had for its working substance a perfect gas, we found that its efficiency was $(\tau_1 - \tau_2)/\tau_1$ (§ 44). We also observed that it was, in the thermodynamic sense, a reversible engine (§ 46). Then we found, by an application of the second law of thermodynamics, that no heat-engine can have a higher efficiency than a reversible engine, when taking in and giving out heat at the same two temperatures τ_1 and τ_2; this was shown by the fact that a contrary assumption would lead to a violation of the second law (§ 47). Hence, we concluded that all reversible heat-engines receiving and rejecting heat at the same temperatures, τ_1 and τ_2 respectively, are equally efficient, and hence that the efficiency

$$\frac{\tau_1 - \tau_2}{\tau_1},$$

already determined for one particular reversible engine, is the efficiency of any reversible engine, and is a limit of efficiency which no engine whatever can exceed.

Another way of stating the performance of a perfect engine evidently is to say that the heat taken in Q_1 is to the heat rejected Q_2 as τ_1 is to τ_2, or

$$\frac{Q_1}{\tau_1} = \frac{Q_2}{\tau_2} \quad \dots\dots\dots\dots\dots\dots\dots(16).$$

The efficiency of any heat-engine may be stated as

$$\frac{Q_1 - Q_2}{Q_1} \text{ or } 1 - \frac{Q_2}{Q_1}:$$

in the perfect engine this becomes

$$1 - \frac{\tau_2}{\tau_1}.$$

51. Conditions of maximum efficiency. The availability of heat for transformation into work depends essentially on the range of temperature through which the heat is let down from that of the hot source to that of the cold body into which heat is rejected; it is only in virtue of a difference of temperature between bodies that conversion of any part of their heat into work becomes possible. No mechanical effect could be produced from heat, however great the amount of heat present, if all bodies were at a dead level of temperature. Again, it is impossible to convert the whole of any supply of heat into work because it is impossible to have a body at the absolute zero of temperature as the sink into which heat is rejected.

If τ_1 and τ_2 are given as the highest and lowest temperatures of the range through which a heat-engine is to work, it is clear that the maximum of efficiency can be reached only when the engine takes in all its heat at τ_1 and rejects at τ_2 all that is rejected. With respect to every portion of heat taken in and rejected the greatest ideal efficiency is

$$\frac{\text{Temperature of reception} - \text{temperature of rejection}}{\text{Temperature of reception}}.$$

Any heat taken in at a temperature below τ_1 or rejected at a temperature above τ_2 will have less availability for conversion into work than if it had been taken in at τ_1 and rejected at τ_2, and hence, with a given pair of limiting temperatures, it is essential to maximum efficiency that no heat be taken in by the engine except at the top of the range, and no heat rejected except at the bottom of the range. Further, as we have seen in § 47, when the temperatures at which heat is received and rejected are assigned, an engine attains the maximum of efficiency if it be reversible.

52. Conditions of reversibility. It is therefore important to inquire more particularly what kinds of action are reversible in the thermodynamic sense. A little consideration will show that a transfer of heat from the source to the working substance, or from the working substance to the receiver, is reversible only when the working substance is at sensibly the same temperature as the

source or the receiver, as the case may be, and an expansion is reversible only when it occurs by the gradual displacement of some part of the containing envelope in such a manner that the expanding fluid does external work on the envelope, and does not waste energy to any sensible extent in setting itself in motion. This excludes what may be termed free expansion, such as that of the gas in Joule's experiment, § 36, and it excludes also what may be called imperfectly-resisted expansion, such as would occur if the fluid were allowed to expand into a chamber in which the pressure was less than that of the fluid, or if the fluid were expanding in a cylinder under a piston which rose so fast as to cause, through the inertia of the expanding fluid, local variations of pressure throughout the cylinder.   A similar condition of course applies in regard to the compression of the working fluid : neither expansion nor compression must take place in such a manner as to set up eddies within the fluid.

To make a heat-engine, working within given limits of temperature, as efficient as possible the conditions to aim at therefore are—(1) to take in no heat except at the highest temperature, and to reject no heat except at the lowest temperature ; (2) to secure that the working substance shall, when receiving heat, be at the temperature of the body from which the heat comes, and that it shall, when giving up heat, be at the temperature of the body to which heat is given up ; (3) to avoid free or imperfectly-resisted expansion.   If these conditions are fulfilled the engine works reversibly and is the most efficient heat-engine possible within the given range of temperatures.

The first and second of these conditions are satisfied if in the action of the engine the working substance changes its temperature from τ_1 to τ_2 by adiabatic expansion, and from τ_2 to τ_1 by adiabatic compression, thereby being enabled to take in and reject heat at the ends of the range without taking in or rejecting any by the way.   This is the action in Carnot's ideal engine (§ 43).

53. Perfect Engine using Regenerator. But there is another way in which the action of a heat-engine may be made reversible.   Suppose that the working substance can be caused to deposit heat in some body within the engine while passing from τ_1 to τ_2, in such a manner that the transfer of heat from the substance to this body is reversible (satisfying the second con-

dition above), then when we wish the working substance to pass from τ_2 to τ_1 we may reverse this transfer and so recover the heat that was deposited in this body. This alternate storing and restoring of heat would serve, instead of adiabatic expansion and compression, to make the temperature of the working substance pass from τ_1 to τ_2 and from τ_2 to τ_1 respectively. The alternate storing and restoring is an action occurring wholly within the engine, and is therefore distinct from the taking in and rejecting of heat by the engine.

In 1827 Robert Stirling designed an apparatus, called a *regenerator*, by which this process of alternate storing and restoring of heat could be actually performed. For the present purpose it will suffice to describe the regenerator as a passage (such as a group of tubes) through which the working fluid can travel in either direction, whose walls have a very large capacity for heat, so that the amount alternately given to or taken from them by the working fluid causes no more than an insensible rise or fall in their temperature. The temperature of the walls at one end of the passage is τ_1, and this falls continuously down to τ_2 at the other end. When the working fluid at temperature τ_1 enters the hot end and passes through, it comes out at the cold end at temperature τ_2, having stored in the walls of the regenerator a quantity of heat which it will pick up again when passing through in the opposite direction. During the return journey of the working fluid through the regenerator from the cold to the hot end its temperature rises from τ_2 to τ_1 by picking up the heat which was deposited when the working fluid passed through from the hot end to the cold. The process is strictly reversible, or rather would be so if the regenerator had an unlimited capacity for heat, if no conduction of heat took place along its walls from the hot to the cold end, and if no loss took place by conduction or radiation from its external surface. A regenerator satisfying these conditions is of course an ideal impossible to realise in practice.

54. Stirling's Regenerative Air-Engine. Using air as the working substance, and employing his regenerator, Stirling made an engine (to be described later) which, allowing for practical imperfections, is the earliest example of a reversible engine. The cycle of operations in Stirling's engine was substantially this:

(1)   Air (which had been heated to τ_1 by passing through the regenerator) was allowed to expand isothermally through a ratio r, taking in heat from a furnace and raising a piston.   Heat taken in (per lb. of air) $= R\tau_1 \log_e r$.

(2)   The air was caused to pass through the regenerator from the hot to the cold end, depositing heat and having its temperature lowered to τ_2, without change of volume.   Heat stored in regenerator $= K_v(\tau_1 - \tau_2)$.   The pressure of course fell in proportion to the fall in temperature.

(3)   The air was then compressed isothermally to its original volume at τ_2 in contact with a cooler (or receiver of heat).   Heat rejected $= R\tau_2 \log_e r$.

(4)   The air was again passed through the regenerator from the cold to the hot end, taking up heat and having its temperature raised to τ_1.   Heat restored by the regenerator $= K_v(\tau_1 - \tau_2)$.   This completed the cycle.

The efficiency is

$$\frac{R\tau_1 \log_e r - R\tau_2 \log_e r}{R\tau_1 \log_e r} = \frac{\tau_1 - \tau_2}{\tau_1}.$$

The indicator diagram of this action is shown in fig. 15.   Stirling's engine is important, not as a present-day heat-engine (though it has been revived in small forms after a long interval of disuse), but because it is typical of the only mode, other than Carnot's plan of adiabatic expansion and compression, by which the action of a heat-engine can be made reversible.
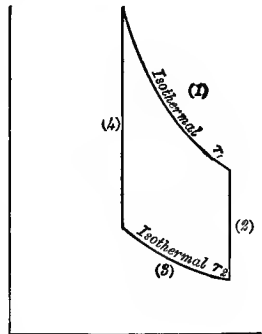
FIG. 15.   Ideal Indicator diagram of Air-Engine with Regenerator (Stirling).

The regenerative principle has been largely used in metallurgy and other industrial processes as a means of economising fuel and attaining high temperatures : the Siemens steel-furnace is an example of its application on a large scale.   It has also supplied a means of reaching the lowest extremes of temperature, as in Linde's method of liquefying air.   Notwithstanding the immensely valuable services which the regenerator has rendered in such processes, its application to heat-engines has hitherto been very limited. Another way of using it in air-engines was tried by Ericsson, who kept the pressure instead of the volume constant while the working

air was passed through the regenerator, thus getting an indicator diagram made up of two isothermal lines and two lines of constant pressure.   Attempts have also been made by Siemens and by Fleeming Jenkin to apply it to steam-engines and to gas-engines. It finds application in some refrigerating machines, or reversed heat-engines, to which reference will be made in a later chapter. But almost all actual heat-engines, in so far as they can be said to approach the condition of reversibility, do so, not by the use of the regenerative principle, but by more or less nearly adiabatic expansion and compression after the manner of Carnot's ideal engine.

CHAPTER III.

PROPERTIES OF STEAM AND ELEMENTARY THEORY OF THE STEAM-ENGINE.

55. Formation of steam under constant pressure. We have now to consider the action of heat-engines in which the working substance is water and water-vapour or steam, and as a preliminary to this it is necessary to give some account of the physical properties of steam as determined by experiment. The properties of steam are most conveniently stated by referring in the first instance to what happens when steam is formed *under constant pressure*. This is substantially the process which occurs in the boiler of a steam-engine when the engine is at work. To fix the ideas we may suppose that the vessel in which steam is to be formed is a long upright cylinder fitted with a frictionless piston which may be loaded so that it exerts a constant pressure on the fluid below. Let there be, to begin with, at the foot of the cylinder a quantity of water (which for convenience of statement we shall take as one unit of mass, namely 1 lb. or 1 kg.), at any temperature t_0; and let the piston rest on the surface of the water with a pressure P (lbs. per square foot or kg. per square metre). Let heat now be applied to the bottom of the cylinder. As it enters the water it will produce the following effects in three stages:—

(1) The temperature of the water rises until a certain temperature t is reached, at which steam begins to be formed. The value of t depends on the particular pressure P which the piston

exerts. Until the temperature t is reached there is nothing but water below the piston.

(2) Steam is formed, more heat being taken in. The piston, which is supposed to continue to exert the same constant pressure, rises. No further increase of temperature occurs during this stage, which continues until all the water is converted into steam. During this stage the steam which is formed is said to be *saturated*. The volume which the piston encloses at the end of this stage,—the volume, namely, of unit mass of saturated steam at pressure P and consequently at temperature t,—will be denoted by V.

(3) If after all the water has been converted into steam more heat be allowed to enter, the volume will increase and the temperature will rise. The steam is then said to be *superheated*: its temperature is above the temperature of saturation.

56. Saturated and superheated steam. The difference between saturated and superheated steam may be expressed by saying that if water (at the temperature of the steam) be mixed with steam some of the water will be evaporated if the steam is superheated, but none if the steam is saturated. Any vapour in contact with its liquid and in thermal equilibrium is necessarily saturated. When saturated its properties differ considerably, as a rule, from those of a perfect gas, but when superheated they approach those of a perfect gas more and more closely the farther the process of superheating is carried, that is to say, the more the temperature is raised above t, the temperature of saturation corresponding to the given pressure P. Saturated steam at a given pressure can have but one temperature; superheated steam at the same pressure can have any temperature higher than that.

57. Relation of pressure and temperature in saturated steam. The temperature t at which steam is formed depends on the value of P. The relation of pressure to temperature was determined with great care by Regnault, in a series of classical experiments to which a great part of our knowledge of the properties of steam is due[1]. Regnault's observations extended from

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[1] *Mem. Inst. France*, 1847, vol. xxi. An account of Regnault's methods of experiment and a statement of his results expressed in British measures will be found in Dixon's *Treatise on Heat* (Dublin, 1849).

temperatures below the zero of the centigrade scale, where the
vapour whose pressure was measured was that given off by ice,
up to 220° C.   The pressures found by him, expressed in milli-
metres of mercury, were as follows, omitting those below 0° C. as
not relevant to steam-engine calculations :—

Temperature °C.	Pressure of saturated steam in mm. of Mercury
0	4·60
25	23·55
40	54·91
50	91·98
75	288·50
100	760·00
130	2030·0
160	4651·6
190	9426
220	17390

It will be seen from these figures that the pressure of saturated
steam rises with the temperature at a rate which increases rapidly
in the upper regions of the scale.   Various empirical formulas have
been devised to express the relation of pressure to temperature
in saturated steam and to allow tables to be calculated in which
intermediate values are shown.   When a table is available, how-
ever, it is more convenient to find the pressure corresponding to
a given temperature, or the temperature corresponding to a given
pressure directly from it, either interpolating or drawing a portion
of the curve connecting P with t when the values concerned lie
between those stated in the table.   In the table given here
the pressures are stated both in English and metric units for
temperatures rising by steps of 5° C.   The values in this table
and in the tables at the end of this book, which deal with other
properties of steam, are those calculated by Mollier[1], whose work
is founded on that of Callendar[2], in a manner which will be
explained in the Appendix.

In the tables printed in the Appendix the relation between
the pressure, temperature, and volume of saturated steam is more
fully stated, numbers being given for a series of pressures as well
as for a series of temperatures, along with values of other pro-
perties which are required in steam-engine calculations.

[1] Dr R. Mollier, *Neue Tabellen und Diagramme für Wasserdampf.*, Berlin, 1906.

[2] H. L. Callendar, *Proc. Roy. Soc.* vol. 67, p. 266, 1900.

Properties of Saturated Steam.

Relation of Temperature, Pressure, and Volume.

Temperature		Pressure		Volume	
Cent.	Fah.	Kilogrammes per sq. cm.	Pounds per sq. inch	Cubic metres per kilogramme	Cubic feet per lb.
0	32	0·0063	0·089	205·0	3283
5	41	0·0089	0·127	146·9	2354
10	50	0·0125	0·178	106·6	1708
15	59	0·0173	0·246	78·2	1253
20	68	0·0236	0·336	58·1	931
25	77	0·0320	0·455	43·67	699·5
30	86	0·0429	0·610	33·13	530·7
35	95	0·0569	0·809	25·39	406·8
40	104	0·0747	1·062	19·65	314·8
45	113	0·0971	1·381	15·35	245·8
50	122	0·125	1·78	12·09	193·7
55	131	0·160	2·28	9·61	153·9
60	140	0·202	2·88	7·69	123·3
65	149	0·254	3·61	6·21	99·5
70	158	0·317	4·51	5·05	80·9
75	167	0·392	5·58	4·135	66·24
80	176	0·482	6·86	3·408	54·60
85	185	0·589	8·38	2·827	45·29
90	194	0·714	10·16	2·359	37·79
95	203	0·862	12·26	1·980	31·71
100	212	1·033	14·70	1·670	26·75
105	221	1·232	17·52	1·417	22·69
110	230	1·462	20·79	1·207	19·34
115	239	1·726	24·55	1·034	16·56
120	248	2·027	28·83	0·889	14·25
125	257	2·371	33·72	0·768	12·30
130	266	2·760	39·26	0·666	10·67
135	275	3·200	45·51	0·580	9·29
140	284	3·695	52·56	0·507	8·12
145	293	4·248	60·42	0·445	7·13
150	302	4·868	69·24	0·3917	6·274
155	311	5·557	79·04	0·3460	5·542
160	320	6·323	89·93	0·3065	4·910
165	329	7·170	101·98	0·2724	4·363
170	338	8·104	115·27	0·2429	3·891
175	347	9·13	129·9	0·2171	3·478
180	356	10·26	145·9	0·1945	3·116
185	365	11·49	163·4	0·1748	2·800
190	374	12·83	182·6	0·1575	2·523
195	383	14·30	203·4	0·1423	2·279
200	392	15·89	226·0	0·1288	2·063
205	401	17·61	250·5	0·1170	1·874
210	410	19·49	277·2	0·1063	1·703
215	419	21·57	306·8	0·0965	1·546

The figures which are given for the pressure of saturated steam are not taken directly from the measurements of Regnault but are inferred from a characteristic equation due to Callendar, the validity of which is demonstrated by the general agreement of the quantities calculated from it with the best experimental results in measurements not only of the pressure but of other properties of steam. The pressures, however, which are stated in the table do agree very closely with the results of Regnault's observations quoted above. It is only at the highest pressures that an appreciable difference will be found, and even there it is not material.

In other respects the tables given here will be found to differ somewhat widely from the earlier tables of such authorities as Rankine[1] or Zeuner[2], which have been accepted as standards and copied into many text-books. When these were calculated the only available data of value were those furnished by the experiments of Regnault. But more recent researches have supplied additional data which in some particulars modify his, and it is now clear that Regnault's figures require revision and in some cases considerable amendment. The various properties of steam are linked together in such a manner that the relations between them can be, and have been, made the subject of thermodynamic analysis. In the light of such investigations the figures given in the old tables are now known to be not even mutually consistent. The Tables of Mollier which are adopted here give a set of values which are consistent amongst themselves and are also in good agreement with the most trustworthy experimental results. Further researches may in time lead to a still closer adjustment of the figures to the results of observation, but the Tables of Mollier may be accepted not only as a notable advance towards accuracy but as certainly correct enough for the purposes of the engineer. The method by which the new values are calculated is explained in the Appendix, where tables of the principal properties of steam will be found.

58. Relation of pressure and volume in saturated steam. The table also shows the volume V, in cubic metres per kilogramme and in cubic feet per lb., of saturated steam at each pressure. The volume of a given quantity of saturated steam

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[1] Rankine, *A Manual of the Steam Engine and other Prime Movers.*

[2] Zeuner, *Technische Thermodynamik*, vol. II.

at any assigned pressure is a quantity difficult to measure by direct experiment, and the volumes which are given in steam tables are generally inferred from the results of experiments on other properties of steam which can be more easily measured. Successful measurements of volume have however been carried out in the laboratory of the Technical High School at Munich[1] and the results are in general agreement with the figures stated in the table. Reference must again be made to the Appendix as to the manner in which the tabulated figures are obtained.

The relation of P to V in saturated steam may be approximately expressed by the formula[2]

$$PV^{\frac{16}{15}} = \text{constant}\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1).$$

When the pressure is stated in kilogrammes per square centimetre and the volume in cubic metres per kilogramme, this may be written

$$PV^{\frac{16}{15}} = 1\cdot786.$$

With P in lbs. per sq. inch and V in cubic feet per lb. it becomes

$$PV^{\frac{16}{15}} = 490.$$

The student will find it useful to draw curves, with the data of the table, showing the relation between the pressure and the temperature of saturated steam and also the relation of pressure to volume, especially within the range usual in steam-engine practice. He will observe that $\dfrac{dP}{dt}$, the rate of change of pressure with respect to change of temperature, increases rapidly as the temperature rises, and hence that in the upper part of the range a very small elevation of temperature in a boiler is necessarily associated with a large increment of pressure. The familiar case of water boiling in a kettle or other open vessel is only a special case of the formation of steam under constant pressure. There the constant pressure is that of the atmosphere, which is 14·7 lbs. per square inch or thereabouts (as indicated by the barometer) and consequently the temperature at which the water boils is about 100° C.[3]

[1] O. Knoblauch, R. Linde and H. Klebe, *Mitteilungen über Forschungsarbeiten herausgegeben vom Verein deutscher Ingenieure*, Heft 21, 1905.

[2] Mollier, *loc. cit.* The index was $\frac{17}{16}$ with the values of V given in old tables.

[3] Water in the open boils at 100° C. when the atmospheric pressure has its standard value, which corresponds to a barometer reading in the latitude of Paris of 760 mm. or 29·921 inches of mercury. This pressure is 1·0333 kg. per

59. Supply of heat in the formation of steam under constant pressure. We have next to consider the supply of heat in the imaginary experiment of § 55 in which unit mass of water initially at some temperature t_0 is first heated to the boiling point and then converted into steam, under a constant pressure P, this constant pressure determining what the temperature of the boiling point shall be. During the first stage, while the temperature is rising from its initial value t_0 to t, no steam is formed, and heat is required only to warm the water. Since the specific heat of water is nearly constant and equal to unity, the amount of heat taken in during the first stage is approximately $t - t_0$ thermal units or $J(t - t_0)$ mechanical units, and except at high temperatures this expression will generally serve with sufficient accuracy in practical calculations. More exactly, however, the heat taken in is in general somewhat greater than this, for Regnault's experiments show that the specific heat of water increases at high temperatures. In stating the amount of heat required for this first stage, t_0 must be taken as a known temperature; for convenience in numerical statement the temperature $0°$ C. is chosen as an arbitrary starting-point from which the reception of heat is to be reckoned. We shall employ the symbol h to designate the heat required to raise unit mass of water from $0°$ C. to the temperature t at which steam begins to form. The value of h in thermal units is given, approximately, by the formula

$$h = t \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(2),$$

which will serve well for low or moderate temperatures. At higher temperatures such as are found in steam boilers more exact values will be obtained by using Regnault's empirical formula

$$h = t + 0\cdot00002t^2 + 0\cdot0000003t^3 \dots\dots\dots\dots\dots(3).$$

The values stated in the Tables (see Appendix) involve the use of this formula[1]. During this first stage, while all the substance still

sq. cm., or 14·697 lbs. per square inch. 1 inch of mercury is equivalent to 0·4912 lb. per sq. inch, and 1 mm. of mercury to 0·0013596 kg. per sq. cm.

[1] The measurements on which Regnault founded this formula were made at temperatures ranging from 110° C. to 192° C. It has no proper application to lower temperatures, and we know from the work of Barnes and others that between 0° C. and 100° C. the specific heat of water varies in a manner which makes a formula of this type inapplicable. It gives values of h in the neighbourhood of 100° C. which are perhaps too high by nearly half a unit, but there is some uncertainty on the point and the differences are in any case not of practical importance.

E.   6

is water, sensibly all the heat that is supplied goes to increase the stock of internal energy which the fluid possesses, for the amount of external work done through the expansion of the water is negligibly small.

60. Latent Heat of Steam. During the second stage water at temperature t is changing into steam at temperature t. Much heat is required to produce this change in physical state, although the temperature of the substance does not alter. The heat taken in during this process is called the *latent heat* of steam: in other words, the latent heat of steam is defined as the amount of heat which is absorbed by unit mass of water while it changes into steam under constant pressure, the water having been previously heated up to the temperature at which steam is formed. We shall denote the latent heat by L. The value of L depends on the particular pressure at which the change takes place, the latent heat of steam being less at high pressures than at low pressures. Values of L are given in the tables.

Part of the heat taken in during this second stage is spent in doing external work, since the piston rises against the constant pressure P. It is only the remainder of the so-called latent heat L that may be said to remain in the fluid and to constitute an increase in its stock of internal energy. The amount spent in doing external work is equal to P multiplied by the change of volume which takes place as the water is converted into steam.

Let w denote the volume of unit mass of water in the original state. At the end of the process when all the water has turned into steam the volume has changed from w to V. The external work done during the formation of steam under constant pressure P is therefore

$$\text{External work} = P\,(V - w)\dots\dots\dots\dots\dots(4).$$

This is the measure of the external work in mechanical units. It may of course be expressed in thermal units by dividing by J. It is convenient to use the symbol A for $\dfrac{1}{J}$. We may accordingly write the external work, measured in thermal units, as

$$A P\,(V - w).$$

61. Total heat of steam. Adding together the heat taken in during the first and second stages of the imaginary experiment we have a quantity which we shall call the *Heat of Formation*, namely

$$H = h + L \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots(5).$$

This is the amount of heat required to raise unit mass of water from the standard temperature (0° C.) to the temperature of evaporation and evaporate it there under constant pressure.

This quantity used to be called the *Total Heat* of saturated steam. But it is preferable to use the name *Total Heat* in a slightly different sense, by including in it another term in addition to h and L, namely the heat-equivalent of the work that is done while the unit of mass of water is being introduced under the loaded piston in the imaginary experiment. We assumed unit mass of water to be already in the cylinder. But suppose instead that the cylinder is empty to begin with and that the water is forced in, while the piston exerts on it the constant pressure P under which steam is afterwards to be formed. The amount of work done in forcing it in is Pw and the equivalent of this in heat units is APw. By adding this term, which is numerically a very small one, to the heat afterwards taken in we obtain a quantity which is of very great importance in steam-engine calculations, and to it we shall apply the name of *Total Heat*, using for it the symbol I. Thus

$$\text{Total Heat of saturated steam } \left. \begin{array}{l} I = H + APw \\ = L + h + APw \end{array} \right\} \; \dots(6).$$

Since the volume of water is only 0·001 cubic metre per kilogramme or 0·016 cubic foot per lb., the numerical values of APw are very small in comparison with L or with h and it is only at high pressures that there is any appreciable difference between I and H. Even at the pressure corresponding to 200° C. it is only about one-third of a thermal unit. The difference is in fact negligible in practice, so far as steam is concerned: but by treating the total heat as including the small term APw much simplification is introduced in the theory of the steam-engine. We may also use the symbol I_w to denote the total heat up to the stage at which steam begins to be formed, but before any formation of steam begins, namely

$$I_w = h + APw$$

Then $\qquad L = I - I_w = H - h$ ....(7).

Values of I for saturated steam and for water at the same temperature (I_w) will be found in the tables. The values of H are not given in the tables. They are very nearly identical with those of I, and may be obtained (if they are wanted) by subtracting APw from I. APw is 0·02 at 100° C., 0·10 at 150° C. and 0·33 at 200° C.[1]

When steam is condensed under constant pressure an amount of heat equal to L is given out during the change of state from steam to water. Regnault's experiments on the latent heat of steam were in fact made by observing the heat given out when steam from a boiler was led to a calorimeter and was there condensed.

Consider next what heat is taken in per unit of mass when feed-water is supplied at a temperature t' and pressure P', is pumped into a boiler in which the pressure is P and is there converted into saturated steam under the constant pressure P. In getting it into the boiler the feed-pump has to expend on it an amount of work $(P - P') w$, the thermal equivalent of which is $APw - AP'w$. The water next takes in heat equal to $h - h'$ before it rises to the temperature at which steam is formed. Finally it takes in heat equal to L during its change of state from water to steam. Therefore the whole heat expended in the process, including the thermal equivalent of the work spent in getting the water in, is

$$APw - AP'w + h - h' + L,$$

or $\qquad H - h' + APw - AP'w.$

But $H + APw$ is I and $h' + AP'w$ is I_w' or the total heat of water at t': hence we may write the expression

$$I - I_w'.$$

Of this total, $H - h'$ is the heat supplied from the furnace exclusive of the work spent in forcing the water in.

[1] Regnault expressed his results for the heat of formation of steam by the formula $H = 606\cdot5 + 0\cdot305t$, but this gives conclusions which are inconsistent with other known properties, and there must have been errors in the observations. The heat of formation cannot be simply a linear function of the temperature. It will be found that values calculated from this formula differ materially from those in the table, except at temperatures in the neighbourhood of 75° C. The formula should be abandoned in favour of the values stated in the table. The same remark applies to a linear formula often given for the latent heat L.

62. Internal energy of steam. Of the whole latent heat of steam L, the part $AP(V-w)$ is, as has been said above, spent in doing external work.   The remainder, namely

$$L - AP(V-w),$$

is the *increase of internal energy* which the substance undergoes during conversion from water at t into steam at t.   Going back to the previous stage in the imaginary experiment, the heat h which was taken in before steam began to form all went to increase the stock of internal energy, for if we neglect the very slight expansion of the water[1] no external work was done.   Thus if we take the condition of water at $0°$ C. as an arbitrary starting-point in reckoning internal energy, the internal energy E of unit mass of saturated steam is given by the expression

$$E = h + L - AP(V-w) \ \dots\dots\dots\dots\dots(8),$$

or, more simply,

$$E = I - APV \ \dots\dots\dots\dots\dots\dots\dots\dots(8a).$$

The notion of internal energy is useful in calculating the heat taken in or rejected by steam during any stage of its expansion or compression in an engine.   When any working substance passes from one condition to another, its gain or loss of heat is determined, as we have already seen, by the equation

Heat taken in = increase of internal energy + external work.

Any of the terms of this equation may be negative; the last term is negative when work is done upon the substance instead of by it.

The relation between the quantities mentioned may be exhibited graphically thus:—

Total heat I

Heat of liquid h	Latent heat L	APw
	Increase of internal energy during change of state from water to steam $L-AP(V-w)$	External work done during change of state from water to steam $AP(V-w)$
	Total internal energy E	Total external work (in heat units) APV

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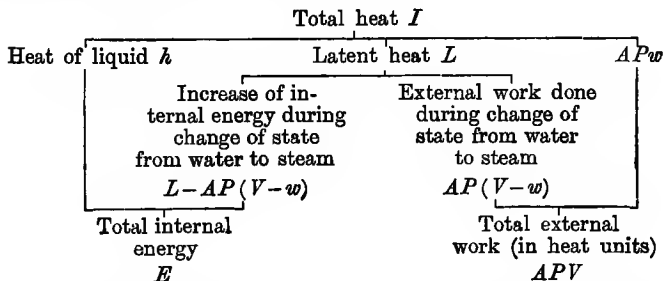
[1] Throughout these statements we are assuming that for practical purposes the volume of water w may be treated as constant.

It will be noticed that the term APw which represents work spent on the water in forcing it into the cylinder in the imaginary experiment, and by definition forms part of the total heat I, is included in the total external work, for it also represents work done by the substance in raising the piston. The whole work done upon the piston, from the time the water begins to enter till all is converted into steam is PV.

For any substance, in any condition whatever, we may *define* the Total Heat I as the sum of the internal energy and the total external work, namely

$$I = E + APV \dots\dots\dots\dots\dots\dots\dots(8\ b),$$

the "total external work" being understood to include the work done while the substance is entering the vessel in our imaginary experiment.

It will be clear that this definition is in agreement with the results already given for steam.

Starting with this definition of I we have, in the case of saturated steam, the following relations:—

Total heat I

Total internal energy acquired E		Total external work done APV	
Internal energy acquired by the water in being heated h	Increase of internal energy during change of state from water to steam $E-h$	External work done during change of state from water to steam $AP(V-w)$	External work done while the water is being forced into the vessel APw

Latent heat L

Total heat I.
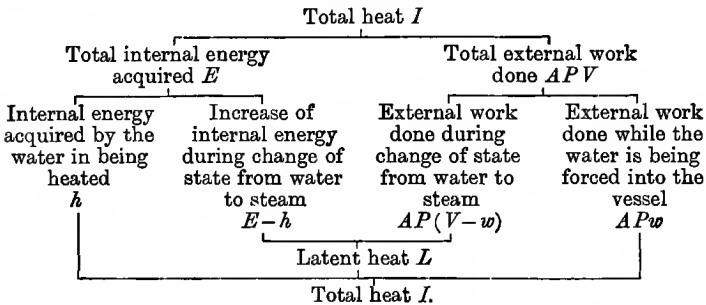
63. Formation of steam otherwise than under constant pressure. The equation

Heat taken in = increase of internal energy + external work

gives a means of finding the amount of heat required to form steam under any assigned conditions, in place of the condition assumed at the beginning of this chapter, where the formation of steam under constant pressure was considered. Whatever be the condition as to pressure under which the process of formation is carried on, the heat required is the sum of the internal energy of the steam when formed and the work done by the expanding fluid during the process.

The internal energy is the same whatever the history of the steam has been : that is to say it depends only on the final state, and not on the manner of formation.

When saturated steam is formed in a closed vessel of constant volume no external work is done; the heat of formation is then equal to the internal energy E, and is less than the total heat I for steam formed at a constant pressure equal to the pressure finally reached in the vessel, by the quantity APV.

64. Wet steam. In calculations which relate to the action of steam in engines we have generally to deal, not with *dry* saturated steam, but with *wet* steam, or steam which either carries in suspension, or is otherwise mixed with, a greater or less proportion of water. In the cylinder of a steam-engine, for example, the working substance is a wet mixture and the proportion of water to steam in it varies as the stroke proceeds. When any such mixture is in a state of thermal equilibrium the steam and water have the same temperature, and the steam is saturated. The *dryness* of wet steam is measured by the fraction q of dry steam in each unit of mass of the mixed substance. When the dryness is known it is easy to determine the other physical constants. Thus, reckoning in every case per unit of mass of the mixture, we have

Latent heat of wet steam $= qL$;

Total heat of wet steam $= I_w + qL$;

Volume of wet steam $= qV + (1 - q)w$

$= qV$ very nearly, unless the steam is so wet as to consist mainly of water ;

Internal energy of wet steam $= I_w + qL - APqV$ very nearly.

65. Superheated steam. Steam is superheated when its temperature is raised, in any manner, above the temperature which corresponds to saturation at the actual pressure. When steam at a low pressure is considerably superheated it behaves very nearly as a perfect gas, approximately satisfying the characteristic equation $\dfrac{PV}{\tau} = $ constant, and having constant specific heat. But under the ordinary conditions in which steam is superheated in practice the pressure is comparatively high, and the amount of superheating is only moderate : and in these conditions the properties differ widely from those of perfect gases.

A characteristic equation will be found in the Appendix which enables the volume to be calculated when the pressure and temperature are known, and also an equation for the total heat, both of which equations are applicable to superheated steam. It may be convenient to calculate the total heat in a different manner, by finding first the total heat of steam in the saturated state at the same pressure, and then adding a quantity which represents the heat taken in during the process of superheating. This requires a knowledge of the specific heat of the vapour, which in general varies as the temperature rises from the point of saturation. What is needed therefore is to know the average specific heat over the range through which superheating takes place. The table below (adapted from Mollier) gives values of this average specific heat for various ranges of superheating at various pressures, or in other words starting from various saturation temperatures. In each case the pressure is assumed to be constant while the steam is being superheated, a condition which is in fact generally fulfilled in steam-engine practice. The values given in the table are accordingly the mean specific heats at constant pressure, in each case for a range of temperature extending from the temperature of saturation t up to the temperature t' to which superheating is carried.

Table of Mean Specific Heat of Steam in superheating at constant pressure from the saturation temperature t to temperature t'.

Temperature of superheat t'	Temperature of Saturation ° C.				
° C.	80°	120°	160°	180°	200°
100°	·49				
150°	·49	·51			
200°	·49	·51	·54	·57	
250°	·48	·50	·53	·56	·59
300°	·48	·50	·52	·54	·57
350°	·48	·49	·51	·53	·56
400°	·48	·49	·51	·52	·55
450°	·48	·49	·51	·52	·54

Taking κ as the mean specific heat from this table we may readily find the total heat of superheated steam by the equation

$$I' = I + \kappa (t' - t) \dots\dots\dots\dots\dots\dots(9),$$

where I' and t' refer to the superheated condition and I and t to the saturated condition at the same pressure. As κ changes only slowly it is easy by inspection of the table to find a value suitable for any assigned condition as to initial pressure and degree of superheat.

In practice, superheating is carried out at constant pressure : the steam on leaving the boiler passes through a group of tubes in which the superheat is given to it while its pressure remains equal (or closely equal) to that in the boiler. Superheating has now become a very usual feature in steam-engine practice, for reasons which will be apparent later when the actual behaviour of steam in the engine is discussed in detail. It is rarely carried further than 400°C. and not often so far.

The value 0·48, based on an experiment by Regnault, is often taken for κ; but it will be seen from the table that though this is nearly right at low pressures a considerably higher value applies at such pressures as are usual in steam-engines.

Instead of using this table we may find the total heat directly, in any condition of superheating, by applying equation (8) of the Appendix, with the aid of the figures given in the Auxiliary Table D. In dealing with superheated steam it is important to realize that when the temperature and pressure are known the material is fully defined and all the other properties, such as volume, total heat, energy, and so forth, can be calculated from the equations in the Appendix without consideration of the particular starting-point from which superheating began.

66. Isothermal Lines for Steam. The expansion of volume which occurs during the conversion of water into steam under constant pressure—the second stage of the process described in § 53—is isothermal. From what has been already said it is obvious that steam, or any other saturated vapour, can be expanded or compressed isothermally only when wet, and that evaporation (in the one case) or condensation (in the other) must accompany the process. Isothermal lines for a working substance which consists of a liquid and its vapour are straight lines of uniform pressure.

67. Adiabatic Lines for Wet Steam. The form of adiabatic lines for a working substance consisting of a vapour or of a mixture of a liquid and its vapour depends not only on the particular fluid, but also on the proportion of liquid to vapour in

the mixture. When steam initially in the dry saturated state is allowed to expand adiabatically it becomes wet, and if initially wet (unless very wet[1]) it becomes wetter. To keep steam dry while it expands, doing work, some heat must be supplied during the process of expansion. If the expansion is adiabatic, so that no heat reaches the expanding fluid, a part of the steam is condensed, forming either minute particles of water suspended throughout the mass or a dew upon the surface of the containing vessel. The temperature and pressure fall ; and, as that part of the substance which remains uncondensed is saturated, the relation of pressure to temperature throughout the expansion is that which holds for saturated steam. The following formula, a proof of which will be given in the next chapter, serves to calculate the extent to which condensation takes place during adiabatic expansion, and so allows the relation of pressure to volume to be determined.

Before expansion, let the initial dryness of the steam be q_1 and its absolute temperature τ_1. Then, if it expand adiabatically until its absolute temperature falls to any value τ, its dryness after expansion is

$$q = \frac{\tau}{L}\left(\frac{q_1 L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau}\right) \quad \ldots\ldots\ldots\ldots\ldots(10).$$

L_1 and L are the latent heats (in thermal units) of unit mass of steam before and after expansion respectively. When the steam is dry to begin with, $q_1 = 1$.

This formula may be called the equation of adiabatic expansion or compression for wet steam. It does not directly give the relation of pressure to volume, but it allows the dryness at any stage of the process to be calculated, and from that (together with the fact that the part which remains in the condition of vapour is saturated) it is easy to find the volume which the mixture will fill when its pressure has changed to any assigned value. An example may help to make this clear. Suppose for instance that originally dry saturated steam at an absolute pressure of 182·6 pounds per square inch is made to expand adiabatically. Its initial tempera-

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[1] When the mixture contains a very large proportion of water to begin with, adiabatic expansion tends to dry it by causing some of the water to evaporate under the reduced pressure which results from the expansion. In Chapter V. graphic methods are described of investigating the changes of dryness that are produced by adiabatic expansion, and these may readily be applied to investigate whether the mixture will become drier or wetter in any given case (see § 95).

ture is 190°C. We wish to find the relation of pressure to volume at any stage in the expansion. Take for example the stage which is reached when the temperature has fallen to 120°C. We then have, for the values of quantities in the adiabatic equation (10),

$$\tau = 120 + 273 = 393,$$
$$L = I - I_w = 647\cdot4 - 120\cdot9 = 526\cdot5.$$

Also
$$q_1 = 1, \quad \tau_1 = 190 + 273 = 463,$$
$$L_1 = I_1 - I_{1w} = 668\cdot8 - 193\cdot1 = 475\cdot7.$$

Hence
$$q = \frac{393}{526\cdot5}\left(\frac{1 \times 475\cdot7}{463} + 2\cdot303 \log_{10}\frac{463}{393}\right)$$
$$= 0\cdot89.$$

This means that by the time the temperature has fallen to 120°C. eleven per cent. of the originally dry steam has become condensed into water. The pressure is then 28·8 pounds per square inch by the table. The volume occupied by that part of the substance which is still in the state of steam is qV per lb. of the mixture, where V is the volume of 1 lb. of dry steam at that pressure. Taking the value of V given in the table, namely, 14·25 cubic feet, qV is 12·68 cubic feet. To obtain the whole volume of 1 lb. of the working substance we ought in strictness to add to this the volume occupied by that part which has been converted into water, namely, by the fraction of a lb. which is represented by $1 - q$. But this is only 0·11 lb., and its volume is 0·11 × 0·017 or 0·0019 cubic foot[1]—a quantity which is negligible in comparison with the volume occupied by the still uncondensed steam. We conclude that 12·68 cubic feet is the volume of the mixture (per lb.) when its pressure has fallen to 28·8 pounds per square inch by adiabatic expansion; in other words, these numbers determine one point on the adiabatic line which begins with dry saturated steam at a pressure of 182·6 pounds per square inch.

In the same way we may go on to find as many points on an adiabatic line as we please, by taking a series of pressures, each lower than the initial pressure, and finding q for each, and then from q finding the volume of the mixture, which in ordinary cases is practically equal to qV, V being the volume which unit mass of saturated steam would occupy at the same pressure and temperature.

[1] The volume of 1 lb. of water, which is 0·016 cub. ft. when cold, rises to 0·017 at about 120° C. and to 0·018 at about 180° C.

The steam may be wet to begin with, and if q_1 have a value much less than unity it will be found on working out examples that q may turn out greater than q_1. This means that in very wet steam adiabatic expansion may reduce the amount of water as the net result of two opposing actions: as the temperature falls during expansion part of the steam initially present becomes condensed; on the other hand, part of the water initially present becomes evaporated because its initial temperature is higher than the temperature which the mixture takes as it expands. With very wet steam the result may be on the whole to make the mixture become drier. An extreme case occurs when all the substance is in the state of water to begin with. Then if adiabatic expansion be allowed to take place steam is formed, and the equation (10) may be applied, by writing $q_1 = 0$, to find how much of the water will be evaporated when the pressure, or the temperature, has fallen to any assigned value.

We shall see, in Chapter V., that by using certain graphic methods of representing the properties of steam the changes which go on during adiabatic expansion may be traced in a very simple manner, and the wetness at any stage in the expansion may be readily determined without the trouble of applying the above equation.

These graphic methods have the additional advantage that they can be applied with great convenience to cases where the steam instead of being saturated to begin with is initially superheated to any extent. Such cases frequently occur in steam-engine practice. When superheated steam expands adiabatically its superheat disappears as expansion proceeds, and after the steam has passed through a stage when it is just saturated any further expansion causes a part of it to condense.

In all cases of adiabatic expansion the fluid is doing work at the expense of its stock of internal energy. After drawing the pressure-volume curve for steam expanding adiabatically the student should calculate the internal energy at the beginning and also at the end of the expansion, taking account of the proportion of water to steam in the mixture, and see that the difference, representing the loss of internal energy which has occurred in the process, agrees with the work done, which is the area under the curve.

68. Formula connecting pressure with volume in the adiabatic expansion of steam. Adiabatic curves for steam, whether initially dry or wet, may be calculated in the way that has just been explained, and may then be represented with considerable accuracy by empirical equations of the form

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

where v is the volume of the wet mixture, by choosing such values for the index n as will give curves approximating closely to the actual adiabatic curves. A formula of this kind is especially useful for application to cases where the data are the initial pressure and the ratio of expansion r, and it is required to find the pressure after expansion. To find P when the substance has expanded to r times its initial volume,

$$P = \frac{P_1}{r^n} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(11)}.$$

The index n has a value which depends on q_1, the initial degree of dryness of the steam. According to the calculations of Zeuner[1] the curve of adiabatic expansion in steam of any degree of dryness not less than 70 per cent. is well represented by making

$$n = 1\cdot035 + 0\cdot1q_1,$$

so that for

$q_1 = 1$	0·95	0·9	0·85	0·8	0·75	0·7
$n = 1\cdot135$	1·130	1·125	1·120	1·115	1·110	1·105.

When it is desired to draw an adiabatic curve for expanding steam, that value of n must be chosen which refers to the degree of dryness at the beginning of the expansion.

It must not be supposed that the expansion of steam in an actual engine is adiabatic, for there is a transfer of heat between the working fluid and the metal of the cylinder and piston, which will be discussed in a later chapter. If it were practicable to make the surfaces of the cylinder and piston non-conducting the idea of adiabatic expansion could be realised. The adiabatic condition is approximated to in cases where the expansion goes on too rapidly to allow any considerable transfer of heat to take place.

69. Carnot's Cycle with steam for working substance. We are now in a position to study the action of a heat-engine employing water and steam (or any other liquid and its vapour)

[1] Zeuner's *Technical Thermodynamics*, vol. II.

as the working substance. To simplify the first consideration of the subject as far as possible, let it be supposed that we have, as before, a long cylinder composed of non-conducting material except at the base, and fitted with a non-conducting piston; also a source of heat A at some temperature τ_1; a receiver of heat, or as we may now call it, a condenser, C, at some lower temperature τ_2; and also a non-conducting cover B (as in § 43). Then Carnot's cycle of operations can be performed as follows. To fix the ideas, suppose that there is unit mass of water in the cylinder to begin with, at the temperature τ_1.

(1) Apply A, and allow the piston to rise against the constant pressure P_1 which corresponds to the temperature τ_1. The water will take in heat and be converted into steam, expanding isothermally at the temperature τ_1. This part of the operation is shown by the line ab in fig. 16.
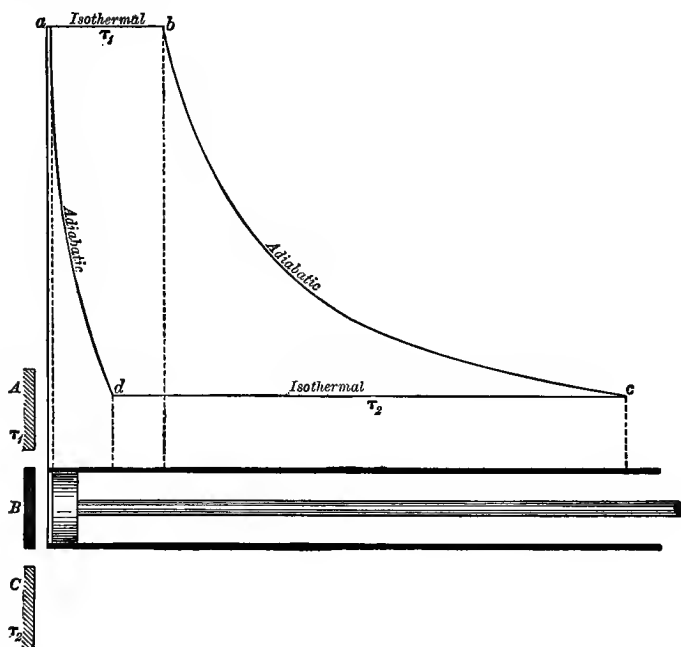
Fig. 16. Carnot's Cycle with water and steam for working substance.

(2) Remove A and apply B. Allow the expansion to continue adiabatically (bc), with falling pressure, until the temperature falls to τ_2. The pressure will then be P_2, namely, the

pressure which corresponds in the steam table to τ_2 which is the temperature of the cold body C.

(3)   Remove B, apply C, and compress.   Steam is condensed by rejecting heat to C.   The action is isothermal, and the pressure remains P_2.   Let this be continued until a certain point d is reached, after which adiabatic compression will complete the cycle.

(4)   Remove C and apply B.   Continue the compression, which is now adiabatic.   If the point d has been rightly chosen, this will complete the cycle by restoring the working fluid to the state of water at temperature τ_1.

The indicator diagram for the cycle is drawn in fig. 16, the lines bc and da having been calculated by the help of the equations in §§ 67 and 68, for a particular example, in which $p_1 = 90$ lbs. per square inch ($\tau_1 = 433$), and the expansion is continued down to the pressure of the atmosphere, 14·7 lbs. per square inch ($\tau_2 = 373$).

Since the process is reversible, and since heat is taken in only at τ_1 and rejected only at τ_2, the efficiency (by § 49) is

$$\frac{\tau_1 - \tau_2}{\tau_1}.$$

The heat taken in per unit mass of the fluid is L_1, and therefore the work done is

$$\frac{L_1 (\tau_1 - \tau_2)}{\tau_1},$$

a result which may be used to check the calculation of the lines in the diagram by comparing it with the area which they enclose. It will be seen that the whole operation is strictly reversible in the thermodynamic sense.

Instead of supposing the working substance to consist wholly of water at a and wholly of steam at b, the operation ab might be taken to represent the partial evaporation of what was originally a mixture of steam and water.   The heat taken in would then be $(q_b - q_a) L$, and as the cycle would still be reversible the area of the diagram would be

$$\frac{L (q_b - q_a) (\tau_1 - \tau_2)}{\tau_1}.$$

70.   Efficiency of a perfect Steam-engine.   Limits of temperature.   If the action here described could be realised in practice, we should have a thermodynamically perfect steam-engine using saturated steam.   Like any other perfect heat-engine

an ideal engine of this kind has an efficiency which depends upon the temperatures between which it works, and upon nothing else. The fraction of the heat supplied to it which such an engine would convert into work would depend simply on the two temperatures, and therefore on the pressures, at which the steam was produced and condensed respectively.

It is interesting therefore to consider what are the limits of temperature between which steam-engines may be made to work. The temperature of condensation is limited by the consideration that there must be an abundant supply of some substance to absorb the rejected heat; water is actually used for this purpose, so that τ_2 has for its lower limit the temperature of the available water-supply.

To the higher temperature τ_1 and pressure P_1 a practical limit is set by the mechanical difficulties, with regard to strength and to lubrication, which attend the use of high-pressure steam. In steam motor cars pressures of 1000 lbs. per square inch are successfully used, but with engines and boilers of the ordinary construction the pressure ranges from about 250 lbs. per square inch downwards.

This means that the upper limit of temperature, so far as the steam is concerned, is about 205° C. A steam-engine, therefore, under the most favourable conditions, comes very far short of taking full advantage of the high temperature at which heat is produced in the combustion of coal. From the thermodynamic point of view the worst thing about a steam-engine is the irreversible drop of temperature between the furnace and the boiler. The combustion of the fuel supplies heat at a high temperature: but a great part of the convertibility of that heat into work is at once sacrificed by the fall in temperature which is allowed to take place before the conversion into work begins.

If the temperature of condensation be taken as 15° C., as a lower limit, the efficiency of a perfect steam-engine, using saturated steam and following the Carnot cycle, would depend on the value of P_1, the absolute pressure of production of the steam, as follows:

Perfect steam-engine, with condensation at 15° C.,

P_1 in lbs. per square inch being	50	100	150	200	250	300
Highest ideal efficiency =	·300	·341	·366	·384	·397	·408

But it must not be supposed that these values of the efficiency are actually attained, or are even attainable. Many causes conspire to prevent steam-engines from being thermodynamically perfect, and some of the causes of imperfection cannot be removed. These numbers will serve, however, as one standard of comparison in judging of the performance of actual engines, and as setting forth the advantage of high-pressure steam from the thermodynamic point of view. We shall see later (§ 73) that there is another standard with which the performance of a real steam-engine may more appropriately be compared.

71. Efficiency of an engine using steam non-expansively. As a contrast to the ideally perfect steam-engine of § 67 we may next consider a cyclic action such as occurred in the early engines of Newcomen or Leupold, when steam was used non-expansively,—or rather, such an action as would have occurred in engines of this type had the cylinder been a perfect non-conductor of heat. In that case the volume of steam formed is equal to the volume swept through by the piston. We may represent the action of such an engine thus:-

(1) Apply the hot body A and evaporate the water as before at P_1. Heat taken in, per unit mass of the working fluid, $= L_1$.

(2) Remove A and apply the cold body C. This at once condenses a part of the steam, and reduces the pressure to P_2.
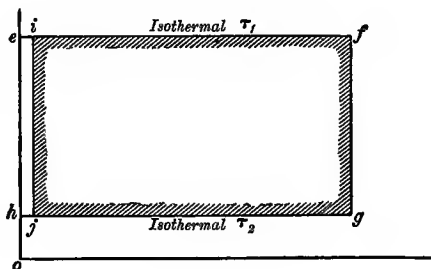
FIG. 17.

(3) Compress at P_2, in contact with C, till condensation is complete, and water at τ_2 is left.

(4) Remove B and apply A. This heats the water again to τ_1 and completes the cycle. Heat taken in $= h_1 - h_2$.

E.   7

The indicator diagram for this series of operations is shown in fig. 17, where $oe = P_1$ and $oh = P_2$.

Here the action is not reversible.   To calculate the efficiency

$$\frac{\text{Work done}}{\text{Heat taken in}} = \frac{(P_1 - P_2)(V_1 - w)}{J(L_1 + h_1 - h_2)}.$$

The values of this will be found to vary from 0·068 to 0·073 for the same range of pressure (50 lbs. to 300 lbs. per square inch) as was taken in the last paragraph, the temperature of condensation being 15° C. as before.   Contrast these numbers with the much higher efficiencies given there for a perfect steam-engine, following Carnot's cycle.

The efficiency of the actual Newcomen engine was much lower even than this calculation indicates, because in every stroke of the piston a large part of the steam entering the cylinder was at once condensed upon the sides, and the volume of steam which had to be supplied from the boiler was therefore much greater than the volume swept through by the piston.

72.   Engine with separate organs.   In the ideal engine whose action is represented in fig. 16 the functions of boiler, cylinder, and condenser are combined in a single vessel; but after what has been said in Chapter II. it is scarcely necessary to remark that, provided the working substance passes through the same cycle of operations, it is indifferent whether these are performed in several vessels or in one.   To approach a little more closely the conditions which hold in practice, we may think of the engine which performs the cycle of § 71 as consisting of a boiler A (fig. 18) kept at τ_1, a non-conducting cylinder and piston B, a surface condenser C kept at τ_2, and a feed-pump D which restores the condensed water to the boiler.   Then for every unit mass of steam supplied and used non-expansively as in § 71, we have

$$\text{work done on the piston} = (P_1 - P_2)V_1;$$

but the amount of work which has to be expended in driving the feed-pump is $(P_1 - P_2)w$.   Deducting this, the net amount of work done per unit mass of steam is the same as before, and the heat taken in is also the same.   An indicator diagram taken from the cylinder would give the area *efgh* (fig. 17), where

$$oe = P_1, \quad ef = V_1, \quad oh = P_2;$$

an indicator diagram taken from the pump would give the negative area *hjie*, where *ei* is the volume of the feed-water. The difference between these two areas, namely, the area *ifgh* which is shaded in the figure, is the diagram of the complete cycle gone through by
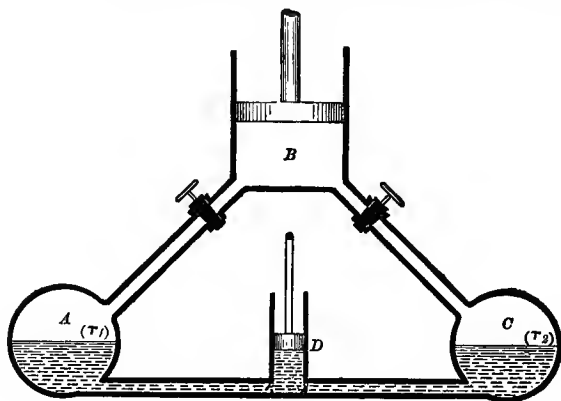
FIG. 18.   Organs of a Steam-Engine.

each unit of the working substance. In experimental measurements of the work done in steam-engines, only the action which occurs within the cylinder is shown on the indicator diagram. From this the work spent on the feed-pump is to be subtracted if we wish to make a rigorous determination of the thermodynamic efficiency. If the feed-water be at any temperature τ_0 other than the temperature of condensation τ_2, it is clear that the heat taken in is $H_1 - h_0$ instead of $H_1 - h_2$.

73.   How nearly may the process in a Steam-engine be reversible?   We have now to inquire how nearly, with the engine of fig. 18, that is to say, with an engine in which the boiler and condenser are separated from the cylinder, we can approach the reversible cycle of § 69. The first stage of that cycle corresponds to the *admission* of steam from the boiler into the cylinder, for during admission of steam to the cylinder a corresponding quantity of steam is being formed in the boiler. Then the point known as the point of *cut-off* is reached, at which admission ceases, and the steam already in the cylinder is allowed to expand, exerting a diminishing pressure on the piston. This is the second stage, or the stage of *expansion*. The process of expansion may be carried on until the pressure falls to that of the

condenser, in which case the expansion is said to be complete. At
the end of the expansion *release* takes place, that is to say, com-
munication is opened with the condenser. Then the return stroke
begins, and a period termed the *exhaust* occurs, that is to say, steam
passes out of the cylinder into the condenser, where it is condensed
at pressure P_2, which is felt as a *back pressure* opposing the return
of the piston. So far, all has been essentially reversible, and identical
with the corresponding parts of Carnot's cycle.

But we cannot complete the cycle as Carnot's cycle was com-
pleted. The existence of a separate condenser makes the fourth
stage, that of adiabatic compression, impracticable, and the best
we can do is to continue the exhaust until condensation is com-
plete, and then return the condensed water to the boiler by means
of the feed-pump.

It is true that we may, and in actual practice do, stop the
exhaust before the return stroke is complete, and compress that
portion of the steam which remains below the piston. It is
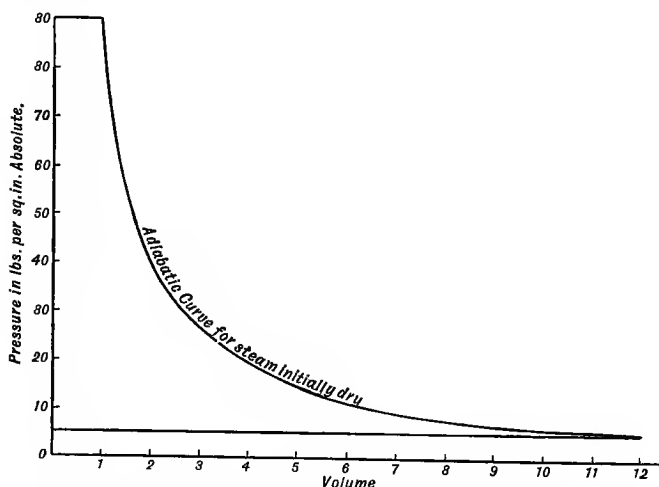
Fɪɢ. 19.   Ideal Indicator Diagram for Steam used expansively.

generally only a small part of the working substance that is so
compressed. This compression does not materially affect the ther-
modynamic efficiency; it is done partly for mechanical reasons, and
partly to avoid loss of power through clearance (see Chap. VII.).
By clearance is meant the small volume which, in a real cylinder,
is left below the piston at the end of the return stroke. In the

present instance it is supposed that there is no clearance, in which case any such compression is out of the question, for there is no volume which the compressed steam could occupy. The indicator diagram given by a cylinder in which steam goes through the action described above is drawn to scale in fig. 19 for a particular example, in which it is supposed that dry saturated steam is admitted to the cylinder at an absolute pressure of 90 lbs. per square inch, and is then expanded adiabatically to twelve times its original volume. This brings it down to a pressure of 5·4 lbs. per square inch, at which pressure it is discharged to the condenser. As we have assumed the cylinder to be non-conducting, and the steam to be initially dry, the expansion curve is calculated by the formula $Pv^{1\cdot135} = $ constant (§ 68). The advantage of expansion is obvious, that part of the diagram which lies under the curve being so much clear gain, as compared with the case dealt with in § 71.

We might proceed to calculate the performance as follows:

Work done during admission $= P_1 V_1$;

$\qquad$ „ $\qquad$ „ $\qquad$ expansion to volume $rV_1 = \dfrac{P_1 V_1 - P_2 r V_1}{n - 1}$

$\qquad\qquad\qquad\qquad$ (by § 39), $= \dfrac{P_1 V_1 - P_2 r V_1}{0\cdot135}$;

Work spent during return stroke $= P_2 r V_1$;

$\qquad$ „ $\qquad$ „ $\qquad$ on the feed-pump $= (P_1 - P_2) w$;

Heat taken in $= H_1 - h_0$.

It is however much more instructive to consider the action from another point of view, namely with reference to the changes of energy and of total heat. In what follows there is no restriction as to the steam being initially in the dry saturated state: it may be dry, wet, or superheated.

Let I_1 represent the total heat of the working substance in the condition before expansion. The numerical value of I_1 will depend not only on the pressure of admission but on whether the steam is initially dry, or wet, or superheated: in any case it is readily calculated when the initial conditions are known. I_1 is made up of the internal energy E_1 which the substance possesses at B (fig. 20) and the heat equivalent of the work done during admission, namely AP_1V_1,

$$I_1 = E_1 + AP_1V_1 = E_1 + A \text{ (area } ABMO).$$

From B to C no heat is taken in, the expansion being assumed to

be adiabatic, but work is done equal to the area under the expansion curve, and therefore the internal energy falls to a value E_2 at C such that $E_1 - E_2 = A$ (area $BCNM$).
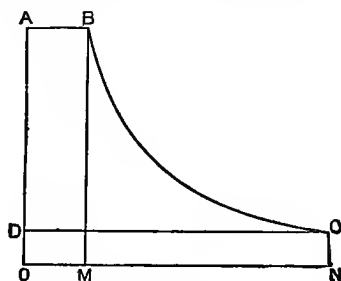
FIG. 20.

At C the total heat of the mixed substance is

$$I_2 = E_2 + A P_2 V_2 = E_2 + A \text{ (area } DCNO).$$

Hence, subtracting,

$$I_1 - I_2 = E_1 - E_2 + A (P_1 V_1 - P_2 V_2)$$
$$= A \text{ (area } BCNM + \text{ area } ABMO - \text{ area } DCNO)$$
$$= A \text{ (area } ABCD).$$

That is to say, the area of the indicator diagram for the whole operation in the cylinder, which measures the work done on the piston, when converted into heat units is equal to $I_1 - I_2$, where I_1 is the total heat of the working substance before expansion and I_2 is the total heat of the working substance after expansion, all the quantities being reckoned per unit of mass of the working substance[1].

74. Ideal Performance measured by the Heat Drop. We have, then, the very important conclusion that the amount of

[1] This result may be obtained more shortly as follows:

The general equation, applicable to any process,

Heat taken in = gain of internal energy + work done,

may be written $dQ = dE + AP dV,$

where dQ represents a small gain of heat on the part of any substance, dE the corresponding gain of internal energy, and dV the change of volume.

From this $dQ = d (E + APV) - AV dP$
$$= dI - AV dP.$$

But in an adiabatic process $dQ = 0$: hence in such a process,
$$dI = AV dP,$$

and, integrating,

$$I_1 - I_2 = A \int_{P_2}^{P_1} V dP,$$

which is the equivalent in heat units of the work done, namely the area $ABCD$.

work ideally obtainable per lb., or per kilogramme, of steam in an ideal engine, using adiabatic expansion from the initial pressure at which the steam is supplied to the final pressure at which it is condensed, is (in heat units)

$$I_1 - I_2.$$

This quantity is often called the *heat drop*. The conclusion holds whatever be the state of the steam on supply, whether wet or saturated or superheated. All that is assumed is that there is a constant pressure of supply, a constant pressure of condensation, that there are no transfers of heat between steam and metal, and that the expansion is complete from the pressure of supply to the pressure of condensation. Both I_1 and I_2 have to be reckoned with reference to the state of the substance in the initial and final stages. As a rule at the end of expansion we have a wet mixture for which to reckon I_2.

By the help of graphic processes, the use of which will be explained in Chapter V., the problem of finding I_2 after any given amount of adiabatic expansion is rendered exceedingly simple, and it becomes easy to work out the theoretical performance $I_1 - I_2$ per unit mass of steam under any assigned con-
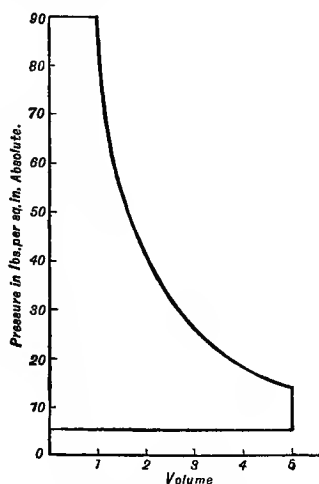
FIG. 21.   Incomplete expansion.

ditions as to initial pressure, initial superheat, and final pressure. In this way we can obtain figures which are of value as standards of comparison with the actual performances found in tests of real

engines. The performance of any real engine, in which the cylinder takes heat from the steam during admission and in which the expansion is not adiabatic, is necessarily less than this theoretical limit $I_1 - I_2$, which however forms a useful criterion of the amount of work ideally obtainable per unit quantity of steam. Another reason why the performance of a real engine is less is that in it the expansion is, in general, not complete. Instead of the steam being caused to expand until its temperature falls to that of the condenser or cold body into which heat is to be rejected it is allowed to escape by the opening of the exhaust valve when its temperature (and pressure) is still somewhat above that of the condenser. The pressure accordingly suffers a sudden drop, and an irreversible transfer of heat takes place. Incomplete expansion is illustrated by fig. 21, where the steam is supposed to escape after expanding to five times its initial volume. It results simply in a loss of the work which is represented by the toe of the diagram, that is to say, by the difference of areas between this and fig. 19.

75. Rankine Cycle. This name is given to the ideal cyclical process of § 73, in which the steam is passed into the cylinder at the initial pressure P_1 at which it is formed, then expands adiabatically to the final pressure P_2, at which it is condensed, and finally is returned to the boiler as water. The net amount of work done by the substance in going through this cycle is the work done by it in the cylinder, which is the area $ABCD$ of fig. 20, less the work spent upon it in the feed pump, which is $(P_1 - P_2)w$. In other words, the net amount of work done by the steam is, in heat units,

$$I_1 - I_2 - A(P_1 - P_2)w.$$

If we divide this by the heat taken in, we have the efficiency of the ideal Rankine cycle.

The term $A(P_1 - P_2)w$ is relatively so small that no material error is introduced by ignoring it, and treating the net amount of work done in the Rankine cycle as equivalent to $I_1 - I_2$ for purposes of practical calculation.

The efficiency of the Rankine cycle is always less than the ideal highest efficiency of a perfect engine working between the same limits of temperature. This is because of the absence of the compression which formed the fourth stage in Carnot's cycle,

and had the effect of bringing the temperature up to the top of the range before the substance began to take in heat. Without compression some of the heat is taken in at temperatures below the highest temperature τ_1, and any heat taken in at a lower temperature cannot contribute so much work as if it had been taken in at τ_1.

The most useful criterion that can be applied to tests of real engines is to compare the work done in the cylinder with the heat drop, in other words to compare the actual amount of indicated work per lb. or per kilogramme of steam passing through the engine with the amount of work ideally obtainable with complete adiabatic expansion but without adiabatic compression, namely $I_1 - I_2$. In this comparison we leave out of account the work spent on the substance in the feed pump, both in the real and the ideal engine.

In Chapter VII. we shall consider more fully why it is that the performance of real engines under the best conditions that hold in practice fall much short of this ideal; in the meantime the fact is mentioned to guard the reader from supposing that calculations based on ideal conditions apply in the working of actual engines.

76. Changes of State in a Liquid and its Vapour : Critical Point. It will assist the student to understand the properties of steam or any other vapour if he will consider what happens when the pressure is gradually reduced under conditions such that the liquid or vapour remains at a constant temperature during the process. Imagine for instance a cylinder to contain a quantity of the liquid under pressure applied by a loaded piston, and let the cylinder stand on a body at a definite constant temperature, which will supply enough heat to it to maintain the temperature unchanged when the pressure of the piston is gradually relaxed and the volume consequently increases. Starting from a condition of very high pressure, say at A_1 (fig. 22), when the contents of the cylinder are wholly liquid, let the load on the piston be slowly reduced so that the pressure gradually falls. The contents at first remain liquid, until the pressure falls to a certain value depending on the temperature, at which vapour begins to form. Thus we have in the pressure-volume diagram a line A_1B_1 to represent what happens while the pressure is falling during this first stage, while the contents are still liquid. The

volume of the liquid increases, but only very slightly, in consequence of the pressure being relaxed, and hence A_1B_1 in the diagram is nearly but not quite vertical. At B_1 steam begins to
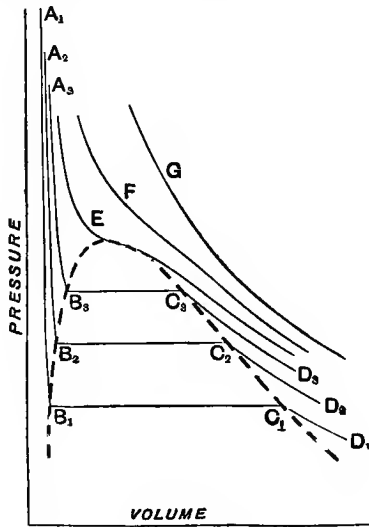
Fɪɢ. 22.

form, and continues forming until all the liquid becomes vapour. This is represented by B_1C_1, a stage during which there is no change of pressure. At C_1 we have saturated vapour. Then, if the fall of pressure continues, a line C_1D_1 is traced, the progressive fall of pressure being associated with a progressive increase of volume. The temperature, by assumption, is kept constant throughout. At D_1, or at any point beyond C_1, the vapour has become superheated, because its pressure is now lower than the pressure corresponding to saturation, and hence its temperature is higher than the temperature corresponding to saturation at the actual pressure. The line $ABCD$ is an *isothermal* for the substance in the successive states of liquid (A to B), liquid and vapour mixed (B to C), saturated vapour (at C), superheated vapour (C to D). Now take a much higher temperature. We get a similar isothermal $A_2B_2C_2D_2$, and a higher temperature still another isothermal $A_3B_3C_3D_3$, and so on. The higher the temperature the nearer do B and C approach each other, and if the temperature is made high enough the line BC vanishes.

A curve (shown by the broken line) drawn through $B_1B_2B_3$, etc. is continuous with one passing through $C_1C_2C_3$, and it is only within the region of which this curve is the upper boundary that any change from liquid to vapour takes place. The branch $B_1B_2B_3$, which shows the volume of the liquid, meets the branch $C_1C_2C_3$, which shows the volume of the vapour, in a rounded top. The temperature for an isothermal line E that would just touch the top of this curve is called the Critical Temperature. We might define the critical temperature in another way by saying that if the temperature of a vapour is above the critical temperature no pressure, however great, will cause it to liquefy. Starting from D and increasing the pressure, the temperature being kept constant, we may trace any of the isothermals backwards. The initial state is then that of a gas (a superheated vapour): as the pressure increases C is reached when it is saturated and condensation begins: at B condensation is complete, and from B upwards towards A we are compressing liquid. At any point between C and B the substance exists in two states; part is liquid and part is vapour. But if the isothermal is one that lies altogether outside of the boundary curve, shown by the broken line, the substance does not suffer any sharp change of state as the pressure rises. It follows a course such as is indicated by the lines F or G, and at no stage in the process is it other than homogeneous.

The critical temperature for steam is about 365° C., and in the action of an ordinary steam-engine we are concerned only with lower temperatures. But with carbonic acid, whose critical temperature is only about 31° C., the behaviour above the critical point is of great practical importance in connection with refrigerating machines which employ carbonic acid as working substance.

Gases such as air, oxygen and so forth, are vapours which under ordinary conditions are very highly superheated. Their critical temperatures are so low that it is only by extreme refrigeration that they can be brought into a condition which makes liquefaction possible. The critical temperature of hydrogen is − 241° C. or 32° absolute. Even helium, the most refractory of the gases, has been liquefied, but only by cooling it to a temperature within about 5 degrees of the absolute zero.

77. Steam at very High Pressures. Of the properties of steam at pressures beyond the range in the Tables but little is

accurately known.   Experiments by Cailletet and Colardeau[1] give
the following figures for the relation of pressure to temperature in
saturated steam up to the critical point (365° C.).   The agreement
between them and the results of other observers leaves a good
deal to be desired[2].   The pressures are in atmospheres, one atmo-
sphere being 1·0333 kilogrammes per sq. cm. or 14·697 lbs. per
sq. inch.

Temperature °C.	Pressure Atmospheres	Temperature °C.	Pressure Atmospheres
225	25·1	300	86·2
230	27·5	305	92·2
235	30·0	310	99·0
240	32·8	315	106·1
245	35·5	320	113·7
250	39·2	325	121·6
255	42·9	330	130·0
260	46·8	335	138·8
265	50·8	340	147·7
270	55·0	345	157·7
275	59·4	350	167·5
280	64·3	355	178·2
285	69·2	360	188·9
290	74·5	365	200·5
295	80·0		

Converting these pressures into pounds per square inch,
plotting the points, and drawing a smooth curve through them,
we have the following:

Saturated Steam at High Pressures.

Pressure lbs. per sq. inch	Temperature °C.	Pressure lbs. per sq. inch	Temperature °C.
300	214·3	1700	321·3
400	229·7	1800	325·5
500	242·3	1900	329·7
600	252·5	2000	333·6
700	261·4	2100	337·4
800	269·5	2200	341·0
900	276·9	2300	344·5
1000	283·7	2400	348·0
1100	290·2	2500	351·4
1200	296·3	2600	354·6
1300	302·0	2700	357·8
1400	307·2	2800	360·9
1500	312·2	2900	363·8
1600	316·9		

[1] Cailletet and Colardeau, *Comptes Rendus*, cxii. 1891, and *Ann. de Chimie et de
Physique*, 6th Ser. xxv. 1892.

[2] Compare with A. Battelli, *Ann. de Chim. et de Phys.* 6th Ser. xxvi. 1892, and
7th Ser. iii. 1894; also Knipp, *Phys. Rev.* 1900.

CHAPTER IV.

78. Rankine's statement of the Second Law. Rankine, to whom with Clausius and Lord Kelvin is due the development of the theory of heat-engines from the point at which it was left by the "Réflexions" of Carnot and the experiments of Joule, has, in his "Manual of the Steam-Engine and other Prime Movers," stated the second law of thermodynamics in a form which is neither easy to understand, nor obvious, as an experimental result, when understood. His statement runs:—

"If the absolute temperature of any uniformly hot substance be divided into any number of equal parts, the effects of those parts in causing work to be performed are equal."

To make this intelligible we may suppose that any quantity Q of heat from a source at temperature τ_1 is taken by the first of a series of perfect heat-engines, and that this engine rejects heat at a temperature τ_2 which is less than τ_1 by a certain interval $\Delta\tau$. Let the heat so rejected by the first engine form the heat-supply of a second perfect engine working from τ_2 to τ_3 through an equal interval $\Delta\tau$; let the heat which it in turn rejects form the heat-supply of a third perfect engine working again through an equal interval from τ_3 to τ_4; and so on. The efficiencies of the several engines are (by § 49)

$$\frac{\Delta\tau}{\tau_1}, \quad \frac{\Delta\tau}{\tau_2}, \quad \frac{\Delta\tau}{\tau_3}, \quad \&c.$$

The amounts of heat supplied to them are

$$Q, \quad Q\frac{\tau_2}{\tau_1}, \quad Q\frac{\tau_3}{\tau_1}, \quad \&c.$$

Hence the amount of work done by each engine is the same, namely,
$$Q\frac{\Delta\tau}{\tau_1}.$$

Thus Rankine's statement is to be understood as meaning that each of the equal intervals into which any range of temperature may be divided is equally effective in allowing work to be

produced from heat when heat is made to pass, doing work in the most efficient possible way, through all the intervals from the top to the bottom of the range.

79. Absolute Temperature : Lord Kelvin's scale. In the preceding chapters we have been using the imaginary perfect gas thermometer as the means of framing a scale of temperatures. In other words, our scale has been such that equal intervals of temperature are defined as those which correspond to equal amounts of expansion of a perfect gas under constant pressure. We have defined τ by means of the formula $V = c\tau$, P being constant. And seeing that air, or hydrogen, behaves as a nearly perfect gas this scale is practically realised by the air thermometer, or by the hydrogen thermometer.

Starting from this definition of temperature we have found by an application of Carnot's principle that a reversible engine working between a hot source A and cold receiver of heat C takes in from the source and gives out to the receiver quantities of heat Q_A and Q_C which are proportional to the absolute temperatures of the source and receiver respectively, as defined by reference to the perfect gas thermometer.

Hence we might have defined temperature in a very different way and still have arrived at just the same scale. We might have said, let the temperatures of A and C be specified by two numbers which shall be proportional to the heat taken in and given out respectively by a reversible heat-engine when working with A for source and C for receiver of heat. This method of defining absolute temperature was proposed by Lord Kelvin. It gives a scale which is truly absolute in the sense of being independent of the properties of any gas or other substance, real or imaginary. The scale so obtained coincides with the scale of the perfect gas thermometer.

Lord Kelvin's method of devising a scale of absolute temperatures may also be put in a somewhat different fashion, thus :—Starting with any arbitrary temperature let a series of intervals be taken such that equal amounts of work will be done by every one of a series of reversible engines, each working with one of these intervals for its range and each handing on to the engine below it the heat which it rejects, so that the heat rejected by the first forms the supply of the second, and so on. Then call these intervals equal. This is only another way of putting the definition of absolute temperature which has just been quoted : it is sug-

gested by what has been said in the last paragraph about
Rankine's statement of the Second Law.

To make this aspect of the matter more intelligible, think of
a quantity of heat supplied at some high temperature and used
to drive a chain of perfect (reversible) heat-engines.   As the heat
goes down from engine to engine in the chain part of it is con-
verted into work at each step and the remainder passes on to form
the heat-supply of the next engine.   We have to think of the
steps as being such that the amount of heat converted into work
is the same for each step.   Thus if we have n engines in the chain
and if the whole quantity of heat supplied to the first engine is Q,
then the steps are such that each engine converts the quantity $\dfrac{Q}{n}$
of heat into work.   When n steps are completed there is no heat
left: all is converted into work.   This means that the absolute
zero of temperature has been reached: we may in fact define the
absolute zero as the temperature which is reached in this manner.
We have reached it by coming down through a finite number of
steps of temperature, and each step represents a finite fall in
temperature.   We define the absolute or thermodynamic scale by
saying that these steps are to be taken as equal to one another.
From this it will be seen that the conception of an absolute zero
and of an absolute thermodynamic scale with uniform intervals does
not depend on any notion about perfect gases or about the proper-
ties of any particular substance.   We reach the absolute zero when,
on going down through the chain of perfect engines, we come to
a point at which the last fraction of the heat has been converted
into work.   That fixes the absolute zero.   And we call the steps
by which we have come equal steps of temperature.   That fixes
the scale.   Moreover the steps can be so taken, by choosing a
suitable number of them, that the scale so obtained will agree at
two fixed points with the ordinary thermometric scale, and will
contain between those fixed points the same number of steps as
the ordinary scale contains degrees.   Thus suppose the initial
temperature, at the top of the chain, is that of the boiling point of
water, and that we have 373 engines in the chain.   Then we find
that it takes 100 steps to come down to the temperature of
melting ice, and 273 more steps to complete the conversion of the
remaining heat into work.   This means that the uniform step of
temperature on the thermodynamic scale is equal to the average

of the intervals called degrees on the centigrade thermometer, when that average is taken between the freezing point and the boiling point (0° and 100°), and that the absolute zero is at a point 373 of such steps below the boiling point, or 273 below the freezing point.

The scale of the actual air thermometer would be in perfect agreement with Lord Kelvin's absolute scale if the laws stated in Chapter II. were rigorously true of air, namely, Boyle's law, Regnault's law, according to which the specific heat is independent of the pressure, and Joule's law, according to which there is no change of temperature when a gas expands without doing external work and without receiving or rejecting heat. The experiments by which Joule established his law have been already described. Reference has also been made to the subsequent experiments of a more searching kind, devised by Lord Kelvin, and carried out by him in conjunction with Joule, in which air was forced slowly through a porous plug to see whether its temperature became changed, which have shown that air does not conform with perfect exactness to Joule's law[1]; but the deviations are so slight that for all practical purposes the scale of the air thermometer may be taken as agreeing with the absolute scale. The agreement is still closer if hydrogen, which is more nearly "perfect," is used in the thermometer instead of air[2].

Actual air or gas thermometers may be made for use in two ways: In one the pressure is kept constant and the volume is allowed to expand or contract as the temperature varies; in the other the volume is kept constant by adapting the pressure to the temperature which is being measured, and the temperature is then taken to be proportional to the pressure. This latter is the more practicable form: it is called the constant-volume gas thermo-

[1] See Lord Kelvin's *Collected Papers*, vol. I. p. 333.

[2] See Callendar, "On the Thermodynamical Correction of the Gas Thermometer," *Phil. Mag.* Jan. 1903. Tables are there given showing the correction to be applied, for air, hydrogen, and other gases, to convert the scale reading of the gas thermometer to the absolute thermodynamic scale. The correction is exceedingly small: in the case of the constant-volume hydrogen thermometer it is not more than 0·001 of a degree at any points throughout the range from $-10°$ C. to 200° C. and even at 1000° C. it is barely one-tenth of a degree. With the constant-volume air thermometer it is about six times as great, but even with air it is only at temperatures below $-100°$ C. or above 300° C. that the correction is so much as one-tenth of a degree. The corrections are somewhat greater when the gas thermometer is of the constant pressure type.

meter.   The air or other gas must be perfectly dry: if there is
any water vapour in it the volume in the one case or the pressure
in the other may be far from proportional to the temperature.

80.   Clapeyron's Equation.   This name is given to an
important relation between the Latent Heat of steam or any
other vapour, the change of volume which it undergoes in being
vaporized, and the rate at which its saturation pressure varies
with the temperature.   To establish it we may revert to the
ideally perfect steam-engine of § 69, in which Carnot's cycle is
followed with water and steam for working substance.   We saw
that this gave an indicator diagram (fig. 16) with two lines of
uniform pressure (isothermals) connected by two adiabatic curves.
The heat taken in was L per unit mass of working substance, and
since the engine was reversible its efficiency was

$$\frac{\tau_1 - \tau_2}{\tau_1},$$

from which it followed that the work done, or the area of the
diagram, was

$$\frac{L(\tau_1 - \tau_2)}{\tau_1}.$$

This is in thermal units: to reduce it to units of work we multiply
by J.   Now suppose that the engine works between two tempera-
tures which differ by only a very small amount.   We may call the
temperatures τ and $\tau - \delta\tau$, $\delta\tau$ being the small interval through
which the engine works.   The above expression for the work done
becomes

$$\frac{JL\delta\tau}{\tau}.$$

The indicator diagram is now a long narrow strip (fig. 23).
Its length ab is $V - w$, V being the
volume of unit mass of steam and w
the volume of unit mass of water.   Its
height is δP, where δP is the difference
between the pressure in ab and that in
cd.   In other words, since the steam
is saturated in cd as well as in ab, δP
is the difference in the pressure of
saturated steam due to the difference
in temperature $\delta\tau$.   When δP is made
very small, the area of the diagram becomes more and more
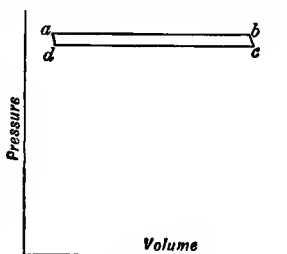

FIG. 23.

nearly equal to the product of the length by the height, namely, $\delta P(V - w)$. This is equal to the work done, whence

$$\delta P (V - w) = \frac{JL\delta\tau}{\tau} \quad\dotfill(1).$$

This equation is only approximate when the interval $\delta\tau$ (or δP) is a small finite interval. In the limit, when the interval is made indefinitely small, it becomes exact and may then be written

$$V - w = \frac{JL}{\tau}\frac{d\tau}{dP} \quad\dotfill(2),$$

$\frac{d\tau}{dP}$ being the rate at which the temperature of saturated steam alters relatively to the pressure when the temperature is τ.

Thus we have the Clapeyron equation

$$V = w + \frac{JL}{\tau}\frac{d\tau}{dP}$$

as a relation between the volume, the latent heat, and $\frac{dP}{d\tau}$. It may be applied to find the volume of saturated vapour when we know the latent heat and also know the relation of pressure to temperature so well as to be able to find $\frac{dP}{d\tau}$ by drawing a tangent to the pressure-temperature curve or by differentiating an expression connecting P and τ. Or it may be applied to find the latent heat when the data are the volume and $\frac{dP}{d\tau}$. The values of pressure, volume, and latent heat, given in steam tables in relation to temperature, must, if the tables are accurate, be such as will satisfy this equation. Take, for example, steam at 100° C. From the table it may be found by plotting the pressure-temperature curve that a rise in temperature of 1 degree at that point gives a rise in pressure of 0·0370 kilogramme per square centimetre or 370 kilogrammes per square metre. The latent heat given in the table is 539·1. Hence, since J is 427 we should have for the volume per kilogramme

$$V - w = \frac{427 \times 539\cdot1}{373} \cdot \frac{1}{370} = 1\cdot668 \text{ cub. metres.}$$

Since w is 0 001 this makes $V = 1\cdot669$, a value which agrees well with that given in the table, namely, 1·670. The agreement would be exact if we took 0·03698 for the rise of pressure per degree,

and the difference is well within the limits of error which apply in the process of finding $\frac{dP}{d\tau}$ from the pressure-temperature curve[1].

81. Extension of the above result to other changes of physical state. In equation (2), above, the left-hand side is positive, since the volume of steam is greater than that of water. The right-hand side must also be positive, and hence it is that $\frac{d\tau}{dP}$ is positive, or in other words, that increasing the pressure under which steam is formed raises the boiling point. The equation might evidently be applied in the reverse way to that indicated above (for finding V); in other words, if the amount by which the volume increases when water changes into steam were given we might employ that to calculate $\frac{d\tau}{dP}$, the rate at which the boiling point is raised by increase of pressure.

Further, the reasoning by which this equation was arrived at was perfectly general and was in no way restricted to the case of steam. The engine whose indicator diagram is sketched in fig. 23 might have anything for working substance, the isothermal line of the first operation, during which heat is taken in, representing in the most general way the change of volume which occurs while any working substance changes its physical state. In the example already dealt with the change is from liquid to vapour. But we might begin with a solid substance previously raised to the temperature τ at which it begins to melt and let the first stage in the cycle consist in the expansion of the substance while it passes from the solid to the liquid state, the substance doing external work by overcoming a constant pressure as it expands. All the steps in the argument remain unaffected, and hence the equation may be written thus with reference to any transformation of state on the part of any substance,

$$U - U' = \frac{J\lambda}{\tau} \frac{d\tau}{dP} \quad\dots\dots\dots\dots\dots\dots\dots(3),$$

where U' is the volume of unit mass of the substance in the

[1] In the steam tables in earlier editions of this book the volumes given were calculated by means of the Clapeyron equation, using the data of Regnault's experiments for L and $\frac{dP}{d\tau}$, but a different procedure is followed in the modern tables (see Appendix).

original state, U is the volume after the transformation has taken place, λ is the heat absorbed while the transformation is going on (the latent heat of fusion or of evaporation as the case may be), and $\dfrac{d\tau}{dP}$ is the rate at which the temperature of the transformation (say the melting-point or the boiling-point) is affected by altering the pressure under which the change of state occurs.

If a solid body expands on melting, U is greater than U', and consequently $\dfrac{d\tau}{dP}$ must be positive: in other words, the melting-point will in that case be raised by applying pressure.

On the other hand if the substance contracts on melting, $U - U'$ is negative and τ must then *decrease* relatively to P, that is to say, the melting-point is then lowered by applying pressure. This is the case with ice. From the known amount by which ice contracts when it melts James Thomson (in 1849) first applied this method of reasoning to show that the melting-point of ice must be lowered to a definite extent when the ice is melted under any assigned pressure, and the result was afterwards verified by an experiment of his brother, Lord Kelvin. The amount by which the melting-point of ice is lowered is about $0.0074°$ C. for each atmosphere of pressure[1].

82. Drying and superheating of steam by throttling. When dry steam is forced through a throttle valve or constricted orifice without receiving or rejecting heat it becomes superheated; and if wet to begin with it becomes drier. This is because the total heat of saturated steam is less at low pressure than at high, and, as we shall see presently, the total heat of the fluid as a whole undergoes no change in the process of throttling. Suppose for instance that steam is flowing through a small pipe or orifice from a chamber where the pressure is P_1 to another where there is a lower pressure P_2. Such an action happens in steam-engines in the movement of steam through contracted pipes or passages,

[1] See Lord Kelvin's *Collected Papers*, vol. I. p. 156 and p. 165. The numerical result stated in the text is obtained as follows:—A pound of water changes its volume in freezing from 0.016 to 0.0174 cub. ft., and gives out 80 (Centigrade) units of heat. Hence

$$\frac{d\tau}{dP} = \frac{0.0014 \times 273}{80 \times 1400} = 0.00000341,$$

and if δP be one atmosphere or 2160 lbs. per sq. ft., $\delta\tau$ is 2160×0.00000341 or $0.0074°$ C.

such as a partially closed stop valve or a reducing valve between the boiler and the engine: the steam becomes reduced in pressure and is said to be throttled or " wire-drawn." Eddies are formed in rushing through the constricted opening and the energy expended in forming them is frittered down into heat as the eddies subside. To calculate the effect of throttling assume, in the first place, that a steady condition exists before and also after the throttling and that the chambers on both sides of the constricted opening are relatively very large, so that the stream of steam has no kinetic energy worth taking account of either before it passes the opening or after it has passed and the eddies have subsided.

To fix the ideas, imagine the substance to be transferred from one side of the opening to the other by piston A (fig. 24) coming up towards the opening, and piston B receding. On the side A there is pressure P_1, on the side B a lower pressure P_2. Let V_1 and V_2 represent the volumes of unit mass of the fluid before and after transfer. Before passing through the opening the fluid has a stock of internal energy (per unit of mass) which we shall call E_1, and as it approaches the orifice the piston A does work upon it equal to P_1V_1. On the other side the fluid does work against the

FIG. 24.

piston B equal to P_2V_2. Hence E_2, the stock of energy which it has after passing the opening, is given by the equation

$$E_2 = E_1 + AP_1V_1 - AP_2V_2$$

or

$$E_2 + AP_2V_2 = E_1 + AP_1V_1,$$

which may be written $I_2 = I_1,$

I_1 and I_2 being the total heats of the fluid, per unit mass, before and after passing respectively. In other words the total heat I does not change in the process. This is a highly important characteristic of I, that it does not change in the kind of expansion which occurs when the fluid passes a throttle valve or along a pipe offering frictional resistance. The imaginary pistons were introduced only to make the argument clear; the result is of general application to all cases of throttling, or frictional drop of pressure, and to any fluid.

Applying this now to the case of wet steam, let q_1 and q_2 be the dryness before and after throttling.   By § 64 we have

$$I_1 = I_{w1} + q_1 L_1 \text{ and } I_2 = I_{w2} + q_2 L_2.$$

Hence the dryness after throttling is found from the equation

$$q_2 = \frac{I_{w1} - I_{w2} + q_1 L_1}{L_2}.$$

To take a numerical example, suppose that the steam in its first state has a pressure of 130 lbs. per sq. inch (temperature $175°$ C.), and contains 6 per cent. of moisture, and suppose it to fall by throttling to atmospheric pressure.   Then q_1 is 0·94 and by the tables $L_1 = 487·6$, $L_2 = 539·1$, $I_{w1} = 177·4$ and $I_{w2} = 100·5$. With these data q_2 becomes 0·993 : in other words the steam after throttling contains only 0·7 per cent. of moisture.   If the initial dryness had been 0·948 a similar calculation shows that with the same drop in pressure the steam would become just dry.

When the steam is dry to begin with or when the drop of pressure is more than sufficient to remove its initial moisture the result of the throttling is to superheat it.   Though the temperature falls, it falls less than corresponds to the fall in pressure and consequently there is superheat.   In that case the total heat after throttling is $I_{s2} + \kappa (t' - t_2)$, where t' is the actual temperature after throttling, t_2 is the temperature of saturation corresponding to the pressure in the second state, I_{s2} the total heat of saturated steam at that pressure, and κ is the mean specific heat in superheating over the range from t_2 to t'.   Equating this to the total heat before throttling, we can calculate the amount of superheating which will result, provided κ is known.

Conversely, if throttling experiments are made and the temperature t' is observed, the observation can be used as a means of determining κ.   We have the equation

$$\kappa = \frac{\text{Total heat before throttling} - I_{s2}}{t' - t_2}.$$

This method of measuring κ has been employed by various investigators[1], but it presents considerable difficulties, due mainly

[1] Ewing and Dunkerley, *Brit. Assoc. Rep.* 1897 ; J. H. Grindley, *Phil. Trans. Roy. Soc.* vol. 194 A ; A. Griessmann, *Mitteilung aus dem Maschinenlaboratorium der Tech. Hochschule in Dresden,* 1903 ; A. H. Peake, *Proc. Roy. Soc.* vol. 76 A, 1905. Observations of the fall of temperature which steam undergoes when it is throttled

to uncertainty as to the exact condition of dryness of the steam. It is difficult to secure that the steam will be initially dry, and even when this difficulty is overcome it appears that errors may arise through the throttled steam holding particles of moisture in suspension although the vapour containing them is itself super-heated, a condition of complete thermal equilibrium not having been reached. We have assumed throughout that the steam neither gains nor loses heat by conduction or radiation, a con-dition very hard to secure in any actual experiment.

Dry steam escaping at high pressure from a boiler into the atmosphere is superheated near the orifice, when its kinetic energy has been frittered down to heat, but further off it becomes wet by condensation through loss of heat to the air. Before the kinetic energy is frittered down the jet may be wet, because work has been done in setting it in motion. We shall return to this point later in dealing with the theory of jets in relation to the steam turbine.

83. Engine receiving heat at various temperatures. In Carnot's cycle it was assumed that the working substance took all its heat in at the higher limit of temperature τ_1. Im-portant cases arise in which heat is taken in partly at one and partly at other temperatures in a single cycle of operations. With regard to every such quantity of heat the result still applies that the greatest fraction that can be converted into work under ideally favourable conditions is represented by the difference between its temperatures of reception and rejection, divided by the absolute temperature of reception.

Thus if Q_1 represents that part of the whole heat which is taken in at τ_1, and Q_2 represents what is taken in at some other temperature τ_2, Q_3 at τ_3 and so on, and if τ_0 be the temperature at which the engine rejects heat, the whole work done, if the processes within the engine are reversible, is

$$W = \frac{Q_1(\tau_1 - \tau_0)}{\tau_1} + \frac{Q_2(\tau_2 - \tau_0)}{\tau_2} + \frac{Q_3(\tau_3 - \tau_0)}{\tau_3} + \dots \text{ etc.} \dots (4).$$

It is perhaps worth while to point out the analogy here to the supposititious case of a water-wheel working by gravity and receiving water into its buckets at different heights above the

have also been made by Callendar and have been used by him in determining the constants of his characteristic equation for water vapour, on which the formulas and tables in the Appendix are based.

level at which water is discharged from them. Let M_1, M_2 and so on be the quantities of water received at heights l_1, l_2 etc. above any datum level, and let l_0 be the height above the same datum level at which the water leaves the wheel. If the wheel is perfectly efficient (and here again the test of perfect efficiency is reversibility) the work done is

$$M_1 (l_1 - l_0) + M_2 (l_2 - l_0) + M_3 (l_3 - l_0) + \dots \text{ etc.}$$

Comparing the two cases we see that the quantity $\dfrac{Q_1}{\tau_1}$ is the analogue in the heat-engine of M_1 in the water-wheel and so on. The amount of work which can be got out of a given quantity of heat by letting it down to an assigned level of temperature is not simply proportional to the product of the quantity of heat by the fall of temperature, but to the product of $\dfrac{Q}{\tau}$ by the fall of temperature. On the strength of this analogy Zeuner has called the quantity $\dfrac{Q}{\tau}$ the "heat weight" of a quantity of heat Q obtainable at a temperature τ.

Another way of expressing the matter has a wider application. Let the engine as before take in quantities of heat represented by Q_1, Q_2, Q_3 etc. at τ_1, τ_2, τ_3 and let $- Q_0$ represent the heat rejected at τ_0, the negative sign being used to distinguish heat rejected from heat received. Then by the principle that in a reversible cycle the heat rejected is to the heat taken in as the absolute temperature of rejection is to the absolute temperature of reception, we have

$$\frac{- Q_0}{\tau_0} = \frac{Q_1}{\tau_1} + \frac{Q_2}{\tau_2} + \frac{Q_3}{\tau_3} + \dots,$$

from which

$$\Sigma \frac{Q}{\tau} = 0 \dots\dots\dots\dots\dots\dots\dots\dots\dots(5),$$

when the summation is effected all round the reversible cycle. It is clear that this result may be at once extended to cases where heat is given out at various temperatures as well as taken in at various temperatures, Q being taken positive or negative according as heat is being received or rejected.

In cases where changes of temperature are going on continuously while heat is being taken in or given out, we cannot divide the reception or rejection of heat into a limited number of steps, as

has been done above. But the equation may be adapted to this most general case by writing it

$$\int \frac{dQ}{\tau} = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(6),$$

integration being performed round the whole cycle.

84. Application to the case of a steam-engine working without compression, but with complete adiabatic expansion. In § 73 we considered the action of an ideal steam-engine following the Rankine cycle, in which the steam formed at τ_1 was expanded adiabatically and fully, that is to say, down to the pressure corresponding to the temperature of the condenser τ_2 and was there condensed, the condensed water being then restored to the boiler by a feed-pump and thus heated again to τ_1 to complete the cycle. This cycle is specially important in the discussion of steam-engines because it forms the ideally best performance of an engine which uses a feed-pump to return the condensed water directly from the condenser to the boiler. It is what such an engine might achieve provided the expansion were complete, so that there should be no sudden drop of pressure at release, and provided the cylinder and piston were perfect non-conductors. The efficiency in this cycle falls short of the Carnot limit

$$\frac{\tau_1 - \tau_2}{\tau_1}$$

because in the fourth stage of the cycle the working substance has its temperature raised from τ_2 to τ_1, not by adiabatic compression, as in Carnot's cycle (§ 69), but by being brought into contact with the contents of the boiler, which are kept at τ_1. Consequently heat enters it in this stage by a non-reversible process: in all other respects however the cycle is reversible.

But we may regard this as a strictly reversible cycle if we think of the feed-water as taking up its heat by infinitesimal instalments at a series of temperatures ranging from τ_2 up to τ_1 from a series of imaginary sources each of which has the same temperature as the water when the water is brought into contact with it. One may realise this notion by thinking of the feed-pipe as passing through a heated channel the temperature of which is τ_1 close to the boiler and tapers down to τ_2 close to the condenser. Thus the feed-water would have its temperature

raised gradually and would nowhere be brought into contact with a source at a temperature different from the temperature which it had itself then reached. With such an arrangement as this it is clear that the engine becomes a strictly reversible engine, receiving portions of its heat, however, at various temperatures. But the action of the engine is in no way altered by this imaginary arrangement of the feed-pipe, nor is the total supply of heat in any way altered. The notion of gradual heating in the feed-pipe has been introduced merely to show that the cycle is a reversible cycle if we take account of the fact that heat is received not all at the top of the range of temperature, but partly at lower temperatures. Every part of the heat which the substance receives is used in the most efficient possible way, *after it has been taken in*, so that the expression

$$\frac{\tau - \tau_2}{\tau}$$

measures the efficiency of the transformation into work of each portion of the heat, τ being the particular temperature at which the working substance happens to be when it takes in that portion of the heat. The only non-reversible feature in the action of this engine is the flow of heat from a source at τ_1 into the feed-water while the temperature of the feed-water is less than τ_1; and we get rid of this partial non-reversibility by taking as the temperature of reception of each portion of the heat that temperature which the working substance has when the portion in question was taken in. It will be evident that these remarks are of general application, and that when this understanding is accepted, both with regard to the temperatures of reception and rejection of heat, the process in any heat-engine is to be taken as reversible provided the expansions and compressions which occur in it are themselves reversible in the sense which has been explained in § 52. With a source of heat at a given temperature the heat can be turned to account most efficiently only when all the heat is taken in while the working substance is at that temperature, and it is only then that the greatest value of the efficiency, namely, $\frac{\tau_1 - \tau_2}{\tau_1}$, can be reached. But the engine may take in part of its supply of heat at temperatures below τ_1 and still act reversibly in the conversion of the heat so received into work, in which case the efficiency of the

whole action will be less than $\dfrac{\tau_1 - \tau_2}{\tau_1}$ though the general formula $\dfrac{\tau - \tau_2}{\tau}$ is still applicable in respect of every separate portion of the heat, when proper values are assigned to τ.

The ideal steam-engine which we are now considering is a case in point. It takes in the greater part of its heat at τ_1, but some is taken in at temperatures ranging between τ_2 and τ_1. So far as actions occurring within the engine are concerned it is reversible. The amount of heat it converts into work is therefore to be found by calculating

$$\Sigma \frac{\delta Q\,(\tau - \tau_2)}{\tau},$$

where δQ represents any part of the heat taken in and τ the temperature at which it is taken in. The whole heat taken in, per lb. of working substance, is, first, the amount of heat which is required to heat the water from τ_2 to τ_1, namely, $h_1 - h_2$, which is taken in while the temperature is varying, and, second, the latent heat L, which is taken in at the temperature τ_1. During the heating of the water through any small interval $d\tau$ the heat taken in may be expressed as dh or $\sigma d\tau$, where σ is the specific heat of water. Hence the whole amount of work done per unit mass of working steam (expressed in thermal units) is

$$W = \int_{\tau_2}^{\tau_1} \frac{\sigma d\tau\,(\tau - \tau_2)}{\tau} + \frac{L_1(\tau_1 - \tau_2)}{\tau_1}$$

$$= \int_{\tau_2}^{\tau_1} \sigma d\tau - \tau_2 \int_{\tau_2}^{\tau_1} \frac{\sigma d\tau}{\tau} + \frac{L_1\,(\tau_1 - \tau_2)}{\tau_1} \;\ldots\ldots\ldots(7).$$

No serious error is introduced in this calculation if we treat the specific heat of water as constant and equal to unity throughout the range usual in steam-engine practice. Writing $\sigma = 1$ we have

$$W = \tau_1 - \tau_2 - \tau_2 \log_e \frac{\tau_1}{\tau_2} + \frac{L_1\,(\tau_1 - \tau_2)}{\tau_1},$$

or
$$W = (\tau_1 - \tau_2)\left(1 + \frac{L_1}{\tau_1}\right) - \tau_2 \log_e \frac{\tau_1}{\tau_2} \ldots\ldots\ldots\ldots(8).$$

This is the greatest amount of work which can be done, per unit mass of steam, under ideally favourable conditions by an engine which takes in saturated steam from a boiler at temperature τ_1 and restores condensed water to the boiler at tem-

perature τ_2. The result is interesting as affording a standard with which the performance of actual steam-engines may be compared.

This is the ideal cycle of operations already considered in § 73 and now usually called the Rankine cycle. It is adopted as a standard for comparison with the performance of real engines by a Committee of the Institution of Civil Engineers[1]. To compare the results of tests with this standard is more logical and useful than to compare them with the standard of the Carnot cycle, because of the absence of compression in the practical steam-engine cycle. If the action in the real steam-engine were strictly adiabatic, and if no losses occurred through leakage, throttling, or unresisted expansion into clearance spaces the ideal efficiency of the Rankine cycle would be actually attained.

The efficiency of a steam-engine working in the Rankine cycle may be found by dividing the above expression for W by the heat taken in per unit mass of steam, which, for an engine using saturated steam, may be written

$$L_1 + \tau_1 - \tau_2,$$

if, as before, we take the specific heat of water to be equal to unity.

The efficiency of the Rankine cycle for saturated steam may accordingly be expressed in this form

$$\frac{(\tau_1 - \tau_2)\left(1 + \dfrac{L_1}{\tau_1}\right) - \tau_2 \log_e \dfrac{\tau_1}{\tau_2}}{L_1 + \tau_1 - \tau_2}.$$

In this expression the error which arises from neglecting the increase in the specific heat of water at high temperatures is very small, for both numerator and denominator are nearly equally affected.

It is easy to extend the consideration of the Rankine cycle to the case of steam that is superheated before admission to the engine. Let κ be the specific heat for the range through which superheating is carried. For the present purpose we shall treat κ as sensibly constant throughout the range. Then the extra heat taken in is $\kappa (\tau_1' - \tau_1)$, where τ_1' is the temperature of superheat and τ_1 the saturation temperature at the pressure

[1] *Report of Committee on the Thermal Efficiency of Steam-Engines*, 1898.

of supply. In the expression for W there is now the additional term $\int_{\tau_1}^{\tau_1'} \frac{\kappa d\tau (\tau - \tau_2)}{\tau}$, and hence

$$W = (\tau_1 - \tau_2)\left(1 + \frac{L_1}{\tau_1}\right) + \kappa(\tau_1' - \tau_1) - \tau_2\left(\log_e \frac{\tau_1}{\tau_2} + \kappa \log_e \frac{\tau_1'}{\tau_1}\right).$$

The efficiency is found by dividing this expression by

$$\tau_1 - \tau_2 + L_1 + \kappa(\tau_1' - \tau_1).$$

As a numerical example of this method of working out the efficiency in a Rankine cycle, take the case of an engine using saturated steam with a supply temperature of 180° C. (pressure 146 lbs. per sq. inch or 10·26 kg. per sq. cm.), and condensing it at 40° C., and calculate the ideal performance when there is complete adiabatic expansion from the top to the bottom of this range. Here τ_1 is 453, τ_2 is 313, L_1 is 483·7, and hence the expression for W (equation 8) gives 173·8 thermal units as the equivalent of the work ideally obtainable per unit mass of steam. The supply of heat, reckoned as $L_1 + \tau_1 - \tau_2$, is 623·7. The efficiency is therefore 0·279. Compare this with the number 0·309 which represents the value of $\frac{\tau_1 - \tau_2}{\tau_1}$; namely, the efficiency of a reversible (Carnot) cycle completed by adiabatic compression as in the engine of § 69. The absence of adiabatic compression has in this case reduced the efficiency by about 11 per cent. This comparison shows what is lost by the partial misapplication of heat which results from letting the feed-water come into the boiler cold, to be heated by contact with the hot water already there, so that the portion of the heat-supply received at that stage is taken in at temperatures lower than the top of the range.

The expression for W in Eq. 8 serves to show how much work it is ideally possible to get from steam when the temperature of the boiler and the temperature of the condenser are assigned. But it is further useful as a standard with which we may compare the action of the steam cylinder, taken by itself, without reference to what has happened before the steam reaches the cylinder, or to what happens in the exhaust pipe and in the condenser. For that purpose τ_1 may be understood as the temperature of the steam on reaching the cylinder and τ_2 the temperature of the steam on leaving the cylinder, although these temperatures may differ from those of the boiler and condenser respectively. Then the formula

serves to show the limiting amount of work obtainable, which the actual performance of the steam must be expected to fall short of, and the calculation furnishes a useful check on the results of engine trials.

We shall see later (§ 100) that it is possible to arrive at the numerical value of W much more simply by a graphic process, taking advantage of the fact already mentioned in § 73 that the work done by the steam in the ideal process is equal to $I_1 - I_2$, where I_1 is the total heat of the steam as supplied and I_2 is the total heat of the fluid after expansion, the fluid being then a mixture of steam and water.

85.   Extension to the case of steam not initially dry. The result arrived at in the last paragraph may be readily extended to cases where the steam is not dry when the adiabatic expansion begins.   Let q_1 be the dryness at this stage: then the heat taken in during evaporation is $q_1 L_1$ per unit mass of working substance, but the heat taken in during the heating of the water up to τ_1 remains what it was before.   The expression for the work done (assuming complete adiabatic expansion as before) is therefore found by substituting $q_1 L_1$ for L_1 in equation (7) or (8), giving,

$$W = \tau_1 - \tau_2 - \tau_2 \log_e \frac{\tau_1}{\tau_2} + \frac{q_1 L_1 (\tau_1 - \tau_2)}{\tau_1} \ldots\ldots\ldots\ldots(9).$$

86.   Derivation of the adiabatic equation from this result.   This result may be applied to prove the equation for the adiabatic expansion of steam which was stated, without proof, in § 67.   The whole heat taken in, per unit of mass, in raising the water from any temperature τ_2 to τ_1 and in evaporating the fraction q_1 of it at the temperature τ_1 is (if we take the specific heat of water as unity)

$$\tau_1 - \tau_2 + q_1 L_1.$$

By expanding this mixture adiabatically to the temperature τ_2 and then condensing it, we get an amount of work equal (by the equation which has just been given) to

$$\tau_1 - \tau_2 + q_1 L_1 - \frac{q_1 L_1 \tau_2}{\tau_1} - \tau_2 \log_e \frac{\tau_1}{\tau_2}.$$

Hence, subtracting the work done from the heat supplied we find that the heat rejected is

$$\frac{q_1 L_1 \tau_2}{\tau_1} + \tau_2 \log_e \frac{\tau_1}{\tau_2}.$$

But the only rejection of heat in the cycle takes place during the condensation at τ_2 after adiabatic expansion, and the amount of heat so rejected is

$$q_2 L_2,$$

where q_2 is the dryness after adiabatic expansion to the temperature τ_2.

Hence
$$q_2 L_2 = \frac{q_1 L_1 \tau_2}{\tau_1} + \tau_2 \log_e \frac{\tau_1}{\tau_2}$$

or
$$\frac{q_2 L_2}{\tau_2} = \frac{q_1 L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau_2}.$$

Now τ_2 may be any temperature lower than τ_1, for the adiabatic expansion might be stopped at any point along the curve and the cycle completed by condensing the mixture at the temperature it had then reached. Hence this equation serves to show in a perfectly general way the change of dryness which takes place during adiabatic expansion, and, dropping the second suffix, we may write it

$$\frac{qL}{\tau} = \frac{q_1 L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau} \dots\dots\dots\dots\dots\dots(10),$$

which is the same as equation (10) in § 67. It is to be noticed that in deriving this expression the specific heat of water has been treated as constant. The result is therefore (to a very small extent) inexact, especially at high temperatures.

87. Heat-Engines employing more than one working substance. So far as general thermodynamic principles are concerned the choice of working substance in a heat-engine is indifferent. The same efficiency is given by one substance as by another provided the character and range of the cycle be the same. But the consideration that the pressure should be neither excessively high nor excessively low often determines whether one or another working substance is to be preferred. Vaporisable liquids have the advantage over air or any other permanent gas that heat can be more readily communicated to and extracted from them; but any such liquid has a comparatively limited range of temperature within which it is practicable for it to work. The efficiency of the steam-engine is, as we have already seen, largely conditioned by the fact that even when steam is highly superheated most of the heat is necessarily taken in at a comparatively low temperature. This

prevents full advantage being taken of the high-temperature heat
which is generated in the combustion of fuel in boiler furnaces;
and in this respect an air-engine has the superiority that in it
a much higher temperature can be reached, since in a gas the
connection of pressure with temperature is arbitrary.   On the
other hand a more volatile liquid than water would be even less
suitable for use at high temperatures, unless indeed a large amount
of superheating were employed.   Going to the other end of the
range, it will be seen by reference to the table of pressure and
temperature for steam that a steam-engine of the cylinder and
piston type is not well fitted to take full benefit of the low
temperature which may be reached when there is condensing
water at hand.   A more volatile liquid would do this better,
because its vapour could be expanded to the bottom of the range
of temperature without making the pressure fall inconveniently
low.   With steam complete expansion in a piston and cylinder
engine would be useless, because in the last stages of the ex-
pansion the pressure would barely suffice to move the piston
against its own frictional resistance; and therefore the *indicated*
work which would be saved by completing the expansion would
contribute nothing to the output of the engine.   Apart from
this fundamental difficulty, complete expansion would be imprac-
ticable owing to the excessive bulk required in the cylinder.
One of the great merits of the steam turbine is that it enables
the last stages of expansion to be effective for the performance
of useful work, for in the turbine the difficulty as to friction
does not arise and it is practicable to deal with the large volume
which the steam assumes when its pressure approaches the
pressure in the condenser.   In the reciprocating steam-engine
considerations both of friction and bulk make it impracticable
to work the steam down nearly so far, and a considerable part
of the theoretically possible output $I_1 - I_2$ is lost because of the
high temperature at which steam is discharged.

For this reason it has been proposed to use what is called
a "binary" heat-engine, that is, an engine with two working fluids,
one to work through the upper part of the range, and another—a
more volatile fluid—to work through the lower part.   The less
volatile fluid, namely water, after being evaporated in a boiler and
after doing work in its cylinder, is condensed by passing through
tubes in a vessel containing the more volatile fluid, to which it

rejects heat. The more volatile fluid is thereby evaporated and does work in another cylinder, after which it is passed into a surface-condenser supplied with cold circulating water. A binary engine using ether as the more volatile fluid was introduced by Du Tremblay about 1850[1], and the type has more than once been revived on a small scale[2].

[1] See *Min. Proc. Inst. C. E.* vol. xviii. p. 233. Also Rankine's *Steam-Engine*, p. 444.

[2] See *Min. Proc. Inst. C. E.* vol. cxii. 1893, pp. 481, 482.

CHAPTER V.

88. Entropy. When a substance receives or rejects heat at any temperature it is said to change its *entropy*. In the definition of entropy any heat generated within the substance itself, as by combustion or friction, as well as any heat taken in from outside, is to be regarded as heat received. The quantity of heat received or rejected, divided by the absolute temperature of the substance at the time, measures the change of entropy. Thus if a substance takes in δQ of heat at any absolute temperature τ, its entropy increases by the amount $\dfrac{\delta Q}{\tau}$. The entropy of a substance in any definite state is a definite quantity, to be reckoned per unit of mass. Just as in dealing with total heat, we take an arbitrary starting point at which the entropy of the substance is reckoned as zero. Thus in reckoning the entropy of water or of steam in any given state, we take the condition of water at 0° C. as the starting point, and calculate the entropy by considering what heat is taken in, and at what temperatures, in passing from that to the state at which the substance is. Each element δQ of the heat taken in is to be divided by τ, which is the absolute temperature of the substance when that element is taken in, and the sum

$$\Sigma \frac{\delta Q}{\tau},$$

reckoned from the starting point till the given state is reached, then measures the entropy. When any quantity of heat δQ is given out at temperature τ the entropy is reduced by the amount $\dfrac{\delta Q}{\tau}$. The entropy of water, per unit of mass, at any temperature τ_1, is $\displaystyle\int_{\tau_0}^{\tau_1} \frac{\sigma d\tau}{\tau}$, where σ is the specific heat of water and τ_0 is 273. In passing from the condition of water at any temperature τ to steam

at the same temperature the substance takes in a quantity of heat equal to L: consequently the entropy increases by $\dfrac{L}{\tau}$.

We shall denote entropy by ϕ. Using ϕ_s for the entropy of saturated steam at any temperature τ and ϕ_w for the entropy of water at the same temperature we have

$$\phi_s = \phi_w + \frac{L}{\tau}.$$

It follows from this definition of entropy that when any substance expands or is compressed in an adiabatic manner its entropy does not change. An adiabatic line in the pressure-volume diagram is consequently a line of constant entropy, or as it is sometimes called, an *isentropic* line.
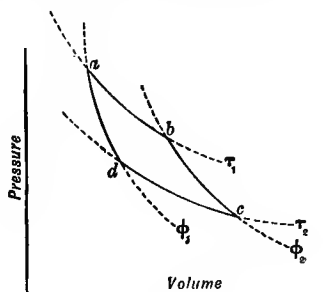
Fig. 25.

Consider now a cycle consisting of two isothermal and two adiabatic operations, fig. 25. In passing from a to b by the isothermal line τ_1 the substance gains entropy $\dfrac{Q_1}{\tau_1}$, where Q_1 is the heat taken in during this operation. Along the adiabatic line from b to c there is no change of entropy. In the isothermal line cd the entropy is reduced by $\dfrac{Q_2}{\tau_2}$, and from d to a there is again no change of entropy. Now $\dfrac{Q_1}{\tau_1} = \dfrac{Q_2}{\tau_2}$, which means that the entropy changes by the same amount whether we pass from one adiabatic line ad to another adiabatic line bc by one isothermal path ab or by any other isothermal path dc. And moreover the change of entropy between one adiabatic and another will be the same whether the cross-path be isothermal or not, for a curve expressing

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any relation between P and V may be regarded as made up of a succession of minute isothermal and adiabatic elements, and the change of entropy along such a curve is the sum of the changes which occur during the isothermal elements of the process, and is still equal to $\dfrac{Q}{\tau}$ for any single isothermal path between the same pair of adiabatic lines.

We see, then, that not only is there no change of entropy during an adiabatic process, but there is a perfectly definite change of entropy when a given substance passes from one adiabatic line to another, by whatever path. Just as isothermal lines can be distinguished by numbers τ_1, τ_2, etc. denoting the particular temperature for which each is drawn, so adiabatic lines can be distinguished by numbers ϕ_1, ϕ_2, etc. denoting the particular value of the entropy on each. The conception of entropy as that characteristic of a substance which does not change during adiabatic expansion or compression is of considerable service in problems relating to heat-engines. We shall see presently some of the uses to which this notion may be put.

When a substance expands in an *irreversible* manner, as by passing through a throttle valve from a region of high pressure to a region of lower pressure it gains entropy. Work is then done by the substance on itself, namely in giving energy of motion to each particle as it passes through the valve, and this energy of motion is frittered down into heat as the motion subsides through internal friction. The effect is therefore equivalent to the communication of a quantity of heat, though no heat reaches the substance from outside, and our definition of entropy is to be understood as including heat acquired in such a way as this. We may regard expansion through a throttle valve as consisting of two stages. The first stage is adiabatic expansion in which the substance does work in setting itself in motion : the second stage is the loss of this motion and the acquiring of an equivalent amount of heat. There is accordingly a gain of entropy, though no heat is taken in from outside.

At the end of any cycle, whether reversible or not, the substance has returned to its initial state in all respects, and its entropy has recovered the initial value. The quantity

$$\Sigma \frac{\delta Q}{\tau}$$

must therefore be zero, when reckoned throughout the complete
cycle, if for every irreversible stage we include in the reckoning
the heat-equivalent of the work dissipated through internal
friction. Any such quantity of heat is necessarily positive, and
therefore if we were to confine our attention to the quantities
of heat actually taken in from outside, or rejected, the sum

$$\Sigma \frac{\delta Q}{\tau}$$

would not be zero, but would be negative for a cycle involving any
irreversible process. We have already seen (§ 83) that when the
cycle is reversible and there are therefore no quantities of heat to
be considered except those that enter the substance from outside
or are given out by it the sum $\Sigma \frac{\delta Q}{\tau}$ is zero in the complete cycle.

89. Sum of the Entropies in a System. It is instruc-
tive, in this connection, to inquire how the sum of the entropies
of all parts of the system is affected when we include not only
the working substance but also the source of heat and the sink
or receiver to which heat is rejected. Consider a cyclic action
in which the working substance takes in a quantity of heat Q_1
from a source at τ_1 and rejects a quantity Q_2 to a sink at τ_2.
When the cycle is completed the source has lost entropy to
the amount $\frac{Q_1}{\tau_1}$: the working substance has returned to the
initial state, and therefore has neither gained nor lost entropy:
the sink has gained entropy to the amount $\frac{Q_2}{\tau_2}$. If the cycle is a
reversible one $\frac{Q_1}{\tau_1} = \frac{Q_2}{\tau_2}$, and therefore the system taken as a whole
consisting of source, substance and sink has suffered no change in
the sum of the entropies of its parts. But if the cycle is not
reversible the action is less efficient, Q_2 bears a larger proportion
to Q_1 and $\frac{Q_2}{\tau_2}$ is greater than $\frac{Q_1}{\tau_1}$. Hence in an irreversible action
the sum of the entropies of the system as a whole becomes
increased. This conclusion is a perfectly general one: it may
be expressed in general terms by saying that when a substance
undergoes any change the sum of the entropies of the substance
and of such other substances as take part in the action remains

unaltered if the action is reversible, but becomes increased if the action is not reversible. No real action is strictly reversible, and hence any real action occurring within a system of bodies has the effect of increasing the sum of the entropies of the bodies which make up the system. This is a statement, in terms of entropy, of the principle that there is in all actual transformations of energy a universal tendency towards what Lord Kelvin called the dissipation of energy. The sum of the entropies in any system, considered as a whole, tends towards a maximum, which would be reached if all the energy of the system were to take the form of uniformly diffused heat, and if this state were reached no further transformations would be possible. Any action within the system, by increasing what may be called the aggregate entropy, which is obtained by summing the entropies of the various parts, brings the system a step nearer to this state, and to that extent diminishes the availability of the energy in the system for further transformations.

As an extreme case of an irreversible action take the direct conduction of a quantity of heat Q from a hot part of the system, at τ_1, to a colder part at τ_2. The hot part loses entropy by the amount $\dfrac{Q}{\tau_1}$: the cold part gains entropy by the amount $\dfrac{Q}{\tau_2}$, and as the latter is greater there is an increase of the aggregate entropy in the system as a whole.

90. Entropy of Wet Steam: Derivation of the Adiabatic Equation. Reckoning from water at any initial temperature τ_0 the entropy of steam (taken wet, for greater generality)

$$\phi = \int_{\tau_0}^{\tau_1} \frac{\sigma d\tau}{\tau} + \frac{q_1 L_1}{\tau_1}.$$

The first term represents the entropy which is acquired during the heating of the water from τ_0 to τ_1, which is the temperature of evaporation, and the second term represents what is acquired during evaporation, q_1 being the dryness of the steam. Treating the specific heat of water as unity we can write $d\tau$ for $\sigma d\tau$; then integrating,

$$\phi = \log_e \tau_1 - \log_e \tau_0 + \frac{q_1 L_1}{\tau_1} \dots\dots\dots\dots(11).$$

Now in adiabatic expansion we have

$$\phi = \text{constant},$$

and hence if the steam be expanded adiabatically to any temperature τ

$$\log_e \tau - \log_e \tau_0 + \frac{qL}{\tau} = \log_e \tau_1 - \log_e \tau_0 + \frac{q_1 L_1}{\tau_1},$$

from which $\qquad \frac{qL}{\tau} = \frac{q_1 L_1}{\tau_1} + \log_e \frac{\tau_1}{\tau_2},$

which is the adiabatic equation of § 67, already derived by another and longer method in § 86.

The entropy of superheated steam will be considered later (§§ 93 and 99).

91. Entropy-Temperature Diagrams. The familiar way to represent graphically those changes which a working substance undergoes in the action of a heat-engine is to draw the indicator diagram, which shows pressure in relation to volume. Another way is to draw a diagram showing the relation of the temperature of the substance to its entropy. Diagrams of this kind form an interesting and often useful alternative to the ordinary indicator diagram[1]. Let $\delta\phi$ be the small change in entropy which a substance undergoes when it takes in any small quantity δQ of heat at any temperature τ. By the definition of entropy

$$\delta\phi = \frac{\delta Q}{\tau},$$

whence $\qquad \tau\delta\phi = \delta Q$

and $\qquad \int\tau d\phi = \int dQ$ ....(12),

the integration being performed between any assigned limits. Now if a curve be drawn with τ and ϕ for ordinates, $\int\tau d\phi$ is the area under the curve. This by the above equation is equal to $\int dQ$, in other words, the area under any portion of the entropy-temperature curve is equal to the whole quantity of heat taken in while the substance passes through the states which that portion of the curve represents. Let ab, fig. 26, be any portion of the curve of ϕ and τ. The area of the cross-hatched strip whose

[1] Entropy-Temperature diagrams were described along with other graphic methods in thermodynamics by Professor J. Willard Gibbs (*Trans. of the Connecticut Acad. of Sciences*, vol. II. 1873, p. 309). Their application to steam-engine problems is mainly due to Mr J. Macfarlane Gray (see *Proc. Inst. Mech. Eng.* 1889, p. 399). Professor Boulvin (*Cours de Mécanique appliquée : Théorie des Machines thermiques*) ascribes their earliest use to M. Th. Belpaire (*Bulletin de l'Académie royale de Belgique*, 1872, v. 34).

breadth is $\delta\phi$ and height τ, is $\tau\delta\phi$, which is equal to δQ, the
heat taken in during the small change
$\delta\phi$. The whole area *mabn* or $\int\tau d\phi$
between the limits *a* and *b* is the
whole heat taken in while the sub-
stance changes from the state repre-
sented by *a* to the state represented
by *b*. Similarly, in changing from
state *b* to state *a* by the line *ba* the
substance rejects an amount of heat
which is measured by the area *nbam*.
The base line *ox* corresponds to the
absolute zero of temperature.
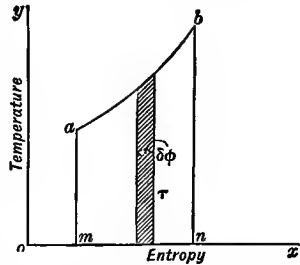
Fig. 26.   Entropy-Temperature
Curve.

When an entropy-temperature curve is drawn for a complete
cycle of changes it forms a closed figure, since the substance
returns to its initial state.   To find the area of the figure we have
to integrate throughout the complete cycle, when

$$\int\tau d\phi = Q_1 - Q_2,$$

Q_1 being the heat taken in and Q_2 the heat rejected[1].   But the
difference between these is the heat converted into work, hence

$$\int\tau d\phi = W \dots\dots\dots\dots\dots\dots\dots(13),$$

when the integration extends round a complete cycle and W is
expressed in thermal units.   Thus entropy-temperature diagrams
have the important property in common with pressure-volume
diagrams that the enclosed area measures the work done in a
complete cycle.

Isothermal lines on an entropy-temperature diagram are
straight lines parallel to *ox* whatever
be the working substance : adiabatic
lines are straight lines parallel to *oy*,
being lines of constant entropy.   Hence
Carnot's cycle, whether with air or
steam or any other substance, would
be represented by a rectangle *abcd*,
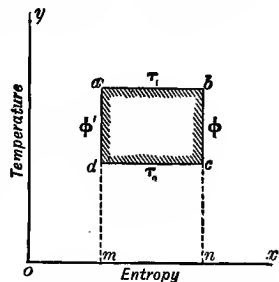fig. 27, in which the heat received

$$Q_1 = \text{area } mabn = \tau_1 (\phi - \phi'),$$

heat rejected

$$Q_2 = \text{area } ncdm = \tau_2 (\phi - \phi'),$$

Fig. 27.   Carnot's Cycle on
the Entropy-Temperature
Diagram.

[1] We are dealing here with cycles in which there is no production of heat
within the fluid, as by throttling.

and work done
$$W = \text{area } abcd = (\tau_1 - \tau_2)(\phi - \phi'),$$
ϕ being the entropy in the adiabatic process of expansion and ϕ' the entropy in the adiabatic process of compression. The efficiency is
$$\frac{\text{area } abcd}{\text{area } mabn} = \frac{\tau_1 - \tau_2}{\tau_1}.$$

92. Entropy-Temperature Diagram for Steam; application to ideal steam-engine working without compression but with complete expansion. A more interesting example of the use of entropy-temperature diagrams is given by the engine of § 73 using the Rankine cycle of operations. In that cycle, after complete adiabatic expansion from τ_1 to τ_2, the steam is condensed isothermally at τ_2, and is then returned as water to the boiler. In drawing the diagram for the cycle we shall begin at the point where the water, at τ_2, is about to be heated. Reckoning from the standard temperature τ_0 and dealing throughout with unit mass of the working fluid, we have

Entropy of water at any temperature $\tau = \displaystyle\int_{\tau_0}^{\tau} \frac{\sigma d\tau}{\tau}$,

where σ is the specific heat of water. The specific heat is sensibly equal to unity at low temperatures and becomes only a very little more than unity at high temperatures. Neglecting this small change, we may write, as a good approximation[1],

Entropy of water $= \displaystyle\int_{\tau_0}^{\tau} \frac{d\tau}{\tau} = \log_e \tau - \log_e \tau_0,$

which relates to any stage in the heating of the feed-water from τ_2 to τ_1. The first part of the diagram is therefore a logarithmic curve, ab, fig. 28, where the temperature at a is τ_2, the temperature at b is τ_1, the entropy at a is $\log_e \tau_2 - \log_e \tau_0$, and the entropy at b is $\log_e \tau_1 - \log_e \tau_0$. Hence the change of entropy which takes place during the heating of the feed-water, $\phi_b - \phi_a$ or mn, is $\log_e \tau_1 - \log_e \tau_2$. It is a matter of indifference, in the drawing of the diagram, at what distance the origin is taken to the left of m; in other words, what value of τ_0 is taken as a datum in reckoning the entropy, but the usual practice is followed of reckoning the entropy of water at 0° C. to be zero, so that τ_0 is 273. At b steam begins to be formed, and bc is the change of entropy which the

[1] In the Appendix it is shown how more exact values of the entropy of water are found by expressing σ as a function of the temperature. The values of the entropy stated in the Tables take account of the variation in the specific heat.

substance undergoes in passing from water to steam at the constant temperature τ_1. bc is therefore equal to $\dfrac{L_1}{\tau_1}$, assuming the evaporation to be complete. If the evaporation were incomplete, bc would be equal to $\dfrac{q_1 L_1}{\tau_1}$. The adiabatic process of complete expansion from the condition of saturated steam at τ_1 down to the temperature τ_2 is represented by cd, and da is the process of condensation which completes the cycle.

The heat taken in during the heating of the feed-water to the boiler temperature is the area $mabn$. The heat taken in during evaporation is $nbcp$. The work done is the enclosed area $abcda$.
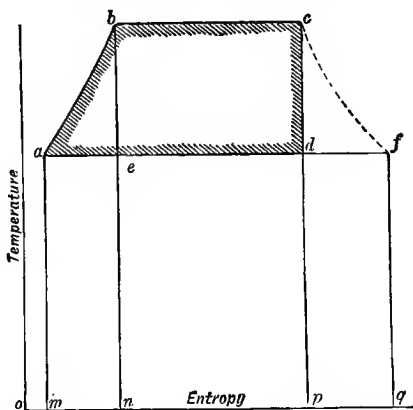
Fɪɢ. 28.   Entropy-Temperature Diagram for Steam.

The heat rejected is $pdam$. Of the heat taken in during the process of evaporation, namely, the area bp, the part measured by the area bd is converted into work: it represents the fraction $\dfrac{be}{bn}$ or $\dfrac{\tau_1 - \tau_2}{\tau_1}$ of the whole, as we should expect. Of the heat taken in during the process of warming the feed-water a smaller fraction is converted into work, namely, the fraction $\dfrac{abe}{mabn}$. This is because the heat is less advantageously supplied during this operation, the temperature being then less than τ_1. An engine going through Carnot's cycle would have the diagram $ebcd$. The engine of the Rankine cycle does more work (by the area abe), but to do this it has to take in a more than proportionally larger amount

of heat and is therefore less efficient. The heat which it takes in is the area *mabcp*, against the area *nbcp* for the Carnot cycle. It will be seen that the diagram exhibits in a very simple way results which we have already arrived at by other routes.

Further, let a curve *cf* be drawn such that the distance from any point in *ab* to it, measured horizontally (that is, parallel to *op*), is equal to the value of $\frac{L}{\tau}$ corresponding to that point. Thus let *af* be equal to $\frac{L_2}{\tau_2}$, *f* being the point where *ad* produced meets this curve. This curve may be called the *saturation* curve, for it shows the entropy of saturated steam at various temperatures. Then the dryness of the steam after the process of
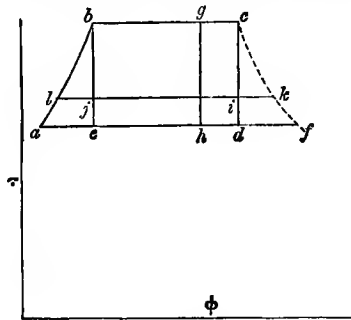
FIG. 29.

adiabatic expansion represented by *cd* is given by the fraction $\frac{ad}{af}$. This follows from the fact that if the steam were perfectly dry at τ_2 the heat given out during its condensation would be equal to the area *qfam*, whereas the heat actually given out is equal to *pdam*. In other words, the former area is L_2 and the later is $q_2 L_2$, q_2 being the dryness when the stage *d* is reached, whence $q_2 = \frac{ad}{af}$. In the same way a straight line drawn horizontally from any point *i* in *cd* (fig. 29) to meet the curves *cf* and *ab* is divided by *cd* into segments *il* and *ik*. These are proportional to the quantities of steam and water respectively which make up the working substance when the expansion has advanced as far as the point *i*. In other words, at *i* the dryness $q = \frac{li}{lk}$. The temperature-

entropy diagram thus affords a convenient method of finding q graphically at any stage in adiabatic expansion.

Further, suppose the steam has not been dry when adiabatic expansion begins. This state of things is represented on the diagram by making the horizontal line from b terminate at a point g such that $bg = \dfrac{q_1 L_1}{\tau_1}$: in other words, $\dfrac{bg}{bc} = q_1$. The line gh now represents the process of adiabatic expansion and the construction just described is still applicable to find q at any stage. Thus at h, $q = \dfrac{ah}{af}$, and $\dfrac{hf}{af}$ is the proportion then present as water.

Again, reverting to the Carnot cycle of fig. 16, § 69, we can use the entropy-temperature diagram to determine the point at which condensation at τ_2 must be stopped in that cycle in order that adiabatic compression may bring the substance to the state of water at τ_1. The process of compression required for this is eb (fig. 28 or 29), and hence compression must begin when the proportion of steam still uncondensed is $\dfrac{ae}{af}$. Similarly the fraction $\dfrac{lj}{lk}$ measures the dryness at any stage j of this adiabatic compression.

93. Application of the entropy-temperature diagram to the case of superheated steam. The entropy of steam superheated to any temperature τ' is greater than the entropy in the saturated state at the same pressure by the amount

$$\int_{\tau_1}^{\tau'} \frac{\kappa d\tau}{\tau},$$

where κ is the specific heat of the steam during superheating, that is to say, the amount of heat required to raise unit mass of the steam $1°C$. when its temperature exceeds τ_1 the temperature of saturation, the process of superheating being conducted at constant pressure. It has been pointed out in § 65 that κ is not strictly constant, but its variations, for any given constant pressure, are not very great, and an approximate reckoning of the entropy may be made by treating κ as a constant with the value given in the table in § 65 for the mean specific heat over the range through which superheating occurs. Using this approximate method we should have, for the entropy after superheating at

constant pressure from τ_1 the temperature of saturation to τ',

$$\phi = \phi_s + \kappa (\log_e \tau' - \log_e \tau_1),$$

where ϕ_s is the entropy of dry saturated steam at the same pressure.

It is, however, preferable to calculate the entropy of superheated steam directly by means of the Callendar-Mollier expression for entropy given in the appendix, namely

$$\phi = 0.477 \log_e \tau - 0.11 \log_e P - ZP - 1.0544,$$

where τ is the absolute temperature, P is the pressure in kilogrammes per square centimetre, and Z is a quantity (depending on the temperature) whose value is given in the Auxiliary Table (Table D, Appendix). Examples of the figures for entropy derived from this equation are given below for pressures of 5, 10, 15, and 20 kilos. per sq. cm. respectively.

P kilos./cm.²	Sat. Temp. ° C.	ϕ_s from table	Entropy in the Superheated state, at Temp.			
			250°	300°	350°	400°
5	151.0	1.636	1.748	1.794	1.835	1.872
10	178.9	1.582	1.665	1.713	1.756	1.794
15	197.2	1.550	1.661	1.664	1.708	1.747
20	211.3	1.527	1.576	1.628	1.673	1.714

Having found the entropy in the superheated state, we can extend the entropy-temperature diagram in the manner shown in fig. 30, where starting from c which represents dry saturated steam at a particular pressure, cr is drawn to show the increased amount of entropy produced by superheating as calculated for a series of temperatures above the temperature of saturation.

We may now apply this entropy-temperature diagram as a means of representing the ideal action of an engine supplied with superheated steam. Starting as before with feed-water at τ_2 whose temperature and entropy are represented by a, ab is the process of heating the feed-water up to the temperature at which steam is formed, bc is the process of evaporation, cr is the subsequent superheating, during which an extra amount of heat is taken in represented by the area under cr, namely $pcru$.

After superheating to any extent let the cycle be completed by the processes rs and sa, namely, by adiabatic expansion to temperature τ_2 and condensation at that temperature. The diagram shows that, in consequence of superheating, the work done by the substance is increased by the area $dcrs$, while the heat taken in is increased by $pcru$. The efficiency is slightly increased, since this additional heat is received at temperatures somewhat higher than those at which the other portions of the heat were received. But unless superheating be carried very far the extra supply of heat is too small a part of the whole to make any large difference in the efficiency of the ideal engine we are dealing with here. In the case sketched in fig. 30 the steam is
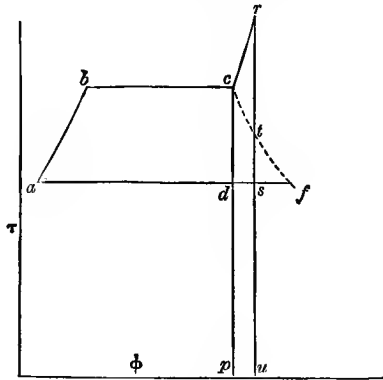
FIG. 30.

supposed to be superheated as much as 110 degrees centigrade above the boiler temperature, but the diagram shows that even this makes but little improvement in the ideal efficiency. In real engines superheating does make a marked difference, but its influence is indirect, and proceeds from the fact that it tends to prevent the steam from being condensed by contact with the metal of the cylinder and piston. This effect of superheating will be considered in the next chapter. Nothing of the kind takes place in the ideal case now dealt with, because here we postulate adiabatic expansion, or, in other words, a perfectly non-conducting cylinder and piston.

It would evidently be fallacious to suppose that when superheating is applied to the steam of the ideal engine the increased range of temperature implies anything like a corresponding gain

of efficiency, for the chief part of the heat is still taken in at the temperature of saturation, and its value for conversion into work depends on the temperature at which it is taken in, not upon the temperature to which the working substance is subsequently raised.

In the diagram, fig. 30, the adiabatic line rs shows by its intersection of the curve cf at t the stage in the expansion at which the steam will cease to be superheated. At the point t it is dry and saturated: as the expansion proceeds it becomes wet, and at the end of expansion the condensed part is $\dfrac{sf}{af}$ of the whole. The extent to which superheating has to be carried if the steam is just to be dry, and no more, at the end of expansion, is readily found by drawing a vertical line through f to meet the continuation of the curve cr.

94. Work done in the Rankine Cycle. We may calculate the work done in the Rankine cycle from the entropy-temperature diagram as follows : Take for greater generality the case where the steam is superheated. The work done in the cyclic process is measured by the area $abcrs$ (fig. 30). This is equal to the area under $abcr$, down to the base line of zero absolute temperature, *minus* the area under af, *plus* the area under sf. Accordingly we may calculate the work done throughout the cycle, per unit mass of the working substance, as

$$H_1 - H_{s2} + \tau_2 (\phi_{s2} - \phi_1)$$

in heat units, where H_1 is the heat of formation of the steam in the state in which it reaches the engine, and ϕ_1 its entropy, and H_{s2} and ϕ_{s2} are the heat of formation and entropy respectively of dry *saturated* steam at the final temperature τ_2.

The work done *in the cylinder* is greater than the work done throughout the cycle by the quantity $A(P_1 - P_2)w$ which measures, in heat units, the work spent on the substance in the feed pump. Adding this to the above expression we obtain, for the work done in the cylinder, the expression

$$H_1 + AP_1w - (H_{s2} + AP_2w) + \tau_2(\phi_{s2} - \phi_1),$$

or $$I_1 - I_{s2} + \tau_2(\phi_{s2} - \phi_1),$$

where I_1 is the total heat of the steam as supplied and I_{s2} is the total heat of saturated steam at the temperature τ_2. This is

equivalent to the general expression already given (§§ 73, 74), namely,

$$I_1 - I_2,$$

because I_2, the total heat of the wet mixture after expansion, is less than the total heat of saturated steam at the same temperature by an amount equal to the area under sf. or $\tau_2(\phi_{s2} - \phi_1)$.

If the steam is not superheated r coincides with c and s with d. The work in the Rankine cycle is the area $abcda$ of fig. 28, and is still given by the expression

$$H_1 - H_{s2} + \tau_2(\phi_{s2} - \phi_1),$$

and the work done in the cylinder is still

$$I_1 - I_{s2} + \tau_2(\phi_{s2} - \phi_1).$$

The dryness after expansion may evidently be expressed thus in terms of the entropies,

$$q = \frac{as}{af} = \frac{\phi_1 - \phi_{w2}}{\phi_{s2} - \phi_{w2}},$$

where ϕ_{w2} is the entropy of water at the lower temperature. This also applies whether the steam is or is not superheated.

95. Values of the Entropy of Water and Steam: Entropy-temperature Chart. In applying this useful graphic method to the investigation of particular cases in the expansion of steam it is convenient to have an entropy-temperature chart for water and steam drawn on section-paper throughout the range of pressures which are found in practice: the construction for particular cases is then readily made by adding horizontal straight lines to correspond with the formation and condensation of the steam, while any adiabatic process is represented by a vertical line. The student will find it instructive to draw for himself, to a large scale, a chart of this kind, using the values given in the table for ϕ_w and ϕ_s, and calculating the entropy of superheated steam at various pressures and temperatures so as sufficiently to map out the region of superheating in the chart.

Referring to the diagram (fig. 31), the curve on the left marked "water" shows the relation of entropy to temperature before steam begins to form: the curve on the right marked "steam boundary curve" shows the same relation when all the water is converted into steam. Between these two curves is what

may be called the wet steam region. The horizontal distance between the two curves at any point, or $\phi_s - \phi_w$, represents the gain of entropy which occurs while the water is changing into steam (namely L/τ). The *Boundary curve* separates the saturated and superheated states. To the right of it lies the region of superheating which may most conveniently be mapped out by drawing a system of *Lines of Constant Pressure*, four re-
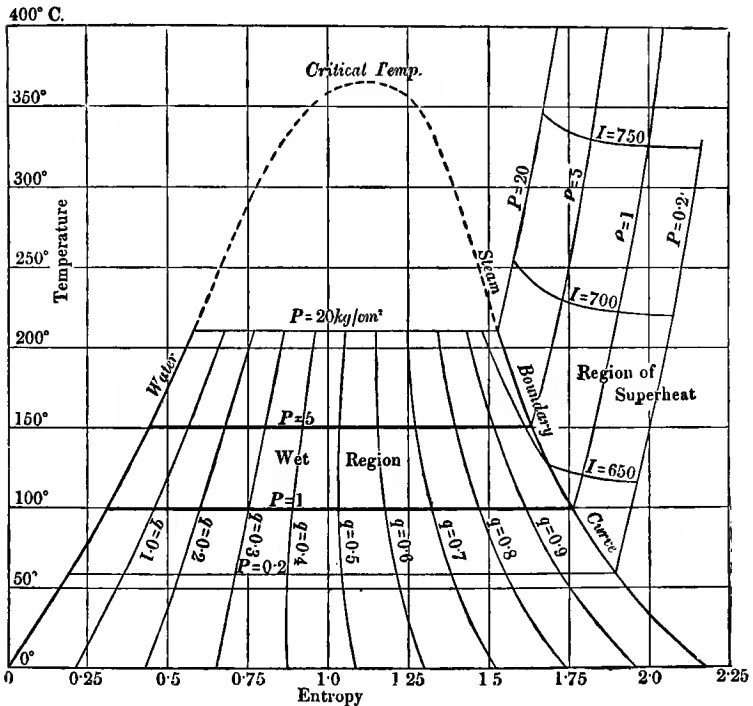

Fig. 31. Entropy-temperature Diagram for Water and Steam.

presentative examples of which are shown in the figure, namely for $P = 20$, 5, 1, and 0·2 kilogrammes per square centimetre, corresponding to 284·5, 71·1, 14·22, and 2·84 lbs. per square inch. In the wet steam region the constant pressure lines are horizontal straight lines (of uniform temperature). To the left of this the constant pressure lines are practically coincident with the "water" curve. Each constant pressure line may be regarded as running up almost along that curve, it then continues horizontally to the steam boundary curve, and then rises in the manner shown in the region of

E. 10

superheating. Such a line shows completely the changes which the entropy undergoes as the water is successively heated, evaporated and superheated, all at constant pressure. When a sufficient number of constant pressure lines are drawn it is easy to mark at once on the chart a point corresponding to any assigned state of pressure and superheat. *Lines of Constant Volume* may be drawn without difficulty (see § 96 below), but they are of less general importance and are omitted from the figure to avoid confusion. *Lines of Constant Total Heat* (I) may also be added to the chart: three examples are shown, for $I = 750$, 700, and 650 respectively. The line for $I = 650$ lies partly in the wet steam region, taking a sudden bend where it crosses the boundary curve. The wet region as well as the region of superheat may be completely mapped out by a system of such lines. In the wet steam region it is very useful to draw a system of *Lines of Constant Dryness*, namely lines showing the proportion of dry steam in the mixture, expressed as a fraction of the whole. These are drawn in the figure for values of $q = 0.9$, 0.8, 0.7, etc., down to $q = 0.1$.

With the aid of such a chart it is easy to trace the changes which steam undergoes in adiabatic expansion. Taking for example steam whose pressure is 5 kg. per sq. cm., initially superheated to 400°C., we mark the corresponding point and draw from it a vertical line towards the base to represent the adiabatic process. This cuts the boundary curve at about $t = 65°$ C., showing that the steam has become just saturated when its pressure has fallen to 0.25 kg. per sq. cm. The line then enters the wet steam region. If the expansion be continued till the temperature falls to 18° C., we find that the adiabatic line cuts the line of constant dryness $q = 0.9$, showing that the proportion of water then present is 10 per cent.

Having found in this way the dryness after any assigned amount of expansion it is easy to calculate the final value of the Total Heat, since from the table we know I for water and steam separately at the final temperature. Then, by subtracting the final from the initial value of the total heat (§ 74) we obtain the ideal amount of work done in the cylinder, and after deducting from this the small amount of work spent in the feed pump we have the ideal performance of an engine following the Rankine cycle with the assumed initial and final pressures and the assumed degree of initial superheat. This should agree with the area measure from the entropy diagram.

In the diagram drawn in fig. 31 a conjectural curve has been added (shown by a broken line) to connect the water and steam boundary curves in the region of high pressures where, at present, there are no data for a precise determination of the entropy. This broken line is simply a smooth curve forming a continuation of each boundary curve and drawn so as to be tangent to the line $t = 365$, which is the critical temperature of water, that being the temperature at which the distinction between ϕ_w and ϕ_s disappears. When exact data become available it will become possible to complete the entropy-temperature chart by some such line. In the meantime the line here sketched will help the student to realize that the steam and water lines are essentially two portions of one continuous curve whose summit is at the critical point, and it may even serve provisionally for estimating probable values of the entropy, the total heat of water and steam, and the latent heat, at temperatures considerably above 200° C.

Not the least merit of the entropy-temperature diagram as a means of representing graphically the cycle of operations in a heat-engine is that it shows the heat taken in and the heat rejected, as well as the work done, and so allows estimates of efficiency to be made by inspection of the diagram itself. The advantage, for instance, which results from raising the initial pressure of the steam is readily shown in a diagram such as fig. 28 by drawing horizontal lines at temperatures corresponding to the initial pressures which are to be compared, and vertical lines through the points where they meet the entropy curve of saturated steam (cf), the vertical lines being continued to meet the base, which is the absolute zero of temperature. Comparison of the enclosed areas then shows that while the heat taken in is but slightly increased with higher boiler pressure there is a more considerable gain of work, a result which is of course to be expected from the fact that the general temperature of reception of the heat is raised.

When a vertical line is drawn to represent the adiabatic expansion of a mixture of steam and water, it is clear from the diagram that if the mixture consists mostly of water to begin with it will become drier as it expands, instead of wetter as is the case when the initial proportion of water is less. In the region of ordinary working pressures the "water" and "steam" curves of fig. 31 have nearly equal inclinations to the vertical

line which represents an adiabatic process. Hence if such a line be drawn starting from a point midway between the two curves it will continue to lie nearly midway between them: in other words, if there is about 50 per cent. of water present at the beginning of adiabatic expansion, nearly the same percentage will be found as the expansion goes on. When the steam is much wetter than this to begin with, adiabatic expansion makes it drier.

It will be shown later that the entropy-temperature diagram is also of service in exhibiting the changes of dryness which occur in real steam-engines, where the action is by no means adiabatic.

In drawing an entropy-temperature chart for the purpose of solving problems on the adiabatic expansion of steam which is initially nearly dry or superheated, or other problems relating to superheated steam, we are concerned only with the region lying between say $\phi = 1\cdot4$ and $\phi = 2$. All the parts to the left of this may accordingly be omitted and a conveniently open scale for
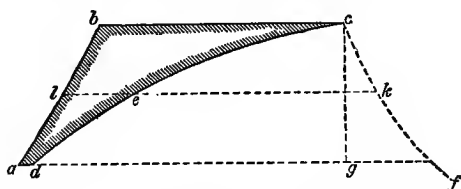
FIG. 32. Entropy-temperature diagram of Steam used Non-expansively.

entropy be adopted in drawing the useful portion, which includes the whole region of superheat along with as much of the wet region as comes into consideration. Within that region it will be found convenient to have the lines of constant dryness drawn for intervals of say 1 per cent.

96. Entropy-temperature diagram for Steam used Non-expansively. By way of contrast with the cases treated in §§ 92 and 93, we may draw the entropy-temperature diagram for a steam-engine working without expansion. The four steps of the cycle have been stated in § 71, and the volume-pressure diagram is drawn there (fig. 17). In the entropy-temperature diagram (fig. 32) we have the four corresponding lines ab, bc, cd, da. ab is the heating of the water from τ_2 to τ_1. bc is the conversion of the water into steam, cd the partial condensation which takes place

when the cold body is applied, the piston meanwhile remaining at the end of its forward stroke, and da is the remainder of the condensation, which occurs while the piston is pressed in, the cold body being still applied. cd is a *Line of Constant Volume*, for throughout the change which it represents the substance as a whole suffers no change of volume, since it remains in the cylinder and there is no movement of the piston. To find points in cd, draw the saturation curve cf as in former examples and at any temperature τ intermediate between τ_1 and τ_2 draw the line lk. We have to divide lk in a point e such that $\dfrac{le}{lk}$ shall represent q, the dryness of the steam at the time its temperature has fallen to τ. The dryness q is determined by the consideration that the volume of the substance as a whole is sensibly equal to qV, where V is the volume of unit mass of saturated steam at τ.
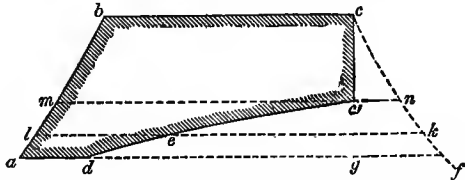
FIG. 33. Entropy-temperature diagram showing incomplete expansion of Steam.

This remains equal to V_1, which is the volume originally occupied by unit mass before the process of condensation began, since there is no change of volume in the process. Hence $q = \dfrac{V_1}{V}$, and e is found by making

$$le = \frac{V_1}{V}\, lk.$$

The work which is lost through the absence of adiabatic expansion is the area cgd.

97. Incomplete expansion. The case of incomplete expansion admits of similar treatment. Let adiabatic expansion be carried on until, at the end of the stroke, the temperature has fallen to the level indicated by c' in the entropy-temperature diagram, fig. 33. This process is represented by the line cc'. Then let the steam be suddenly cooled by applying the cold body. The constant-volume curve $c'd$ shows this cooling; after which the

return stroke takes place, which is shown by da. To draw the curve $c'd$ take a line lk at any lower temperature τ and take e in it, such that

$$le = \frac{q'V'}{V} lk,$$

where V' is the volume of unit mass of saturated steam at the temperature corresponding to c', and q' is the dryness at c', which is equal to $\frac{mc'}{mn}$, and V is the volume of unit mass of saturated steam at temperature τ.

In dealing with the process of sudden condensation represented by the line cd in fig. 32 and $c'd$ in fig. 33 we have supposed, to simplify the statement, that the steam is retained in the cylinder and the cold body is applied to it. But it makes no difference if the steam be allowed to escape into a separate vessel, to be condensed there. Just the same amount of work is done, for the pressure on the piston is the same in that case as in the other. Hence the area of the entropy-temperature diagram is unaffected, and since that is true whatever be the value of τ_2 the form of the curve cd or $c'd$ is unchanged[1].

98. Derivation of Clapeyron's Equation. As another example of the use of the entropy-temperature diagram we may apply it in obtaining the Clapeyron equation of § 80. Take as

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[1] The constant-volume curve cd or $c'd$ in the entropy-temperature diagram may be drawn as follows by an application of equation (3) of § 81. Let U represent the volume of the mixture of steam and water at any stage in the process of condensation, the temperature then being τ. Let λ represent the heat which would be given out if the condensation of the mixture were completed at the temperature τ. Then by that equation

$$U - w = \frac{J\lambda}{\tau} \frac{d\tau}{dP},$$

w being the volume when the substance is all water. Hence

$$\frac{\lambda}{\tau} = \frac{U - w}{J} \frac{dP}{d\tau}.$$

But $\frac{\lambda}{\tau}$ is the length le, if the line le be drawn at the level τ, and U is the volume of the cylinder, which is constant. We therefore have a relation which allows le at any level of temperature to be readily determined when the values of $\frac{dP}{d\tau}$ for saturated steam are known. These may be found by measurement of the slope of the pressure-temperature curve, or approximately from the tables of pressure and temperature in the Appendix by dividing small differences of pressure by corresponding differences of temperature.

before the action of an engine following Carnot's cycle and working through a very small interval of temperature $\delta\tau$. The indicator diagram, sketched in fig. 23, has the area $\delta P\,(V-w)$. The entropy-temperature diagram is a long narrow strip whose length is $\dfrac{L}{\tau}$ and height $\delta\tau$. Its area is $\dfrac{L\delta\tau}{\tau}$, equivalent in work units to $\dfrac{JL\delta\tau}{\tau}$. Hence

$$\delta P\,(V-w) = \frac{JL\delta\tau}{\tau},$$

and

$$V - w = \frac{JL}{\tau}\frac{d\tau}{dP},$$

which is Clapeyron's equation.

99. Adiabatic Equation for Superheated Steam. In § 90 we saw that an equation expressing the change of condition which wet steam undergoes while expanding adiabatically was readily found from the consideration that the entropy does not change in an adiabatic process. Similarly, the expression for the entropy of superheated steam given in § 93 may serve to furnish a relation between the pressure and temperature when superheated steam expands adiabatically. We have only, in that expression, to give to ϕ the numerical value proper to the initial state, and since that remains constant during the expansion, the expression then forms an equation connecting the temperature and the pressure during the process which may be applied to calculate the pressure for any assigned temperature lower than the initial temperature, until the stage of saturation is reached, after which the equation of § 90 applies. This method of calculation may be used to trace stage by stage the effects of an adiabatic process: but it will be apparent that in point of convenience the graphic method has very great advantages. One such graphic method is furnished by the Entropy-Temperature diagram as already noticed. Another diagram which is even more useful for the purpose is described in the next paragraph.

100. Mollier's Diagrams. In 1904 Dr R. Mollier[1] introduced two novel graphic methods of representing the properties

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[1] R. Mollier, "Neue Diagramme zur technischen Wärmelehre," *Zeitschrift des Vereins deutscher Ingenieure*, 1904, p. 271. See also his *Neue Tabellen und Diagramme für Wasserdampf*, Berlin (Julius Springer), 1906.

of steam which furnish very valuable alternatives to the diagram
of entropy and temperature. Both are interesting and useful, and
one of them in particular may be said to offer advantages in point
of convenience which entitle it to the first place among these
devices. It puts into the hands of engineers an extremely simple
method of solving the important problem of determining the state
of steam which has expanded adiabatically from any initial con-
dition, superheated or not, and of finding the greatest theoretical
output attainable from steam when the initial condition and the
lower limit of temperature are assigned. We have seen that this
problem may be solved by aid of the entropy-temperature diagram.
But Mollier's diagram allows the solution to be obtained still more
easily. It is also, as will appear later, specially convenient in
steam-turbine applications, as a means of finding the velocity
resulting from any assigned drop of pressure when steam expends
its energy in setting itself in motion.

In this diagram Mollier takes for coordinates the Entropy and
the Total Heat. The diagram is shown in a skeleton form in
fig. 34 and is reproduced on a working scale as a folding plate at
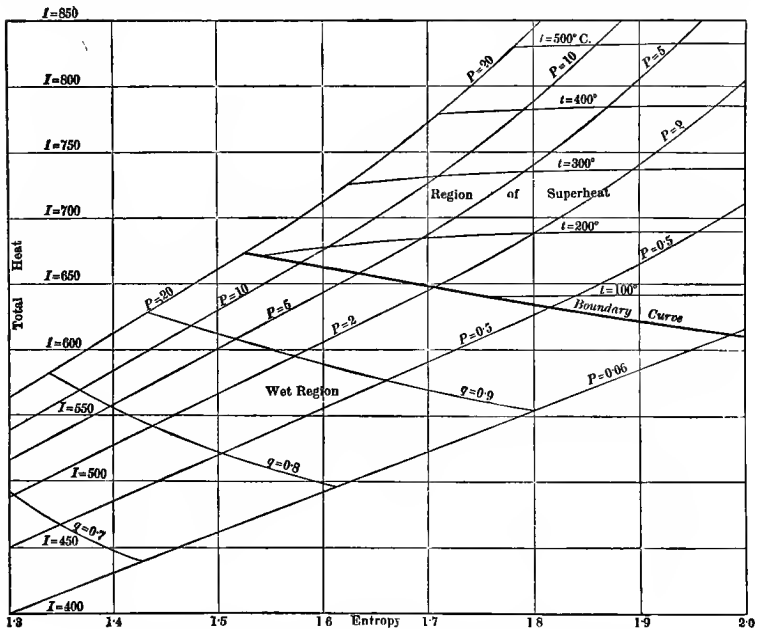
Fig. 34. Mollier's Diagram of Entropy and Total Heat.

the end of this book. Only the useful region, near the boundary curve, is shown in the plate, namely from $\phi = 1\cdot 4$ to $\phi = 2\cdot 0$. The vertical lines being lines of constant entropy are adiabatics. Lines of Constant Pressure are drawn extending throughout the wet steam region, where they are straight, over the boundary curve into the region of superheat where they become curved, but with no abrupt change of direction in crossing the boundary. In the skeleton diagram (fig. 34) these are drawn only for pressures of 20, 10, 5, 2, 0·5, and 0·06 kilogrammes per sq. cm.; in the large scale diagram many more such lines are included. The pressures marked on these lines are in kilogrammes per sq. cm. In the wet region there is also a system of Lines of Constant Dryness, drawn for intervals of 10 per cent. in fig. 34 and 1 per cent. in the large diagram. In the region of superheat there is a system of Lines of Constant Temperature drawn for intervals of 100° C. in fig. 34 and 5° in the large diagram. The combination of pressure and temperature lines in the region of superheat allows the point to be found corresponding to any assigned initial condition. Once this point is marked, a vertical line drawn down through it shows an adiabatic process of expansion and allows the consequent changes of condition to be traced, giving the temperature for each pressure so long as there is superheat, and also the wetness for each pressure in the final stages after the boundary curve has been crossed. Further, the length of this vertical line is the "heat drop"; it is a direct measure of the work done in the cylinder in an adiabatic process, being $I_1 - I_2$. This fact alone gives the diagram much value as an aid in calculations of theoretical performance. Again, since in a throttling process I is constant, the diagram exhibits such processes by horizontal straight lines, from which the changes of condition that occur in throttling are readily determined.

In Mollier's other diagram the coordinates are the Pressure and the Total Heat. This diagram misses the feature possessed by diagrams in which the Entropy is one of the coordinates, of exhibiting adiabatic processes by straight lines, but it has certain advantages, and forms in any case a very useful adjunct to the ϕ-I diagram. Any of these diagrams may be regarded as a graphic way of stating the properties of steam, superheated as well as saturated, with a completeness that would be scarcely practicable in numerical tables. The P-I diagram appeals to

engineers as being a graphic table with a pressure base, and is particularly convenient not only in giving the total heat and the heat of superheating for given pressure and temperature, but also in tracing changes of volume. It has the property that Isotherms or Lines of Constant Temperature, and also Lines of Constant Volume are straight, and consequently systems of such lines are very readily drawn.

A skeleton of the P-I diagram is shown in fig. 35 for the region that is useful in practice, and Mollier's large version is re-produced as a folding plate at the end of the book. The space below the boundary curve is the wet region, and is mapped out as in the other charts by Lines of Constant Dryness. Throughout
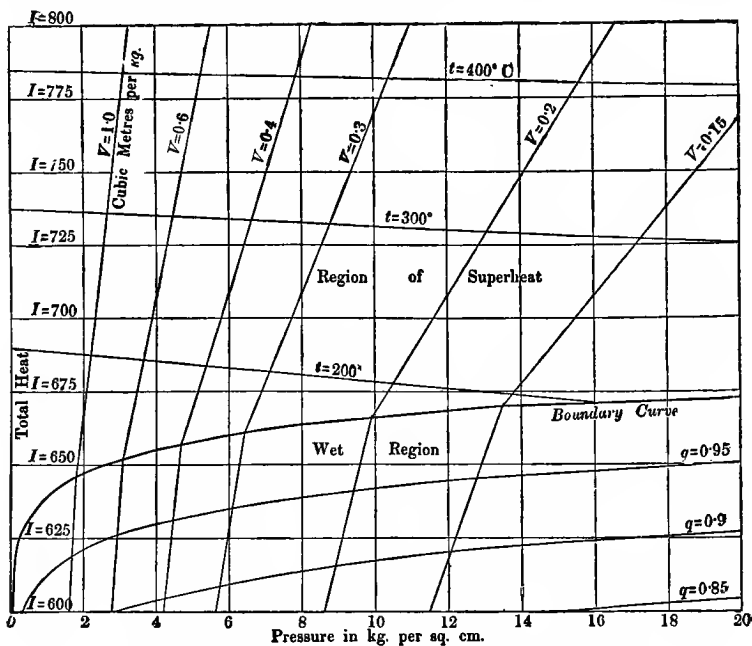
FIG. 35. Mollier's Diagram of Pressure and Total Heat.

this space, as well as in the region of superheat above, Lines of Constant Volume are drawn. These are straight lines in the region of superheat (see Appendix), and they remain sensibly straight in the region of wetness but suffer an abrupt change of direction on crossing the boundary curve. A few representa-tive lines of constant volume are sketched in the figure. The

Lines of Constant Temperature are also straight and slope down to the right with gradients equal to the values of Y in the auxiliary table (Table D, Appendix).

101. Entropy-temperature diagrams in engines using a Regenerator. An engine such as Stirling's, which substitutes the use of a regenerator for the adiabatic expansion and compression in Carnot's cycle, has an entropy diagram of the type shown in fig. 36. The isothermal operation of taking in heat at τ_1 is represented by ab; bc is the cooling of the substance from τ_1 to τ_2 in its passage through the regenerator, where it deposits heat: cd is the isothermal rejection of heat at τ_2; and da is the restoration of heat by the regenerator while the substance passes through it in the opposite direction, by which the temperature is raised from τ_2 to τ_1. Assuming the action of the regenerator to be ideally perfect, bc and ad are precisely similar curves whatever be their form. The area of the figure is then equal to the area of the rectangle which would represent the ordinary Carnot cycle (fig. 27). The equal areas $pbcq$ and $ndam$ measure the heat stored and restored by the regenerator.
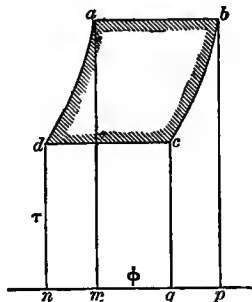
FIG. 36. Entropy-temperature diagram of perfect engine using a Regenerator.

When the working substance is air and the regenerative changes take place either under constant volume, as in Stirling's engine, or under constant pressure, as in Ericsson's, the specific heat K being treated as constant, ad and bc are logarithmic curves with the equation

$$\phi = \int \frac{Kd\tau}{\tau} = K \log_e \tau,$$

K being K_v in one case and K_p in the other.

102. Joule's Air-Engine. A type of air-engine was proposed by Joule which, for several reasons, possesses much theoretical interest. Imagine a chamber C (fig. 37) full of air (temperature τ_2), which is kept cold by circulating water or otherwise; another chamber A heated by a furnace and full of hot air in a state of

compression (temperature τ_1); a compressing cylinder M by which
air may be pumped from C into A, and a working cylinder N in
which air from A may be allowed to expand before passing back
into the cold chamber C.   We shall suppose the chambers A and
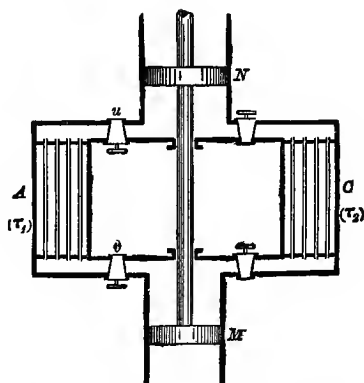
Fig. 37.   Joule's proposed Air-Engine.

C to be large, in comparison with the volume of air that passes in
each stroke, so that the pressure in each of them may be taken as
sensibly constant.   The pump M takes in air from C, compresses
it adiabatically until its pressure becomes equal to the pressure
in A, and then, the valve v being opened, delivers it into A.   The
indicator diagram for this action on the part of the pump is the
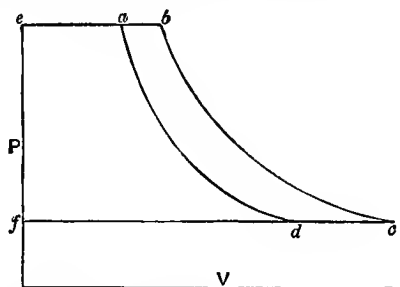
Fig. 38.   Indicator diagram in Joule's Air-Engine.

diagram $fdae$ in fig. 38.   While this is going on, the same quantity
of hot air from A is admitted to the cylinder N, the valve u is then
closed, and the air is allowed to expand adiabatically in N until
its pressure falls to the pressure in the cold chamber C.   During

the back stroke of N this air is discharged into C. The operation of N is shown by the indicator diagram $ebcf$ in fig. 38. The area $fdae$ measures the work spent in driving the pump; the area $ebcf$ is the work done by the air in the working cylinder N. The difference, namely, the area $abcd$, is the net amount of work obtained by carrying the given quantity of air through a complete cycle. Heat is taken in when the air has its temperature raised on entering the hot chamber A. Since this happens at a pressure which is sensibly constant,

$$Q_A = K_p (\tau_b - \tau_a),$$

where τ_b is τ_1, the temperature of A, and τ_a is the temperature reached by adiabatic compression in the pump. Similarly, the heat rejected

$$Q_C = K_p (\tau_c - \tau_d),$$

where $\tau_d = \tau_2$, the temperature of C, and τ_c is the temperature reached by adiabatic expansion in N. Since the expansion and compression both take place between the same terminal pressures, the ratio of expansion and compression is the same. Calling it r, we have

$$\frac{\tau_a}{\tau_d} = \frac{\tau_b}{\tau_c} = r^{\gamma - 1}$$

($\S$ 41), and hence also

$$\frac{\tau_b}{\tau_a} = \frac{\tau_c}{\tau_d}, \quad \text{and} \quad \frac{\tau_b - \tau_a}{\tau_a} = \frac{\tau_c - \tau_d}{\tau_d}.$$

Hence

$$\frac{Q_A}{Q_C} = \frac{\tau_a}{\tau_d} = \frac{\tau_b}{\tau_c}.$$

and the efficiency

$$\frac{Q_A - Q_C}{Q_A} = \frac{\tau_a - \tau_d}{\tau_a} = \frac{\tau_b - \tau_c}{\tau_b}.$$

This is less than the efficiency of a perfect engine working between the same limits of temperature $\left(\dfrac{\tau_1 - \tau_2}{\tau_1} \right)$ because the heat is not taken in and rejected at the extreme temperatures.

The atmosphere may take the place of the chamber C: that is to say, instead of having a cold chamber, with circulating water to absorb the rejected heat, the engine may draw a

fresh supply at each stroke from the atmosphere and discharge
into the atmosphere the air which has been expanded adiabatically
in N.

The entropy-temperature diagram for this cycle is drawn in
fig. 39, where the letters refer to the same
stages as in fig. 32. After adiabatic compression
da, the air is heated in the hot chamber A
and the curve ab for this process has the
equation

$$\phi = \int_{\tau_a}^{\tau} \frac{K_p d\tau}{\tau} = K_p (\log_\epsilon \tau - \log_\epsilon \tau_a).$$

Then adiabatic expansion gives the line bc,
and cd is another logarithmic curve for the re-
jection of heat to C by cooling under constant

pressure. The ratio $\dfrac{\tau_a}{\tau_b}$ which is represented by

$\dfrac{ea}{eb}$ in fig. 38 and by $\dfrac{ma}{nb}$ in fig. 39, shows the

Fig. 39. Entropy-
temperature dia-
gram in Joule's
Air-Engine.

proportion which the volume of the pump M must bear to the
volume of the working cylinder N. The need of a large pump
would be a serious drawback in practice, for it would not only
make the engine bulky but would cause a relatively large part
of the net indicated work to be expended in overcoming friction
within the engine itself.

In the original conception of this engine by Joule it was
intended that the heat should reach the working air through the
walls of the hot chamber, from an external source. But instead
of this we may have combustion of fuel going on within the hot
chamber itself, the combustion being kept up by the supply of
fresh air which comes in through the compressing pump, and, of
course, by supplying fuel either in a solid form from time to time
through a hopper, or in a gaseous or liquid form. In other words,
the engine may take the form of an *internal combustion* engine.
Internal combustion engines, essentially of the Joule type, em-
ploying solid fuel have been used on a small scale, but by far
the most important development of this type is the explosive
gas or oil engine. Its cycle is substantially Joule's, considerably
modified, however, by features which will be noticed in a later
chapter.

This, however, is not the only reason why Joule's cycle is now interesting. It has found application in the reversed form as a practical process for cooling air. Refrigerating machines in which air is the working substance have been extensively used to keep the temperature of rooms on board ship below the freezing point, to allow frozen meat to be carried over seas, and such machines work, as we shall see in the next chapter, by reversing the cycle suggested by Joule.

CHAPTER VI.

REVERSAL OF THE HEAT ENGINE: MECHANICAL REFRIGERATION[1].

103. Reversal of the cycle in heat-engines: Refrigerating Machines or Heat-Pumps. By a refrigerating machine or heat-pump is meant a machine which will carry heat from a cold to a hotter body. This, as the second law of thermo-dynamics asserts, cannot be done by a self-acting process, but it can be done by the expenditure of mechanical work. Any heat-engine will serve as a heat-pump if it be forced to trace its indicator diagram backwards, so that the area of the diagram represents work spent on, instead of done by, the working substance. Heat is then taken in from the cold body and heat is rejected to the hot body.

Take for instance the Carnot cycle, using air as working substance (fig. 40), and let the cycle be performed in the order $dcba$, so that the area of the diagram is negative, and represents work spent upon the machine. In stage dc, which is isothermal expansion in contact with the cold body C, the gas takes in a quantity of heat from C equal to $R\tau_2 \log_e r$ (§ 42), and in stage ba it gives out to the hotter body A a quantity of heat equal to $R\tau_1 \log_e r$. There is no transfer of heat in stages cb and ad.

FIG. 40.

Thus C, the cold body, is constantly being drawn upon for heat and can therefore be maintained at a temperature lower than its

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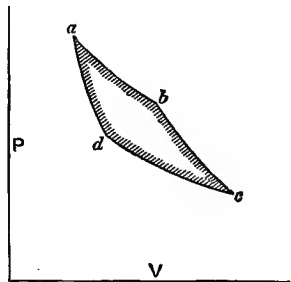
[1] On the subject of refrigerating processes generally reference should be made to the author's book on the *Mechanical Production of Cold* (Camb. Univ. Press, 1908), where fuller details will be found.

surroundings. Suppose that such a machine were to be applied to the making of ice, then C might consist of a coil of pipe immersed in brine. The brine could in this way be kept by the action of the machine at a temperature below $0°$ C., and be used, in its turn, to extract heat by conduction from the water which is to be frozen. The " cooler " A, which is the relatively hot body, is kept at as low a temperature as possible by means of circulating water, which absorbs the heat rejected to A by the working air. This is substantially the process which is used in actual ice-making machines, except that the cycle of operations is not a reversed Carnot cycle, but more nearly a reversed Rankine cycle, and the working substance is a vaporisable liquid instead of air.

A machine using air as working substance and following Carnot's cycle would be exceedingly bulky. Its size would be considerably reduced if a regenerator, as in Stirling's engine, were resorted to in place of the two adiabatic stages of the Carnot cycle. Refrigerating machines of this kind, using air as working substance, with a regenerator, were introduced by Dr A. C. Kirk, and were at one time considerably used[1]. The working air was completely enclosed, which allowed it to be in a compressed state throughout, so that even its lowest pressure was much above that of the atmosphere. This made a greater mass of air pass through the cycle in each revolution of the machine, and hence increased the performance of a machine of given size.

This type of refrigerating machine has not survived, and those machines which now use air as working substance follow the reversed Joule cycle as described below in § 106.

104. Vapour Compression Refrigerating Machines. In most modern refrigerating machines, however, the working substance, instead of being air, consists of a liquid and its vapour, and the action proceeds by alternate evaporation under a low pressure and condensation under a relatively high pressure. A liquid must be chosen which evaporates at the lower extreme of temperature under a pressure which is not so low as to make the bulk of the engine excessive. Sulphuric ether was one oı the earliest liquids

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[1] See Kirk, "On the Mechanical Production of Cold," *Min. Proc. Inst. C. E.* vol. xxxvii. 1874.

to be used in this way, but ether machines were inconveniently bulky and could not be used to produce intense cold, for the pressure of that vapour is only about 1·3 lbs. per square inch at −15° C. and to make it evaporate at any temperature considerably below zero would require the cylinder to be excessively large in proportion to the performance. This would not only make the machine clumsy and costly but would involve much waste of power in mechanical friction. The tendency of the air outside to leak into the machine is another practical objection to the use of so low a pressure. The liquids now used are carbonic acid, ammonia, and sulphurous acid. Of these ammonia and carbonic acid are the most common. With ammonia it is easy to reach as low a limit of temperature as is required in any of the usual industrial
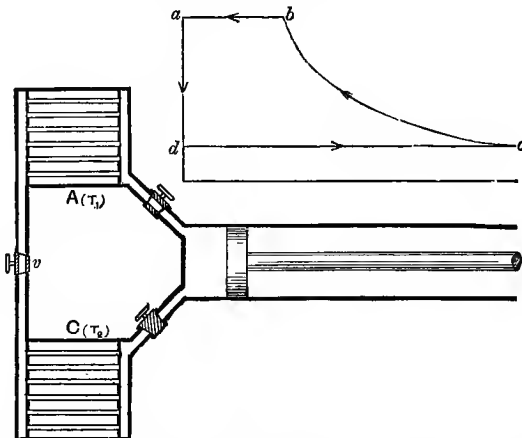
Fig. 41. Refrigerating Machine using the vapour of a liquid.

applications of cold: the pressures are fairly but not excessively high, and the apparatus is compact. With carbonic acid the apparatus is even more compact, but the pressures are so high as to make packing a matter of greater difficulty, and the thermodynamic efficiency of the operation is not quite so good. Carbonic acid however is frequently preferred, especially on board ship, on account of its being more harmless should any of the working substance escape by leakage into the atmosphere of the room containing the machine.

Engines of this type are usually arranged to act as follows, in a cycle which is almost exactly the reverse of the Rankine cycle.

The organs, which are shown diagrammatically in fig. 41, are (1) a compressing cylinder, (2) a cold body C which serves as boiler for the volatile working fluid and allows heat to pass into the working fluid from the water or other substance that is to be made cold, and (3) a cooler A such as a coil of pipe surrounded by circulating water, in which the working fluid is condensed under pressure.   The steps of the cycle are shown by the indicator diagram in the same figure; dc is the forward stroke, during which the cylinder is taking in vapour from C at a uniform pressure corresponding to the lower limit of temperature τ_2.   Compression of the vapour occurs during cb, which is the first part of the back stroke, and during which the valves leading to both chambers are shut.   This continues till the pressure in the cylinder becomes equal to the pressure in A.

Next, the communication with A is opened and the back stroke is completed under a uniform pressure which corresponds to the temperature in A, the working substance passing into A and being condensed there (ba).

To complete the cycle, the same quantity of the substance is allowed to pass through the valve v directly from A to C (ad).

This last step in the process is not reversible, but it is a simpler way of completing the cycle than to complete it reversibly by letting the fluid do work in an expansion cylinder in passing from A to C, and the amount of work which would be saved if that were done is inconsiderable.

The operation of such machines may be represented by an entropy-temperature diagram like that of fig. 42, taking the lines in the reverse of the former order.   If the evaporation were complete, a horizontal line extending to f in that figure would represent the process of evaporation, during which heat is being taken in from the body to be cooled.   More generally however evaporation is incomplete; what is taken into the cylinder and compressed is a mixture of vapour with some of the unevaporated liquid.   This reduces the superheating which compression would otherwise cause, and may even prevent superheating entirely, provided enough liquid be present in the mixture.   Thus, in fig. 42, if d represents the condition as regards wetness of the mixture which is taken into the cylinder, the adiabatic process of compression dc will bring the substance to a dry saturated state. On the other hand, if the original condition of the mixture is

represented by the point s, adiabatic compression, represented by
str, will superheat it to some extent, though not so much as
if the evaporation had been complete before compression.   In
Dr Linde's form of ammonia compression machines superheating
is usually prevented or at least reduced by using a wet mixture.
In the entropy diagram the compression process is represented by
dc, fig. 42, when the mixture taken into the compressor is so wet
as to escape superheating by compression.   In such a case the
machine is said to employ wet compression.   In other ammonia
machines the mixture taken in is less wet, and superheating
occurs, but in some its effects are reduced by using a water-
jacket to cool the cylinder.
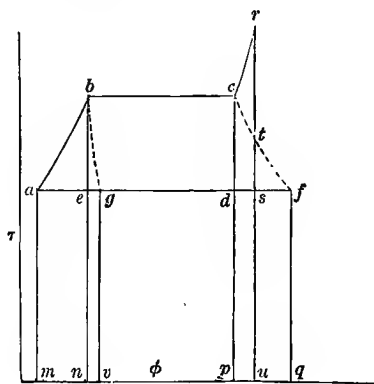
Fig. 42.

The action is substantially a reversal of the Rankine cycle.
Taking the entropy diagram (fig. 42) drawn for the particular
substance and particular temperatures used, the work spent in
the compressing cylinder is the area $dcba$ *plus* the small correction
$A\,(P_1 - P_2)\,w$ (§ 94).   We may regard the area under ad as the
gross amount of heat to be taken up during evaporation of the
working substance after it has been completely condensed.   But
in passing through the regulating valve from the condenser A to
the refrigerator or evaporator C the substance conveys with it an
amount of heat equal to the area under the line ba, namely $nbam$,
together with the heat-equivalent of the work $(P_1 - P_2)\,w$ spent
upon it as it streams through the valve, consequent on the fall in
pressure, and hence the net refrigerating effect is considerably

less. What happens to the substance in streaming through the valve is that its total heat I remains constant, in accordance with the principle explained in § 82. The change of state in passing through the valve may therefore be represented by the line bg, which is a line of constant total heat, to be drawn by consideration that at any point in it the proportion q of vapour is such that $qL + I_w = I_{w1}$. By the time the substance has passed the valve the part $\frac{ag}{af}$ is already in the state of vapour and only the remainder is available for effective evaporation. Thus the net cooling effect, if evaporation is stopped at d is the area under gd, namely $vgdp$.

Comparing this with the reversed Carnot cycle which would represent an ideally efficient refrigerating process we see that the net refrigerating effect is less, being the area under gd instead of the area under ed; and also that the work spent is greater, being the area $dcba$ instead of $dcbe$. For both these reasons the efficiency is less. The difference represents what is sacrificed by using a throttle valve instead of an expansion cylinder between the condenser and the refrigerator. From a study of the diagram it will be evident that the difference in question will be comparatively unimportant if the specific heat of the liquid is very small compared with the latent heat of the vapour. Ammonia is from this point of view the most efficient of the substances used in refrigerating machines. With carbonic acid, especially when the upper temperature approaches the critical point, the loss resulting from the absence of an expansion cylinder is much greater.

105. Coefficient of Performance of Refrigerating Machines. The ratio

$$\frac{\text{Heat extracted from the cold body}}{\text{Work expended}}$$

may be taken as a coefficient of performance in estimating the merit of a refrigerating machine from the thermodynamic point of view. When the limits of temperature τ_1 and τ_2 are assigned it is easy to show by a slight variation of the argument used in § 47 that no refrigerating machine can have a higher coefficient of performance than one which is reversible in Carnot's sense. For let a refrigerating machine S be driven by another R which is reversible and is used as a heat-engine in driving S. Then if S

had a higher coefficient of performance than R it would take from the cold body more heat than R (working reversed) rejects to the cold body, and hence the double machine, though purely self-acting, would go on extracting heat from the cold body in violation of the Second Law. Reversibility, then, is the test of perfection in a refrigerating machine just as it is in a heat-engine.

When a reversible refrigerating machine takes in all its heat, namely Q_C, at τ_2 and rejects all, namely Q_A, at τ_1, $\dfrac{Q_C}{\tau_2} = \dfrac{Q_A}{\tau_1}$ and the coefficient of performance

$$\frac{Q_C}{W} = \frac{Q_C}{Q_A - Q_C} = \frac{\tau_2}{\tau_1 - \tau_2}.$$

Hence—and the inference is highly important in practice—the smaller the range of temperature is the better. To cool a large mass of any substance through a few degrees will require much less expenditure of energy than to cool one-tenth of the mass through ten times as many degrees, though the amount of heat extracted is the same in both cases. If we wish to cool a large quantity, say of water or of air, it is better to do it by the direct action of a refrigerating engine working through the desired range of temperature, than to cool a portion through a wider range and then let this mix with the rest. This is only another instance of a wide general principle, of which we have had examples before, that any mixture or contact of substances at different temperatures is thermodynamically wasteful because the interchange of heat between them is irreversible. An ice-making machine, for example, should have for its lower limit a temperature only so much below the freezing point as will allow heat to be conducted with sufficient rapidity to the working fluid from the water that is to be frozen.

106. Reversed Joule Engine: the Bell-Coleman Refrigerating Machine. This machine was briefly mentioned in § 102 as one which has been employed to maintain a cold atmosphere in the frozen-meat chambers of ocean steamships. It acts by drawing in a small portion of the air of the chamber, compressing that and extracting as far as possible by means of a cooler the heat developed by compression, then expanding the air until its pressure falls to that of the chamber. The temperature of the

expanded air is then lower than the temperature of the chamber in consequence of the removal of heat which took place while it was compressed. The air thus chilled by expansion is returned to the chamber, and in this way the temperature of the chamber is kept down notwithstanding the heat which reaches it by conduction from outside. The chamber has a thick lining of poorly conducting matter in order to reduce as far as may be the work which has to be spent on refrigeration.

The sketch, fig. 43, shows the organs diagrammatically. C is part of the cold chamber, which is at or about atmospheric pres-
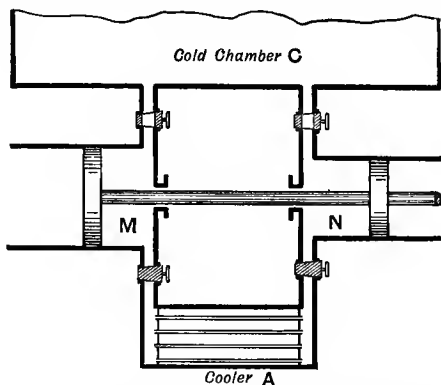
Fig. 43.   Organs of the Bell-Coleman Refrigerating Machine.

sure, and A is the cooler, a set of pipes with circulating water. Compression takes place in M and expansion in N. M takes in air from C at temperature τ_2 during its out-stroke, and compresses that during part of its in-stroke till the pressure becomes equal to the pressure in A. These two operations are represented by the lines fc and cb in the indicator diagram, fig. 44. The compression
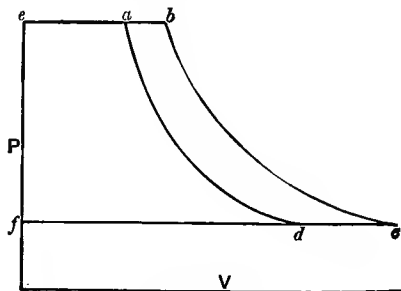
Fig. 44.

cb has the effect of raising the temperature of the air above that of A. Consequently when the pump delivers the compressed air into A, by completing its return stroke (bc), which is the next operation, the temperature of the air falls and a quantity of heat is rejected to A, namely

$$K_p (\tau_b - \tau_a),$$

where τ_b is the temperature reached by compressing, and τ_a is τ_1 the temperature of A. While this is going on, the cylinder N takes an equal quantity of air from A at τ_1 or τ_a, and expands it to the pressure of C: these operations are shown by the lines ea and ad in the indicator diagram. At the end of this expansion the temperature τ_d is lower than that of the cold chamber. Finally the chilled air is discharged into C during the return stroke of N, which is shown by the line df in the indicator diagram. The net amount of work expended is $badc$, $fcbe$ being the indicator diagram of work spent upon the pump M and $eadf$ being the diagram of work recovered in the expansion cylinder N. The net amount of heat taken from the cold chamber is $K_p (\tau_c - \tau_d)$. Assuming the processes cb and ad to be adiabatic, the ratio of expansion in N is equal to the ratio of compression in M, and hence $\dfrac{\tau_a}{\tau_d} = \dfrac{\tau_b}{\tau_c}$, as we have already seen in treating of the Joule cycle (§ 102) of which this is simply a reversal. Also $\dfrac{Q_A}{Q_C} = \dfrac{\tau_a}{\tau_d}$ and the coefficient of performance $\dfrac{Q_C}{Q_A - Q_C} = \dfrac{\tau_d}{\tau_a - \tau_d}$, a value less than $\dfrac{\tau_2}{\tau_1 - \tau_2}$ for the same reason that Joule's engine is less efficient than Carnot's.

In practice the compression in M is not adiabatic: by using a water-jacket, or by injecting water into the cylinder itself, the compression may be made to follow a curve which lies between an adiabatic and an isothermal line. This has the thermodynamic advantage that some heat is extracted at a lower temperature than would be the case if compression were completed before the action of the cooler began. Or, to take another point of view, the net expenditure of work is reduced, since the compression curve from c rises less steeply than the adiabatic line cb.

A difficulty attending the use of machines of this type arises from the fact that the working substance is not *dry* air. It contains water-vapour in solution, as all air does except when

specially dried, and when the temperature falls this tends to be condensed and even frozen. Difficulty has therefore in some cases been experienced from the clogging of valves and passages by snow or hoar-frost deposited by the working air. The air is generally saturated when it is cooled after compression (at the point a in the diagram), even when no injection-water is used to assist the cooling. It was to meet this objection that Mr Lightfoot introduced a form of the machine, in which the expansion was performed in two stages by means of a compound pair of expansion cylinders. In the first the temperature of the air was reduced to only about 2° C., stopping just short of the freezing point. In this first stage the greater part of the water-vapour was deposited as water, which was drained away, and the air then went on to the second cylinder, where it completed its expansion with very little further deposit. In Mr Coleman's form of the machine the compressed air after giving up heat through tubes to water in the ordinary cooler, is further cooled by passing through pipes which are exposed to the action of chilled air from the chamber, and is thus forced to give up its suspended moisture before it is allowed to expand[1]. This arrangement of "drying pipes" has been employed by Messrs Haslam in their construction of the Coleman apparatus, but other makers of refrigerating machines which use air are content with a mechanical separation of the air from any water which has been deposited in the cooling which precedes admission to the expansion cylinder. Provided the air entering that cylinder is merely saturated and does not carry with it water in a state of mechanical suspension the deposit of snow is not so great as to be seriously troublesome.

The actual coefficient of performance of a machine of this class is much less than that of a machine using for working substance a vaporisable liquid such as ammonia. This is partly due to the relatively great waste of power, through friction, in air machines, and partly due to the practical necessity of using, in them, a much wider range of temperature than the range through which refrigeration is to be carried on. To keep the dimensions of the machine within reasonable bounds, the air is cooled by expansion to a temperature much lower than that of the cold chamber, and is

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[1] For particulars of the construction and performance of these machines see Coleman, *Min. Proc. Inst. C. E.* vol. LXVIII. 1882, p. 146; Lightfoot, *Proc. Inst. Mech. Eng.* 1881, p. 105, and 1886, p. 201.

heated by compression to a temperature considerably higher than that of the cooling water. When the working substance is a liquid which is being alternately vaporised and condensed the heat is much more easily got into and out of it. The efficiency of a vapour machine can be made to approach more closely to the ideal of a perfect refrigerator, and the coefficient of performance of a machine using ammonia is found in practice to be about five times that of an air machine. In consequence of their higher efficiency and small bulk vapour compression machines have now to a great extent displaced air machines in all applications of cold storage both on land and on board ship.

107.   The Reversed Heat-Engine as a Warming Machine. It was pointed out by Lord Kelvin in 1852 that the reversed heat-engine cycle might serve not only as a means of cooling but as a means of warming[1]. Let it be required for instance to raise and keep the temperature of a room above the temperature of the surrounding air. A machine of the Bell-Coleman type may take in air from the atmosphere, expand it so as to lower the temperature somewhat, and allow the temperature to rise again by conduction from external air. Then let it compress the air so as to restore it to atmospheric pressure. The temperature of the air will be thereby raised above the temperature of its surroundings, and it may then be discharged into the room which is to be warmed. The effect is, that by expending some mechanical work a quantity of heat is transferred from the cold atmosphere to the warmer room,—a quantity which may be far greater than the thermal equivalent of the work spent in driving the machine. For if the machine were reversible the heat rejected to the room A, namely Q_A, would be to the heat extracted from the atmosphere, namely Q_C, as τ_1 is to τ_2, and

$$\frac{Q_A}{W} = \frac{Q_A}{Q_A - Q_C} = \frac{\tau_1}{\tau_1 - \tau_2},$$

where W is the work expended, expressed in thermal units. When the range of temperature is small Q_A may be many times greater than W, that is to say, a very large amount of heating through a small range may be achieved with but little expenditure of mechanical work.

[1] *Proc. of the Phil. Soc. of Glasgow*, vol. III. p. 269, or *Collected Papers*, vol. I. p. 515.

The importance of the suggestion lies in the fact that the necessary power may be obtained, by means of a heat-engine, with a smaller supply of heat than would be required to effect the warming directly, provided the range of temperature of the warming be less than the range through which the heat-engine works in generating the required power. Burning fuel to warm a room by a few degrees is a wasteful way to utilise heat, even if all the heat of combustion be conceived to pass into the air of the room. The high-temperature heat produced in the combustion of coal or gas could warm a much larger volume of air to the same extent if it were applied to drive an efficient heat-engine, which in its turn drove a reversed heat-engine or warming machine to pump up heat through a short range of temperature from the diffused store of heat which is contained in the atmosphere or in the ocean. This is because a heat-engine can be arranged to take advantage of the high temperature at which heat is produced in the burning of fuel, whereas any direct communication of this high-temperature heat to a comparatively cool body, such as the air of a room, is thermodynamically bad. It is interesting, and may some day be useful, to recognize that even the most economical of the usual methods employed to heat buildings, with all their advantages in respect of simplicity and absence of mechanism, are in the thermodynamic sense spendthrift modes of treating fuel.

108. Transmission of Power by Compressed Air. A brief reference may be made in passing to the process, used on a large scale in Paris and elsewhere, of distributing power from central stations by compressing air there and conveying the compressed air through pipes to the places where it is to be used in driving engines, which are generally of the piston and cylinder type. Though the purpose of the process is not to produce refrigeration it may conveniently be noticed here.

Imagine the compression to be performed exceedingly slowly, in a conducting cylinder, so that the air within may lose heat by conduction to the atmosphere as fast as heat is generated by compression; the process will in that case be isothermal, at the temperature of the atmosphere. Imagine further that the compressed air is distributed without change of temperature, and that the process of expansion in the consumer's engine is also indefinitely slow and consequently isothermal. In that case (if we

neglect the losses caused by friction in the pipes) there would
be no waste of power in the whole process of transmission.   The
indicator diagram would be the same, per lb. of air, in the com-
pressing engine as in the consumer's engine, namely *fcae* (fig. 45)
in one and *eacf* in the other, *ac* being an isothermal line.
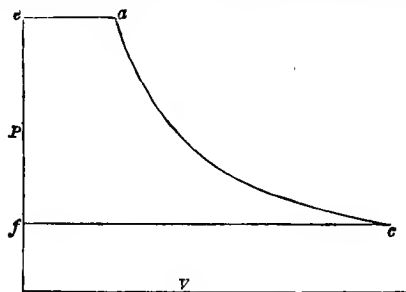
Fig. 45.

Imagine, on the other hand, that the compression and expansion
are both adiabatic—a state of things which would be approximated
to if they were performed very quickly.   Then the diagram of the
compression is *fcbe* (fig. 46) and that of the consumer's engine is
eadf (fig. 47), *cb* and *ad* being adiabatic lines.   The change of
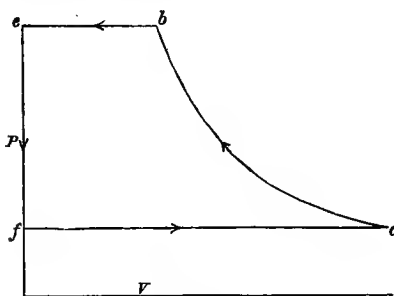
Fig. 46.
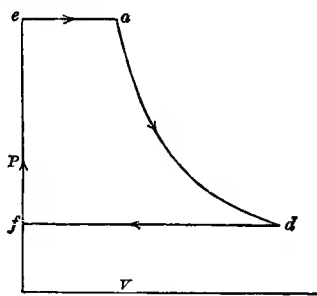
Fig. 47.

volume of the compressed air from *eb* to *ea* occurs through its
cooling in the distributing pipes, from the temperature produced
by adiabatic compression down to the temperature of the atmo-
sphere.   Superposing the diagrams as in fig. 48 and sketching an
isothermal line between *a* and *c* (both of which are points at
atmospheric temperature) we see that the use of adiabatic com-
pression involves a waste of power which is measured by the area

cba, while the use of adiabatic expansion by the consumer involves a further waste measured by *acd*.

When the expansion and compression are both adiabatic, the efficiency of the process is readily found from the fact that a horizontal line drawn at any pressure to meet the curves *ad* and *bc* is divided by *ad* into segments the ratio of which is constant
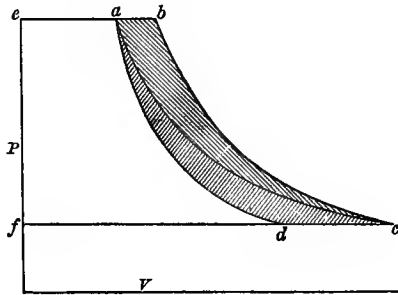
Fig. 48.

and equal to the ratio of *ea* to *ab* or of *fd* to *dc*. Hence the efficiency, which is the ratio of the area of the expansion diagram to that of the compression diagram, is $\frac{ea}{eb}$ or $\frac{fd}{fc}$. This may be expressed as $\frac{\tau_d}{\tau_o}$, or $\frac{\tau_d}{\tau_a}$ since $\tau_a = \tau_c$. Hence the efficiency is $\left(\frac{P_d}{P_a}\right)^{\frac{\gamma-1}{\gamma}}$. As an example, say that the air is compressed from 1 to 4 atmospheres: the efficiency is then $\frac{1}{4^{0.29}}$ or 0·67. When the expansion and compression are isothermal the efficiency, so far as these processes are concerned, is unity.

In practice the compression cannot be made strictly isothermal for want of time. The temperature of the air is prevented as far as possible from rising during compression by injecting water into the compressing cylinder, and in this way the curve which would be $PV = $ const. if isothermal and $PV^{1\cdot4} = $ const. if adiabatic takes an intermediate position between *ca* and *cb* (as examination of the actual indicator diagram shows), and may be roughly expressed by the equation

$$PV^{1\cdot2} = \text{const.}$$

Again, the waste of power in compression may be reduced by dividing the process into two or more stages (performed in two or

more successive cylinders) and cooling the air between one stage
and the next.   In this way a stepped compression curve such as
cghijk (fig. 49) can be obtained which approximates more nearly
to the isothermal curve *ca*, and the loss is consequently reduced
by the amount of the cross-hatched area.   The saving so effected
is considerable when air is highly compressed.
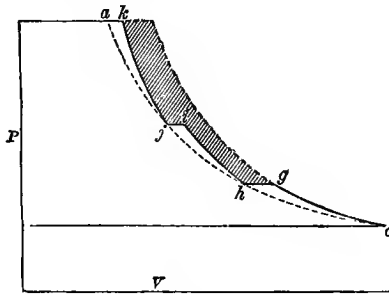
FIG. 49.

Similar devices may be used by the consumer to make the
expansion curve in his engine approximate more nearly to the
isothermal line : that is, he may inject water or use a compound
engine, allowing the air time to take up enough heat to restore it
more or less nearly to atmospheric temperature between one stage
of expansion and the next.   By these means the efficiency of the
transmitting system as a whole (neglecting all losses due to friction
in the distributing pipes, in the valves of the engines, etc.) may be
made to approximate to unity.

There is, however, another point to be considered.   If the
temperature be allowed to fall materially during expansion the
same difficulties present themselves as were referred to above in
speaking of refrigerating machines : the expanding air tends to
deposit dew or even snow.   To prevent this the practice is often
followed of passing the compressed air through a stove or
"preheater" in order to raise its temperature just before it is
allowed to expand, and so prevent the deposit of frozen moisture.
When "preheaters" are used the extra heat which they supply is
of course itself partly converted into work[1].

[1] On the subject of transmission of power by compressed air reference should
be made to papers by Sir A. B. W. Kennedy, *Brit. Assoc. Rep.* 1889, p. 448, and
Prof. Nicolson, *Engineering*, July 7, 1893. See also Prof. Peabody's *Thermody-
namics of the Steam-Engine*, 5th ed. chap. **xv.**

CHAPTER VII.

109. Comparison of actual and ideal indicator diagrams.
We have now to consider in what respects the action of steam in
a real engine differs from the ideal action described in § 73 of
Chapter III., where a hypothetical engine was considered in which
the Rankine Cycle of operation was followed, which is as near an
approximation to Carnot's Cycle as can be reached without the
use of adiabatic compression. An engine imagined to work in the
manner there described, and having an indicator diagram of
the type shown in fig. 19, where the expansion is adiabatic and
complete, forms a useful standard with which to compare the
action of any real engine. The efficiency of the real engine
is always less, for reasons which will be discussed in this
chapter.

In the first place, the expansion in real engines is not (except
in rare cases) complete: the steam at release has a pressure which
is higher than the pressure in the condenser if the engine is a
condensing engine, or higher than the pressure of the atmosphere
if the engine is non-condensing. Reasons for this have been
already indicated: complete expansion would increase unduly the
bulk and weight of the engine; the work done by the steam in
the last stages would add nothing to the net mechanical output
for it would be used up in overcoming the friction of the piston;
further, complete expansion would aggravate certain evils to be
described later which arise from the cooling of the cylinder during
expansion and exhaust. For these reasons it is practically desirable
to cut off the toe of the ideal diagram sketched in fig. 19. The
effect which incompleteness in the expansion produces by itself on
the efficiency of the ideal process has already been considered in
reference to the indicator diagram, fig. 21, and to the entropy-
temperature diagram, fig. 33 (§ 97).

Other features of difference are most conveniently noticed by
comparing stage by stage the ideal diagram of fig. 21 with a

diagram taken from a real engine.   In the action to which figs. 19 and 21 refer it was assumed—(1) that the steam was supplied in the dry saturated state, and had during admission the full (uniform) pressure of the boiler P_1; (2) that there was no transfer of heat to or from the steam except in the boiler and in the condenser; (3) that after more or less complete expansion all the steam was discharged by the return stroke of the piston, during which the back pressure was the (uniform) pressure in the condenser P_2; (4) that the whole volume of the cylinder was swept through by the piston.   It remains to be seen how far these assumptions are untrue in practice, and how the efficiency is affected in consequence.
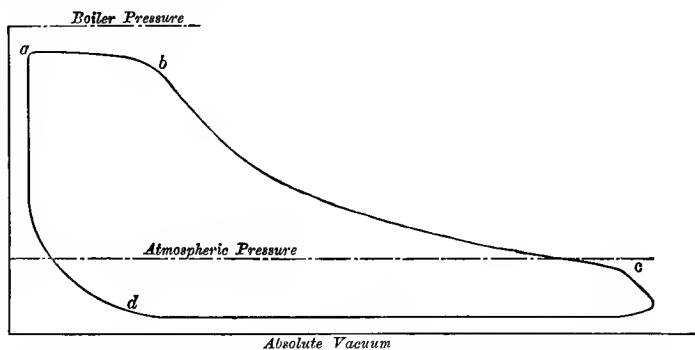
FIG. 50.   Typical Indicator Diagram from a Condensing Steam-Engine.

The actual conditions of working differ from these in the following main respects, some of which are illustrated by the practical indicator diagram of fig. 50, which is taken from an actual engine.

110. Wire-drawing during Admission and Exhaust. Owing to the resistance of the ports and passages, and to the inertia of the steam, the pressure within the cylinder is less than P_1 during admission and greater than P_2 during exhaust.

Moreover P_1 and P_2 are themselves not absolutely uniform, and P_2 is greater than the pressure of steam at the temperature of the condenser, on account of the presence of some air in the condenser.   The presence of air is accounted for partly by its entering the boiler dissolved in the feed water, and partly by its leaking into the cylinder and other parts of the engine at times when the pressure within is less than the pressure of the atmosphere.

During admission the pressure of steam in the cylinder is less than the boiler pressure by an amount which often increases a little as the piston advances, on account of the increased velocity of the piston's motion and the consequently increased demand for steam. When the ports and passages offer much resistance the steam is expressively said to be "throttled" or "wire-drawn." The steam is dried by the process to a small extent, as was shown in § 82, and if initially dry it becomes superheated. In an indicator diagram wire-drawing causes the line of admission to lie below a line drawn at the boiler pressure, and generally to slope a little downwards. In fairly good practical instances the mean absolute pressure during admission is about nine-tenths of the pressure in the boiler. With a long steam-pipe or a badly designed valve the fall of pressure may be greater, and the effect is aggravated when the steam is allowed to become wet by leaving the steam-pipe bare or insufficiently covered, instead of having the pipe properly "lagged" with some material which is a poor conductor of heat. Even under the best conditions some of the steam is condensed on its way to the engine by loss of heat from the pipe. There is in general some additional water present in the steam through what is called "priming" on the part of the boiler, that is to say the delivery of steam in which particles of water are mechanically suspended. Whatever water is present, from either cause, may be more or less completely removed by the use of what is called a "separator," but usually the steam is to some extent wet when it enters the cylinder, notwithstanding the slight tendency which wire-drawing has to dry it. The separator is a vessel through which the steam passes on its way to the engine and in which the suspended particles of moisture settle, the accumulated water being drained off from time to time. In many cases the steam is made to take such a course through the separator that the centrifugal action assists in causing the particles of water to be thrown off.

Again, during the exhaust the actual back-pressure exceeds the pressure in the condenser by an amount that depends on the freedom with which the steam makes its exit from the cylinder. In condensing engines with a good vacuum the back-pressure is often as much as 3 lbs. per square inch and even more, and in non-condensing engines it is 16 to 18 lbs. in place of the 14·7 lbs. or so which is the pressure of the atmosphere.

The excess of back-pressure may be greatly increased by the presence of water in the cylinder. The effects of wire-drawing do not stop here. The valves open and close more or less slowly; the points of cut-off and release are therefore not absolutely sharp, and the diagram has rounded corners at b and c in place of the sharp angles which mark those events in fig. 21. For this reason release is allowed in practice to begin a little before the end of the forward stroke, hence the toe of the diagram takes a form like that shown in fig. 50. The sharpness of the cut-off, and to a less extent the sharpness of the release, depends greatly on the kind of valves and valve-gear used; valves operated by a trip gear, for instance, such as will be described in a later chapter, stop the admission of steam more suddenly than the ordinary slide valve does and therefore produce a diagram in which the events of the stroke are more sharply defined.

111. Clearance. When the piston is at either end of its stroke there is a small space left between it and the cylinder cover. This space, together with the volume of the passage or passages leading thence to the steam and exhaust valves, is called the *clearance*. It constitutes a volume through which the piston does not sweep, but which is nevertheless filled with steam when admission occurs, and the steam in the clearance forms a part of the whole steam which expands after the supply from the boiler is cut off. If AC be the volume swept through by the piston up to release, OA the volume of the clearance, and AB the volume swept through during admission, the apparent ratio of expansion is $\dfrac{AC}{AB}$, but the real ratio is $\dfrac{OA + AC}{OA + AB}$.

Clearance must obviously be taken account of in any calculation of curves of expansion. It is conveniently allowed for in indicator diagrams by shifting the line of no volume back through a distance corresponding to the clearance in the manner illustrated in fig. 51. In actual engines the volume of the clearance OA is usually from $\frac{1}{10}$ to $\frac{1}{50}$ of the volume of the cylinder. Its size

Fig. 51. Effect of Clearance.

depends largely on the kind of valve that is used. As a rule small engines have relatively more clearance than large ones.
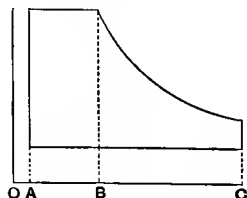
112. Compression. Clearance affects the thermodynamic efficiency of the engine chiefly by altering the amount of steam that is consumed per stroke, and its influence depends materially on the extent to which the *compression* of part of the steam during the return stroke, referred to in § 73, is carried on. If there were no compression: if, in other words, the exhaust pipe leading to the condenser or to the atmosphere were left open throughout the whole of the back stroke, at the end of that stroke the clearance space would have nothing more in it than steam at a pressure equal to the back-pressure, and consequently at the next admission enough steam would have to be drawn from the boiler to bring up the pressure in the clearance as well as to fill the volume which is swept through by the piston up to the point of cut-off. With compression this cause of waste is more or less completely avoided. During the back stroke the process of exhaust is discontinued before the end as at *d* in fig. 50, and the steam remaining in the cylinder is compressed. The cushion of steam thus shut in finally occupies the volume of the clearance; and by a proper selection of the point at which compression begins the pressure of this cushion may be made to rise just up to the pressure at which steam is admitted when the valve opens. This may be called complete compression, and when it occurs the existence of clearance has no direct effect on the consumption of steam nor on the efficiency; for there is then simply a permanent cushion which is alternately expanded and compressed without net gain or loss of work, in addition to the working steam proper, which on admission fills the volume AB (fig. 51), and which enters and leaves the cylinder in each stroke. But if compression be incomplete or absent there is, on the opening of the admission valve, an inrush of steam to fill up the clearance space. This increases the consumption to an extent which is only partly counterbalanced by the increased area of the diagram, and the result is that the efficiency is reduced. The action is, in fact, a case of unresisted expansion (§ 52), and consequently tends, so far as its direct effects go, to make the engine less than ever reversible. Incidentally, compression has the mechanical advantage that it obviates the shock which the admission of steam would otherwise produce, and increases the smoothness of running by giving the piston work to do while its velocity is being rapidly reduced—an action which receives the name of "cushioning."

12—2

The opening of the steam valve for admission being a somewhat gradual process, it generally begins before the back stroke is quite complete, in order that the valve may be widely enough open to let the steam in freely when the piston begins to move forwards. The valve is then said to have *lead,* and the effect is to produce what is called *pre-admission.* Pre-admission tends to increase the mechanical effect of cushioning which has just been referred to.

113. Cushion Steam and Cylinder Feed. In dealing with the influence of clearance, whether the compression be complete or incomplete or even altogether wanting, it is convenient to think of the working substance in the cylinder as made up of two parts, namely, (1) the part that has been shut up in the clearance from the previous stroke, and (2) the part that is freshly supplied from the boiler. For brevity we shall refer to these in what follows as (1) the cushion steam, and (2) the cylinder feed. During expansion the whole quantity of working substance in the cylinder is the sum of these two; during compression the cushion steam only is present. If the steam which leaves the engine is condensed and the condensed water weighed, its quantity forms a measure of the cylinder feed, from which the amount of steam passing through the cylinder per stroke may be deduced. But to this amount the cushion steam must be added when it is desired to know the whole quantity of steam (or rather of working substance) present in the cylinder.

114. Influence of the Cylinder Walls. Condensation and Re-evaporation in the Cylinder. The Missing Quantity. Generally by far the most important element of difference between the action of a real engine and that of our hypothetical engine is that which was alluded to at the end of Chapter I., the difference, namely, which proceeds from the fact that the cylinder and piston are not non-conductors. As the steam fluctuates in temperature in the phases of admission, expansion and exhaust there is a complex give-and-take of heat between it and the metal it touches, and the effects of this, though not very conspicuous on the apparent form of the indicator diagram, have an enormous influence in reducing the efficiency by increasing the consumption of steam. Attention

was drawn to this action by Mr D. K. Clark as early as 1855[1], and the results of his experiments on locomotives were confirmed and extended in 1860 by Mr Isherwood's trials of the engines of the United States steamer "Michigan[2]." Rankine in his classical work on the steam-engine notices the subject only very briefly, and takes no account of the action of the cylinder walls in his calculations. Its importance became established beyond dispute, notably, among early experiments, by those of Messrs Loring and Emery on the engines of certain revenue steamers of the United States[3], and by a protracted series of investigations carried out by M. Hallauer and other Alsatian engineers under the direction of Hirn[4], whose name should be specially associated with the rational analysis of engine tests, and who was one of the first to recognize the losses that result from condensation of steam on the surface of the cylinder. The evidence afforded by these experiments has been confirmed by trials made on all kinds of engines and under every variety of working. The following is, in general terms, what experiments with actual engines show to take place.

When the amount of steam that has passed through the engine is measured, by weighing either the feed water or the condensed steam discharged from the condenser, it is found to be greatly in excess of the quantity of dry steam that would suffice to fill the cylinder volume up to the point of cut-off, at the pressure which the steam then has, even when the effects of clearance are fully allowed for. The difference between the two is called the *missing quantity*. It is to be accounted for in two ways:—

(1) Some steam leaks directly across from the steam side to the exhaust side and so escapes being measured by the displacement of the piston.

(2) At the point of cut-off the steam in the cylinder is by no means dry: consequently more working substance passes through

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[1] *Railway Machinery*, or art. STEAM-ENGINE, *Ency. Brit.* 8th edition. See also *Min. Proc. Inst. C. E.* vol. LXXII. p. 275.

[2] See Isherwood's *Experimental Researches in Steam Engineering*, Philadelphia, 1863. This important work describes a great number of experiments, undertaken at a time when engineers in general were but little alive to their value.

[3] An abstract of Messrs Loring and Emery's reports is given in *Engineering*, vols. XIX. and XXI., and in Mr Maw's *Recent Practice in Marine Engineering*.

[4] *Bull. Soc. Industr. de Mulhouse*, from 1877.

the engine at each stroke than corresponds to the volume of dry steam that would fill the admission space.

The missing quantity is rarely less than 20 per cent. of the steam supplied by the boiler, often as much as 30 per cent., and is sometimes as much as 50 per cent. Even 69 per cent. has been recorded in trials of a small engine[1].

Of the two factors which make it up the second is in general the more important, namely the effect of initial condensation in making the steam wet at the point of cut-off. The excessive amount of the missing quantity in some engines, especially where slide valves are used, has been shown by the experiments of Callendar and Nicolson[2] to be partly due to direct leakage, but in most cases it is the give and take of heat between steam and metal to which the chief effect is to be ascribed.

When steam is admitted at the beginning of the stroke, it finds the metallic surfaces of the cylinder and piston chilled by having been exposed to low-pressure steam during the exhaust of the previous stroke. A portion of it is therefore at once condensed, and, as the piston advances, more and more of the chilled cylinder surface is exposed and more and more of the hot steam is condensed. At the end of the admission, when communication with the boiler is cut off, the cylinder consequently contains a film of water spread over the exposed surface, in addition to saturated steam. The boiler has therefore been drawn upon for a supply of steam greater by perhaps 20 or 25 per cent. than that which corresponds to the volume of the admission space.

Then, as expansion begins, more cold metal is uncovered, and some of the remaining steam is condensed upon it. There is in addition a further condensation which takes place in consequence of the work the steam is doing during expansion—a condensation which would be found even if the walls were perfect non-conductors and the process were strictly adiabatic. So far as these two actions are concerned, the mixture is getting wetter as it begins to expand. But the pressure of the steam now falls, and the layer of water

[1] See papers by Col. English (*Proc. Inst. Mech. Eng.* Sept. 1887, Oct. 1889, May 1892), which describe experiments on this subject. In several cases examined by him the amount is over 60 per cent.

[2] H. L. Callendar and J. T. Nicolson, "On the law of condensation of steam deduced from measurements of temperature-cycles of the walls and steam in the cylinder of a steam-engine," *Min. Proc. Inst. C. E.* vol. cxxxi. 1897.

which has been previously deposited begins to be re-evaporated as soon as the temperature of the expanding steam falls below that of the liquid layer.   Hence, on the whole, the amount of water present increases during the earliest part of the expansion, but a stage is soon reached when the condensation which occurs on the newly exposed metal or throughout the steam as a whole in consequence of expansion is balanced by re-evaporation of older portions of the layer.   The percentage of water present is then a maximum ; and from this point onwards the mixture of steam and water in the cylinder becomes more and more dried by re-evaporation of the layer.

If the amount of initial condensation has been small this re-evaporation may be complete before release occurs.   Very usually, however, there is still an undried layer at the end of the forward stroke, and the process of re-evaporation continues during the return stroke, while exhaust is taking place.   In extreme cases, if the amount of initial condensation has been very great, the cylinder walls may fail to become quite dry even during the exhaust, and a residue of the layer of condensed water may either be carried over as water into the condenser, or, if the exhaust valves are not arranged so that it can be discharged, this un-evaporated residue may gather in the clearance space, and in very bad cases may even require the drain-cocks to be left open to allow of its escape.   When any water is retained in this way it may be conjectured that the initial condensation will be in-creased, for the hot steam then meets not only comparatively cold metal but comparatively cold water when it enters the cylinder. The latter tends to cause much condensation, partly because of its high specific heat, and partly because it is brought into intimate mixture with the entering steam.

Apart, however, from this extreme case, whatever water is re-evaporated during expansion and exhaust takes heat from the metal of the cylinder, and so brings it into a state that makes condensation inevitable when steam is next admitted from the boiler.   It is in fact the condensation of the layer and its re-evaporation, whether during expansion or during exhaust, that is the means of exchange of heat between the metal of the cylinder and the working substance.   Mere contact with low-pressure steam during the later stages of expansion and during the ex-haust stroke would cool the metal but little, for communication

of heat between dry metal and any gaseous substance is slow even when the difference of temperature between them is large. The cooling of the cylinder walls which actually occurs is due mainly to the re-evaporation of the condensed water. Thus if an engine were set in action, after being heated beforehand to the boiler temperature, the cylinder would be only slightly cooled during the first exhaust stroke, and little condensation would occur during the next admission. But the metal would be more cooled in the subsequent expansion and exhaust, since it would part with heat in re-evaporating this water. In the third admission more still would be condensed, and so on, until a permanent *régime* would be established in which condensation and re-evaporation were exactly balanced. The same permanent *régime* is reached when the engine starts cold.

However early the re-evaporation of the condensed film is completed it results in some chilling of the cylinder walls, leaving them to be re-heated by condensation of fresh steam in the next stroke. The evils of initial condensation are greater the later this re-evaporation is completed. If the steam in the condensed layer is all evaporated before the release but little further cooling of the metal will occur during the exhaust stroke: if water remains to be evaporated during exhaust the whole action of the walls is intensified. It is only in exceptionally favourable cases that the water condensed during admission is completely evaporated before release.

115.   The Balance of Heat in the action of the Cylinder Walls. When steam is alternately condensed and evaporated by contact with the cylinder walls in the action of a steam-engine more heat is given to the metal by each pound that is condensed than is taken from the metal by each pound that is evaporated. This is because the condensation takes place at a higher temperature and pressure than the re-evaporation. The heat given up to the metal for each unit quantity of steam condensed upon it by contact at any temperature t_1 is the latent heat L_1 together with the heat which the layer of condensed water gives up in falling from t_1 to the temperature t_2 at which re-evaporation occurs, namely $h_1 - h_2$. The heat taken from the metal, by re-evaporation of the layer, is L_2. But $L_1 + h_1 - h_2$ is greater than L_2, and consequently the process could not go on unless the metal were

losing heat in some other way, for so far as this alternate condensation and re-evaporation is concerned the result would be an accumulation of heat in the metal which would increase without limit.

This accumulation does in fact go on for a short time when the engine starts cold and while the metal is being warmed up. But when a permanent *régime* has become established it is clear that in each revolution of the engine the total gains and losses of heat on the part of the metal must just balance one another. This implies that the metal is losing more heat than is accounted for by the re-evaporation of the initially condensed layer. In an unjacketed engine, and especially if the cylinder is not well lagged, there is a loss by conduction to the outside, which is an important item in the general balance of losses and gains. But by using a steam jacket the metal may be prevented from losing heat to the outside, and may even be made to take up heat. Under these conditions it is on the fact that more water is re-evaporated than was condensed by contact that the balance depends, and it is this that makes alternate condensation and re-evaporation possible.

This excess of water to be re-evaporated may arise simply as a consequence of the work done during expansion. We have seen that in a strictly adiabatic process steam becomes partially condensed as it expands doing work. Consequently, in an actual engine there is a greater quantity of water available for evaporation in the later stages of the cycle than was condensed by contact with the walls in the early stages. But the excess may also arise from the presence of water in the steam supply. Any water that enters the cylinder as water will be available for evaporation in the later stages of the cycle and may therefore be a factor in establishing the balance of give-and-take of heat to which initial condensation is due.

On the other hand, it is important to notice that if the cylinder valves are so situated as to facilitate drainage of water from the cylinder there will be a tendency to check the whole action by removing the excess of water before re-evaporation has taken place.

A comparatively small excess of water to be re-evaporated is competent to make a large amount of alternate condensation and re-evaporation possible. To illustrate this take a numerical

example. Suppose we have steam admitted at a temperature of 160° C. (say 90 lbs. per sq. inch absolute), and expanded to 70° C. (4·5 lbs. per sq. inch). To simplify the problem we shall ignore the action at intermediate temperatures and think of the condensation on the walls as taking place at 160° C. and the re-evaporation as taking place at 70° C. On that basis L_1 is 491·4 and $h_1 - h_2$ is 91·6, so that for each unit of steam condensed by contact the metal gains 583 units of heat. It loses heat equal to L_2, or say 557 units, by the re-evaporation of the same quantity of water. Hence to preserve the balance 26 additional units of heat must be taken from the metal for every unit quantity of water that is initially condensed by contact. The proportion of 26 to 557 is barely one-twentieth, and hence under these conditions any extra quantity of water that is available to be evaporated at the lower temperature, in addition to the quantity initially condensed by contact, is competent to account for the alternate condensation and re-evaporation of fully twenty times its own weight of the working steam.

The work done during expansion, between the limits of temperature chosen in this example, would itself produce 14 per cent. of wetness if the steam were dry at cut-off, and 8 per cent. if the steam at cut-off contained 20 per cent. of water: it would therefore be much more than sufficient to supply the amount of extra water for re-evaporation which is necessary to account for a large amount of initial condensation. It is however open to question how far the water that is due to the work done during expansion takes a form which permits of re-evaporation in the later stages of the cycle. In so far as it is deposited on the walls it is of course available, but whatever part of it is mechanically suspended as a mist throughout the cylinder would not contribute to the action.

The investigations of Callendar and Nicolson led them to conclude that the amount of cylinder condensation is in general limited by the time rate at which condensation can occur on the metallic surfaces with which the steam comes in contact, having regard to the difference of temperature between the metal and the steam. By examining the fluctuations of temperature in the metal, and also in the steam, during the revolution of the engine they deduced a limit to the amount of condensation that should occur in the cycle. When there is a sufficient excess of water

available for re-evaporation this limit may be expected to be reached in the action of the engine, and the amount of initial condensation occurring per stroke will then not be much affected by small changes in the conditions of operation.

But the case is different if the amount of the action is restricted to something short of that limit by being dependent on the heat balance. In that case a very small change in the conditions may effect a very large change in the amount of the initial condensation. The balance, as Callendar and Nicolson remark, is extremely delicate and is very easily turned. A little water in the steam supply may mean much increase in the whole effect: on the other hand, a little removal of water by drainage, or a little addition of heat by superheating or by jacketing may mean much reduction in the whole effect. It is to considerations of this kind that much of the advantage actually found in superheating or in the use of the steam jacket is to be ascribed.

116. Leakage affecting the Missing Quantity. In Callendar and Nicolson's experiments the missing quantity was much larger than they could attribute to cylinder condensation, in view of the limit which their observations of the temperature of the cylinder walls imposed on the amount of that condensation. The discrepancy was explained by the discovery that a large part of the missing quantity arose from leakage past the valve, direct from the steam chest to the exhaust. A slide valve of the ordinary type was used. When tested in a stationary position it was steam-tight, but experiments devised to examine the leakage when it moved showed that while in action it allowed a large quantity of steam to pass to the exhaust without entering the cylinder at all. The leakage appears to take place not as steam but as water, a film of which is formed on the metal surface over which the valve slides in consequence of the surface having been chilled by previous contact with exhaust steam. Then the film is driven through under the valve face by the difference of pressure between the steam and exhaust sides. These experiments suggest that in some cases when the missing quantity is very large the greater part of it may be due to valve leakage : they also show the practical advantage of having separate valves for admission and exhaust, an arrangement which is in fact adopted in engines of the most efficient class. Apart from questions of leakage, the use of separate valves has the advantage of reducing

condensation by not exposing the surface of the same steam port alternately to hot and colder steam.

117. Graphic Representation, on the Indicator Diagram, of the Water present during Expansion. In testing engines the amount of steam is measured which passes through the cylinder per stroke—that is, the quantity which we have called the "cylinder feed." The whole quantity of steam and water present during expansion is the cylinder feed *plus* the cushion steam. To estimate the amount of the cushion steam we take, on the indicator diagram, a point after compression has begun, when the exhaust valve has become completely closed, and note the pressure and the volume there, remembering that the true volume is the sum of the uncompleted portion of the stroke and the clearance. From this pressure and volume the quantity of the cushion steam is readily calculated, if we may assume that the steam is simply saturated and that no water is present when compression begins. As a rule, this assumption is probably correct : occasionally the cushion steam may be wet, which would make its amount greater, but in most cases the supposition that the steam is dry when compression begins may be accepted as involving at least no serious error. The total quantity of steam which is or should be in the cylinder during expansion is next found by adding the amount of this cushion steam to the cylinder feed. A "saturation curve" can then be drawn on the indicator diagram to show the volume which this total quantity would fill if it were dry and saturated at each pressure reached during the expansion. An example is shown in the indicator diagram of fig. 52, where SS is the saturation curve. In drawing this line the axis of no volume is to be taken to the left of the diagram which the indicator traces, by a distance which represents the volume of the clearance. Then if a horizontal line ABS be drawn to intersect the expansion curve at any point B, AB is the actual volume which the expanding mixture filled at this pressure, AS is the volume it would have filled if dry and saturated ; BS is the volume of the missing quantity, which is due to wetness and to valve leakage. If it is all due to wetness the proportion of water in the mixture is sensibly $\dfrac{BS}{AS}$, and the dryness q is $\dfrac{AB}{AS}$. Thus the proportion of water present at any stage of the expansion is determined and is shown in the diagram.

Fig. 52 relates to a real case—a trial, by the author, of a small engine of the marine type. The amount of cylinder feed per single stroke was 0·0404 lb. The pressure at the point D was found to be 4 pounds per square inch, and the volume there was 0·12 cub. ft. Since the volume of 1 lb. at that pressure is 90 cub. ft., it follows that the amount of cushion steam was 0·0013 lb. This gives a total of 0·0417 lb., for which the curve SS is drawn. By measuring values of $\dfrac{BS}{AS}$ at points along the curve it is found that the proportion of water in the mixture was 52 per cent. at cut-off, then increased to about 55 per cent. during the early stages of expansion, then became less, and finally sank to 37 per cent. just before release. These figures assume
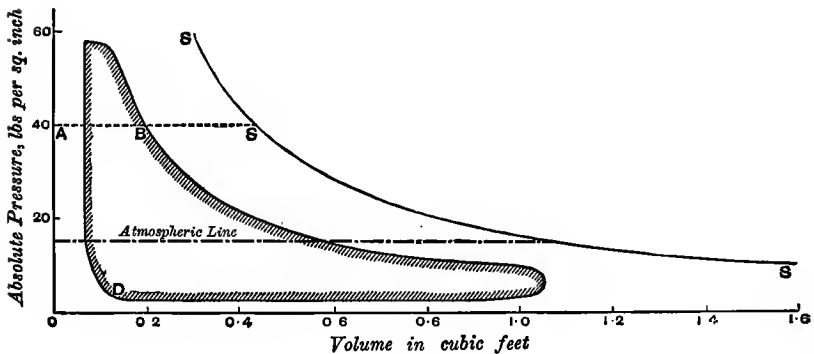
Fig. 52.

the missing quantity to be all due to wetness, but valve-leakage may account for part.

Again, knowing the wetness of the mixture at the point of cut-off we may draw an adiabatic line through that point using the equation $Pv^n = $ constant with a suitable value of n (see § 68). This curve will in general be found to lie a trifle above the actual expansion curve at first, but to cross it early and lie distinctly below it towards the end of expansion. This is because the metal continues for some time after cut-off to take heat from the working fluid, but later gives up heat to it through the re-evaporation of the condensed film.

By comparing the adiabatic with the actual expansion curve it is possible to examine the give-and-take of heat between the metal and the working fluid. But this is more conveniently done after

the entropy-temperature curve has been drawn, as will be presently described.

When tests of compound engines are in question it is useful to modify the construction shown in fig. 52 by separating the cylinder feed from the cushion steam, and drawing the diagram for the former. This allows a combined diagram for the several cylinders to be drawn, along with a single saturation curve. The reason is that the amount of cylinder feed is the same for both or all the cylinders, whereas the amount of cushion steam may be very different. An example of this construction will be given later in dealing with compound engine trials.

118. Use of the Entropy-Temperature diagram in exhibiting the behaviour of steam during expansion and the exchanges of heat between it and the cylinder walls. In the entropy-temperature diagram, fig. 53, let *ab* be drawn at
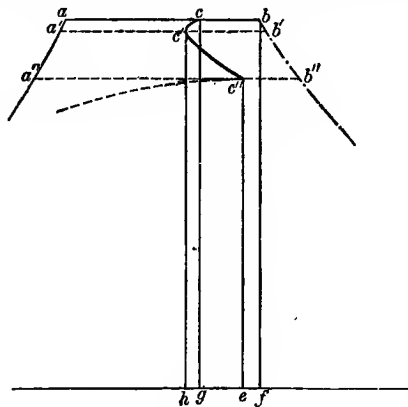
Fig. 53.

the temperature which corresponds to the pressure at the point of cut-off, and let it be divided at *c* so that $\dfrac{ac}{cb}$ represents the proportion of dry steam to water in the total quantity of working fluid present in the cylinder. Similarly, at any lower temperatures reached during expansion let lines $a'b'$, $a''b''$ be divided at points c', c'' in the proportion of steam to water then present, making

$$\frac{a'c'}{a'b'} = \frac{AB}{AS}$$

at the corresponding pressure in the indicator diagram (fig. 52). In this way the curve $cc'c''$ is determined, which represents the real process of expansion, and this is readily compared with the ideal adiabatic process represented by the straight vertical line cg. Taking c'' as the point of release the diagram may be continued by drawing a constant-volume curve as described in § 97. In the first stages of expansion, namely from c to c' in the sketch, the proportion of water in the cylinder is increasing, and the heat abstracted by the cylinder walls from the steam is the area $cghc'$. From this point onwards the steam becomes drier, and takes up heat from the metal, the whole amount recovered up to the point of release being the area $c'c''eh$. It will be seen that a diagram of this type is particularly well fitted to allow the transfer of heat between metal and fluid to be traced throughout all stages of the expansion, the heat given up or recovered in any part of the process being equal to the area under the corresponding portion of the expansion curve $cc'c''$. When this curve slopes down to the left heat is passing from the steam to the metal; when it slopes down to the right the exchange is the other way. The heat abstracted from the steam during compression and admission is nearly equal to the area $fbcg$—nearly, but not exactly, because all the condensation in these stages does not occur at the pressure of cut-off. During compression condensation is going on at lower pressures because the temperature of the cushion steam—necessarily rising with the pressure—is being raised above the temperature to which the walls have been chilled during exhaust.

119. Thermodynamic Loss due to Initial Condensation. From a thermodynamic point of view all initial condensation of the steam is bad, for, however early the film of water be re-evaporated, this can take place only after its temperature has cooled below that of the boiler. The process consequently involves a misapplication of heat, since the substance, after parting with high-temperature heat, takes it up again at a temperature lower than the top of its range. This causes a loss of efficiency, and the loss is greater the later in the stroke re-evaporation occurs. The heat that is drawn from the cylinder by re-evaporation of the condensed film becomes less and less effective for doing work as the end of the expansion is approached, and finally, whatever evaporation continues during the back stroke is an unmitigated source of waste. The heat it

takes from the cylinder does no work[1]; its only effect, indeed, is to increase the back-pressure by augmenting the volume of steam to be expelled. A small amount of initial condensation reduces the efficiency of the engine but little; a large amount causes a much more than proportionally larger loss.

120. Action of a Steam-jacket. The action of the cylinder walls is increased by any loss of heat which the engine may suffer by radiation and conduction from its external surface. We have already seen how any such loss tends to disturb the balance of heat in a cyclical process of condensation and re-evaporation, and consequently to promote condensation. More steam is initially condensed in a cylinder which is losing heat externally. The loss of efficiency due to the action of the cylinder walls will therefore be greater in an unprotected cylinder than in one which is well lagged or covered with non-conducting material. On the other hand, if the engine have a steam-jacket the deleterious action of the walls is reduced. The working substance is then on the whole gaining instead of losing heat by conduction during its passage through the cylinder. The jacket maintains a higher mean temperature on the inner surface of the cylinder, reduces condensation, and accelerates the process of re-evaporation, tending to make it occur while the temperature and pressure of the steam are still comparatively high. After the process of re-evaporation is complete the jacket cannot superheat the steam in the cylinder to any material extent, for conduction and radiation between dry steam and the metal of the cylinder are incompetent to cause any considerable exchange of heat. The earlier, therefore, that re-evaporation is complete the less is the metal chilled, and the less is the subsequent condensation. But after re-evaporation is completed the steam in the jacket continues to give heat to the metal during the remainder of the cycle, and so warms it to a temperature more nearly equal to that of the boiler steam before the next admission takes place.

Thus a steam-jacket, though in itself a thermodynamically imperfect contrivance, inasmuch as it supplies heat to the working substance at temperatures lower than the top of the range, acts

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[1] Unless, of course, the cylinder in question is one of a compound series, and the steam that leaves it passes on to another cylinder to undergo further expansion there.

beneficially by counteracting, to some extent, the more serious misapplication of heat which occurs through the alternate cooling and heating of the cylinder walls. The heat which a jacket communicates to the working steam often increases the power of the engine to an extent far greater than corresponds to the extra supply of heat which the jacket itself requires. A jacket has the obvious drawback that it increases waste by external radiation, since it both enlarges the area of radiating surface and raises its temperature; notwithstanding this, however, many experiments have shown that the influence of a steam-jacket on the efficiency is good, especially in slow running engines and in engines where there is a large ratio of expansion in a single cylinder. This is to be ascribed to the fact that it reduces, though it does not entirely remove, the evils of initial condensation. To quote once more Watt's words, the jacket does good by helping to keep the cylinder as hot as the steam that enters it. To be effective, however, jackets must be well drained and kept full of " live " steam, instead of being, as many are, traps for condensed water or for air. The action is kept up by condensation of steam in the jacket itself. When the jacket is acting effectively the amount of steam which is condensed in it generally ranges from about 7 to 12 per cent. of the whole steam supply. The most economical treatment of the jacket-water is to allow it to drain directly back into the boiler. In some cases the activity of the jacket has been secured by letting all the steam supply pass through the jacket on its way to the cylinder, an arrangement which makes particular care necessary to prevent the water which is formed in the jacket from passing into the cylinder. The parts of the cylinder where the application of a steam-jacket is most beneficial are the ends, for it is on the end surfaces rather than on the cylindrical surface that initial condensation mainly takes place.

We shall refer presently to experiments which show the influence of steam-jackets on the efficiency of engines of various types. Meanwhile it may be said that in no trials has it appeared that a jacket has done harm : in other words, the saving of steam in the cylinder-feed brought about by the use of a jacket is always greater than the amount of steam which the jacket itself uses, and in many instances the net saving is as much as 10 or 20 per cent. The best results are found in cases where, if the jacket were absent, the conditions are such as would give rise to much initial

condensation. In engines which make a great number of strokes per minute the influence of the jacket is necessarily small[1].

The advantage of the jacket may be increased by making its temperature higher than that of the steam during admission to the cylinder. Re-evaporation of the condensed layer is further hastened, and after it is over the jacket gives up but little heat. Mr Bryan Donkin has obtained good results in experiments where the cylinder of a small engine was kept hot by gas flames, and it has been proposed to jacket engines with the hot gases from the furnace after these have passed through the boiler-flues[2].

121. Influence of Speed, Size, and Ratio of Expansion. It is interesting to notice, if only in general terms, the effects which the particular conditions of working in different engines may be expected to produce on the loss that occurs through the action of the cylinder walls. Initial condensation will be increased by anything that augments the range of temperature through which the inner surface of the cylinder fluctuates in each stroke, or that exposes a larger surface of metal to the action of a given quantity of steam, or that prolongs the contacts in which heat is exchanged. The influence of time is specially important; for the whole action depends on the rate at which heat is taken up and given up by the substance of the metal. The changes of temperature which the metal undergoes are in every case mainly superficial; the alternate heating and cooling of the inner surface initiates waves of high and low temperature in the iron whose effects are sensible only to a small depth; and the faster the alternate states succeed each other the more superficial are the effects[3]. In an engine making an indefinitely large number of strokes per minute the cylinder sides would behave like non-conductors and the action of the working substance would be adiabatic.

We may conclude, then, that in general an engine running at a high speed will have a higher thermodynamic efficiency than

[1] See the Reports of the Inst. of Mechanical Engineers' Research Committee on the Value of the Steam-jacket. *Proc. Inst. Mech. Eng.* 1889, 1892, 1895.

[2] For Mr Donkin's experiments see *Min. Proc. Inst. C. E.* 1889, vol. xcviii.

[3] The temperature of the cylinder walls has formed the subject of an interesting experimental study by Mr Bryan Donkin, who has examined the general gradient of temperature across the walls, both with and without steam in the jacket. See his papers, *Min. Proc. Inst. C. E.* 1890 and 1891, also *Proc. Inst. Mech. Eng.* 1895.

the same engine running at a low speed, all the other conditions of working being the same in both cases.

Again, as regards range of temperature, the influence of the cylinder walls will be greater (other things being equal) with high than with low pressure steam, and in condensing than in non-condensing engines.

In large engines the action of the walls will be less than in small engines, since the proportion of wall surface to cylinder volume is less. This conclusion agrees with the well-known fact that small engines do not readily achieve the economy that is reached in many larger forms.

Cylinder condensation is increased when the ratio of expansion is increased, all the other circumstances of working being left unaltered. The quantity of water formed by adiabatic condensation is then greater, and it is on this that the whole action mainly depends. Further, the metal is then brought into rather more prolonged contact with low-temperature steam. The volume of admission is reduced to a greater extent than the surface that is exposed to the entering steam, since that surface includes two constant quantities, the surface of the cylinder-cover and of the piston. For these and perhaps other reasons we may expect that with an early cut-off the initial condensation will be relatively large, and this conclusion is amply borne out by experiment. An important result is that increase of expansion does not, beyond a certain limit, involve increase of thermodynamic efficiency; when that limit is passed the augmentation of waste through the action of the cylinder walls more than balances the increased economy to which, on general principles, expansion should give rise, and the result is a net loss. For this reason (as well as for mechanical reasons) it is not wise in practice to carry expansion too far—not, in general, nearly so far as to be complete. With a given engine, boiler pressure, and speed, a certain ratio of expansion will give maximum efficiency. But the conditions on which this maximum depends are too complex to admit of theoretical solution; the best ratio is a matter rather for experiment. It may even happen that an engine which is required to work at a specified power will give better results, in point of efficiency, with moderate steam-pressure and moderate expansion, than with higher steam-pressure and a very early cut-off.

122. Results of Experiments with various ratios of Expansion. The effect of increased expansion in augmenting the action of the sides and so reducing the efficiency, when carried beyond a certain moderate grade, was clearly shown by the American and Alsatian experiments alluded to above. The following figures, relating to a single-cylinder Corliss engine, are reduced from one of Hallauer's papers[1]:—

Single-cylinder Corliss Engine: Effect of Varying the Expansion.

Ratio of Expansion	Percentage of Water present		Consumption of Steam per Hour per Indicated Horse Power lbs.
	At End of Admission	At End of Expansion	
7·3	24·2	17·8	17·8
9·4	30·8	18·6	17·6
15·1	37·5	20·8	17·7

Here, in consequence of the amount of initial condensation increasing with increased expansion, a maximum of efficiency lies between the extreme grades of expansion to which the test extends, but the efficiency varies exceedingly little even through

U. S. Revenue steamer "Bache": Effect of Varying the Expansion.

Ratio of Total Expansion	Consumption of Steam per Hour per I.H.P. lbs.
4·2	21·2
5·7	20·0
7·0	20·3
9·2	20·7
16·8	25·1

this wide range. In the American experiments the best results were obtained with even more moderate ratios of expansion. The compound engines of the United States revenue steamer "Bache,"

[1] *Bull. Soc. Industr. de Mulhouse*, May 26, 1880.

when tested with steam in the jacket of the large cylinder, with the boiler pressure nearly uniform at 80 lb. by gauge, or 95 lb. per square inch absolute, and the speed not greatly varied, gave results which are shown in the Table.   Here the efficiency is very little affected by a large variation in the position of the cut-off, but when the ratio of expansion becomes excessive a distinct loss is incurred.

Again—to take an instance relating to a very different type of machine—trials made by Willans with one of his high-speed compound non-condensing single-acting engines, using steam with an absolute initial pressure of 130 lbs., gave these results.

Willans' Engine (Non-condensing): Effect of Varying the Expansion, the initial pressure and speed being constant [1].

Ratio of Total Expansion	Percentage of Water present at end of admission in high-pressure cylinder	Consumption of Steam per Hour per I.H.P. lbs.
4	8·9	20·7
4·4	10·2	20·5
4·8	11·7	20·35
5·2	14·2	20·26
5·6	14·3	20·0
6	18·4	20·3
8	25·0	23·1

The initial condensation is comparatively small here, mainly in consequence of the exceptional speed (404 revolutions per minute), and for the same reason the economy in steam consumption is remarkably high for a small non-condensing engine.   In another series of trials in which a compound engine of this type was worked with a condenser [2], and with steam at about 170 lbs. (absolute), Willans found a slight increase in the steam consumption from 14·26 to 14·72 lbs. per hour per I.H.P. when the ratio of expansion was increased from $15\frac{1}{2}$ to 20; at the same time the percentage of water present at cut-off in the high-pressure cylinder increased from 31 to 37.   All these results agree in showing that the ratio

[1] Willans on Non-Condensing Steam-Engine Trials, *Min. Proc. Inst. C. E.* March, 1888.

[2] Willans on Steam-Engine Trials, *Min. Proc. Inst. C. E.* April, 1893.

of expansion may be varied through a large range with but little influence on the efficiency, because the gain that comes of making the expansion more complete is counterbalanced by the bad effects of increased initial condensation.   The ratio of expansion which gives a maximum of efficiency is never sharply defined, and its value depends much on the initial steam-pressure and the particular features of the engine under trial.

123.   Advantage of high speed.   The advantage of high speed in making the action of an engine more nearly adiabatic has been demonstrated by experiment.   Among the trials described by Willans in his earlier paper are the following two sets made with one of his compound non-condensing engines, in the first set with an absolute admission pressure of 90 lbs. per square inch and 3·2 as the ratio of expansion; in the second set with 130 lbs. pressure and 4·8 as the ratio.   In the three trials of each set the only condition varied was the speed.

Willans' Non-condensing Engine Trials: Influence of Speed.

	I. Trials with Steam of 90 lbs. pressure			II. Trials with Steam of 130 lbs. pressure		
Speed: revolutions per minute	401	211	122	405	216	131
Percentage of Water present at cut-off in the high-pressure cylinder	5·0	12·6	20·2	11·7	19·1	29·7
Consumption of Steam per Hour per I.H.P. (lbs.)	24·2	25·3	27·0	20·3	21·3	23·7

The increase of steam consumption as the speed is reduced is considerable, and still more marked is the greater initial condensation.   The same features are apparent in the trials quoted below from an extensive series in Willans' second paper; they relate to a condensing engine with an absolute admission pressure of 90 lbs. and a very moderate ratio of expansion (4·8).

Willans' Condensing Engine Trials: Influence of Speed.

Speed : revolutions per minute	401	301	198	116
Percentage of Water present at cut-off in the high-pressure cylinder	8·9	12·2	17·9	20·9
Consumption of Steam per Hour per I.H.P. (lbs.)	17·3	17·6	18·9	20·0

124. Experiments on the value of the Steam-jacket. Abundant evidence of the advantage of the steam-jacket is given in Reports by the committee appointed by the Institution of Mechanical Engineers to inquire into the subject. Individual figures vary widely, but it appears that the saving usually secured by jackets in condensing engines is something like 12 or 15 per cent. In non-condensing engines it is less. The following results of special trials with condensing engines are stated in the Report. In several of these cases, notably in the last, it is remarkable how

Influence of Steam-jacket.

Engine	Total Steam per Hour per I.H.P. lbs.		Percentage less with Jackets	Proportion of Jacket feed to total consumption per cent.
	Without Jackets	With Jackets		
Two-cylinder compound[1]	18·2	16·6	9	7
Two-cylinder compound[1]	24·7	20·0	19	6
Triple compound[1]	17·2	15·4	10	11
Triple compound[2]	16·4	13·6	17	
Two-cylinder compound[3]	21·1	19·5	7	12
Same engine run non-compound, the large cylinder only being used	32·1	26·7	17	7
Small single-cylinder[4] engine	39	29	25	7

[1] *Proceedings Inst. Mech. Eng.* 1889, 1892, and 1895.
[2] Prof. O. Reynolds' tests. For particulars see *Min. Proc. Inst. C.E.* vol. xcix. 1889.
[3] Prof. Unwin's tests: *Proc. Inst. Mech. Eng.* 1892, p. 460.
[4] Mr B. Donkin's tests: *Proc. Inst. Mech. Eng.* 1892, p. 464.

large a net saving of steam is secured by a comparatively small consumption in the jackets. In other trials of the same small engine, using an earlier cut-off, Mr Donkin found that 8 or 9 per cent. used in the jackets was capable of saving as much as 40 per cent. of the whole steam. In this instance there was excessive initial condensation when the jackets were out of use.

In compound engines the jackets are most effective when both or all of them are filled with steam at the boiler pressure. In Prof. Reynolds' triple engine trials, it was found that steam of the full boiler pressure (200 lbs. per sq. inch) in all the jackets reduced the initial condensation in the second cylinder to about one-third or one-fourth of the amount that occurred without jackets, made the steam practically dry before the end of expansion in the second cylinder, and almost entirely prevented condensation in the third cylinder. Without steam in the jackets the second and third cylinder had been very wet, the proportion of water in them being about 40 per cent. of the whole. Indicator diagrams relating to these trials will be found in the chapter on Compound Expansion.

125.   Superheating.   Superheating the steam before its admission reduces the amount of initial condensation. Exohange of heat between the gaseous fluid and the metal is difficult, and each pound that is condensed has more heat to give up. This in turn lessens the subsequent cooling by re-evaporation. That superheating has a marked advantage in this respect was first experimentally demonstrated by Hirn, who found that the consumption of steam was reduced from 19·4 to 16·2 lbs. per horse-power-hour in a condensing engine by superheating the steam some 45° C. On general thermodynamic grounds superheating has a very slight direct theoretical advantage because it allows a small part of the whole heat supply to be taken in at a higher temperature than that of the boiler. But the indirect advantage is much more considerable. About the year 1860 superheating was frequently used in marine practice, but it was abandoned, mainly on account of difficulties in regard to lubrication. The importance of taking means to avoid or rather to reduce initial condensation was less generally understood in those days than it is now, and with the lubricants that are now used the old objections to superheating have little force.

So far as land engines are concerned, a revival in the use of

superheated steam has taken place.   Many engines are furnished with superheaters and in some a large amount of superheating is successfully applied.   Superheating is easily carried out in cases where there is a steady consumption of steam : it is less applicable where there is much variation.   Experiments made in 1892 by the Alsatian Association of Steam Users on a large number of engines furnished with superheaters showed that super-heating effected a saving of coal to the extent of about 20 per cent. in cases where the superheater was simply placed in the boiler flue, so that it enabled what would otherwise be waste heat to be utilized, and about 12 per cent., on the average, in cases where the superheater was separately fired.   Several of the engines tested were fairly large, indicating 500 or 600 horse-power, and the superheating, which usually amounted to about 40° C., was carried on without inconvenience.   One of the trials, dealing with a triple-expansion Sulzer engine of 300 horse-power, records a consumption of 14·6 lbs. of steam per I.H.P.-hour without superheating, and 11·6 lbs. when the steam was superheated 55° C.

The following experiments made by Willans with and without superheating in a high-speed single-cylinder condensing engine cutting off steam at half stroke show that some advantage results even under conditions which are not such that much advantage can be expected.

Willans' Engine with and without Superheating.

Initial pressure, lbs. per sq. in. absolute	45		35		25	
Temp. of Steam at admission °C.	134°	152°	126°	148°	115°	143°
Amount of Superheating	none	18°	none	22°	none	28°
Percentage of Water at cut-off	21	17	24	19	25	15
Consumption of Steam per Hour per I.H.P.	26·7	24·6	28·9	26·4	30·0	26·4

Important trials of engines using superheated steam have been carried out in Germany by Prof. Schröter and Prof. Gutermuth.

In one of Prof. Schröter's trials[1] a triple expansion factory engine indicating 1000 H.P. and supplied with steam at 100 lbs. per sq. inch was tested with saturated steam and with steam superheated by 52°C. In the former case it used 13·2 lbs. per I.H.P.-hour, in the latter case only 12·0 lbs.[2] Allowing for the additional heat required the results show a thermodynamic advantage of $5\frac{1}{2}$ per cent. in favour of superheating.

Prof. Schröter's analysis of this test shows that the amount of the superheating was insufficient to prevent the steam from becoming somewhat wet during admission.   At the point of cut-off in the first cylinder the steam was nearly, but not quite dry, and as expansion went on in this cylinder its wetness increased.   The advantage, such as it is, of moderate superheating lies in reducing the losses which proceed from exchanges of heat between the steam and the cylinder walls.   Although the steam retains its superheat until it reaches the engine, it at once falls to the temperature of saturation when it meets the cylinder walls unless the amount of initial superheat is large.   To keep it dry during admission would require an amount of superheating probably never less than 60° C. and often much more, the amount that is necessary depending on the ratio of expansion in the cylinder and on the rate at which heat is lost from the external surfaces.   To keep the steam dry during expansion a much higher degree of superheat would be needed.   Superheating, in any moderate degree, may be regarded as a device for reducing the action of the cylinder walls by bringing the expansion curve of the indicator diagram nearer to the saturation curve (fig. 52) than it would otherwise come.   In ordinary cases it barely makes the expansion curve come out so far as to reach the saturation curve even at cut-off, and as expansion proceeds the interval between the two increases.   The ideal diagram sketched in fig. 30 is widely departed from in real superheated steam-engines, for the action of the cylinder walls in general keeps the whole process of expansion to the left of the saturation line cf.   The working substance, after being taken up the line cr before it reaches the engine, immediately comes down the line cr when it is admitted

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[1] *Zeitschrift des Vereins deutscher Ingenieure*, vol. XL. 1896.

[2] In reducing to British measure the results of tests stated in the metric system the number of kilogrammes per metric H.P.-hour has to be multiplied by 2·235 to bring it to lbs. per British H.P.-hour, the metric H.P. being 0·9863 of a British H.P.

to the cylinder and before it begins to expand, giving up to the walls the heat (represented by the area under *cr*) which it has received in being superheated, and often going on to give up a part of its latent heat by becoming slightly wet. In cases where the superheat is high the steam comes only part of the way back from *r* towards *c* before it begins to expand.

126. Use of highly superheated steam. More striking advantages are found when steam is highly superheated. In the Schmidt engine, which is specially designed for use with high-temperature steam, the steam is superheated to about 400° C. by passing it through a coil of tubes in the uptake of the boiler furnace, and the consumption of heat is much lower than in any ordinary steam-engine of similar power.

In many of Schmidt's engines there is a re-heater in the intermediate receiver between the high- and low-pressure cylinders, consisting of a number of pipes through which part of the supply of highly superheated steam is made to pass before it reaches the high-pressure valve-chest. The effect is to transfer a portion of the heat of the highly superheated steam to the steam in the receiver, which has already performed the first stage of its expansion, and consequently to reduce the degree of superheat on admission to the high-pressure cylinder.

In a two-cylinder compound condensing Schmidt engine tested by the present writer the boiler pressure was 140 lbs. per sq. inch by gauge and the steam was initially superheated to 396° C. Before entering the first cylinder, however, the steam lost 104° of its superheat, by being used in the reheating device, so that its temperature in the high-pressure valve-chest was 292°C. The engine developed 184 indicated horse-power. Under these conditions 10·4 lbs. of steam were used per hour, per I.H.P. The same engine using saturated steam of the same pressure consumed 17·2 lbs. per hour per I.H.P. In the superheated trial the steam remained superheated throughout the whole course of expansion in the high-pressure cylinder, but was somewhat wet from the point of cut-off onwards in the low-pressure cylinder. In the trial with saturated steam the missing quantity at cut-off in the high-pressure cylinder was fully 30 per cent.

In another of the writer's tests, a two-cylinder compound Schmidt engine developing 300 horse-power took steam of

142 lbs. pressure by gauge, which was initially superheated to 425° C. but reduced to 317° C. after passing the reheater. The consumption was 9·0 lbs. of steam per hour per I.H.P. and 14·9 lbs. per electrical unit generated by the dynamo which the engine was employed to drive. Taking the mean specific heat in the superheating process to be 0·52 (see § 65) the extra heat required for superheating was 126 units. As the feed-water was supplied at 24° C. the heat taken in by the steam was consequently $643 + 126$ or 769 units, compared with 643 for saturated steam. Thus the actual consumption of 9·0 lbs. per horse-power-hour is equivalent in point of heat to $\dfrac{9·0 \times 769}{643}$ or 10·8 lbs. of saturated steam per horse-power-hour at the same pressure and with the same temperature of feed, a figure which is much better than the best results that have been obtained in trials of engines of comparable size using saturated steam. In trials of Schmidt engines by Prof. Lewicki and others like results have been observed, the consumption of steam being in some instances rather less than 9 lbs. per horse-power-hour.

Taking the above result it is interesting to compare the actual performance with the ideal performance of an engine in which there is complete adiabatic expansion and no losses. That, as we have seen in § 73, is expressed by $I_1 - I_2$. In the example just cited I_1 is 793 and I_2 is 558, when the ideal expansion is reckoned as being carried down to the pressure in the condenser, which was 1 lb. per square inch. These figures for I_1 and I_2 are most readily found by measurement from the Mollier diagram of Entropy and Total Heat. Hence in the ideal process each unit quantity of steam would do work equivalent to 235 thermal units, reckoning from the boiler pressure and the full initial condition of superheat. Since one horse-power-hour is equivalent to 1414 thermal units (lb.-deg. Cent.) the indicated work actually done per lb. of steam was $\dfrac{1414}{9·0}$ or 157 units. The observed performance therefore represents 67 per cent. of the ideal.

A slightly more favourable comparative estimate is arrived at if we reckon the ideal performance with reference to the condition of the steam as it reached the engine. Its pressure had then fallen to 154 lbs. per sq. inch and its temperature to 405°, through losses in the steam-pipe. This was before it entered the reheating

device.   Reckoning I_1 with reference to this state it is 786 and I_2 is 555, by the Mollier diagram : the ideal performance is therefore 231.   The actual performance was accordingly 68 per cent. of this ideal, which forms a fairer criterion than the other in judging of the losses for which the engine itself is responsible, the difference between the two being due to losses in a rather long steam-pipe.

127. Advantage of Compound Expansion.   The most important means in general use of preventing cylinder condensation from becoming excessive is the use of compound expansion. If the vessels were perfect non-conductors of heat it would be, from the thermodynamic point of view, a matter of indifference whether expansion was completed in a single vessel or divided between two or more, provided the passage of steam from one to the other was performed without introducing unresisted expansion. In practice, the transfer of steam from one cylinder to another during its expansion cannot be accomplished without some more or less of wasteful drop in pressure.   But the loss which this entails is more than counterbalanced by the gain that results from the reduced influence of the cylinder walls.   Compound working acts beneficially by diminishing the range through which the temperature of any part of the cylinder-metal varies.   For this reason the amount of steam initially condensed in the high-pressure cylinder of a compound engine is less than if admission were to take place at once into the low-pressure cylinder and the whole expansion were to be performed there.   Further, the steam which is re-evaporated from the first cylinder during its exhaust does work in the second, and it is only the re-evaporation that occurs during the exhaust from the last cylinder that is absolutely wasteful.   The same remark applies to the effects of valve leakage. When a compound engine is tested first with compound expansion and then with the same grade of expansion in the large cylinder alone it is found that more steam is required per horse-power-hour in the second case.   Professor Unwin's tests referred to in the Table of § 124 furnish an instance : an engine taking 21 lbs. of steam per horse-power-hour when working compound required 32 lbs. when the large cylinder only was used, no jackets being then in action.   With steam in the jackets the difference was rather less, for the jacket checked that excessive cylinder condensation which reduced the efficiency in the non-compound trials.

The general subject of compound expansion will be considered more particularly in a later chapter: at present we are concerned with the influence of compounding on efficiency. Experience shows that it is only by resorting to compound expansion that the economical advantages of high-pressure steam are to be secured. When high-pressure steam is used in a non-compound engine the waste due to initial condensation is excessive because of the great range of temperature through which the metallic surfaces fluctuate in every stroke. The advantage of compounding becomes greater the higher is the boiler pressure. So long as the initial pressure does not much exceed 100 lbs. per square inch it suffices to reduce the range of temperature into two parts by employing two-cylinder compound engines; with the considerably higher pressures now common triple expansion is economical, and even quadruple expansion has advantages when pressures of 200 lbs. or more are employed.

128. Summary of Sources of Loss. The principal reasons have now been named which make the actual results of engine performance differ from the results which would be obtained if the steam conformed in every respect to the ideal Rankine Cycle. The sources of loss may be summarised as follows:—

(1) Wire-drawing in admission and exhaust.

(2) Incomplete expansion before release.

(3) Incomplete compression of the cushion steam, through which the clearance becomes a cause of waste.

(4) The action of the metallic surfaces of the cylinder and piston, causing condensation during compression and admission, with re-evaporation during expansion and exhaust.

(5) Radiation and aerial convection of heat from steam-pipe, valve-chest and all hot parts of the engine.

(6) Escape of the working fluid by leakage to the atmosphere, and leakage of air into the condenser.

(7) Leakage of steam to the exhaust, so that it escapes acting on the piston; and leakage past the piston.

(8) In compound engines, additional wire-drawing or un-resisted expansion in the transfer of steam from one cylinder to another.

If in drawing a comparison between the real engine and the ideal we take as the lower limit of temperature that of the

discharge from the condenser, we have a further item, namely, the loss that comes from the pressure in the condenser being higher than the pressure corresponding to this lower limit.

Some representative examples will be given later of results obtained in trials of steam-engines of the most efficient kinds. When we come to examine these we shall find that in the most favourable cases, namely in engines working with the very highest recorded efficiency, some 20 per cent. of the heat supplied to the steam is converted into work. In other words, the thermodynamic efficiency of the actual cycle is found to be, at the best, about 0·20. In such cases it will be found that the work actually got from the steam is, in round numbers, 70 per cent. of the work which would be theoretically obtainable if the ideal conditions of the Rankine Cycle were fulfilled.   The real performance falls short of the ideal by some 30 per cent. through the joint effect of the sources of loss here enumerated.   Of these various sources of loss the second and the fourth are in general the most important, in engines of the piston and cylinder type.   In steam turbines the chief causes of loss are different, but it is remarkable that the aggregate amount of loss is much the same. In them also about 70 per cent. of the work theoretically obtainable marks, at present, the highest level of actual performance.

In ordinary engine practice the actual performance of condensing engines is more like 60 per cent. of the Rankine ideal.

CHAPTER VIII.

129. Theory of the Steam Jet. The essential feature which distinguishes steam turbines from steam-engines of other types is that in the action of the turbine the heat energy present in the steam is first employed to set the steam itself in motion, giving it kinetic energy, and this in turn is made to do work on the turbine blades. There is thus a double transformation of energy. In treating of the theory we have first to consider the manner in which a steam jet is formed by the discharge of steam under pressure. To simplify matters we shall assume that no heat is taken in from or given out to external bodies during the operation.

Consider the flow of steam, or any other gas, through a nozzle or channel of any form, from a region where the pressure is relatively high to one where it is lower. The steam acquires velocity in the process and also increases in volume. Let A and B (fig. 54) be imaginary partitions, across which it flows, taken at
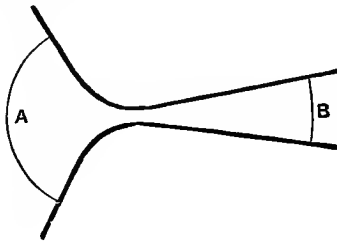
Fig. 54.

right angles to the direction of the stream lines, A being in the region of higher pressure. Let P_a be the pressure at A, v_a the velocity there, and V_a the volume which unit quantity of the gas has as it passes the imaginary partition at A. Similarly let P_b,

v_b and V_b be the pressure, velocity, and volume of unit quantity at B. Let E_a and E_b be the internal energy of the gas at A and B respectively. In flowing from A to B the velocity changes from v_a to v_b and there is consequently a gain of kinetic energy amounting, per unit of mass, to $\dfrac{v_b^2 - v_a^2}{2g}$.

Each unit quantity of gas that enters the space between A and B has work done upon it by the gas behind amounting to $P_a V_a$. In passing out of this space at B it does work on the gas in front amounting to $P_b V_b$. In flowing from A to B it loses internal energy amounting to $E_a - E_b$. Hence by the principle of the conservation of energy, since by assumption no heat is taken in or given out,

$$\frac{v_b^2 - v_a^2}{2g} = E_a - E_b + P_a V_a - P_b V_b \quad \ldots\ldots\ldots\ldots(1).$$

But $E_a + P_a V_a$ is I_a, the total heat at A and $E_b + P_b V_b$ is I_b, the total heat at B, and the equation may consequently be written

$$\frac{v_b^2 - v_a^2}{2g} = I_a - I_b \quad \ldots\ldots\ldots\ldots\ldots\ldots(2).$$

The gain in kinetic energy is therefore equal to the loss of total heat, or what is commonly called the " heat drop." We are treating E and I as if they were expressed in work units: when expressed in heat units they have to be multiplied by the mechanical equivalent J.

The equation applies as between any two places in the flow, and taking the process as a whole, from the initial condition in which the velocity is v_1 and total heat I_1 to the final condition in which the velocity is v_2 and total heat I_2 we have

$$\frac{v_2^2 - v_1^2}{2g} = I_1 - I_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots(3).$$

In the particular case where the initial velocity is zero or negligibly small

$$\frac{v^2}{2g} = I_1 - I_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

where v is the velocity acquired in consequence of the heat drop.

So far there has been no assumption as to absence of losses through friction or eddy currents. If we now consider the case of steam forming a jet by expanding under adiabatic conditions, with

no communication of heat and with no losses through friction or eddies, the heat drop in the equation

$$\frac{v_2{}^2 - v_1{}^2}{2g} = I_1 - I_2$$

is that which occurs in expansion with constant entropy, and the value of $I_1 - I_2$ is readily found from the Mollier diagram. The initial condition as to pressure and superheat (if any) determines I_1 and the final pressure then allows I_2 to be found by drawing a vertical line ($\phi = \text{const.}$) through the point representing the initial condition, until the final pressure is reached.

We have already seen (§ 73) that this heat drop is equal to the
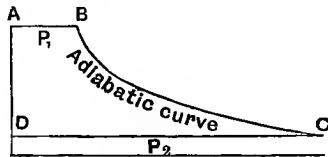
Fig. 55.

area $ABCD$ of the ideal indicator diagram (fig. 55) for complete adiabatic expansion from the initial to the final state, or $\int_{P_2}^{P_1} V dP$.

Hence $\qquad \dfrac{v_2{}^2 - v_1{}^2}{2g} = \text{area } ABCD = \int_{P_2}^{P_1} V dP \ldots\ldots\ldots\ldots(5).$

This result might also be obtained from the consideration that under the assumed conditions the steam or other gas is expending all the work of which it is ideally capable, as it expands from the first to the second state, in giving itself kinetic energy, and the gain of kinetic energy is therefore equal to the area of the ideal diagram.

Expressing the expansion in the ideal diagram by a formula of the type $PV^n = \text{const.}$, we have for the area of the diagram

$$\int_{P_2}^{P_1} V dP = \frac{n}{n-1}(P_1 V_1 - P_2 V_2)$$

$$= \frac{n}{n-1}(1 - D^{\frac{n-1}{n}}) P_1 V_1 \ \ldots\ldots\ldots\ldots(6),$$

where D is the ratio in which the pressure falls, namely $\dfrac{P_2}{P_1}$.

In the case where the substance is steam, initially dry and

saturated n may be taken as 1·135 (§ 68) and the area of the diagram then becomes

$$8\cdot41\,(1 - D^{0\cdot119})\,P_1V_1.$$

This, when reduced to heat units, will be found to agree very closely with the values of the heat drop $I_1 - I_2$ measured on the Mollier diagram, or calculated by first finding the wetness after expansion as a preliminary step towards determining I_2.

130. Form of the Jet. As expansion proceeds the volume and velocity of the steam both increase. It is easy in frictionless adiabatic flow to calculate both, and in that way to determine the proper form to give to the nozzle or channel, to make provision for the increased volume, having regard to the increased velocity. At any stage the area of cross-section of the channel required for the discharge is equal to the volume divided by the velocity. It is convenient to reckon this per lb. or kilogramme of steam in the discharge, and afterwards multiply by the number of lbs. or kilogrammes.

On making the calculation for steam starting from rest and discharged into a region of much lower pressure it will be found that in the earliest stages the gain in velocity is relatively great, but as expansion proceeds the increase in volume outstrips the increase in velocity. The result is that the ratio of volume to velocity at first diminishes, passes a minimum value, and then increases; and hence the channel to be provided for the discharge, after passing a minimum, expands in the later stages. The proper form for the nozzle to allow the heat drop to be duly utilized in giving kinetic energy to the steam is therefore one in which the area of section at first contracts to a narrow neck or "throat" and afterwards becomes enlarged to an extent determined by the final pressure.

Starting from rest, the velocity when the pressure has fallen to any lower value is found by making $\dfrac{v^2}{2g}$ equal to the heat drop down to that pressure, and the volume is got by finding the dryness q for adiabatic expansion down to the same pressure.

Taking a numerical example, let steam initially dry and saturated be discharged from a vessel where the pressure is 15 kg. per sq. cm. absolute. From the Mollier diagram the dryness q and the velocity due to the heat drop are found at successive points

14—2

in the expansion as follows, for pressures of 12, 10, 9, 8 etc., kg. per sq. cm. Metric units are used throughout, and in these units the formula

$$\frac{v^2}{2g} = J(I_1 - I_2)$$

becomes $\qquad v^2 = 8378\,(I_1 - I_2),$

or $\qquad v = 91.53\,\sqrt{I_1 - I_2},$

v being expressed in metres per second.

P kg./cm.^{2}	Dryness q	Vol. in cub. metres per kg.		Velocity in metres per second	Area of jet per kg. in sq. metres $= \dfrac{\text{volume}}{\text{velocity}}$
		For saturated steam	Actual		
15	1·000	0·1360	0·1360	0	
12	0·983	0·1678	0·1649	295	0·000559
10	0·970	0·1993	0·1933	400	0·000483
9	0·963	0·2200	0·2119	444	0·000477
8	0·955	0·2458	0·2347	492	0·000477
7	0·946	0·2786	0·2636	541	0·000487
6	0·937	0·3220	0·3017	590	0·000512
5	0·927	0·3820	0·3538	644	0·000549
1·033	0·856	1·6702	1·430	957	0·001494
0·12	0·784	12·568	9·85	1214	0·00812

The table is continued down for pressures of 1·033 kg. per sq. cm. (atmospheric pressure) and 0·12 which represents an ordinary condenser vacuum.

It will be seen from these figures that the minimum area of section of the jet is reached at a pressure between 8 and 9 kg. per sq. cm. By taking intermediate values it may be shown that the actual minimum is 0·000476 and that it occurs when the pressure is about 8·6 kg. This therefore is the size of the " throat " per kilogramme of steam discharged: it is determined simply by reference to the initial pressure. From that point onwards the jet enlarges, and if the steam is to escape at atmospheric pressure the channel should continue to enlarge until its area of section is 0·001494 or 3·14 times that of the throat, but if the steam is to escape at a condenser pressure of 0·12 kg. (1·7 lbs. per sq. inch) the enlargement of the channel should continue till its area is 0·00812 or say seventeen times that of the throat. These calculations relate to frictionless adiabatic flow.

131. De Laval's Divergent Nozzle. It is on this principle that De Laval's divergent nozzle (fig. 56) is designed. The throat or smallest section is approached through a more or less rounded entrance which allows the stream lines to converge, and from the throat outwards to the discharge end the nozzle expands in any gradual manner, generally in fact as a simple cone, until an area of section is reached which will correspond to the proper area of discharge for the final pressure, reckoned in the manner here exemplified.
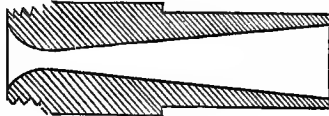
FIG. 56.

The taper from the throat onwards is made sufficiently gradual to prevent, as far as possible, the formation of eddies which would dissipate the kinetic energy of the stream. A very short rounded entrance to the throat is sufficient to guard against eddies in the convergent portion of the stream, but where the stream is diverging a much more gradual change of section is required. The nozzle shown in the figure is designed for an initial pressure of 250 lbs. per sq. inch and a back pressure of about $1\frac{1}{2}$ lbs. per sq. inch. By the back pressure is meant the pressure in the space into which the fluid is discharged.

The equation for the velocity

$$\frac{v^2}{2g} = \frac{n}{n-1}(1 - D^{\frac{n-1}{n}})P_1 V_1 \quad \dots\dots\dots\dots(6a)$$

may be applied to calculate generally the discharge per unit of area of stream section, and hence to find at what pressure in the stream the discharge per unit of area becomes a maximum, in other words to determine the pressure at the throat.

Since at any stage in the expansion $PV^n = P_2 V_2^n$, the volume at any stage $V = \dfrac{V_1}{D^{\frac{1}{n}}}$, where $D = \dfrac{P}{P_1}$.

The mass of the discharge per unit of area when the volume is V and the velocity v is

$$Q = \frac{v}{V} = \frac{vD^{\frac{1}{n}}}{V_1} = \sqrt{\frac{2gn}{n-1} \cdot \frac{P_1}{V_1}(D^{\frac{2}{n}} - D^{\frac{n+1}{n}})} \dots\dots\dots(7).$$

This will be a maximum when $\dfrac{dQ}{dD}$ is zero, which occurs when

$$D = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}.$$

This result is general for any gas, n being the index for adiabatic expansion. With saturated steam, initially dry, n is 1·135 and Q is therefore a maximum when $D = 0·577$. That is to say the pressure in the throat is 0·577 of the initial pressure, a result which agrees with the figures in the example calculated above.

From these considerations it follows that provided the final pressure is less than 0·577 times the initial pressure the amount of the discharge depends simply on the least area of section of the nozzle or orifice and on the initial pressure, and is independent of the final pressure. By continuing the expansion in a divergent nozzle after the throat is passed, the amount of the discharge is not increased, but the steam is caused to acquire a greater velocity of exit when it leaves the nozzle, because the range of pressure through which the expansion is effective for producing velocity is increased. To put it in another way we may say that the heat drop down to the pressure at the throat determines the amount of the discharge, and the remainder of the heat drop, which would be wasted if there were no divergent nozzle, is caused by the divergent nozzle to give additional velocity to the escaping stream.

When the pressure drop under which the discharge takes place is small, so that P_2 is more than $0·577P_1$, the full velocity due to the drop is obtained without the use of a divergent nozzle. In a Parsons turbine, for example, the whole range through which the pressure falls is divided into many separate stages, and in each stage the drop is so small as to come well within this limit, so that the channels in which the steam acquires velocity at each stage have no divergent part.

132. Maximum Discharge from an orifice or nozzle. The velocity at any point in the jet may (from Eq. 6a) be written

$$v = \sqrt{\frac{2gn}{n-1}\left(1 - D^{\frac{n-1}{n}}\right)P_1 V_1} \quad \ldots\ldots\ldots\ldots(6b).$$

Hence at the throat, where $D = \left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$, the velocity becomes

$$v_t = \sqrt{\dfrac{2gn}{n+1} P_1 V_1} \quad\dots\dots\dots\dots\dots\dots(8).$$

To find the discharge per unit of area at the throat we divide the velocity there by the volume, which is

$$\dfrac{V_1}{D^{\frac{1}{n}}} \text{ or } \dfrac{V_1}{\left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}},$$

and obtain

$$Q_t = \left(\dfrac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{\dfrac{2gn}{n+1} \cdot \dfrac{P_1}{V_1}} \quad\dots\dots\dots\dots(9).$$

In the case of saturated steam, dry to begin with, these formulas give

$$v_t = 3 \cdot 23 \sqrt{P_1 V_1} \quad\dots\dots\dots\dots\dots\dots(8a)$$

$$Q_t = 1 \cdot 99 \sqrt{\dfrac{P_1}{V_1}} \quad\dots\dots\dots\dots\dots(9a)$$

where V_1 is the initial volume in cubic metres per kilogramme, P_1 the initial pressure in kilogrammes per square metre, v_t the velocity at the throat in metres per second, and Q_t the discharge, in kilogrammes, per square metre of sectional area at the throat. If the pressure is expressed in the more usual unit, kilogrammes per square centimetre, the numerical constants become 323 and 199.

The greatest discharge that can be obtained with an orifice or channel of any form is found by multiplying Q_t by the least area of the orifice. In any channel the least section may for this purpose be regarded as a throat: if there is a divergent portion beyond it the amount of the discharge is not affected though the final velocity of the steam is affected.

It is interesting to notice that the greatest discharge depends on the initial pressure and not upon the back pressure provided that the drop in pressure is sufficient to make the ratio of pressures D not less than $\left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$. Taking an orifice or nozzle of any form, and a constant initial pressure P_1, if we reduce the back pressure from values which to begin with are only a little below P_1 we find that the discharge increases with reduced back

pressure until D reaches the above value, but after that any further reduction of the back pressure does nothing to augment the discharge. With a simple orifice, or a channel which has no divergent part, discharging into a space at pressure P_2, the velocity of discharge is that due to the full drop of pressure so long as P_2 is not less than $\left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$ times P_1, and the amount of the discharge is, in that case, readily calculated as the product of that velocity into the area of the orifice and the density of the fluid when expanded to pressure P_2. When P_2 is reduced to the above limit equations 8 and 9 apply, and no further increase in the discharge takes place with further reduction in the back pressure.

But the velocity of efflux may then be augmented by adding a divergent or expansion nozzle. The nozzle will be rightly designed when it provides for just enough expansion to make the pressure when the fluid leaves it equal to the back pressure. In that case the jet escapes as a smooth stream, and the energy of expansion is utilized to the full. If the back pressure is too low for the nozzle, or in other words if the nozzle does not carry expansion far enough, vibrations are set up in the fluid and energy is wasted. If on the other hand the back pressure is too high for the nozzle, so that the nozzle provides for more expansion than can properly take place, there is also some vibration resulting from the fact that the last part of the nozzle is too big for the stream, though the loss is much less considerable than in the other case[1].

133. Effects of Friction. In these calculations we have been treating an ideal case. It has been assumed that no energy is wasted through friction and hence the whole of the work represented by adiabatic expansion has been available for giving velocity to the jet. But in any real jet there is some friction between the fluid and the sides of the channel, and eddies are formed which fritter down a portion of the kinetic energy into heat. The consequences are twofold. The jet is discharged with less velocity than it would have possessed if there had been no friction because of the loss of kinetic energy, but its volume after expansion to any given pressure is greater because the lost energy has been restored as heat. At any stage we may think of the jet

[1] For a discussion of the effects of nozzles which are too long or too short to suit the pressure range, see Stodola's book on the Steam Turbine.

as dissipating a part of its kinetic energy through the formation of eddies at the sides, and these subsiding to form heat which goes to raise the temperature of the fluid if it is a dry gas, or to re-evaporate a portion of the liquid in it if it is a wet mixture, as would be the case if the fluid were steam free of initial superheat. In either case the volume is increased, in comparison with the volume that would be found in frictionless adiabatic expansion, but the kinetic energy of the jet as a whole is reduced, in comparison with the same standard.

Taking the pressure-volume diagram the effect of friction is to make the actual expansion curve take a form such as *BC'*, the adiabatic curve being *BC*. But though this apparently implies a
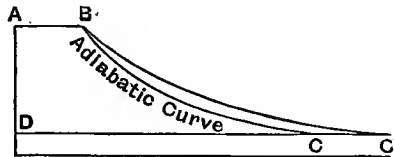
Fig. 57.

gain of work there is really a loss. The pressure-volume diagram does not measure the net effective output of work. Out of the whole work *ABC'D* indicated by the diagram, a part has been reconverted into heat, namely a quantity sufficient to supply enough heat at each stage to bring the expansion curve out from *BC* to *BC'*. At the end of the operation the net amount of work got from the expansion, so far from being greater than the adiabatic area *ABCD*, is less than that area by the equivalent of the heat required to change the condition of the expanded fluid from *C* to *C'*.

This point will be clear if we refer to the entropy-temperature diagram (fig. 58). We shall, as before, assume that no heat comes from outside. Taking first the case of saturated steam, the ideal case with a frictionless nozzle is represented[1] by *ABCD*. In the actual case with friction the expansion curve has a form such as *BC'*, the effect of friction being to bring about a progressive increase of entropy as the expansion proceeds. The area *MBC'M'* represents

[1] It may be well to remind the student that this is subject to a small correction, for the area *ABCD* of the entropy diagram is not exactly equivalent to the area *ABCD* of the indicator diagram (fig. 57), being less by the amount $A(P_1 - P_2)w$ (see § 94).

the heat produced by friction: it is the heat required to give the expansion curve its actual form, and since no heat comes in from outside sources, it is supplied at the expense of the kinetic energy of the jet. This conversion of work into heat is going on progressively from the beginning to the end of the expansion. The gross work done is represented by the area *DABC'*: but from this
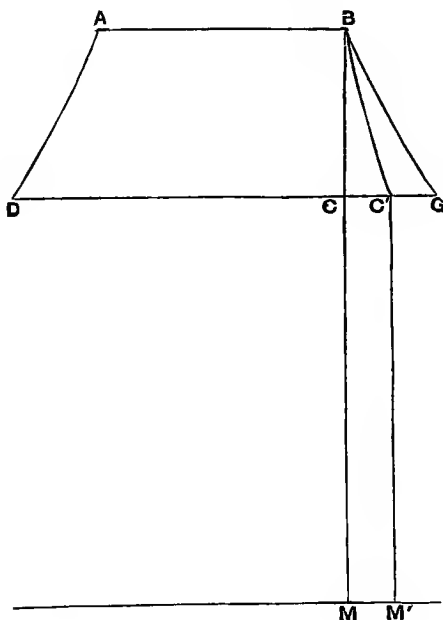
FIG. 58.

we have to deduct the area *MBC'M'* to find the net amount of work which finally appears as kinetic energy in the jet. Thus allowing for friction the net amount of work is

<div align="center">

Area *DABC'* − Area *MBC'M'*

</div>

or   Area *DABC* − Area *MCC'M'*.

Hence the *net* loss, as compared with the work that would be got from frictionless adiabatic expansion is the area *MCC'M'*, which is the heat required to change the condition of the expanded fluid from the ideal final state indicated by *C*, namely the state which would have been reached in frictionless adiabatic expansion, to the actual state indicated by *C'*. The gross amount of work

converted by friction into heat is $MBC'M'$; but of this the area CBC' is recovered through the augmentation of volume that occurs in consequence of friction as the expansion proceeds, and the net loss in comparison with the ideal performance $ABCD$ is only the area $MCC'M'$.

Again, taking the case of superheated steam (fig. 59), let E be the initial condition, ABE being the constant pressure line for the initial pressure P_1 and $DGFF'$ the constant pressure line for P_2.
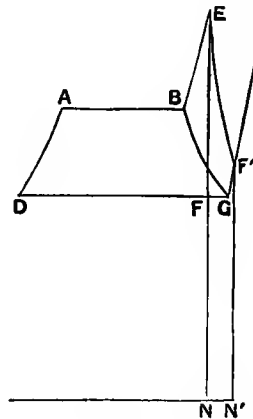
FIG. 59.

Frictionless expansion would be represented by EF: the actual expansion is along some such line as EF'. The gross effect of friction is to develop heat equal to the area $NEF'N'$, and the effective energy is found by deducting that area from the area $DABEF'GD$. Or we may regard the net effect of friction as a reduction of the ideal performance $DABEFD$ by the amount $NFGF'N'$.

Still another useful way of representing the effect of friction is by means of the Mollier diagram of total heat and entropy.

In that diagram (fig. 60) let A be the initial condition and AC the actual expansion process, AB being the ideal frictionless or isentropic process. I_1 is the initial total heat, I_2 the total heat which would be left after ideal expansion to the final pressure P_2 and I_2' is the total heat actually possessed by the steam after expansion to that pressure. The effective heat drop, from which the velocity is to be calculated, is AB' or $I_1 - I_2'$, and the net loss resulting from friction is BB' or $I_2' - I_2$.

When the proportion is known, by experiment, of the frictional loss to the whole theoretically available heat drop, or BB' to AB, we can mark the point B' on the line of adiabatic expansion AB and find C by drawing a horizontal line from B' to the constant pressure curve for the final pressure. Thus when experimental data are available regarding the frictional losses in expansion
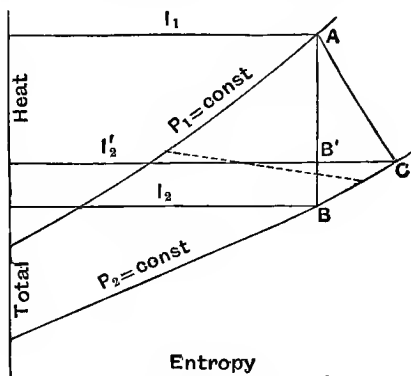
Fɪɢ. 60.

down to various intermediate pressures we can by this construction trace the actual expansion curve AC. The horizontal line $B'C$ on the Mollier diagram represents a *throttling* process ($I = $ const.). The effect of friction is as if we first had adiabatic frictionless expansion to the pressure corresponding to B' and then throttling from that to the lower pressure P_2, which brings the steam into the final condition C[1].

---

[1] It may assist the student to put these results in another form. The equation
$$dQ = dE + PdV$$
is applicable, dQ being the heat generated by friction. Since
$$I = E + PV, \quad dI = dE + PdV + VdP,$$
or
$$dI - V'dP = dQ,$$
using V' for the volume found when there is friction. Hence
$$\int_2^1 dI - \int_2^1 V'dP = \int_2^1 dQ,$$
$$I_1 - I_2' = \int_1^2 V'dP - Q.$$
Also $I_1 - I_2 = \int_2^1 VdP$ for the case of no friction.

Hence the net loss due to friction, which is $I_2' - I_2$, is equal to
$$Q - \int_1^2 (V' - V)\, dP,$$
namely the whole heat generated by friction less the area CBC' of fig. 57 which is recovered as work in consequence of the augmented volume.

134. Application to Steam Turbines. It is only in one type of turbine—De Laval's—that the whole heat drop is turned to account in a single operation. In all other steam turbines there are a number of stages, each involving a separate expansion of the steam, and the heat drop is divided into a corresponding number of parts, each of which is applied in turn to give the steam a new impulse after it has parted with kinetic energy to those turbine blades against which it has already acted.

In some turbines, such as the Curtis, the number of separate stages is small, sometimes only two or three, in others, namely those of the Parsons type the number of stages is very large, and each stage is associated with only a small drop in pressure.

Whether the number of stages be many or few the turbine, in its ideal action free from frictional and other losses, may be regarded as utilizing the whole heat drop in a corresponding number of successive parts or steps, which may be represented on the pressure-volume diagram or on the entropy-temperature diagram by drawing a series of constant pressure lines, at the pressures reached in the successive stages. These lines divide the diagram into strips and the area of each strip represents the work ideally obtainable in the corresponding stage of the action. From this the velocity which the steam may, under the ideal conditions, acquire at each stage is readily calculated, and the diagram also serves to determine the volume which under the ideal conditions is to be handled at each stage. In the ideal case each strip of the diagram is measured out to the line representing adiabatic expansion (the line *BC* in fig. 58 and the line *EF* in fig. 59).

Passing from the ideal to the real turbine we have at each stage losses due to friction and to other causes, such as throttling by leakage over the tips of the turbine blades, which may be regarded as equivalent to friction in their effects. The proportion in which these losses reduce the actual output below the ideal output is a matter of experience, determined by trials with machines of the various types. At each stage we may apply the considerations set forth in the preceding article to investigate the effects of frictional losses: the apparent area of the strip is increased by a shifting out of the real expansion curve beyond the adiabatic line, but the effective area is reduced to a value less than that which it would have in the corresponding step of an ideal frictionless process. Taking the entropy-temperature diagram

(fig. 61), where the whole range is from P_1 to P_2, any intermediate step from a pressure P to $P - \delta P$ would in the ideal engine give a work area *abcd*, but since the actual expansion curve is shifted out to BC' by the effects of friction, the effective
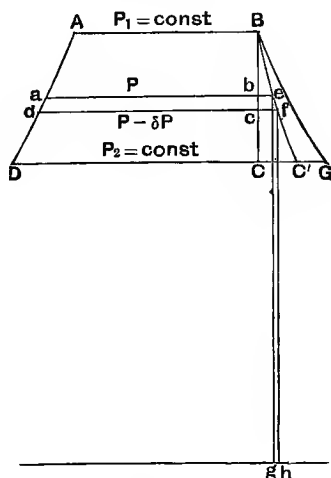
Fig. 61.

work in the step is *aefd minus gefh*, the area *gefh* representing that part of the gross work *aefd* which is re-converted by friction into heat. Thus the effective work done in the step is less than the ideal quantity *abcd*.

The volume at any stage is greater than the volume in the ideal process, corresponding as it does to the actual expansion curve.

135. Importance of Division into Stages. Taking the whole available heat drop, from such pressures as are usual in steam supply down to the vacuum obtainable in a good condenser, we find that $I_1 - I_2$ may easily be as much as 200 with saturated steam or even 250 when the steam is considerably superheated. If the whole of this drop were employed to give velocity to the steam in a single operation the jet would leave the nozzle at a speed of about 1300 to 1450 metres per second, or say 4200 to 4700 feet per second. Making allowance for frictional losses we have still to reckon on velocities of 4000 feet per second or more. If the kinetic energy of the jet is to be extracted by making it impinge on the blades or buckets of a simple turbine wheel, the condition

for efficiency is that the velocity of the blades should be not far short of one-half the velocity of the jet. To get the best return we should therefore require to have a speed of something like 2000 feet per second at the periphery of the wheel which carries the blades. This is an impracticable speed. Apart from the difficulties which arise in gearing down from an excessive speed of rotation to such a speed as would serve for driving the machines in which the power is to be utilized, there are no materials of construction fitted to withstand the forces which are involved in a peripheral speed approaching this magnitude. De Laval's turbine does indeed run at very high speeds, but they are far short of this and involve the sacrifice of a considerable part of the available energy of the jet. There is in addition a loss through friction, which becomes important when the jet rubs against the blade surfaces at a very high speed. Another point to be considered is that steam impinging with great velocity on turbine blades exercises a cutting action, especially when it contains particles of water in suspension, an action which is practically absent at lower speeds : consequently low or moderate steam velocities are not only better from the point of view of efficiency, but tend to secure greater durability in the machine.

On these grounds it will be clear that there is an advantage in dividing the heat drop into stages, and this is in fact done in all turbines designed for large output and high economy of steam. Though the inventions of De Laval have secured for his single wheel turbine a degree of efficiency which for a single-stage action is remarkably good, it is only in small sizes that his construction can compete with other types, for in large sizes the consumption of steam in compound or many stage turbines can be brought down to a much lower figure.

136. De Laval Turbine. In De Laval's turbine the steam expands in a single step from the initial to the final pressure in a divergent nozzle of the type already described. The jet thus formed is directed against a ring of blades grouped round the circumference of a wheel which is designed to permit of very rapid rotation. With this object the wheel is much thickened in the neighbourhood of the axis to provide strength there for the stresses to which rotation gives rise. This is seen in fig. 62 which is a section through the wheel, showing the thickened

centre and the blades projecting at the circumference.   Further
the shaft is made so thin as to be
flexible, the period of transverse vibra-
tion of the shaft, loaded with the wheel,
being much longer than the time taken
to complete a single revolution.   This
device enables the wheel to run with
great steadiness at the highest speeds,
and protects the bearings and founda-
tion from the vibration which would be
caused by any want of balance if the
shaft were rigid.   The rim of the wheel
has slots cut in it, which hold the blades
firmly by clasping an enlarged root.
Near the circumference, just under the
blade roots, the thickness of the wheel
is reduced (see fig. 62) by turning a
groove on either side.   This is done to
prevent serious damage in the event of
the turbine running away and bursting
the wheel.   The grooved rim forms a
relatively weak place which will give
way before any other part of the wheel,
and the rim with the blades will be

Fig. 62.

thrown off in pieces too small to be capable of breaking their
way through the containing case.   Should rupture of this kind
occur the unbroken portion of the wheel would be relieved of
much of the stress and would speedily be brought to rest by
rubbing on the case through the disturbance of balance resulting
from the breaking of the rim.

The position of a steam nozzle relatively to the blades is
illustrated in fig. 63, where the arrow on the left shows the
direction of rotation of the wheel, and the other arrow shows how
the steam escapes to the exhaust after having had its course
diverted by acting on the blades, only a few of which are shown in
the figure.   The nozzle delivers steam against one side of the ring
of blades, and exhaust takes place from the other side.   In all
except the smallest sizes of these turbines there are several
nozzles, placed at intervals round one side of the wheel, each
delivering an independent jet, and the power is conveniently
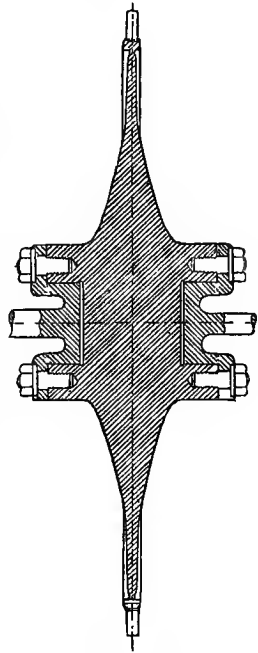
regulated by opening more or fewer of the nozzles as may be required. In a small De Laval turbine, developing 5 horse-power with a single nozzle, the wheel makes some 30,000 revolutions per minute, giving a blade speed of about 500 feet per second. For a turbine developing 300 horse-power a wheel 30 inches in diameter is used, making 10,000 revolutions per minute, and the blade speed is consequently about 1300 feet per second. The high
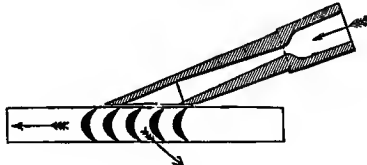
Fig. 63.

speed shaft which carries the wheel in these turbines is geared, by means of double-helical gear wheels with teeth of specially fine pitch, to a second motion shaft running at one-tenth of the speed, and from this the motion is taken by direct coupling or otherwise to the machine which the turbine has to drive. Trials of De Laval turbines using condensers have shown an average steam consumption at the rate of about 20 lbs. per hour per brake horse-power, in a 60 horse-power turbine, and between 15 and 16 lbs. in a turbine of 300 horse-power.

137. Action on the Blades in a De Laval or other Impulse Turbine. In entering the De Laval turbine the jet is inclined at an angle α to the plane of rotation of the wheel. Calling the velocity of the jet v_1 as it enters the wheel, and the velocity of the blades u, we may compound v_1 with u as is shown in fig. 64, to find v_2 which is the velocity of the steam relative to the wheel on admission. A line AB parallel to v_2 determines the proper angle of the blade, on the entrance side, if the steam is to enter without shock. During the passage of the steam through the blade channel this relative velocity v_2 does not change in magnitude, except that it is a little reduced on account of friction between the steam and the blade. The action is one of pure impulse; there is no change of pressure during the passage and consequently no acceleration of the steam through drop in pressure, after the jet has left the nozzle. Hence, if we neglect friction, the

relative velocity with which the steam leaves the blades, namely v_3 in the diagram, is equal in amount to v_2. The direction of v_2 is
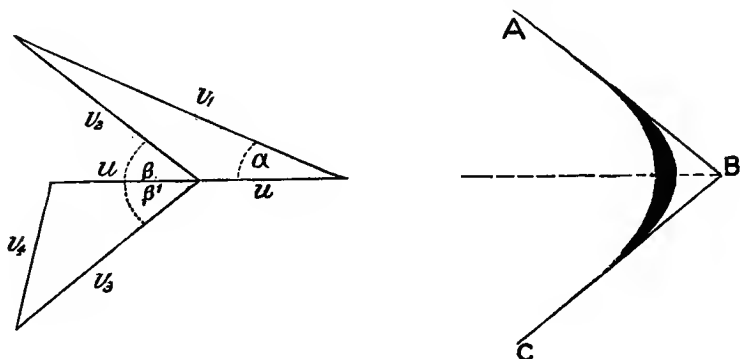
that of BC, which is drawn tangent to the exit side of the blade. Compounding v_3 with u we find v_4 which is the absolute velocity of the steam on leaving the blades, and this should be no greater than is required to get the steam clear of the wheel, for the kinetic energy which the steam retains in consequence of this final velocity is lost.

The most favourable condition of running would be reached by making the blade speed u such that v_4 is perpendicular to the plane of rotation of the wheel, for v_4 would then have the least value for given blade angles of admission and discharge. Assuming the blade angle of discharge β' to be equal to β (as it is in fact usually made for the sake of getting the resultant impulse to act in the plane of the wheel) we should get v_4 perpendicular to the plane of rotation by making $u = \frac{1}{2}v_1 \cos \alpha$. Hence the most favourable blade speed u approximates more and more closely to $\frac{1}{2}v_1$ the smaller α is made. The ideal efficiency, in the action of the jet on the blades, would be achieved if the jet entered the blades without shock and travelled over them without friction. In that case the only loss would be that due to the final absolute velocity v_4, and the efficiency of the action would therefore be

$$\frac{v_1{}^2 - v_4{}^2}{v_1{}^2},$$

which is equal to $1 - \sin^2 \alpha$ when the condition that $u = \frac{1}{2}v_1 \cos \alpha$ is satisfied. In practice α is about $20°$ and therefore the highest

ideal efficiency in the action of the steam on the blades would be 0·883. But owing to the practical impossibility of making the blade speed so high as the above condition requires, the steam in a De Laval turbine enters the blades with some shock and leaves them with a velocity which is inclined to the plane of rotation, with a backward component. Hence further losses arise from the formation of eddies through shock on admission, and also from the fact that the steam escapes with an absolute velocity in excess of what is ideally required. These losses are, of course, additional to those that occur through nozzle friction in the formation of the jet.

Taking a test of a 300 horse-power De Laval turbine, in which the consumption of steam was 15·6 lbs. per horse-power-hour, Stodola estimates that the losses in the nozzle amount to about 15 per cent. of the total heat drop ; the losses in the blades, due to friction and to eddy currents set up by shock, to 21 per cent.; and the loss due to the final velocity in the steam at exit to 5 per cent. The losses due to friction in the mechanism consume about 5 per cent. more, leaving finally a net return of about 54 per cent. of the theoretically available energy.

138. Curtis Turbine. The Curtis turbine, like that of De Laval, is a pure impulse turbine, but the velocity of the jet is extracted not by a single ring of blades but by a series of two or more rings each of which absorbs part of the energy of flow. Between the first and second rings there is fixed a ring of guide blades which serve to alter the direction of the jet in such a manner as to adapt it for acting on the next ring. Each jet, having to begin with acquired its velocity in a nozzle, impinges first on a moving ring of blades and gives up part of its energy of flow to them. It leaves them with a reduced but still considerable absolute velocity and then impinges on a ring of fixed guide blades which turn it into a suitable direction for impinging on a second ring of moving blades to which it gives up a further portion of its energy of flow. In many cases it goes on to a second ring of fixed guide blades and to a third ring of moving blades before the action is completed. The function of these fixed guide blades is merely to alter the direction of the stream: it suffers no drop of pressure in passing over them, or over the moving blades.

Besides this, the Curtis turbine is generally compound in the sense that the whole heat drop is divided into two, three, or more

stages.   Thus the steam in expanding through the first set of
nozzles is brought down to an intermediate pressure in the
chamber in which the first sets of moving and fixed guide blades are
situated.   From this chamber it passes on to a second, at a lower
pressure, through nozzles in which it acquires velocity afresh, and
this velocity is in turn extracted by a second series of moving and
fixed and moving blades.   From this it may pass to a third
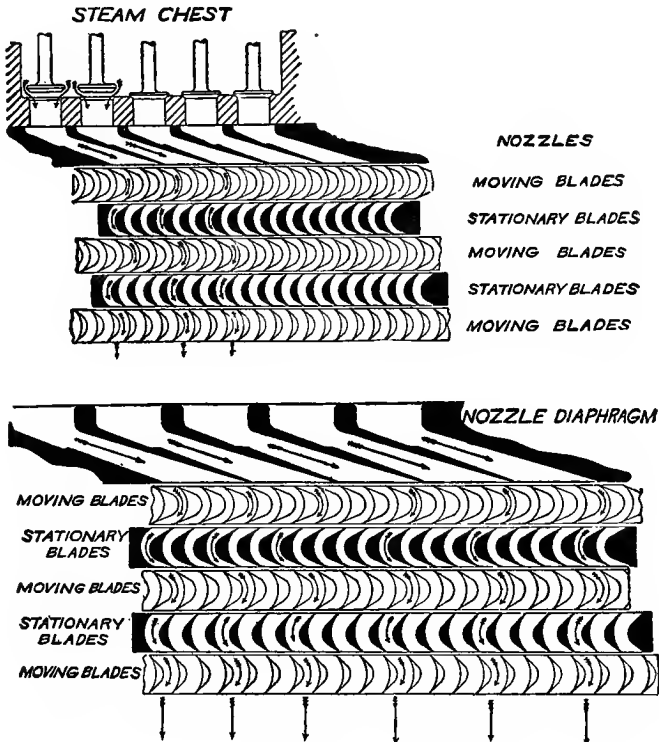
Fig. 65.   Diagram of Steam Nozzles and Blades, Curtis Steam Turbine.

chamber and so on.   The number of such stages varies in
different cases: very commonly four are employed, and at each
stage two successive rings of moving blades are used to extract
the velocity which, if the division is equal, is that due to one-
fourth of the total heat drop.   This secures that the velocity at
each stage is moderate, and the blade speed is still more moderate
because it is by successive impulses in two or more rings of blades

that the kinetic energy acquired at each stage is transferred to the wheel. In the division of the heat drop into stages Curtis follows Parsons: the distinctive feature of the Curtis turbine is the multi-impulse action which occurs at each stage.

The arrangement is illustrated in fig. 65, which shows the nozzles and blades of a two-stage Curtis turbine with three rings of moving blades in each stage. The moving blades are arranged round the periphery of a wheel, and between the three rings of moving blades in each wheel there are two rings of fixed guide blades which are supported by the case in which the wheel is enclosed. Across the case extends a diaphragm separating the first stage from the second, and in this the nozzles are situated in which the steam acquires velocity in the second portion of the heat drop. In each stage the jets of steam proceeding from the nozzles pursue a sinuous course through the moving and fixed blades, and when they leave the third ring of moving blades the jets have only a small residual velocity in a direction nearly parallel to the axis of the wheel. Owing to the progressive reduction which takes place in the absolute velocity of the steam from ring to ring the area of blade opening has to increase and this is provided for by making the blades higher in each successive ring.

The changes of velocity are illustrated in fig. 66 for an ideal turbine with no friction, in which there are three rings of moving blades and two intermediate rings of fixed blades. There u is the velocity of the blades, which is the same for all three rings, and v_1 is the initial velocity of the jet. Compounding these in the triangle OAB we find OB or v_2 which is the initial relative velocity of steam to blade on entry into the first moving ring. This determines the angle of the first set of blades, the form of which is shown alongside, the upper edge, at which the steam enters, being parallel to OB. The line BP is drawn to represent the relative velocity at exit from the first set of blades; it is (neglecting the effect of friction) equal to v_2, and is drawn parallel to the exit edge of the blades. In this example the same inclination is taken at entry and at exit. Compounding BP with u in the triangle PBC we find v_3 the absolute velocity of the jet as it leaves the first set of moving blades. The lines CP and PD, which is equal to CP and is drawn at the same inclination, determine the form of the first set of fixed blades. The jet leaves

them with velocity PD (equal to v_3), and impinges on the second
set of moving blades, over which its relative velocity at entrance is
PE or v_4 and at exit EQ. Similarly FQ or v_5 is the absolute
velocity at exit from the second set of moving blades and is the
velocity over the second set of fixed blades. The jet reaches the
last set of moving blades with the absolute velocity QG, and relative
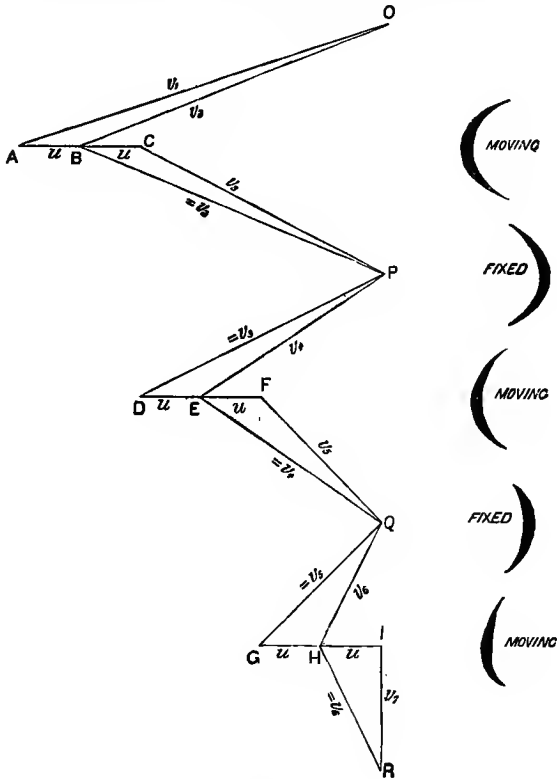
Fig. 66.

velocity v_6, and leaves them with absolute velocity IR or v_7. It
will be seen that in this example the blade speed u has been
so chosen with regard to the magnitude and direction of the initial
velocity v_1 that the final absolute velocity IR is perpendicular to
the plane of the wheel. The edges of the moving blades are
parallel respectively to OB, BP; PE, EQ; QH, HR. Those of the
fixed blades are parallel to CP, PD and FQ, QG. Thus the blades

become progressively flatter in form throughout the series. The efficiency, in the operation of the jet upon the blades, neglecting friction, is $\dfrac{v_1^2 - v_7^2}{v_1^2}$, which in this example is $1 - \sin^2 \alpha$, α being the initial inclination OAB. If we were to take account of friction
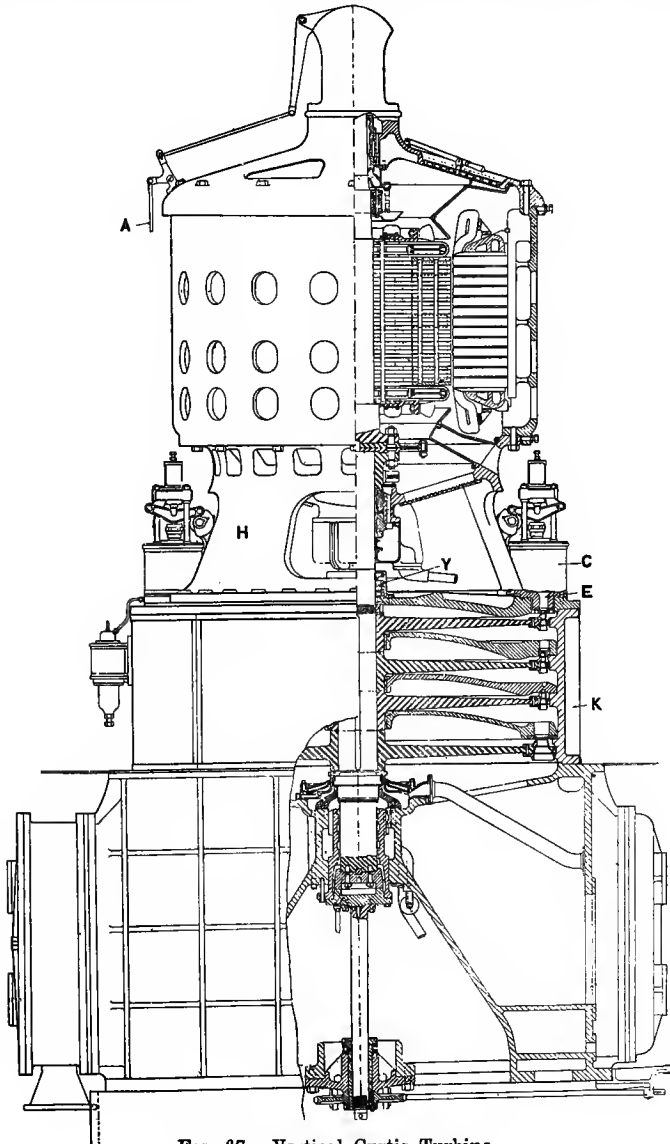
FIG. 67. Vertical Curtis Turbine.

the diagram would be modified first by a reduction in the initial velocity as the steam leaves the nozzle, and then by a loss of velocity in the movement over each fixed or moving blade, with the effect that BP would be less than OB, PD less than CP, and so on.

139. Vertical form of Curtis Turbine. Large Curtis turbines used for driving electric generators are often built with the axis vertical, standing on the condenser as a base and carrying the electric generator on the top. The weight of the revolving part is taken by a foot-step bearing at the bottom end of the shaft,
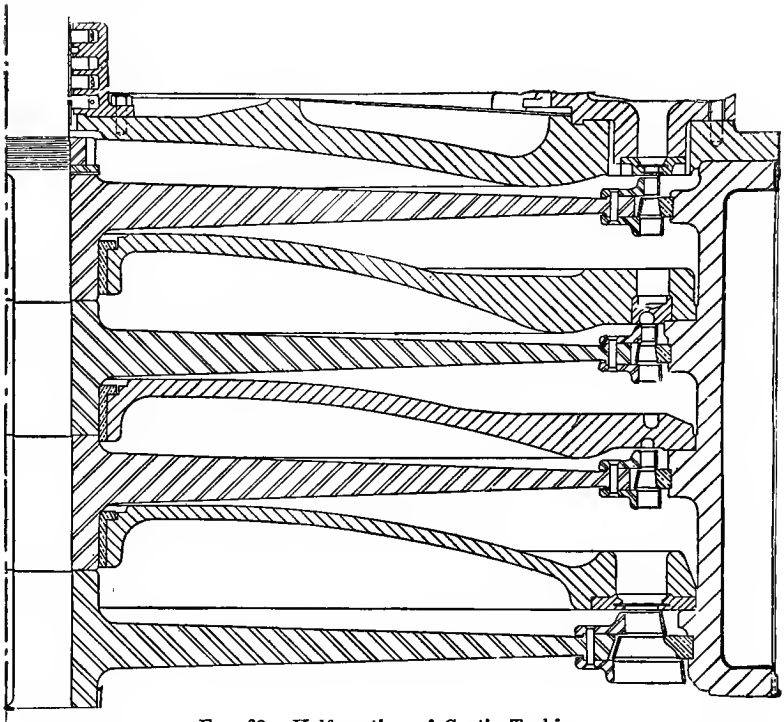
FIG. 68.   Half section of Curtis Turbine.

the bearing surface being a film of oil, which is kept in continuous circulation and is forced in under sufficient pressure to sustain the whole weight of the revolving parts. Contact of metal to metal is in this way avoided in the foot-step which carries the weight. This device is so effective in escaping friction that it is said the turbine will continue to revolve for some hours after steam is shut

off unless it is stopped by applying a load. It is usual to have
four successive pressure stages, with two sets of moving blades
and one set of fixed blades in each stage. Fig. 67 shows a four
stage vertical turbine of this class designed to run at 1000 revolu-
tions per minute with an electric alternator on top developing
3000 kilowatts. Fig. 68 shows a half section of the turbine
wheels to a larger scale. The turbine proper is enclosed in the
part K (fig 67) with the condenser below, and on top of it is the
base H of the electric generator. A centrifugal governor at the
summit of the shaft acts through the rod A on a relay valve
which controls the opening of the valves admitting steam to the
first stage turbine wheel through the steam chamber C and
passages E. The nozzles in the first stage extend round a part
of the circumference only, and this is also true of the stationary
blades, which are supported by a flange in the turbine case so
that they project inwards between the two sets of revolving blades
on the circumference of the wheel. In the later stages the nozzles
and stationary blades extend more nearly all the way round. The
wheels are tapered to increase the strength towards the axis, where
the stresses due to rotation are greatest. The manner in which
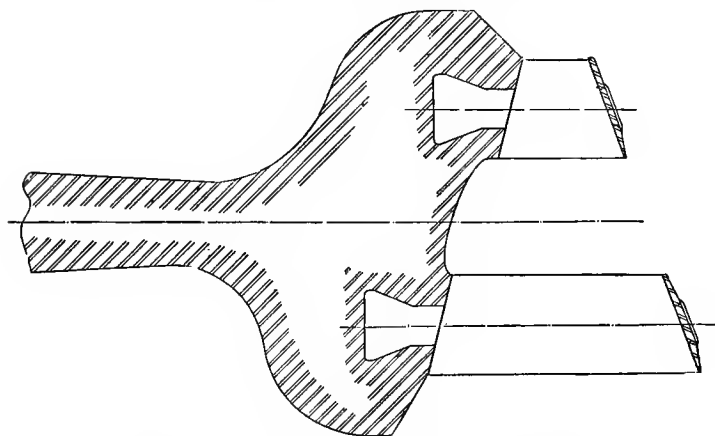
Fig. 69. Rim and Blades of Curtis Turbine wheel.

the blades are attached to them will be apparent from fig. 68.
The end of each blade projects through a steel shrouding strip
and is riveted over on it, so that the strip forms a ring enclosing
and securing the blades. Another mode of fixing the blade is
illustrated in fig. 69 : in this case also the ends are riveted into a

shrouding ring. Each stage is separated from the next by a dia-
phragm plate, the edge of which rests on a flange ring in the
casing: the centre of each plate is bushed to take the revolving
part. The nozzles through which the steam expands acquiring
fresh velocity at each stage are formed by holes in these plates.
In fig. 70 which is from a photograph with part of the turbine
casing removed the nozzle holes in the diaphragm plates are seen,

Fig. 70.

and also the edges of the revolving wheels carrying their blades
with the ends of the blades riveted into shrouding rings. The
shaft passes out of the top of the turbine case through a carbon-
packed gland Y (fig. 67). Between the last stage wheel and the
footstep bearing there is a gland arranged to prevent admission of
air into the vacuous space and also any mixture of oil with the
exhaust steam. Overload is provided for by opening supple-
mentary nozzles in the first stage.

140. Curtis Marine Turbine. Horizontal forms of the
Curtis turbine are also built, for land as well as for marine use.
The applications of this turbine to marine propulsion have hitherto
been somewhat limited. In the Curtis turbines of the steamer
"Creole," with steam of 250 lbs. pressure, and a superheat of
about 35° C., there are seven stages. The first stage has four
rows of moving blades and takes one-fourth of the whole heat

drop; the six remaining each take one-eighth, with three rows of moving blades in each. The first set of nozzles are divergent, the pressure-ratio being so large in that stage as to require divergence according to the theory already explained (§ 131): after the first stage non-divergent orifices are used. The two turbines of this ship develop 4000 horse-power each. The blade rings are about 10 feet in diameter. The first stage has 12 nozzles, each operated by a separate valve. Nine are opened to give full power, leaving three for overload. In each of the other stages valves are fitted, each closing one-fifth of the nozzle area, to enable proper pressures to be maintained at each stage when running at reduced speed. For running astern there is a separate two-stage turbine at the low pressure end, each stage carrying three rows of moving blades.

In the Curtis turbines of the U. S. Cruiser "Salem," which develop about 20,000 horse-power, there are also seven stages, with four rows of moving blades on the first stage wheel and three on each of the others. Two stage wheels each carrying three rows of moving blades at the low pressure end serve for reversing. Steam is admitted for running ahead by 20 nozzles each opened by a separate valve. The propeller thrust is taken by a thrust block of the normal kind, which also serves to fix the axial position of the turbine rotor. The "Salem" is one of three sister ships, the "Birmingham," "Salem" and "Chester," which were engined with reciprocating engines, Curtis turbines, and Parsons turbines respectively for the purpose of comparing performance. On trials made early in 1909 it was found that at full power the "Birmingham" with reciprocating engines achieved a speed of 24·3 knots, the "Salem" 25·9 knots, and the "Chester" (with Parsons turbines) 26·5 knots. The coal consumption in the two turbine ships was much alike, the miles per ton of coal being 1·51 for the "Salem" and 1·55 for the "Chester." When all three vessels were run at $22\frac{1}{2}$ knots, with an expenditure of about 10,000 horse-power, the miles per ton of coal were 2·47 for the "Birmingham," 2·73 for the "Salem" and 2·83 for the "Chester." The results show a marked superiority on the part of both turbines over the reciprocating engines, with a slight advantage in the Parsons turbine over the Curtis[1].

[1] A description of the turbines of the "Salem" and particulars of the results of the trials will be found in Mr Sothern's book on *The Marine Steam Turbine* (3rd Edition, 1909).

141. Performance of Curtis Turbines. Curtis turbines, especially in the vertical forms which are employed in very large sizes for generating electric power, give a high thermal efficiency under tests.   The following figures, supplied by the British Thomson-Houston Company, show that in a large turbine of this class, rated for 9000 kilowatts and actually developing nearly 14000 kilowatts when overloaded, the consumption of steam, superheated to the extent of about 80 degrees Centigrade, was about 13 lbs. per kilowatt-hour throughout a wide range of loads, under the favouring condition of an unusually high vacuum.   The machine ran at 750 revs. per minute.

Load in kilowatts	Steam Pressure lbs. per sq. inch	Superheat °Cent.	Vacuum in inches (Bar. 30 in.)	Steam consumption in lbs. per k.w.-hour
5374	162	74	29·43	13·15
8070	197	64	29·55	13·0
10156	176	82	29·47	12·9
12108	182	80	29·34	13·05
13900	196	78	29·31	13·6

142. Parsons Turbine. In its simplest form the steam turbine may work purely by impulse, as is the case in De Laval's, or purely by reaction as in the primitive contrivance described by Hero of Alexandria (§ 2).   The Parsons turbine works partly by impulse and partly by reaction.   Each ring of moving blades receives the impact of jets directed against it by a preceding ring of fixed blades and to that extent the turbine acts by impulse. But in passing through the channels in the ring of moving blades the steam becomes accelerated.   Its velocity relatively to the blades is increased as well as altered in direction, and it consequently exerts on the blades, by reaction, a greater driving force than it would exert if it passed over them without increase of relative speed.   There is a drop of pressure in the moving blade channels as well as in the fixed blade channels.   Parsons makes the action in both alike, so that the steam loses pressure and acquires relative velocity to just the same extent in a ring of moving blades as it does in a ring of fixed blades.   Each pair of fixed and moving rings of blades makes up what is called a " stage ": the whole is a series of many such stages, which succeed

each other from end to end of the turbine. The blades are in an annular space between the stator and rotor, that is to say between the "cylinder," or enclosing case, and the "spindle," or rotating drum. The fixed blades are secured in grooves in the cylinder and project inwards, nearly touching the surface of the spindle: the moving blades are secured in grooves on the spindle and project outwards, nearly touching the inner surface of the cylinder. The cylinder is stepped so that the annular space becomes progressively larger towards the low pressure end, to give longer blades and so provide a greater area of flow for the steam as its volume increases. The spindle also (in land turbines) is generally stepped in one or two places, to give a higher velocity to the low pressure blades and thereby allow the steam velocity to be greater at the low pressure end, so that less enlargement of the area of flow may suffice, for otherwise the blades would have to be made unduly long.

The number of stages depends on what peripheral speed it is convenient to use. When comparatively high blade speeds are practicable, as in turbines for driving electric generators, the steam is allowed to acquire a fairly high velocity in passing through each ring of blades, and in these cases it suffices to divide the whole heat drop into a moderate number of stages, such as forty-five. This is the number of stages in the example shown in fig. 72, where there are 45 rings of moving blades and 45 rings of fixed blades. In large marine turbines, on the
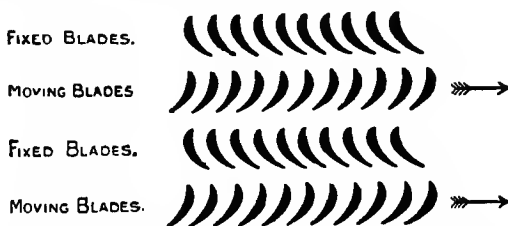
Fig. 71. Fixed and Moving Blades of Parsons Turbine.

other hand, where the number of revolutions per minute has to be kept low in the interests of propeller efficiency, an equally high blade speed could be obtained only by greatly increasing the diameters of the spindle and cylinder. This would be bad, not only because it would make the turbine very bulky, but especially because the blades would have to be very short, to keep

the area of flow of the proper size, and consequently the radial clearance spaces would become relatively too great and the leakage over the tips of the blades would be excessive. To avoid this the number of stages has to be increased, and hence in marine turbines the number is generally between 100 and 200.

The shape and relative position of the fixed and moving blades is indicated in fig. 71, where a few of the blades in two successive stages are shown. It will be noticed that fixed and moving blades have the same form, and are set with the same spacing and with the same entrance and exit angles. From the fact that the steam acquires relative velocity to an equal extent in passing through the fixed and moving channels, it follows that half of the drop for each stage occurs in the fixed blades and half in the moving blades.

Fig. 72 shows in longitudinal section a complete Parsons turbine of 1000 kilowatts capacity (say 1400 horse-power), designed for a speed of 1500 revolutions per minute. Steam enters at A, passes successively through the fixed and moving blades in 45 stages and escapes at B to the condenser, its volume becoming increased about one hundred-fold in the process. The rings of blades are arranged in groups, or "expansions" as they are sometimes called; the first group has seven stages with blades of equal height; the next has also seven but with rather taller blades, and so on. After the third group the spindle is increased in diameter, and on the larger diameter are three more groups, each of four stages. A larger diameter still carries the remaining groups or "expansions," of which there are six, each consisting of two stages. In the last six stages the blades are of the same height, because it would be undesirable, as a matter of construction, to make blades in the final stages any higher, but the effective opening in the last two stages is enlarged by using what are called "wing blades," which are set less obliquely than other blades to the diameter of the axis.

In the ideal turbine there should be a continuous enlargement of the steam passage-way from row to row of the blades, corresponding to the continuous increase in volume which is undergone by the steam as it passes from row to row. But as a matter of convenience in construction it is judged better to confine the changes to a limited number of steps, making the size of the blades the same for several rows in each step, and in this way a sufficiently good approximation to the ideal form is obtained.
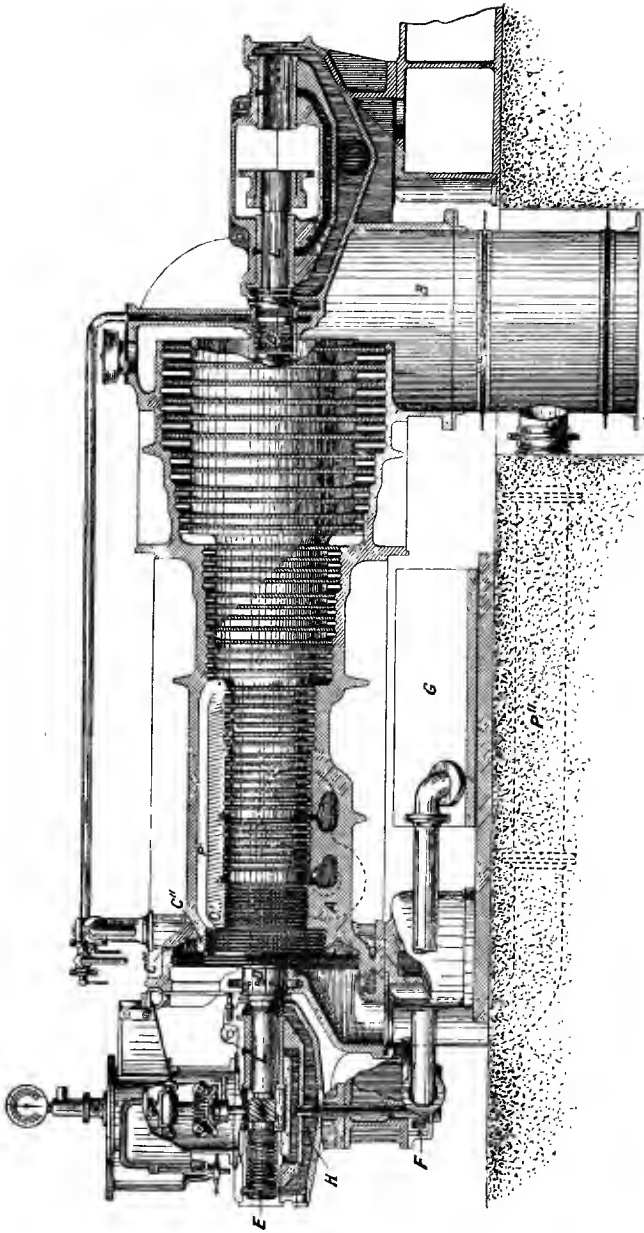
Fig. 72. Section of a Parsons Steam Turbine (Land type).

143. Balance of Longitudinal Forces. Dummies. Since the pressure of the steam falls progressively from left to right, there is a resultant longitudinal thrust on the spindle forcing it to the right. In the earliest Parsons turbines this was avoided by admitting steam in the middle of the length, and making the construction symmetrical on either side, so that steam flowed both ways to an exhaust at each end. In modern turbines, such as the one shown in the figure, a balance is effected by the use of the "dummy" rings C', C'', C'''. These are stepped so that they correspond in diameter to the several portions of the bladed spindle, and steam passages are provided which secure that the same pressure shall act on the dummy, forcing it to the left, as acts on the corresponding portion of the bladed spindle, forcing it to the right. One of these passages P' is shown in the figure, equalizing the pressures which act on the first step in the spindle and on the dummy C''. A similar passage connects the next step with the larger dummy C'''. For the same reason the back of the largest dummy is connected by the pipe P'' to the main exhaust. No steam-tight packing is permissible between the dummy rings and the cylinder, but leakage there is minimized by a device called by Parsons a "labyrinth," which consists in furnishing the circumference of each dummy with a series of collars or rings, the projecting parts of which revolve between a corresponding series of fixed rings projecting inwards from the cylinder. The dummy rings on the rotor and the fixed rings do not touch, but the clearances are made as fine as possible, so that any escaping steam suffers a series of throttlings as it passes each pair of rings, and the result is to offer much resistance to leakage. The device will be illustrated when we come to speak of marine turbines. A similar device is used to prevent or rather to reduce leakage at the glands DD, where the spindle shaft passes out of the cylinder, and the gland at the low pressure end is supplied with steam, at a suitable pressure, so that such leakage as takes place into the turbine is a leakage of steam and not of air. There is a further device resembling on a small scale the thrust-block of a marine engine at one end of the shaft (at E) which keeps the spindle from moving endways, but has provision on it to allow an endways adjustment of its position to be made so as to regulate the amount of the fine longitudinal clearance between the fixed and revolving dummy rings.

144. Bearings and Lubrication. In the early Parsons turbines, which were of small size and adapted to make a very great number of revolutions per minute, the difficulty of obtaining perfectly balanced running was met by means of a bearing which permitted a small amount of lateral movement, but at the same time damped out lateral oscillations by opposing frictional or viscous resistance to them. In the smaller sizes this device is still used, taking the form of a series of three concentric sleeves which fit over a bearing bush on each of the two journals, each sleeve fitting somewhat loosely over the one beneath it. The whole runs in a bath of oil, which is caused to circulate under pressure. Films of oil form over the journal and between the sleeves and so provide a viscous cushion giving the small amount of freedom that is required for lateral displacement and at the same time opposing suitable resistance. The device is remarkably successful in securing steady running at high speeds. In large turbines, where the speeds are more moderate, it is not required, and the bearing is formed of a simple bush of brass, lined with white metal.

The main bearings LL and the thrust block E are supplied with oil under pressure which is kept in continuous circulation by a small rotary pump F, which draws the oil from a tank G. The pump shaft H, which also carries a centrifugal governor to control the speed, is driven from the main shaft by a worm and worm-wheel. This gearing runs in an oil-bath which is included in the oil circulatory system. The same oil is cooled and circulated over and over again with a very small consumption. There is no lubrication of any parts to which steam has access, as these parts have no rubbing surfaces, and consequently the steam is kept free from oil—an important advantage over reciprocating engines which steam turbines in general share.

145. Cylinder, Spindle and Blades. In small turbines the spindles or rotors are solid forgings: in large turbines they are hollow cylinders of forged steel, turned true inside and out, and supported on end shafts, to which they are attached by wheel-shaped steel castings, over which they are shrunk and to which they are fastened by studs, the heads of which are riveted over. The cylinder is of cast-iron, stiffened in the large sizes by external

ribs, with a longitudinal joint on the middle plane which allows the upper half to be lifted off to give access to the rotor.

The blades are generally of drawn brass, polished to make the friction as small as possible, but copper is used for the first few rows to fit the turbine for use with superheated steam. The speed is so moderate that there is no cutting action : blades which have been in use for years show no sign of wear. The most usual method of fixing the blades is to put them one by one into the groove along with distance-pieces which hold them at the proper angle and at the proper distance apart, and secure them by caulking. The height of the blades ranges from a fraction of an inch upwards : in the largest marine turbines (those of the "Lusitania" and "Mauretania") it is as much as 22 inches. When the height exceeds an inch or so the blades are strengthened by means of a ring of stout wire which is let into a notch near the top and extends round the circumference. Each blade is laced to this by binding wire, and the lacing is brazed to hold it fast. In long blades two and even three such rings of stiffening wire, secured by brazed lacing, are introduced at various distances between root and tip. The tips are fined down to little more than a knife-edge in order that if contact should accidentally take place between the fixed and moving parts they may simply grind or burr over without being stripped off. The possible causes of such contact are wear in the bearings and bending or warping of the spindle or cylinder through unequal expansion. With a proper circulation of oil there is practically no wear : unequal expansion can, to a great extent, be avoided by care in the design of the cylinder and in the running, but its effects cannot be entirely escaped except by leaving sufficient clearances. Various methods of assembling and fixing the blades are adopted by Parsons and other makers of his turbines. In one method they are strung with distance-pieces on a wire passing through holes near the root, and having been assembled before-hand in a curved chuck so as to form a sector of the required ring, are brazed together before the sector is put in place. In another plan, the blades are secured in a root-strip of solid metal in which cuts have been machined to receive them. In another the root-strip is cast about steel cores which form in it holes of the right shape to receive the blade ends : the cores are then removed, and the blades are fixed in the holes.

146. Governing of Parsons Turbine. In turbines fitted with a governor Parsons usually arranges that the supply of steam will take place not continuously, but in the form of a series of blasts, the duration of which is automatically adjusted to suit the demand for power. When the turbine is working at full power the admission is practically continuous: at lower powers it is intermittent, the steam valve being opened and closed to give a series of rapidly recurring blasts. To effect this, each revolution of the governor shaft causes a cam attached to the governor to open and close a relay valve which admits steam to a cylinder controlling the position of the main steam valve, and consequently the main valve opens and closes in unison with the relay. The position of the governor determines how long the relay will admit steam to the controlling cylinder in each periodic admission, and consequently determines the duration of the blast. In turbines driving electric generators, the control of the relay valve is often made to depend on variation of electric pressure, instead of directly on variation of speed. In either case the arrangement gives a control that is remarkably free from frictional interference, and therefore secures a high degree of uniformity in speed or in electric pressure, as the case may be.

A conspicuous merit of steam turbines is that they admit readily of overload, that is, of working considerably in excess of the power for which the turbine is designed, with no drawback beyond a certain reduction of thermodynamic efficiency for the time being. To allow overloading to take place provision is often made in Parsons turbines for letting steam enter at full admission pressure beyond the first group of rows of blades : this increases the quantity admitted and causes more power to be developed. An orifice Q will be seen in fig. 72 through which steam can be admitted between the first and second group of rows, in case a larger amount of power is wanted than the turbine will give with full normal admission to the first set of blades.

147. Blade Velocity and Area for flow. Experience has shown that the most economical results are obtained when the velocity of steam through the blades of a Parsons turbine is about twice, or two and a half times, the velocity of the blades themselves. In the design of a turbine a ratio such as this may be assumed as a basis on which the calculation of the principal

dimensions will proceed. From a knowledge of the intended power, and of the steam consumption usual in turbines of the class under consideration, we can judge of the quantity of steam which will pass through the turbine per second. Call the volume of this steam V in the initial conditions, or rather (to be more exact) in the conditions which apply when it escapes from the first ring of guide blades, with a pressure slightly reduced below the pressure at admission. Say that the ratio of steam velocity to blade velocity is to be 2·5. Then $2\cdot5u$ is the velocity of the steam, u being the velocity chosen for the blades, and $\dfrac{V}{2\cdot5u}$ is the area required to be provided for the flow. The passage area between each blade and its neighbour is to be measured at right angles to the direction of the blades at the exit edge, and consequently the whole area available for flow is the area of the annulus between the spindle and the cylinder, multiplied by $\sin\alpha$ where α is the inclination of the blade at exit, subject to a deduction for the space occupied by the blades themselves and also to a correction for leakage over the tips. Generally the angle of discharge is about $20°$ or $22°$, and the effective area for the passage of steam is about one-third of the area of the annulus.

Fig. 73 shows a representative example of the fixed and moving blades in the relative positions they occupy in a Parsons turbine, and alongside of them is drawn a corresponding velocity
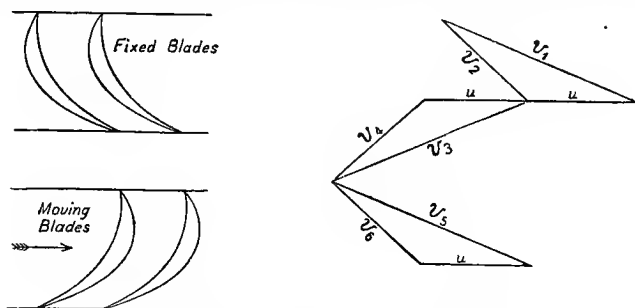
Fig. 73.   Blades of Parsons Turbine.

diagram, in which v_1, the exit velocity from the first ring of fixed blades, is taken as twice the blade velocity u. Compounding v_1 and u we obtain v_2, which is the relative velocity with which the steam enters the moving blades. The design of

the blades is such that their back surface is approximately parallel to v_2, but the blades are so thick near the entrance edge that their front faces have a considerably different slope, and there is therefore some shock at entrance. In passing through the moving blades the velocity of the steam relative to them changes from v_2 to v_3. Allowing for the velocity u of the blades themselves this corresponds to an absolute velocity v_4, with which the steam enters the next set of fixed blades: in these it is again accelerated to v_5, and so on. Except for the fact that the volume of the steam undergoes a slight increase in passing from stage to stage, we should have the same velocity diagram repeated ; but as the annulus is not enlarged until several successive stages are passed, the velocity necessarily increases a little to allow the augmented volume of steam to flow through the same area of passage-way.

In an example of a Parsons marine turbine, for which calculations have been published[1], the blade velocity u for the first ring is 96 ft. per second and the steam velocity v_1 is $2·5 \times 96$ or 240 ft. per second. The first group or "expansion" consists of 11 stages, and in these the steam velocity progressively rises, owing to the augmented volume, until at the end of the group it reaches 321 ft. per second, or $3·34u$. The area of the annulus is then increased by lengthening the blades, so as to bring the ratio of v/u back to 2·5, and this is repeated at each of the succeeding groups or "expansions," of which there are in all 13. In designing the blade height and consequent area of the annulus it is to be borne in mind that the volume of steam for which passage-way has to be provided is the volume actually reached during expansion which, as we have seen, is intermediate between the ideal volume due to adiabatic expansion and the volume which would correspond to saturation. In the case of a turbine of the highest efficiency, utilizing say 70 per cent. of the energy of the Rankine cycle, the actual wetness of the steam, as affected by friction and other modifying causes, is, according to Mr Parsons' experience, from 55 to 60 per cent. of the wetness which would be found if the expansion were adiabatic.

The blade speeds used in Parsons' turbines rarely exceed 350 ft. per second, and are generally a good deal less. In marine forms the blade speeds usually range from about 100 to 150 ft.

[1] "The Design of Marine Steam Turbines," by S. J. Reid, *Min. Proc. Inst. C. E.* Feb. 1909.

per second, though speeds as low as 80 ft. per second have been
used. A high blade speed has the advantage that comparatively
few stages are required, for the velocity gained and given up by
the steam at each stage will then be comparatively great, and
hence the turbine may be short. But, as has been pointed out
above, in a marine turbine, when the number of revolutions is
fixed by other considerations, high blade speed can only be got by
using a large diameter of turbine, and this, apart from serious
drawbacks due to its bulk, would have the disadvantage that the
blades would have to be very short and the loss through leakage
would tend to be excessive.

148. Number of Stages : Division of the Heat Drop.
The number of stages has to be such that the share which each
takes of the whole heat drop will give the desired velocity to the
steam at each stage. This requires an adjustment of the design
which may be arrived at in various ways. Taking the whole heat
drop, from the admission to the exhaust, we may divide it into as
many steps or parts as there are to be " expansions." These parts
may or may not be equal. At the beginning of each step we
calculate, knowing the volume which the steam will then occupy,
what is the proper area of passage-way to be provided, subject to
the condition already explained that the ratio of steam velocity
to blade velocity is to have an assigned value which is the same
at the beginning of each step. This determines the area of the
annulus and the suitable blade height for each " expansion." The
number of stages for each expansion is then found by considering
what is the effective heat drop for each expansion and sub-
dividing it into as many parts as may be necessary to give the
required velocity at each stage. By the effective heat drop is
meant a quantity which is less than the ideal heat drop, corre-
sponding to adiabatic action, by an amount which allows for
frictional and other losses. The ratio of effective to ideal heat
drop is about 0·7 in favourable cases.

The difference between the real action and the ideal is of the
same general character as we have already discussed in speaking
of the De Laval nozzle. In the Parsons turbine we have losses by
leakage as well as by friction. A portion of the steam leaks past
the tips of the blades without doing work : in other words it is
throttled and carries its total heat with it to the region of lower

pressure, without suffering heat drop. The consequence, at each stage of the expansion, is identical in kind with that due to friction of steam against the blades. At each stage the steam is drier than it would be if the ideal adiabatic process were followed, and its volume is greater for any given pressure. The real expansion curve, as represented either on the pressure-volume diagram or the entropy-temperature diagram, or the Mollier (entropy and total-heat) diagram is a curve sloping to the right of the adiabatic line, as the pressure falls, in the manner sketched in figs. 57 to 61.

In each stage the steam is twice accelerated, once on passing through the fixed blades and once on passing through the moving blades, and thus in each semi-stage a definite fraction of the effective heat drop available is used up. Using the notation of fig. 73 and reckoning per unit mass of steam we have

$$\text{Effective heat drop per semi-stage} = \frac{v_3{}^2 - v_2{}^2}{2g} = \frac{v_5{}^2 - v_4{}^2}{2g},$$

and twice this amount measures the effective heat drop per stage, from which the number of stages in the expansion is to be found.

In the example already referred to[1], the division of the heat drop for the successive expansions is effected by dividing the entropy-temperature diagram into a series of horizontal strips. The volume at each step is determined for the actual expansion line in that diagram, in which the drying effect of friction and leakage is included. The area (in Mr Reid's example) is also measured out to the actual expansion line to obtain what may be called a gross heat drop (greater than the ideal) which is then reduced by the use of a suitable factor to give it the effective value.

A slightly different procedure is described by Mr Gerald Stoney[2] who has been associated with Mr Parsons in the development of the turbine from its earliest forms. Taking the pressure-volume diagram we may regard it as divided into horizontal strips. Each acceleration of the velocity of the steam is due to the work

[1] S. J. Reid, *loc. cit.* Reference should be made to this paper for further particulars of the process by which the principal dimensions of steam turbines may be determined.

[2] G. Stoney, Lectures on Steam Turbines, *Journal of the Society of Arts*, 1909.

done by the steam upon itself during a drop δP in the pressure. The amount of this work is $V \delta P$ where V is the actual volume. It is convenient in practice to write this $\dfrac{(PV) \delta P}{P}$, for the product PV changes only slowly as the expansion proceeds. A table of the values of PV throughout the range of pressures from admission to exhaust is prepared, and from these numbers it is easy to calculate the work done at each stage, the pressure P and the drop in pressure δP being known. In the ideal case, with no losses, we should have

$$\frac{v_3{}^2 - v_2{}^2}{2g} = \frac{(PV) \delta P}{P},$$

or

$$v_3{}^2 = \frac{2g (PV) \delta P}{P} + v_2{}^2.$$

v_3 being (as before) the velocity at exit from a ring of blades, and v_2 the velocity of admission to the same ring. But under actual conditions the velocity acquired is less than this, owing to steam friction on the blades, shock, and other sources of loss. It appears that under the conditions which hold in practice in Parsons turbines the frictional and other losses are such that approximately

$$v_3{}^2 = \frac{2g (PV) \delta P}{P},$$

that is to say, the correction for friction and leakage is made with sufficient accuracy by omitting the $v_2{}^2$ term. In this reckoning the volume V is the actual volume at each stage, as augmented by friction and leakage, and the area of the strip in the pressure-volume diagram or $V \delta P$ is what we may call the gross area, measured out to the actual expansion curve. This represents a quantity of work which is greater than the ideal performance in adiabatic expansion, and consequently still greater than the true performance under actual conditions. The omission of the $v_2{}^2$ term is a convenient, if somewhat rough, way of allowing for the difference between the gross area and the area which would represent the effective work, taking losses into account.

149. Parsons Marine Steam Turbines. It is in marine use that the largest examples of the Parsons turbine are to be found. Reference has already been made to the large amounts

PARSONS MARINE TURBINE.

A	Cylinder
B	Rotor
C	Balance Piston
D	Bearing
E	Adjusting Block
F	Steam Packed Glands
G	Worm for Actuating Governor
H	Exhaust to Low Pressure Turbine
J	Steam Inlet
K	Turbine Drain
L	Oil Inlet
M	Oil Drain

Fɪɢ. 74. High Pressure Turbine.

A	Cylinder
B	Rotor
C	Ahead Balance Piston
D	Astern Balance Piston
E	Bearing
F	Adjusting Block
G	Steam Packed Gland
H	Worm for Actuating Governor
J	Low Pressure Turbine Steam Inlet from High Pressure Turbine
K	Astern Turbine Steam Inlet
L	Exhaust to Condenser
M	Low Pressure Manœuvring Steam Inlet
N	Turbine Drain
O	Oil Inlet
P	Oil Drain

Fɪɢ. 75. Low Pressure and rn Turbine.

of power developed by these turbines in various warships and merchant vessels, such as the 41,000 horse-power of the "Indomitable" and her sister cruisers, and the 72,000 horse-power of the "Lusitania" and "Mauretania." In nearly all examples of Parsons turbines for use on land the turbine forms a single drum, the whole expansion being completed within one chamber, though in exceptional instances the turbine has been divided into two distinct high-pressure and low-pressure parts. In marine examples the turbine is always divided into two or more parts which are placed on separate propeller shafts. A common arrangement is to have three shafts, with the high-pressure portion of the turbine on the centre shaft, and the low-pressure portion divided into two similar parts on the two side shafts. In other examples there are four propeller shafts forming two independent sets with a high-pressure turbine on one shaft of each set and a low-pressure turbine on the other. To provide for going astern reversing turbines are added, the blades of which are inclined to the axis in the opposite manner from those of the ahead turbines. The astern turbines are often placed in the same casing with the low-pressure turbines, at the exhaust end so that they run *in vacuo* when the ship is steaming ahead. They consist of a few rows of blades only, designed to develop enough power for going astern at a moderate speed, but with no pretensions to high efficiency. This is a matter of comparatively little consequence since the reversing arrangement is used only occasionally and for short periods. Marine turbines have no speed governor except a safety device for cutting off the supply of steam in the event of a runaway, should the speed approach a dangerous limit.

150. Example of a Parsons Marine Steam Turbine. A characteristic example of a Parsons marine steam turbine, designed for three shafts to develop 12,500 horse-power at 450 revolutions per minute, is shown in the folding plate, figs. 74 and 75. Fig. 74 is the high-pressure part on the central shaft, and fig. 75 is one of the two similar low-pressure parts, along with the reversing turbine which is contained in the same casing. In both figures the upper half is shown in section. In both turbines A is the cylinder or case from which the fixed blades project inwards and B is the rotor or spindle drum carrying the moving blades. B has the same diameter throughout its length

except at the astern turbine (in fig. 75) where it is reduced. Provision for the increased steam volume as expansion proceeds is made by using blades of greater height, stepping the case but not the drum.

In the high-pressure turbine, fig. 74, there are four "expansions" or groups of blade rings, containing respectively 9, 9, 8, and 8 "stages." Entering at J the steam flows to the left through the annulus containing the blades and escapes at H to the low-pressure turbines. The dummy or balance piston C has a diameter smaller than the effective diameter of the turbine, with the result that it leaves unbalanced a part of the steam thrust. This unbalanced resultant is as nearly as possible equal to the thrust of the propeller, and consequently the whole system of longitudinal forces is approximately in equilibrium and scarcely anything is left to be taken up by the thrust-block E. A small thrust-block accordingly suffices: its function is not only to take up whatever difference of thrust is left unbalanced but also to fix the longitudinal position of the rotor and give a means of adjusting that.

The exact adjustment of longitudinal position is a matter of importance on account of the method that is employed in making the dummy steam-tight, by what is known as the labyrinth. This
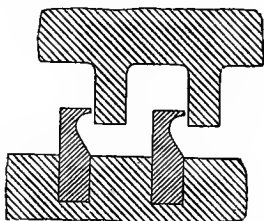
FIG. 76. Dummy rings with facial clearance.

consists of a long series of rings fixed in the case and projecting inwards with their faces just clear of a corresponding series of projecting collars turned on the dummy drum. The construction will be clear from fig. 76, where two of the rings are shown. In the dummy at C in fig. 74 there are 30 such rings. Their effect is to check the leakage of steam by acting as successive throttles, and for this purpose the clearance between the faces is not in general allowed to exceed 15 or 20 thousandths of an inch. The

facing portion of the dummy rings is turned down to an edge about 0·015 inch wide, so that if contact should occur they may grind down without causing trouble.

To permit of the adjustment of these clearances the bush of the thrust-block, which is divided into an upper and a lower half, carries two sets of half-rings of brass which bear against the sides of a corresponding number of collars turned on the shaft. The rings in the upper half of the bush bear against the after faces of the collars and those in the lower half bear against the forward faces. Each half of the bush has its position set by means of a taper key or screw adjusting studs, so as to prevent longitudinal play of the shaft and to provide no more than the desired clearance between the faces of the dummy rings and collars.

In the low-pressure turbine, fig. 75, there is a similar thrust-block F and a similar dummy C on the ahead turbine. The dummy D of the astern turbine has a different form of labyrinth, namely a series of rings with sharp edges and small radial clearance, secured alternately in the case and the dummy drum. This radial form of dummy is shown in fig. 77. It is adopted
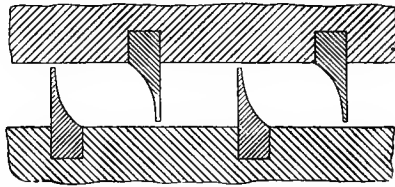
FIG. 77. Dummy rings with radial clearance.

instead of the form shown in fig. 76 because at a point so far from the thrust-block the longitudinal position of the dummy rings is too uncertain, owing to the effect of longitudinal expansion, to make it practicable to have a labyrinth in which the rings have facial clearance. In the radial dummy the rings, which are of brass, have a clearance round the edge of 15 or 20 thousandths of an inch, and are turned to a sharp edge to allow them to grind down readily in case of contact.

The same device is used in the glands (FF, fig. 74; GG, fig. 75) through which the shaft carrying the rotor drum enters the turbine case. In each gland the part nearest the interior consists

of a labyrinth of sharp-edged rings with small radial clearance, and outside of that there are four grooves fitted with split rings of the Ramsbottom type sprung into place and working against a smooth cast-iron sleeve fixed to the case.   Any steam which leaks over the tips of the radial rings is drained off from a space between the labyrinth and the Ramsbottom rings ; and when the pressure within the case is less than that of the atmosphere steam is admitted into the space so that the leakage inwards may be a leakage of steam and not of air.

The spindle is carried by main bearings at DD, fig. 74; EE, fig. 75, which are supplied with oil circulated under pressure from an oil-pump.   The thrust-block is included in the oil circulation. At the extreme forward end of each shaft, beyond the thrust-block, is a worm which drives the runaway safety governor.

In the low-pressure ahead turbine there are eight expansions with six stages in each.   In the last three expansions the blades are of the same height, but are distinguished by wider spacing and less obliquity in the seventh and eighth.   Steam enters at J (fig. 75) and escapes to the condenser by the large exhaust pipe L. Under L is the astern turbine consisting of four expansions with ten stages in each.   For astern running steam enters from the boiler at K.

The rotors consist of steel drums, carefully turned inside and out, shrunk on cast wheels which are carried by end shafts.   In the low-pressure rotor an intermediate wheel serves to connect the drum of the ahead turbine with the smaller drum of the astern turbine.   On the extreme left in each figure is a coupling to connect the rotor shaft with the propeller shaft.

151.   Arrangement of Turbines in the Ship.   The general arrangement of turbines of this class in the ship is shown diagrammatically in fig. 78.   It is the arrangement usually adopted in cross-channel packets and other merchant vessels of moderate size.   Fig. 79 shows a modified three-shaft arrangement which has been adopted in torpedo destroyers and some other war-vessels, in which what are called " cruiser" turbines are added with the object of promoting economy when the vessel is running at a " cruising" speed much less than the speed corresponding to full power.   This is the usual condition under which a war-vessel steams, and it is highly important as regards consumption of

coal that the efficiency of its turbines shall not be unduly low
when the speed is half or less than half the full speed attainable.
The reduced speed requires, for efficiency, a greater number of
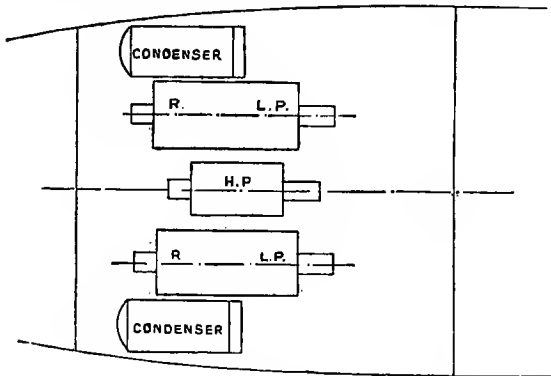
FIG. 78.

stages in order that the velocity of the blades may continue to
bear a proper proportion to the velocity of the steam at each
stage.   Accordingly one or (as in this example) two cruising
turbines are added, which are not used at all in steaming at full
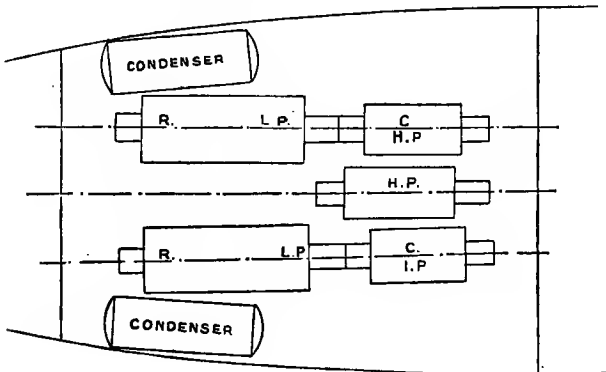
FIG. 79.

power, but are used in series with the others when the speed is
reduced.   Thus at low speeds the steam passes through the first
of these (C.H.P. in fig. 79), and then through the second (C.I.P.)
before entering the main high-pressure turbine on the central

shaft. At a higher speed the first turbine, C.H.P., would be cut out, and boiler steam be admitted to C.I.P.; at a higher speed still C.I.P. would also be cut out. When not in use the cruising turbines revolve *in vacuo*, and provision is made for disconnecting them if desired, when the full speed run is a long one. The cruising turbines contain a large number of stages with short blades. In some of the fastest torpedo-boat destroyers there are three expansions each with 24 stages in C.H.P. and three expansions each with 20 stages in C.I.P., making 132 stages in all, to supplement the 33 H.P. and the 37 L.P. stages of the main turbines; and the cruising turbine blades range from $\frac{3}{4}$ of an inch in height to 2 inches, against 11 inches for the final blades of each of the two L.P. turbines. The shortness of the blades in cruising turbines, by making leakage relatively important, tends to limit their value as an aid to efficiency, and in some instances their advantage in practice has proved so doubtful that they have been disused, and cruising has been carried out with the main turbines only, the
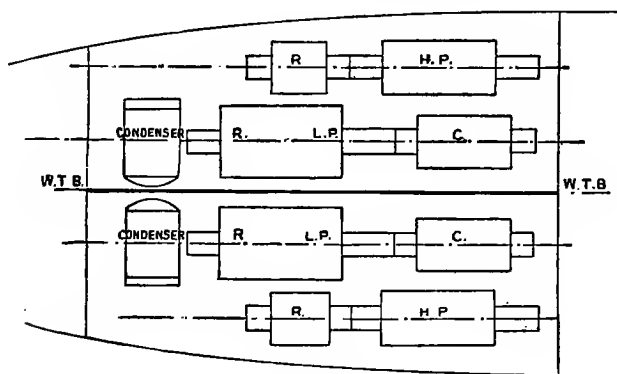
Fig. 80.

steam being throttled before admission to a pressure which is only a small fraction of the pressure in the boilers. The cruising turbines have not been retained in recent designs of battleships or large cruisers[1].

Another arrangement of turbines for warships is illustrated in fig. 80. There are four shafts, with turbines in two independent

[1] See Engineer Vice-Admiral Oram's address on "Propelling Machinery for Warships," *Junior Institution of Engineers*, Nov. 1909, which contains much valuable information on the whole subject.

sets separated by a longitudinal water-tight bulkhead. Each of the two shafts nearest the centre carries a cruiser turbine. All four shafts reverse, a separate reversing turbine being fitted on each wing shaft in addition to a reversing turbine inside the low-pressure casing on each central shaft. The turbines of the "Dreadnought" and "Indomitable" and their respective sister ships furnish examples of this arrangement. In the "Lusitania" and "Mauretania" there are also four shafts, arranged in two pairs. The outside shafts carry the high-pressure turbines, and have no provision for reversing. Each of the inside shafts carries a reversing turbine in a separate case as well as the low-pressure ahead turbine: there are of course no cruising turbines. The low-pressure drums in these ships are 188 inches in diameter without the blades, and the speed is 180 revolutions per minute. The blades range up to a height of 22 inches and a width of $1\frac{3}{16}$ inches at the exhaust end. The low-pressure turbines are about $17\frac{1}{2}$ feet in diameter over all, 50 feet long, and weigh 300 tons.

Various proposals and experiments have been made towards allowing the marine steam turbine to have a different speed of revolution from the propellers, so that the advantages of high speed may be secured in the turbines while the propellers run at the comparatively low speed best adapted for efficiency of propulsion[1]. In some of these arrangements there is mechanical gearing between the turbine and propeller shafts: in others hydraulic gearing: in others still electrical gearing. In the electrical arrangement the turbine drives a dynamo which gives current to a motor driving the propeller shaft. More than one of these devices has the advantage that the propeller may be reversed without reversing the turbine. But the drawbacks of increased weight and general complication stand seriously in the way of the adoption of any system of indirect coupling. A simple gearing, using pinion and spur wheel with helical teeth appears to be the most promising plan; and it is probable that this will enable the steam turbine to be applied to merchant vessels of low speed.

152. Performance of Parsons Turbines. It is now well established not only as a result of special trials but by experience in running in the usual circumstances of practice that in large sizes the Parsons turbine takes less steam per horse-power-hour

[1] See *Engineering*, Sept. 17 and Nov. 5, 1909; *Inst. Nav. Architects*, March 1910.

than any form of reciprocating engine using steam under similar conditions. Trials made in April, 1900, by the present writer, of a 2000 horse-power Parsons turbine coupled to an electric generator showed a consumption of $18\frac{1}{4}$ lbs. per kilowatt hour, with steam at a pressure of 155 lbs. per square inch, superheated 47° C. Since 1 kilowatt is 1·34 horse-power, this consumption was equal to 13·6 lbs. per electrical horse-power-hour. The best piston engines when driving dynamos convert about 84 per cent. of their indicated power into electrical power. Hence the above result is as good, in the relation of the electrical power developed to the consumption of steam, as would be got from a piston-engine using only 11·4 lbs. of steam per indicated horse-power-hour. This figure, good as it is, has been much surpassed in more recent instances where tests have been carried out on turbines of later design and larger size.

An important characteristic of the steam turbine is that it maintains a high efficiency under comparatively light loads, so long as the speed of running is not reduced. This is illustrated by the following figures, which give the results of a series of trials of the machine just referred to, under various loads :

<div align="center">Load in kilowatts.</div>

1450	1250	1000	750	500	250

<div align="center">Steam used per kilowatt-hour in lbs.</div>

18·1	18·5	19·2	20·3	22·6	34·0

A Parsons turbine driving an electric generator rated as of 3500 kilowatts, but working up to 5000, tested in 1907 at the Carville generating station of the Newcastle-on-Tyne Electric Supply Company, showed a steam consumption of only 13·19 lbs. per kilowatt-hour, with steam of 200 lbs. pressure by gauge, superheated 67° C. to a temperature of 264·7° C., the vacuum being 29·04 inches (Barometer 30 inches). We may compare this with the work, theoretically obtainable in an ideal engine following the Rankine cycle, or rather with the "heat drop" in adiabatic expansion which is substantially the same thing. At the temperature and pressure of supply the Total Heat I is 709 units. After adiabatic expansion to the stated vacuum (0·96 inch of mercury) it would be 486 units, the dryness being then 0·792. Hence the heat drop is 203 units ; and this represents the thermal equivalent of the work theoretically obtainable in an ideal operation, but with

the actual conditions as to pressure and temperature of supply and exhaust. Since one kilowatt hour is 1896 thermal units, each lb. of steam was generating an amount of electrical energy equivalent to $\frac{1896}{13\cdot19}$ or 143·7 thermal units. The electrical output therefore corresponds to $\frac{143\cdot7}{203}$ or 64½ per cent. of the ideal work. If we allow for the loss in the generator by taking the electrical output as 92 per cent. of the mechanical work expended it follows that 70 per cent. of the heat drop was mechanically utilized.

An important merit in the Parsons turbine is the absence of wear, and in consequence of this the efficiency remains unimpaired by use. This has been confirmed by comparison of tests made on a new machine and on the same machine after prolonged working. At very high velocities of impact a steam jet will, especially when wet, exert much cutting action in metals, but in the Parsons turbine the velocity of the steam is everywhere too low for this to occur. The blade surface remains smooth and unaffected, and with reasonable attention to the bearings the clearances can be kept from undergoing change.

As regards the steam economy of Parsons turbines used for marine propulsion, the "Dreadnought," which was the earliest example of their application on a large scale, showed on trial at full power a consumption of less than 13½ lbs. of steam per shaft horse-power-hour. With subsequent experience this result was improved on, the average for the three succeeding battleships of the same class being 13·01 lbs. In the still larger turbines fitted in cruisers of the "Indomitable" class, developing 41,000 shaft horse-power, a still lower consumption is recorded, the average at full power being only 12·03 lbs. per shaft horse-power-hour. This is with a pressure of 123 lbs. by gauge on admission to the high-pressure turbine, and without superheat[1].

153. Dummyless Parsons Turbine. In some of the early examples of Parsons turbines, as we have seen, no dummies were required, for the steam entered in the middle of the length and the turbine consisted of two symmetrical systems of blading extending from the middle towards each end and balanced as regards the effects of longitudinal thrust.

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[1] Oram, *loc. cit.*

Messrs White of Cowes in designing Parsons turbines for marine propulsion have devised a form of dummyless turbine in which the idea is carried out of balancing the thrust by making the steam flow both ways, but without making the turbine symmetrical about a central inlet. For marine uses a complete balance is not wanted in the longitudinal forces exerted by the steam, since the propeller thrust has to be reckoned with. Accordingly in Messrs White's design the principle is carried out only so far as to neutralize that part of the steam thrust which is not wanted for balancing the thrust of the propeller. This is done by making the steam flow both ways only in the first group or "expansion." Part of the steam from the inlet flows aft, through a group of blades forming the first expansion, and part of it flows forward, through a group which acts in parallel with these. The latter part then returns through a hollow space in the interior of the spindle and joins the other part of the steam before passing on to the next expansion. All the steam then goes on together through the remainder of the turbine. The group of blades forward of the steam inlet are on a part of the drum having a larger diameter than the others. By a suitable choice of this diameter enough thrust forward is obtained to secure a condition of balance with the thrust aft, when account is taken of the additional thrust forward exerted by the propeller.

154. Utilization of Low-Pressure Steam. Compared with a piston and cylinder engine the turbine has a disadvantage in its comparative inability to make the best use of high-pressure steam in the early stages of the expansion. While the volume of the steam remains small and its density great, the effects of leakage and friction are relatively so considerable that in this part of the range the turbine is notably less efficient. At low pressures, on the other hand, it has the advantage, and in the final stages of expansion it continues to extract useful work out of the steam after the volume has become so large and the pressure so low that the piston-engine is entirely out of court. It is because the turbine can make an effective use of steam in these latest stages, while its pressure approaches the lowest limit attainable in a good condenser, that the general average of its performance is so high. With the turbine a good vacuum is essential to efficiency. In a reciprocating engine expansion is seldom in

practice carried beyond an absolute pressure of 7 lbs. per square inch, and not often beyond 10 lbs. Provided the vacuum is not less than 25 inches or so its amount is a matter of comparative indifference. With the turbine the expansion is continued right down to the condenser pressure, and it is a matter of the utmost consequence to make that as low as may be practicable. A reduction of condenser pressure from 2½ lbs. absolute (25 inches) to 1 lb. absolute (28 inches) raises the efficiency of a turbine by about 13 per cent. when the supply takes place at usual boiler pressures, and raises it by a still larger percentage when the turbine is driven by low-pressure steam.
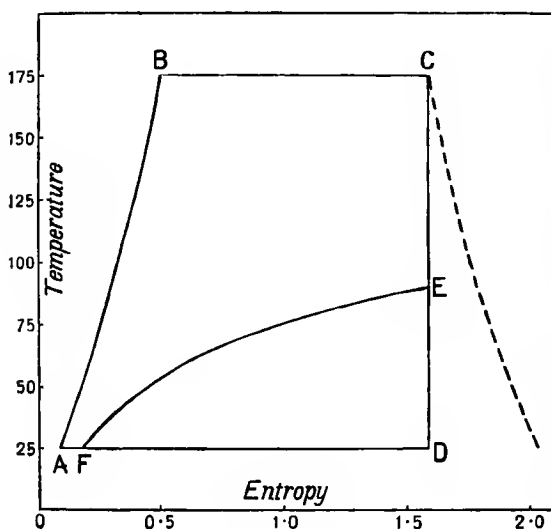
Fig. 81.

This point may be better appreciated if we use the entropy-temperature diagram to compare the work theoretically obtainable when expansion is completed with that obtainable when release takes place at a pressure of say 10 lbs. absolute. In the diagram (fig. 81) the area $ABCD$ represents the work obtainable with complete (adiabatic) expansion from an admission pressure of 226 lbs. absolute down to a condenser pressure of 29 inches or ½ lb. per square inch absolute; while $ABCEF$ represents the work obtainable when release takes place at a pressure of 10 lbs. absolute, EF being a line of constant volume. The same con-

17—2

denser pressure is assumed in both cases. The area *FED* represents what is lost by the incompleteness of the expansion in an engine of the reciprocating type. The figure applies to an ideal performance, but in the conditions of actual work the steam turbine saves most of the area *FED*. It will be noticed that any reduction of vacuum will diminish the performance of the turbine to a much greater degree than it diminishes the performance of the reciprocating engine. To raise the line *AD* affects the turbine along the whole length of the line, but it affects the piston-engine only along the short portion *AF*.

155. Vacuum Augmenter. In view of the importance of a high vacuum the condensers of steam turbines are fitted with air-pumps of large volumetric capacity; and in some cases the ordinary wet air-pump, which removes the water of condensation, is supplemented by a dry air-pump run at a high speed.
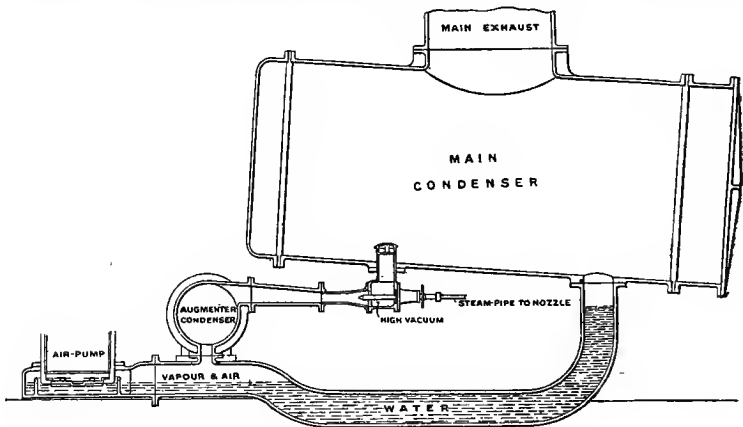
Fig. 82. Parsons Vacuum Augmenter.

Mr Parsons has introduced another device called a *vacuum augmenter,* which is shown in fig. 82. The condensed water passes to the air-pump through a pipe bent to form a water seal. The air from the condenser is extracted by means of a small steam jet pump which delivers it into a chamber called the "augmenter condenser" in which the steam of this jet is condensed. The vacuum in the augmenter condenser is directly produced by the action of the air-pump. The steam jet delivers

air against pressure : its effect is therefore to maintain in the main condenser a vacuum higher than that in the augmenter condenser, and consequently higher than the vacuum the air-pump is itself competent to produce. This is done at the expense of a small quantity of steam supplied to the jet, but the beneficial influence of a high vacuum on the efficiency of the turbine is so great that there is a net saving.

156. Exhaust steam Turbines. The efficient use which the turbine makes of low-pressure steam has led in many cases to the application of what are called exhaust steam turbines, that is to say, turbines which take their supply from the exhaust of other engines. In this way a large amount of power is saved which formerly ran to waste. In some instances such turbines are installed as auxiliaries to non-condensing steam plant: but even when the existing plant has been worked with a condenser it may be well worth while to interpose a turbine designed to operate with steam supplied at the pressure of release, and so recover what would otherwise be lost through incomplete expansion.

An exhaust steam turbine may even be applied to engines which work intermittently, such as the winding engines of collieries or the rolling mill engines of steel works, by adding a device for storing the thermal energy of the exhaust steam in such a manner as to enable a constant supply to be drawn off. Professor Rateau, who is himself the designer of a form of turbine to be mentioned later, has given much attention to this subject and has designed various forms of thermal accumulators which receive the steam from the high-pressure plant irregularly as it comes, and utilize it to generate a steady supply for a turbine designed to work with low-pressure steam. Such thermal accumulators take the simple form of a horizontal boiler-shell, 7 or 8 or even 12 feet in diameter, which is kept about two-thirds full of water. Longitudinal "flues" or pipes are fitted in it, through which the steam bringing in the heat enters. This steam escapes into the water through holes in the sides of these pipes and establishes a vigorous circulation. About 250 lbs. of water are allowed for each lb. of steam, and the pressure ranges from about 15 to 18 lbs. per square inch. In a plant of this type installed at the works of the Steel Company of Scotland, the steam from rolling mill engines which was formerly allowed to blow off

without doing further work, is delivered to a thermal accumulator and furnishes a low-pressure supply for two turbines generating 900 kilowatts of electric power[1].

In connection with the use in turbines of exhaust steam mention may be made of the "closed exhaust" system in steamships, by which the steam used in all the auxiliary engines returns to do useful work in the main engines, being admitted at a place appropriate to its pressure, such as the supply pipe of the low-pressure turbine in the case of a turbine-driven ship. In warships especially the consumption of steam by auxiliary engines is so considerable that when the ship is running at cruising speed, the closed exhaust of the auxiliaries furnishes a large part of the whole supply to the main turbines. In some large turbine-driven battleships the exhaust steam from the auxiliaries is sufficient to drive the ship at five or six knots without drawing on the boilers for any direct supply to the main engines.

157. Combined Reciprocating and Turbine Engines. Apart from the use of exhaust steam turbines to save what would otherwise be wasted, the combination of a reciprocating engine with a turbine presents advantages in certain cases as the most efficient method of employing high-pressure steam. At the low-pressure end of the scale the turbine is beyond question the more efficient type: at the high-pressure end the reciprocating engine has advantages which are especially conspicuous when the speed at which the shaft may run is limited by other considerations. Thus in marine propulsion there would, in general, be a gain in thermal efficiency if the earlier stages of the expansion were performed in a reciprocating engine, and the later stages in a turbine. Whether this gain in efficiency is worth the extra complication and extra weight of the combined machines is a question the answer to which will depend on the circumstances of each individual case. The combination appears to be specially suitable for large ships designed to run at no more than a moderate speed. It is already used in some steamships, and it appears probable that by such a mixture of types the turbine may be adapted to the service of certain classes of vessels in which the speed is too low to admit of a satisfactory design of turbine alone. With a low speed of propeller shaft, the turbine can be made efficient

[1] See a paper by Mr R. F. Halliwell, *Engineering*, 5 Feb. 1909.

only by increasing greatly the number of stages, which makes the turbine unduly long, or by increasing the drum diameter to get greater blade speed, which not only adds seriously to the bulk and weight but enlarges the circumference of the annulus so much that the blades have to be made very short, with the result that the clearance over the tips is relatively too great and the leakage becomes excessive. For this reason turbines alone are not applicable (without gearing) to vessels whose speed is less than about 15 knots. But this limit would not apply if the turbine were used for the lower part of the pressure range only, and the upper part were taken by an engine of the piston and cylinder type[1].

158. Rateau and Zölly Turbines. Professor Rateau has designed a form of steam turbine which combines some of the features of the Parsons turbine with those of the De Laval. He divides the whole drop into a limited number of stages, sometimes as many as 24, and at each stage employs an impulse wheel substantially of the De Laval type, the steam passing from one stage to the next through a diaphragm with nozzles. The turbine is in effect a series of impulse wheels on the same shaft, each enclosed in a separate chamber, for each wheel is separated from the next by a diaphragm containing groups of fixed vanes forming the nozzles in which the steam acquires fresh velocity before impinging on the moving vanes of the next wheel. The fixed vanes or nozzles in the diaphragm do not, in the earlier members of the series, extend all round the circumference, but the arc over which each group extends becomes larger in successive diaphragms, and they are so set that, as far as possible, the steam which has traversed one wheel passes on through the next set of nozzles without having its velocity checked by striking a blind portion of the diaphragm.

The Zölly turbine is of the same general type, differing from Rateau's only in mechanical details. It has come into extensive use, and on a large scale. Tests at Essen of a 5000 kilowatt Zölly turbine, running at 1000 revolutions per minute, show a steam consumption of about 15·2 lbs. per kilowatt hour with steam supplied at 162 lbs. per square inch absolute, superheated 100° C.,

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[1] See a paper "On the Combination System of Reciprocating Engines and Steam Turbines," by the Hon. C. A. Parsons and Mr R. J. Walker, *Trans. Inst. Nav. Arch.* April 1908.

the condenser pressure being 1·16 lbs. per square inch. A 6000 kilowatt Zölly turbine installed at the Manchester Corporation Electricity Works in 1909 has a horizontal shaft 11 inches in diameter carrying 20 turbine wheels, forming 20 stages. They are in two groups, the first group being 67¾ inches in diameter and the second group about 90 inches measured to the tips of the blades. It runs at 1000 revolutions per minute. By the help of a bye-pass admitting steam direct to the sixth stage the output may on an emergency be raised to 10,000 kilowatts. Tests at the normal full load gave a consumption of 14·6 lbs. per kilowatt hour, and at half load 15·9 lbs., the steam being superheated through about 78° C.[1]

Among other forms of turbine not requiring special notice here is the Riedler-Stumpf, in which a jet acts on Pelton buckets formed on the circumference of a single wheel. In some instances two sets of such buckets are placed side by side on the circumference of the wheel, and fixed guiding channels are arranged outside the circumference in such a manner that the jet, after acting on one set of buckets has its direction reversed and is returned to the wheel to continue its action on the adjoining set. The effect is to have one pressure and two velocity-stages.

Combinations of two types of turbine are occasionally found, as when a Curtis wheel is used for the first stage or stages of the expansion, and the action is completed in a Parsons turbine. This combination is favoured by more than one maker, on the ground that it escapes the leakage losses which tend to make the early stages of a Parsons turbine comparatively inefficient, and also facilitates the use of steam having a considerable degree of initial superheat, for the relatively large clearances of the impulse wheel escape the risk to which a Parsons turbine is liable through unequal expansion under high superheat.

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[1] *Electrical Engineering*, Oct. 28, 1909. For a detailed account of the Zölly turbine see *Engineering*, July 3, 1908.

CHAPTER IX.

THE TESTING OF STEAM-ENGINES.

159. The Indicator. In this chapter we have first to describe the ordinary process of taking indicator diagrams, whether for the purpose of finding the horse-power of an engine or of examining the action of the steam; then to speak of those further measurements that have to be made when the thermodynamic efficiency of the engine is under trial; and finally to mention a method of finding the brake horse-power which, by comparison with the indicated power, gives the mechanical efficiency. The indicator diagram, apart from its use in determining power, is invaluable as an index of what is going on within the cylinder. It shows the time and manner of the four events of the stroke, namely, the admission, cut-off, release and compression, which together make up what is called the "distribution" of the steam; it detects faults in the setting or in the working of the valves and suggests changes by which the distribution may be improved. When the information which it gives is supplemented by a knowledge of how much steam is passing through the cylinder per stroke a complete analysis of the action becomes possible; the wetness of the steam at any stage may then be determined, as well as the exchanges of heat that take place between the steam and the cylinder walls.

The indicator, invented by Watt and improved by M'Naught and by Richards, consists in its simplest form of a small steam cylinder, fitted with a piston which slides easily within it and is pressed down by a spiral spring of steel wire. The cylinder of the indicator is connected by a pipe below this piston to one or other end of the cylinder of the engine, so that steam from the engine cylinder has free access, and the piston of the indicator consequently rises and falls in response to the fluctuations of pressure which occur in the engine cylinder. The indicator piston actuates

a pencil, which rises and falls with it and traces the diagram on
a sheet of paper fixed to a drum that is caused to turn back and
forth about its axis through a certain angle, in unison with the
motion of the engine piston.   In M'Naught's indicator the pencil
was directly attached to the indicator piston, in Richards' the
pencil is moved by means of a system of links so that it copies the
motion of the indicator piston on a magnified scale.   This has the
advantage that an equally large diagram is drawn with much less
movement of the indicator piston, and errors which are caused
by the piston's inertia are consequently reduced.   In high-speed
engines especially it is important to minimize the inertia of the
indicator piston and the parts connected with it.   In Richards'
indicator the linkage employed to multiply the indicator piston's
motion is an arrangement similar to the parallel motion which
was introduced by Watt as a means of guiding the piston-rod in
beam engines.   In several forms of indicator lighter linkages are
adopted, and other changes have been made with the object of
fitting the instrument better for high-speed work.   One of the
best of these modified forms of Richards' indicator is that made
by the Crosby Company, which is shown in figs. 83 and 84.
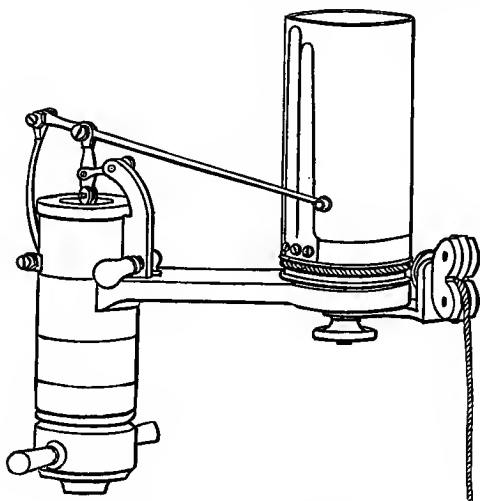
Fig. 83.   Crosby Indicator.

The pressure of steam in the engine cylinder raises the piston F
(which is shown in section in fig. 84), compressing the spring

above it and causing the pencil to rise in a nearly straight line through a distance proportional, on a magnified scale, to the compression of the spring, and therefore to the pressure of the steam. At the same time the drum *D*, which carries the paper,
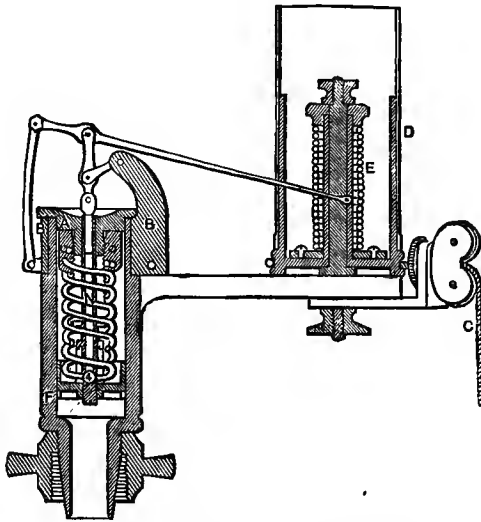
Fig. 84.   Crosby Indicator.   Sectional view.

receives motion through the cord *C* from the cross-head of the engine. Inside this drum there is a spiral spring *E* which becomes wound up when the cord is pulled, and serves to turn the drum in the reverse direction during the back stroke when the cord is relaxed. The cap *A* of the indicator cylinder has holes in it which admit air freely to the top of the piston, keeping the upper side at the pressure of the atmosphere, and the piston has room to descend, extending its spring, when the pressure of the steam is less than that of the atmosphere. The spring is easily taken out and replaced by a stiffer or less stiff one when higher or lower pressures have to be dealt with. Springs adapted to various ranges of pressure are supplied with the indicator and are marked with a number which states the pressure, in lbs. per square inch, which will raise the pencil through a distance of one inch on the paper. The accuracy of this number should be verified by testing the indicator under steam against a standard pressure-gauge, or against a mercury column. Tests made by applying water under pressure are not suitable, unless a proper

allowance be made for the change of elasticity of the spring through change of temperature. The spring is stiffer (generally by two or three per cent.) when cold than at the temperature (about 100°C.) which it takes up when in use. In testing an indicator the pressure of the steam should be raised slowly or by steps, and the test should be made with rising and also with falling pressure, to see that there is no material friction-error, which would show itself by causing the indicator pencil to stand higher during the fall than during the rise when the steam had the same pressure.

A tap, placed just below the indicator but not shown in the diagram, allows the communication with the cylinder to be closed at pleasure, and also puts the space below the indicator piston into communication with the atmosphere, thereby allowing the "atmospheric line" to be drawn on the diagram.

The pencil is withdrawn from the paper by turning back the piece *BB* which is separate from the rest of the indicator cylinder. The small handle shown in fig. 83 is provided for this purpose, and a stop behind the handle prevents the pressure of the pencil against the paper from exceeding a regulated amount.

Fig. 85 shows another arrangement of pencil gear and spring used in one of the forms of indicator made by Messrs Elliott Brothers. Here the spring is outside, away from the action of the steam, and its form is such that it is easily adjusted and easily removed and replaced. In several other indicators an external spiral spring is used.

160. Optical Indicator. In the indicating of gas-engines and oil- or petrol-engines, where the ignition of the combustible mixture causes a very sudden rise of pressure, it is specially important to avoid the effects of inertia which not only tend to cause oscillations in the diagram but to loosen the joints of the indicating mechanism and
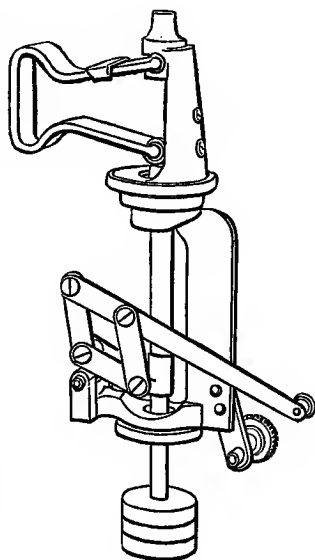
Fig. 85.

so to produce error due to backlash. By using a very stiff spring the bad effects of inertia may be escaped, but this is done at the expense of reducing the height of the diagram, with the result that any error due to backlash becomes relatively more important. Accordingly optical methods of magnifying the motion have been resorted to, in place of mechanical methods. By using a very stiff spring to keep the motion of the piston within narrow limits and magnifying this small motion by an optical device which is free from inertia and backlash, an indicator is obtained which will work satisfactorily at the highest speeds, and in conditions under which indicators of the older kind are inapplicable.

In some forms of optical indicator an elastic diaphragm has been substituted for the piston, the elasticity of the diaphragm itself furnishing the control without a separate spring. This was the case in an early form designed for laboratory use by Professor Perry. But it appears preferable to retain the use of a piston with a separate spring, which can be changed at will to suit the different ranges of pressure. This has a uniform scale, and has the
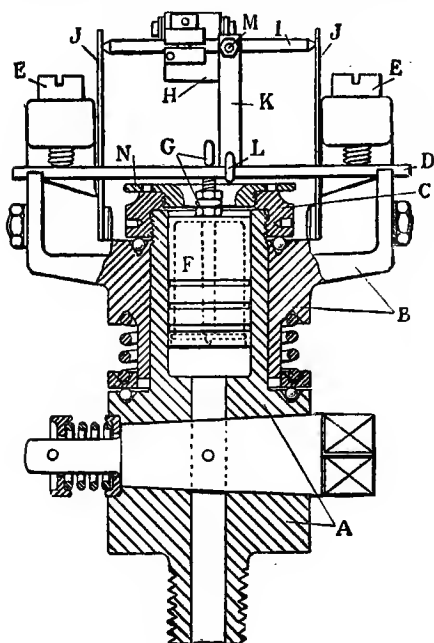
FIG. 86.
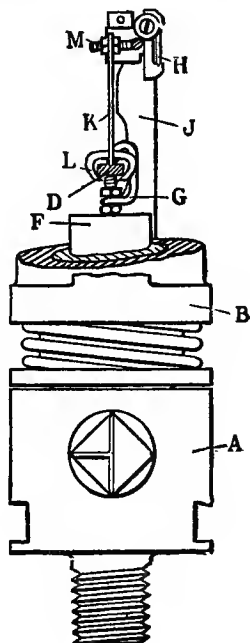
3 Inches
FIG. 87.

further advantage that it escapes the variations of temperature, and therefore of elastic stiffness to which a diaphragm is subject in use. A form of optical indicator with a piston designed by Professor Hopkinson for gas-engine researches is shown in figs. 86 and 87. The block A which is screwed into the indicator hole of the engine is bored to receive a piston F, made hollow for the sake of lightness, and furnished with three grooves to reduce leakage. Round the outside of A fits a frame B abutting against a shoulder at the top and pressed up against the shoulder by a spiral spring in compression. The shoulder is furnished with a ball-race to make the frame B turn readily about the axis of the piston. The spring which controls the motion of the indicator piston is a flat bar D clamped to the frame B by screws EE. The part between the screws is free to bend as a beam and is connected with the piston by the hook G. The movement of the piston and spring is communicated to a small mirror M, pivoted by an axle I between flexible side supports JJ, through a flexible arm K which makes the mirror tilt by an amount that is proportional to the small movement of the piston. A beam of light reflected from the mirror is accordingly displaced in proportion to the changes of pressure in the engine cylinder. At the same time the mirror receives another motion, at right angles to the first, proportional to the displacement of the piston of the engine. This is done by making the frame B rock about its axis, by connecting it with an excentric which causes this rocking motion to correspond exactly with the motion of the main piston. The beam of light consequently traces out an indicator diagram. The diagram is made visible by the optical device shown in Fig. 88. Light falls on the indicator
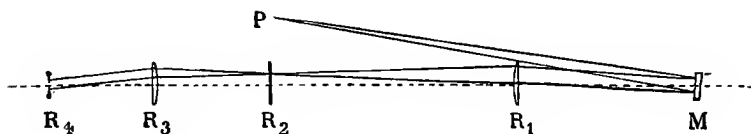
Fig. 88.

mirror M, from a source P, consisting of a small hole illuminated by an incandescent lamp. The reflected beam is brought to a focus at R_2 by the lens R_1. This lens being 4 inches in diameter is big enough to include the largest displacements of the beam. A transparent screen is placed at R_2 and a second lens R_3 is interposed between it and the eye, which is placed at R_4,

the principal focus of R_3. The transparent screen is engraved with horizontal and vertical lines on which the diagram is seen projected. As an alternative a photographic plate may be placed at R_2 to give a permanent record [1].

161. Conditions of accurate working. To register correctly, an indicator must satisfy two conditions: (1) the motion of the piston must be proportional to the change of steam-pressure in the engine cylinder; and (2) the motion of the drum must be proportional to that of the engine piston.

The first of these conditions requires that the pipe which connects the indicator with the cylinder shall be short and of sufficient bore, and that it shall open in the cylinder at a place where the pressure in it will not be affected by the kinetic action of the steam during admission or exhaust. Frequently pipes are led from both ends of the cylinder to a central position where the indicator is set, so that diagrams may be taken from either end without shifting the instrument. This arrangement is convenient and shows the double action prettily; but except with small cylinders it makes the connecting pipes so long as to give rise to serious errors. In large engines it is therefore not admissible: a pair of indicators should rather be used, each fixed with the shortest possible connecting pipe, or the diagrams should be taken successively from the two ends of the cylinder with a single instrument set first at one end and then at the other. The general effect of an insufficiently free connection between the indicator and the engine cylinder is to make the diagram too small.

The first condition of correct working is also invalidated to some extent by the friction of the indicator piston, of the joints in the linkage, and of the pencil on the paper. The piston must slide very freely; nothing of the nature of packing is permissible, and any steam that leaks past it must have a free exit through the cover. The pencil pressure must not exceed the minimum which is necessary for clear marking. By careful use of a well-made instrument the error due to friction in the piston and connected parts may be kept so small as not to be serious. Another source of disturbance is the inertia of these parts, which

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[1] For a fuller account of this indicator, with examples of diagrams given by it, see Professor Hopkinson's paper on the "Indicated Power and Mechanical Efficiency of the Gas-Engine" (*Proc. Inst. Mech. Eng.* Oct. 1907).

tends to set them into oscillation whenever the indicator piston
suffers a comparatively sudden displacement.   These oscillations,
superposed upon the legitimate motions of the piston, give a wavy
outline to parts of the diagram, especially when the speed is great
and when the last-named source of error (the friction) is small.
When they appear on the diagram a continuous curve may be
sketched by hand midway between the crests and hollows of the
undulations.   To keep the oscillations within reasonable compass
in high-speed work a stiff spring must be used and an indicator
with light parts has to be selected.   Still another possible source
of error is backlash through too great looseness in the joints.
Finally, to secure accuracy in the pencil's movement, the strain
of the spring must be kept well within the limit of elasticity,
so that the strain may be as nearly as possible proportional to
the steam-pressure.

With regard to the motion of the drum, it is, in the first place,
necessary to have a reducing mechanism which will give a suffi-
ciently accurate copy, on a small scale, of the engine piston's stroke.
Many contrivances are used for this purpose; in some a rigorous
geometrical solution of the problem is aimed at—as for instance by
adapting some form of pantagraph—but it is not unusual to find
that the multiplicity of joints in such mechanisms gives rise, through
backlash, to greater errors than would occur in simpler forms of
gear designed to produce no more than a close approximation.
Of these simpler gears a very usual form is that shown in fig. 89.
AB is a long pendulum rod, pivoted on a fixed centre at A.
BC is a short link connecting B with the cross-head C.   The
indicator drum receives its motion from any suitable point D on
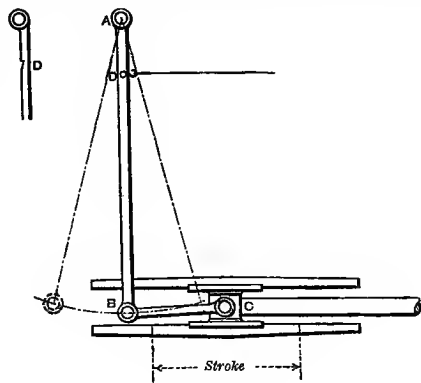
Fig. 89.

the rod *AB* by a cord which is led over pulleys if necessary. A very convenient arrangement, shown by the separate sketch at the side, is to have a notch in the rod *AB* at *D* in which a hook at the end of the cord will engage when the hook is slipped along from *A*: this gives a ready means of throwing the indicator drum in or out of gear. To make the mechanism fairly accurate the length of the pendulum rod *AB* should be considerably greater than that of the stroke. Another form of indicator gear is obtained by omitting *BC*, joining *AB* directly to the cross-head and setting the pin *A* on a sliding block in fixed guides which allow it to slide towards or away from the piston as the rod oscillates. This is geometrically a better form, but it requires careful construction to escape backlash at *A*.

Even when the cord which is to move the indicator drum is connected to the piston-rod in such a way as to copy its motion correctly, the motion of the drum itself may become incorrect because the length of the cord is not strictly constant. The varying pull causes varying amounts of extension, and when the cord is a long one the error which this involves may be serious. The tension in the cord varies from three causes:—(1) the varying resistance of the drum spring, (2) the varying acceleration of the drum, and (3) the friction of the drum and of the guide pulleys, if there are any. Causes (1) and (2) may be arranged to counteract each other, but the friction of the drum necessarily tends to make the cord longer during the forward motion of the drum than during the backward motion. Hence it is important to see that the drum and pulleys turn readily with little friction, and still more important to make the cord short. Where there is a long distance between the indicator and the point from which motion is taken—as will generally be the case in large engines—cord should be used only at places where flexibility is required and stout wire should as far as possible be substituted. Even in comparatively small engines wire may be used with advantage[1]. Of all the errors to which indicator diagrams are liable perhaps none are so often neglected as those that come from the stretching of long driving cords[2].

[1] See for example the device used by Prof. O. Reynolds on the experimental engine at Owens College, by which the length of the cord is reduced to a few inches (*Min. Proc. Inst. C. E.* vol. xc. 1889).

[2] For a discussion and experimental investigation of the errors of the indicator, see papers by Prof. O. Reynolds and Mr H. W. Brightmore (*Min. Proc. Inst. C. E.* vol. lxxxiii. 1896).

162. Directions for taking Indicator Diagrams. In taking indicator diagrams the following practical hints may be found useful:—Before attaching the indicator to the engine, see that the indicator is clean and in good order; that the piston moves very freely; that the joints of the lever and links are oiled with fine oil and are sufficiently slack to avoid friction, but not so slack as to allow the pencil to shake; that the pencil point is sharp, and that it is adjusted to press lightly upon the paper drum; and that the paper drum turns freely without shaking. The spindle on which the drum turns needs oil now and then.

Select a spring appropriate to the pressure within the cylinder and to the speed of the engine. With the Crosby indicator the diagram should not be more than $1\frac{3}{4}$ inches high; thus a 50 spring should not be used if the range of pressure to be indicated exceeds 87 lbs. per sq. in. When the engine runs fast it is necessary to use a still stiffer spring, to prevent the diagram from showing an inconvenient amount of oscillation. If large oscillations occur the process of smoothing the diagram by sketching a line midway between the crests and hollows is unsatisfactory, and a new diagram must be taken with a stiffer spring.

In putting a spring in and screwing the parts together, try whether there is any backlash or shake between the spring and the indicator piston. If there is any it is to be taken up (in the Crosby instrument) by means of the set screw under the piston.

Screw the indicator cock to the pipe on the engine cylinder, and couple up the indicator, taking care to tighten up the coupling collar in such a position that it leaves the handle of the cock free to turn. See that the cord from the drum has a clear course to the oscillating lever, and that its mean position during the oscillation is about perpendicular to the lever. Adjust the length of the cord and the amount of its motion so that when the cord is in gear the drum turns backwards and forwards without coming up against a stop at either end of its travel. If it touches one stop or the other the cord is too long or too short: if it touches both stops the travel of the drum is too great and a point nearer the fulcrum of the oscillating lever must be taken for the attachment of the driving cord.

Do not keep the indicator drum moving except while diagrams are being taken. Stop the drum by disconnecting the cord from the oscillating lever before attempting to put a paper on the drum. In putting on the paper see that it is taut and clear of wrinkles, and fold down the projecting edges so that they may not touch the lever which carries the marking pencil.

Turn on steam to the indicator for a minute or so before taking the diagram. Then press the pencil lightly on the paper, keeping it on long enough to complete a single diagram. Withdraw the pencil. Shut the cock leading to one end of the cylinder and open the cock from the other end (if pipes from both ends come to the same indicator). Touch the pencil to the paper again to take the other diagram. Withdraw it and shut the indicator cock. Touch the pencil again to the paper to draw the atmospheric line. Stop the drum by disconnecting the cord. Remove the paper and mark the diagrams to show which end of the cylinder each refers to. Note the scale number of the spring, and the speed of the engine, with the date and hour and any other particulars that may be wanted.

163. Calculation of the Indicated Horse-power. By measuring the mean height of the diagram between the top and bottom lines we find the *mean effective pressure*, which when multiplied by the area of the piston and the length of the stroke give the work done per stroke.

The mean height of the diagram is most accurately found by measuring the area of the diagram with a planimeter or otherwise (as for instance by applying Simpson's rule) and dividing that area by the length of the base, namely, the distance between lines drawn perpendicular to the atmospheric line and touching the diagram at its extremities. More usually the mean height is found by dividing the base into ten or twelve equal parts, drawing a perpendicular to the atmospheric line through the middle of each of these parts, measuring the lengths of these perpendiculars between the top and bottom lines and taking the mean of these lengths. The lengths of these perpendiculars are most conveniently measured by applying to each line in succession a scale graduated in inches and tenths of an inch: with a little practice it is easy to estimate to hundredths of an inch. The mean height in inches is multiplied by the scale number of the

18—2

spring to find the mean effective pressure in lbs. per square inch. The mean effective pressure for the other side of the piston is found from the other diagram in the same way. Calling these mean effective pressures p_m and p_m', in lbs. per square inch, and the net areas of the corresponding sides of the piston a and a' in square inches, and the length of the stroke l in feet, the work done by the steam per revolution is

$$l\,(p_m a + p_m'a')$$

in foot-pounds.

The work done per minute is

$$nl(p_m a + p_m'a'),$$

n being the number of revolutions per minute; and the indicated horse-power

$$\text{I.H.P.} = \frac{nl\,(p_m a + p_m'a')}{33000}.$$

In general a and a' are nearly equal. The sum of them may then be taken and multiplied by the mean pressure $\frac{1}{2}\,(p_m + p_m')$, as a substitute for the quantity within brackets. And it is convenient when many diagrams are to be worked out for one engine to express the quantity $\frac{l\,(a + a')}{33000}$ as a single constant factor which has only to be multiplied by the mean pressure $\frac{1}{2}\,(p_m + p_m')$ and by n to find the indicated power.

In place of the ordinary indicator an apparatus is occasionally used which integrates the two coordinates which it is the business of the indicator diagram to represent, and exhibits the power developed from stroke to stroke by the progressive movement of an index round a dial.

164. Examples of Indicator Diagrams. Fig. 90 shows a pair of indicator diagrams taken from a Corliss condensing engine, in which after a very early cut-off the whole expansion is performed in a single cylinder. In these and subsequent diagrams lines are added at either end which show the amounts of the respective clearances, and as base line the line of absolute vacuum is drawn, the distance of which below the atmospheric line is determined by reading the barometer. The numbers are pressures in lbs. per square inch above the atmosphere. Inspection of the diagram shows that the distribution of steam is very symmetrical as regards

the two ends of the cylinder; also that the amount of compression might be increased with advantage. If an adiabatic curve be
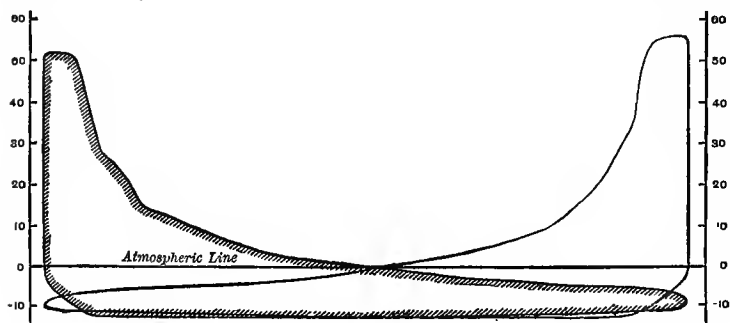
FIG. 90. Indicator diagrams from Corliss Engine.

drawn through the point of cut-off (assuming a reasonable percentage of wetness) it will be found that the actual curve of expansion at first lies below the adiabatic curve but afterwards rises above it
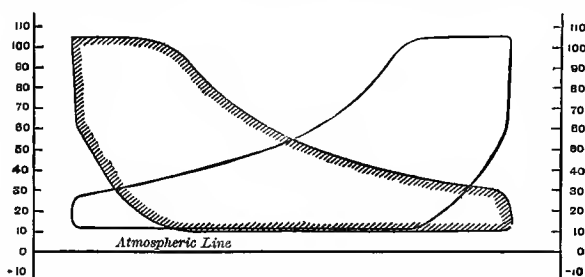
FIG. 91. Indicator diagrams from Compound Engine: High-pressure cylinder.
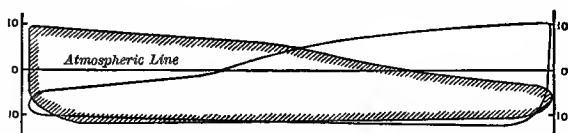
FIG. 92. Low-pressure cylinder.

in consequence of the re-evaporation of the condensed water. The irregularities in the expansion curve are an effect of the inertia of the indicator. Figs. 91 and 92 show a set of diagrams taken from a small compound engine using slide valves. Fig. 91 is the high-pressure pair of diagrams, and fig. 92 is the low-pressure pair. In the former the cut-off is a little sharper on one side than on the other, but the distribution is on the whole symmetrical and

good. The points of release and compression are well marked, showing that there is a free exhaust. Other examples of compound diagrams will be given later.

Indicator diagrams are often taken for the purpose of testing the setting of the valves, although the circumstances may be such that the engine is not doing external work. Fig. 93, for instance,
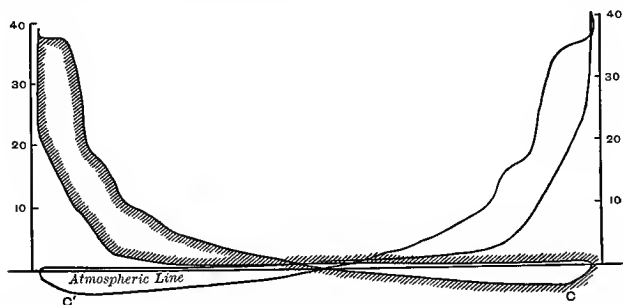
FIG. 93.   Indicator diagrams taken to examine the action of the Valves.

is a pair of diagrams taken from a Corliss engine when first erected by the makers without having the condenser in action and with no external load.   The exhaust is into the atmosphere, and as expansion has made the pressure in the cylinder less than that of the atmosphere the pressure rises at release (C, C' in the figure). This produces a loop on the diagram representing negative work: the excess of the positive over the negative portion represents the net amount of work which is done by the steam in overcoming the friction of the engine.

165.   Thermodynamic Tests.   Measurement of the Supply of Steam by means of the Feed.   When engine trials are to serve as tests of thermodynamic performance, either the heat supplied or the heat rejected has to be measured, for comparison with the work done.   Measurements of the supply of heat are most usual.   Sometimes, however, this method of testing may be impracticable and a test by means of the rejected heat may be easy. In any case a measurement of the rejected heat furnishes a valuable check on the accuracy of the other method, and the most satisfactory trials are made by measuring the heat supplied as well as the heat rejected; this allows a species of balance-sheet to be drawn up in which the heat given to the engine is more or less completely accounted for.

To determine the supply of heat the quantity of steam used by the engine is measured. Except when the engine has a surface-condenser, this has to be done by measuring the amount of feed-water that is required to keep the level of water in the boiler constant during a prolonged run. A somewhat long run is necessary in a trial of this kind because the level of water in the boiler cannot be read very exactly and the whole consumption of feed-water should be so great that any error due to this cause will become negligible. With an ordinary Cornish or Lancashire boiler a run of six or eight hours may be desirable and even essential if an accurate result is to be got: on the other hand if the engine is getting its steam from a small tubular boiler working hard under forced draught, or from a water-tube boiler, the evaporation may be so rapid that a single hour or even less will suffice. Care should be taken to have all the conditions of the experiment as closely as possible the same at the end as at the beginning of the trial: if for instance the feed-pump is working at the beginning it should be working at the end at the same rate, and the pressure in the boiler should be the same. In these circumstances the quantity of water in the boiler, for a given reading in the gauge-glass, may be taken to be the same at the end as at the beginning of the run, and the quantity of feed-water that has been supplied in the interval is therefore equal to the quantity of steam (dry or wet) that has left the boiler. If there has been no leakage and no blowing off at the safety-valve or otherwise, this quantity of steam has been delivered to the engine.

To measure the feed-water a very convenient plan is to have two tanks, one a small tank (A, fig. 94) set above the other (B) so that it may drain into B. The weight of water contained by A when full must be accurately known, and it should be furnished with a gauge-glass C to let fractions of the whole contents be read. B must have a float or a
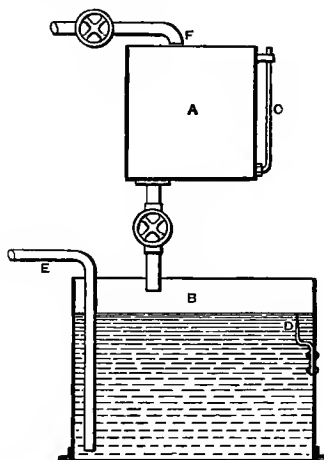
Fig. 94. Arrangement of tanks for measuring Feed-water.

point gauge D or other mark in it to indicate when the water
reaches some one standard level. The feed-pump draws water
from B by the pipe E; fresh water can be run into A at
pleasure from the supply pipe F, and there is a stop-cock be-
tween A and B. At the beginning of the test let it be seen
that the water in B is at the standard level and that the stop-
cock between the tanks is shut. Supply water during the test by
completely filling A as often as may be necessary, letting its con-
tents drain completely into B each time and noting the hour and
minute at which each fill of A is emptied into B. At the end of
the trial, after filling A for the last time, let just enough of its
contents pass into B to bring the level of water in B up to the
standard, and read on the gauge-glass of A the fraction which com-
pletes the whole supply. In a long run it is useful to check the
work by dividing the whole period into a series of parts, in each
of which the supply of feed-water is separately noted. The boiler
pressure, the speed, and all other conditions of working must of
course be kept as nearly uniform as may be throughout and
should all be noted at regular intervals during the trial. It is
useful to exhibit the log of the trial graphically by plotting all
the observed quantities on section-paper with time as the base.

The engine should work for some time under the prescribed
conditions as to speed, pressure, and load before the period of the
test begins, in order that it may get thoroughly warmed up and
that a uniform action may be established. During the trial
indicator diagrams are taken from time to time and the times are
noted. Where there is a mechanical counter the whole number of
revolutions made during the period of trial is found by reading
the counter at the beginning and at the end.

**166. Measurement of the Supply of Steam by means
of the Condensed Water**. In engines which are fitted with a
surface condenser the amount of steam passing through the engine
in a given time is readily determined by measuring or weighing
the condensed water discharged from the air-pump. An important
advantage of this method is that a satisfactory trial of the engine
can be made in much less time than is necessary when the steam
used is to be determined from the feed-water. Provided the engine
has been running long enough for the action to become uniform
before the trial begins, the air-pump discharge need not be collected

during more than ten or fifteen minutes, and thus a series of distinct trials under different conditions can be made in a single day.

167. Measurement of Jacket steam. If the engine has jackets the water condensed in them must be measured in addition to the water discharged by the air-pump; and even when the whole supply of steam is inferred from the feed it may be desirable to determine separately the amount that is used in the jackets. This is done by draining them into a tank or tanks so that the condensed water may be weighed. The water must escape freely enough to prevent its accumulating in the jacket and yet not so freely as to let steam blow through. This is readily secured by means of one or other of the two devices shown in fig. 95. A gauge-glass is inserted in the jacket drain, or is fitted to the drain as in the left-hand figure, with a throttle valve below it. By adjusting this valve the escape of the condensed water can be regulated so that the surface of the water will show itself in the glass at a constant height; the water is then passing off just as fast as it is condensed. To prevent evaporation of the discharged water the continuation of the drain may pass in the form of a bend or worm through a tank of cold water so that the jacket water may be cooled before it reaches the vessel in which it is to be measured[1].
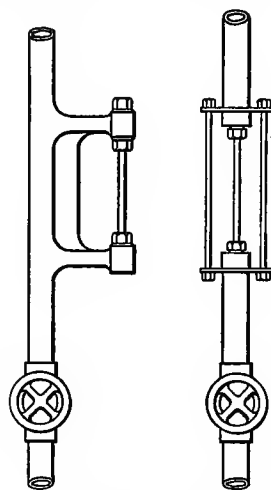
Fig. 95.   Gauge on Jacket drain.

168. Comparison of Feed-water with Discharged Water. In many trials the quantity of steam used by the engine is measured by both of the means that have been described, namely, by finding on the one hand how much feed is supplied to the boiler, and on the other hand how much is discharged by the air-pump and the jacket drains. In most cases some discrepancy

[1] This device was shown to the author by Mr Bryan Donkin, who had used it in some of his engine tests. It is also used in the Barrus Calorimeter described below (§ 169).

is observed: the feed-water may be as much as five per cent. more than the discharged water. This apparent loss of substance is due in part to water-vapour being discharged from the air-pump without being included in the measurement, but it is mainly due to leakage. Steam may escape at joints in small quantities showing little trace of its presence, and there is often some leakage within the boiler, as for instance at the ends of the tubes into the firebox in a boiler of the locomotive or the marine type. In most cases the measurement of the water discharged by an engine gives a fairer test of its performance than is given by measuring the feed. Should a serious discrepancy between the two quantities be found its causes are of course to be searched for and remedied.

169. Estimation of Heat supplied. Measurement of Dryness of the Steam. Knowing the amounts of steam supplied to the cylinder and jackets we may go on to calculate the amount of heat which the working substance takes up. In the absence of information as to the proportion of water in the steam as supplied to the engine the assumption that the steam is dry is the only safe one, though this may do some injustice to the engine by over-estimating the supply of heat. When the dryness q is known the heat supplied per lb. is

$$qL + h - h_0,$$

h_0 being the heat already present in the feed-water. Direct measurement of q is difficult mainly because it is difficult to secure that the steam used in any test of dryness is of the same quality as that which is delivered to the engine. One method is to blow steam from the boiler into a barrel or other vessel containing water, allowing the steam to be condensed, and noting the amounts by which (1) the temperature and (2) the weight of the contents have become increased after a suitable time. The former shows how much heat has been given up in condensing the steam that is blown in; the latter shows what the quantity of that steam is. Let the temperature rise from t_1 to t_2 while the weight increases from W_1 to W_2. Then q is found from the following equation, which expresses the fact that the water W_1 originally present gains an amount of heat equal to that which the entering steam loses:

$$(W_2 - W_1)(qL + h - h_2) = W_1(h_2 - h_1).$$

Here h_1 and h_2 refer to the temperatures t_1 and t_2, and h and L refer to the condition of the steam as supplied. This is subject to corrections (1) for loss of heat by radiation and (2) for the thermal capacity of the barrel itself. Accurate results are not easily got on account of the large error which is introduced by any inexactness in the measurement of the weight, and the measurement of dryness in this way is not recommended as a practical operation in engine testing.

Another form of calorimeter has been devised by Prof. Barrus, which also determines the wetness of steam by measuring the heat given out during its condensation, but the condensed steam is not allowed to mix with the condensing water. The steam to be examined flows into a pipe which passes through a vessel of water and so forms a surface-condenser. A steady circulation of water is kept up in the vessel, cold water flowing in and passing off after having been warmed by the condensation of steam within the pipe. The temperatures t_1 and t_2 of the water at the inlet and outlet respectively are noted. The water formed by condensation in the pipe is weighed after allowing it to escape through a stop-cock furnished with a gauge-glass as in fig. 95 (§ 167) and its temperature t_3 is noted. The quantity of cooling water which passes through the vessel in a given time has also to be weighed. Before an observation is made the apparatus is kept running long enough to let the temperatures all take steady values. Then, if W be the quantity of cooling water which passes while the quantity w is condensed,

$$w\,(qL + h - h_2) = W\,(t_2 - t_1),$$

subject to a small correction for radiation as before, the amount of which can be determined by noting the rate at which the calorimeter cools when it stands full of water at temperatures intermediate between t_1 and t_2.

170. Measurement of the Wetness of Steam by means of Wire-drawing. Professor Peabody[1] describes a simple apparatus of his own design for measuring the proportion of water in steam, which acts by throttling the wet steam until it becomes dry or slightly superheated (see § 82). In this apparatus, which is commonly called a wire-drawing calorimeter, but to which the name calorimeter is scarcely appropriate, the steam passes through

[1] Peabody, *Thermodynamics of the Steam-Engine.*

an adjustable throttle-valve A, fig. 96, into a chamber B lagged with non-conducting material, in which its temperature and its pressure are observed by thermometer C and gauge D. From this it escapes through another adjustable valve E to the atmosphere or to a condenser. The valves are adjusted until the steam in the chamber is seen to be slightly superheated, by comparing the observed temperature with the temperature which, in saturated steam, would correspond to the observed pressure. The amount of superheating, and the drop in pressure which has caused it are noted. In the process of throttling the total heat does not change. Before the throttling the total heat is $I_{w1} + q_1 L_1$, where q_1 is the dryness which is to be determined. Let t' be the observed temperature after throttling, and t_2 the temperature which saturated steam at the same pressure would have. The total heat after throttling is

$$I_{s2} + \kappa (t_1 - t_2).$$

Equating the total heats we have

$$q_1 = \frac{I_{s2} + \kappa (t' - t_2) - I_{w1}}{L_1}.$$

An appropriate value of κ should be taken from the table in § 65.

The throttling should be regulated so that the temperature t' is not greater than t_2 by more than a

FIG. 96. "Throttling Calorimeter" (Peabody).

trifling amount—just enough to ensure that the steam in the chamber is perfectly dry. It is only when the steam is nearly dry to begin with that it can be superheated or even completely dried by throttling. Even when the steam is no more than dried by throttling the limit of wetness beyond which the apparatus cannot be used is not high. With steam at 100 lbs. pressure, for instance, only 4 per cent. of moisture can be removed by throttling if the pressure in the chamber is as low as that of the atmosphere: but if a condenser is available the pressure in the chamber may be reduced far enough to deal with about 6 per cent.

In a modified form of the apparatus by Prof. Barrus[1] the

[1] *American Soc. Mech. Eng.* 1890.

vessel B becomes a mere tube, separated from the steam-pipe by a diaphragm with a small aperture through which the steam is wire-drawn. A separator is added, between the throttling apparatus and the steam-pipe, to allow as much as possible of the original moisture to be deposited before throttling takes place. With this addition it becomes possible to apply the method to steam that is originally very wet, for the separator leaves so little moisture still in the steam as to make throttling suffice to dry it completely. The water collected by the separator is to be added in reckoning the original wetness.

A porous plug forms a better means of throttling than the stop-valves and pin-hole orifices which have been used in instruments of this kind. The thermometer by which the temperature is taken after throttling should be placed as close as possible to the plug, for the steam quickly loses its superheat by conduction to the outside, and this cannot so well be done when a pin-hole orifice is used instead of a plug, since the kinetic energy of the stream through the orifice must be destroyed before it is allowed to come into contact with the thermometer. When all precautions are taken to secure that there shall be no losses of heat between the point of throttling and the thermometer, the method only serves, at the best, to show what was the wetness of the steam when it was on the point of entering the throttling plug. Whether its state then is the same as the average state of steam in the steam-pipe is another question. Unless special precautions be taken in connecting the apparatus to the steam-pipe, the sample of steam taken off for examination is liable to suffer some condensation before it reaches the plug and therefore to give an exaggerated impression of the wetness of the supply. On the other hand the steam supplied by a steam-pipe to an engine is liable to carry along with it a film of water on the inner surface of the pipe, and this wetness would not be represented in a sample taken off for the purpose of a test by means of a branch pipe opening into the interior of the steam-pipe in such a way as not to catch any of the water that is dragged along the surface by the current of steam. The consideration of these points will serve to show that little or no reliance can in general be placed on determinations of the general wetness of a steam supply by tests of a sample, whether the tests are made by the wire-drawing calorimeter or otherwise. In the writer's opinion none of these appliances are of much practical value in connection with engine trials. With a properly

designed boiler using clean feed-water the steam as it leaves the boiler is practically dry, and any wetness subsequently detected is to be ascribed to condensation before the measuring apparatus is reached.   When water is carried over in the steam a separator should be used, and under ordinary conditions the steam leaving the separator may be taken as practically dry.

171. Measurement of Heat rejected by an Engine. The rejected heat is measured by observing the quantity of the condensing water and the amount by which its temperature rises as it passes through the condenser. With small engines the quantity may be found by direct weighing or measuring in a large tank, or by the use of a pair of measuring tanks arranged so that one fills while the other empties. But in general the quantity of condensing water is too great to be easily treated in this way, and it has rather to be gauged as a stream, by observing the *head* under which it flows through an orifice of known size, or over a weir. This gauging is generally done after the water leaves the condenser, in which case, if the condenser is of the injection type, the quantity that is measured is the sum of the cooling water and the condensed steam, and the amount of the cooling water alone can be inferred by deducting from the whole a measured or estimated allowance to represent the feed.

When the stream to be gauged is large an open weir with a rectangular or V-shaped notch will be found most convenient: but for small streams a submerged circular orifice has the advantage that the accuracy of the result is less affected by any small error that may be made in measuring the head.   The stream to be gauged enters at *A* (fig. 97), a box containing baffle plates and perforated
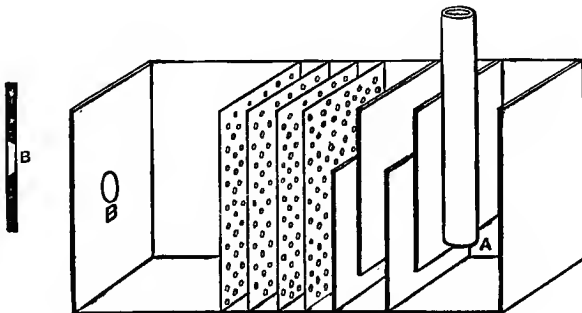
Fig. 97.   Weir-box with circular orifice.

diaphragms (sheets of perforated zinc or gauze will do well), which reduce it to stillness before it reaches the orifice B. This is a circular hole in a flat plate, and is bevelled to a sharp edge with the bevel outside. The head of water in the chamber close to the orifice is to be observed by means of a float and scale which are not shown in the diagram. If h is the head in feet, measured from the surface to the centre of the hole, and s is the area of the hole in square feet, the discharge Q in cubic feet per second is given by the formula

$$Q = cs \sqrt{2gh},$$

where c is a "coefficient of discharge," the ordinary value of which for a circular hole in a large flat surface is 0·62, when the head is sufficient to bring the surface of the water to a considerable height above the hole[1].

A compact and simple weir-box may easily be made by using a tall rectangular sheet-iron tank, a foot or so square in horizontal section and 3 or 4 feet high. The pipe bringing in the water is brought down inside the tank, in a corner, and opens close to the bottom. The lower part of the tank is filled by sheets of gauze serving as baffle plates. The water rises steadily above these, and escapes by one or more sharp-edged circular orifices cut in the side of the tank about a foot from the top.

In trials of marine engines where the weir-box method of measuring is impracticable, the feed-water may be measured with considerable accuracy by interposing a water-meter between the feed-pump and the boiler.

172. Example of an Engine Trial. To illustrate the reduction of the observations and the comparison of the heat supplied with the heat rejected and the work done we may take the data of a test by Mr Mair-Rumley[2]. The engine under trial

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[1] When an open rectangular notch with sharp edges in a vertical plate is used for a weir, h is to be measured from the bottom of the notch to the free level of the surface, at a distance far enough back to give practically still water ; then

$$Q = 3 \cdot 33 \, (b - 0 \cdot 2h) \, h^{\frac{3}{2}},$$

where b is the breadth of the notch.

With a triangular notch cut so that the breadth is twice the depth $Q = 2 \cdot 54 h^{\frac{5}{2}}$, where h is the depth of the bottom of the notch below the still-water surface level. For the justification of these formulas reference must be made to books on hydraulics or to papers by James Thomson, *Rep. Brit. Assoc.* 1858, 1861, and 1876, p. 243.

[2] See two papers (J. G. Mair), *Min. Proc. Inst. C. E.*, vols LXX. and LXXIX.

was a compound beam engine with steam-jackets and with a jet-condenser. The cylinders were 21 and 36 inches in diameter, and the stroke of each piston was $5\frac{1}{2}$ feet. The feed-water was measured during a period of 6 hours and the air-pump discharge was gauged by means of a weir. The following are the data of the trial :—

Pressure in boiler, 76 lbs. per sq. in., absolute (for which $L = 503.7$ and $h = 155.0$).

Duration of trial, 6 hours.

Revolutions, 8632, or 24.0 per min.

Indicated horse-power, 127.4.

Feed-water, 12032 lbs.

Air-pump discharge, 1226 lbs. per min.

Water drained from jackets, 1605 lbs.

Dryness of steam as supplied, 0.96.

Temperature of feed, $t_0 = 15°$ C.

      „   „ injection, $t_1 = 10°$ C.

      „   „ air-pump discharge, $t_2 = 23°$ C.

These give the following results :—

Total feed per revolution   = 1.394 lbs.

Jacket feed per revolution   = 0.186 lb.

Cylinder feed per revolution = 1.208 lbs.

Injection water per revolution $= \frac{1226}{24} - 1.208 = 49.9$ lbs.

Heat taken in by the working substance per revolution

$$= 1.394 \, (qL + h - h_0)$$

$$= 1.394 \, (0.96 \times 503.7 + 155.0 - 15.0 = 1.394 \times 623.5$$

$$= 869.2 \text{ thermal units.}$$

Heat converted into work per revolution

$$= \frac{127.4 \times 23.57^*}{24} = 125.1 \text{ thermal units.}$$

The whole heat rejected per revolution should therefore be 744.1 thermal units.

That part of the working substance which is cylinder feed

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* 23.57 is the thermal equivalent of 1 horse-power acting for 1 minute, namely, $\frac{33000}{1400}$ thermal units.

rejects heat first and chiefly to the injection water, and secondly by becoming itself cooled from t_2 the temperature of the air-pump discharge to t_0 the temperature at which it returns to the boiler. The heat it rejects per revolution in these two ways is therefore

$$49.9 \, (t_2 - t_1) + 1.208 \, (t_2 - t_0),$$

or $49.9 \times 13 + 1.208 \times 8 = 658.4$ thermal units.

That part of the substance which is jacket steam rejects heat by becoming cooled from the temperature at which it is condensed in the jacket to the temperature at which it is returned to the boiler. In the present case the jackets drained into the hot-well, and the temperature of the water draining from them therefore fell to 59°, the temperature of the feed. The heat rejected in this way per revolution was

$0.186 \, (h - h_0) = 0.186 \, (155 - 15) = 26.0$ thermal units.

Adding these we have 684.4 units of rejected heat. A balance of 59.7 units remains to be accounted for. It is made up partly of heat carried away by the air and vapour of the air-pump discharge, partly of losses through radiation from the engine and pipes, and partly of heat lost in steam which escapes by leakage. In the example cited the loss by radiation was estimated to amount to 25 units[1]; allowing for this the discrepancy between the two sides of the account is reduced to 34.7 units or 4 per cent. of the whole supply.

The consumption of steam per indicated horse-power-hour calculated from the whole amount of the feed is

$$\frac{12032}{127.4 \times 6} \text{ or } 15.7 \text{ lbs.}$$

This makes the indicated work done per lb. of steam equivalent to

$$\frac{33000 \times 60}{1400 \times 15.7} \text{ or } 90.1 \text{ thermal units.}$$

In considering the efficiency of a cycle as a whole we should in strictness deduct from this the net amount of work which has to be expended in returning the condensed steam from the condenser to the boiler, or say $0.017 \times 76 \times 144$ foot-lbs. per lb. As this is the equivalent of only 0.13 thermal unit per lb. the correction is

[1] The loss by radiation is approximately estimated by letting the engine stand still with the jackets and steam-chest full of steam and noting the amount that is condensed in a given time.

unimportant. Since the heat taken in per lb. is 623·5 units the efficiency of the cycle is 0·144*.

173.   Wetness of the steam during expansion. In § 117 it was explained how to find the proportion of water present in the cylinder at any stage of the expansion and to represent the results of this calculation graphically by means of a "saturation curve" upon the indicator diagram—namely, a curve which represents the volume which the steam in the cylinder should fill at any pressure if it were dry throughout. To draw this curve for either side of the piston we should in strictness know how the whole amount of the cylinder feed is shared by the two ends of the cylinder— a matter which the test does not determine. But in general the action in the two ends is so nearly symmetrical that results which are practically correct may be obtained by combining the indicator diagrams for the two into a mean diagram, taking for clearance the mean of the two actual clearances, and taking half the cylinder feed per revolution as the quantity of steam that enters the cylinder per stroke. The diagram shown in § 117, is in fact a combination diagram drawn in this way. To determine the wetness of the steam during expansion is an important part of an engine test, and it is useful to exhibit the results, so far as this particular is concerned, by showing the saturation curve in its relation to the actual curve of pressure and volume. In dealing with compound engines a saturation curve may be drawn separately for each cylinder, or the diagrams for the several cylinders may be combined into one by means of a device which will be described in the next chapter. The variations of wetness which take place during expansion may also be conveniently exhibited by drawing a diagram showing the weight of dry steam present at each stage in the stroke.

This process of estimating the water present during expansion by comparing the saturation volume with the volume actually filled by the working substance depends on the assumption that the whole quantity of substance does not change from the time that cut-off is complete until release begins. Any leakage of steam, in or out, through the valve or past the piston will invalidate the calculation.

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* For further illustrations of engine trials and the reduction of results reference should be made to the excellent examples contained in several of Mr M. Longridge's *Reports* as Engineer of the Engine, Boiler, and Employers' Liability Association from 1880.

Having determined what proportion of the working substance is steam and what is water throughout the expansion we may go on to calculate how much heat is taken from or given to the walls of the cylinder and piston during any stage of its action. This analysis of the transfers of heat, introduced by Hirn and developed by his pupils and followers, has been pursued at great length in some engine tests[1]. It proceeds on the basis that if any two points in the action be taken and the internal energy calculated for each, the heat taken in or given out between the two is measured by the change of internal energy together with the work done by or spent upon the substance between the two points, which is determined by reference to the pressure-volume curve of the indicator diagram. The data are not in general very exact, and the analysis is not particularly instructive. As regards the heat stored and restored during expansion, this is readily found from the entropy-temperature diagram in a manner which has already been sufficiently explained.

174. Tests of mechanical efficiency. Measurement of Brake Horse-power. In tests of mechanical efficiency the engine is commonly set to work against some form of friction brake arranged to serve as an absorption dynamometer. For engines of small or moderately small power no form is so simple or so easy of application as a rope or band brake of the type shown in fig. 98. Two, three, or more parallel turns of rope with a few wood blocks to hold them apart (the number of ropes depending on the quantity of power that is to be absorbed) are made to clasp the fly-wheel in the manner sketched; the slack end is attached to a spring balance and the other end is loaded with weights, either directly, or through a lever if the amount of load is inconveniently great. A little grease applied to the surface of the
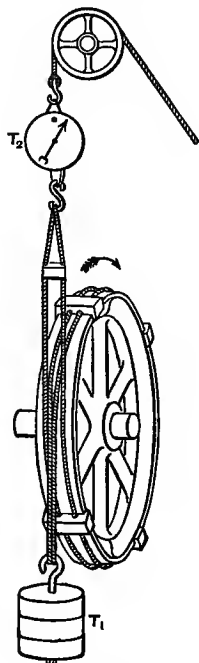
FIG. 98.

[1] See Dwelshauvers-Dery, "*Étude Calorimétrique de la Machine à Vapeur*." Also Mr Mair-Rumley's papers already cited.

19—2

metal makes the brake work quietly and steadily. The wheel may be kept cool by water; a wheel with internal flanges on the rim, forming an internal channel to which cold water may be supplied, is convenient. The resistance is adjusted by varying the amount of the weight T_1. A platform or stop fixed a little way below this weight allows the brake to be applied and removed by pulling or slacking away the rope by which the spring balance is suspended; by pulling the rope which is shown at the top of the figure the weights are lifted off their platform and the brake comes into action. When the brake is in action the pull T_2 indicated by the spring balance is noted from time to time. The effective resistance is $T_1 - T_2$, and the work done against the brake per revolution is $2\pi r (T_1 - T_2)$ where r is the radius measured from the axis of rotation to the middle of the rope's thickness. Hence the brake horse-power

$$\text{B.H.P.} = \frac{2\pi n r (T_1 - T_2)}{33,000}.$$

The mechanical efficiency is the fraction $\frac{\text{B.H.P.}}{\text{I.H.P.}}$; or, without reducing to horse-power, it is the ratio of the work done on the brake per revolution to the work done by the steam per revolution, namely,

$$\frac{2\pi r (T_1 - T_2)}{l (p_m a + p_m' a')},$$

in the notation of § 163.

A flexible band such as may be made by using a few strips of cotton listing has the advantage as compared with rope of working smoothly and silently without any lubrication and is to be preferred to rope for small engines. When only two or three horse-power have to be measured a single strip of listing will be found to make an excellent brake.

In dealing with large powers the most effective and accurate absorption dynamometer is one in which the work of the engine is spent in churning water by turning a species of turbine wheel in a casing through which water is continuously passed. The casing is held from turning by applying weights to a lever arm, and this measures the moment exerted by the engine-shaft, on which the turbine wheel is fixed. Prof. Reynolds designed and used in his experiments a very perfect brake of this kind, a full description of which will be found in the *Philosophical Transactions of the*

Royal Society for 1897. He applied this brake not only in engine trials but also, as was mentioned in Chapter II., to measure the mechanical equivalent of heat by observing, along with the work done, the quantity of water which in passing through the brake had its temperature raised from 0° C. to 100° C.

In testing engines employed for pumping, the useful output may be determined by measuring, in the one case, the volume of water delivered and the pressure against which it is delivered, the product of these quantities giving the effective work got from the pump. This however is less than the brake horse-power by an amount which depends on the losses that occur in the pump itself. Similarly, in testing engines which generate electricity, the useful output of the dynamo is readily measured by electrical appliances, but here again the quantity measured is less than the brake horse-power by an amount which represents the loss in converting mechanical into electrical energy. In either case the efficiency determined, when the effective output is divided by the indicated power of the engine, represents a joint efficiency of engine and pump or engine and dynamo as the case may be, and is necessarily less than the mechanical efficiency of the engine alone.

175. Torsion Dynamometer for measuring the mechanical output of an engine. Various forms of dynamometer have been devised for measuring the work done by an engine by finding the torque or twisting moment transmitted through the shaft in the working of the engine. In particular these have been applied in marine engines to find the horse-power employed in driving the propeller. Such measurements are specially desirable in marine steam turbines, where no means of "indicating" can be used. A portion of the length of the shaft is taken, and by static experiments or by calculation from the known dimensions and known modulus of rigidity of the material a relation is determined beforehand between the torque and the angle of twist. Then when the engine is running the angle of twist is observed, the corresponding torque is inferred, and the work done per revolution is calculated as $2\pi M$, where M is the torque. Several devices have been employed for measuring the angle of twist, by finding the relative angular displacement of two wheels or collars which are clamped to the shaft at a certain distance apart. In the Denny and Johnson torsion meter an electrical appliance is used for measuring the

relative displacement of the collars, and a length between them of
15 or 20 feet is the least that will give fairly satisfactory results.
With such torsional strains as occur in propeller-shafting the
displacement under full load is only about one two-thousandth part
of the distance between the collars.   Refined means of measure-
ment are therefore necessary if a reasonable percentage of accu-
racy is to be attained on a short length of shaft.   In Hopkinson
and Thring's torsion meter the relative displacement of the two
collars is measured by the tilting of a mirror set between them,
and its tilting is observed by using a scale and lamp (or telescope)
at the side.   The arrangement is illustrated in figs. 99 and 100.
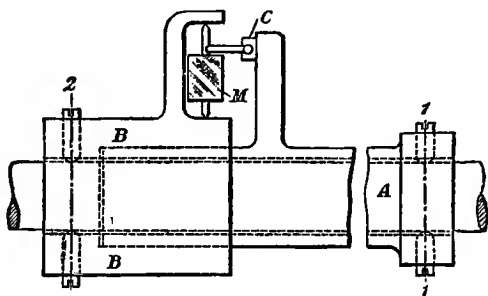
FIG. 99.   Hopkinson-Thring Torsion Meter.

The collars A and BB are clamped to the shaft by screws 11 and
22, and have sleeves which bring them together: the mirror M is
pivoted on B and carries a short arm which abuts against a pro-
jection C on the sleeve of the collar A.   Light from the lamp E is
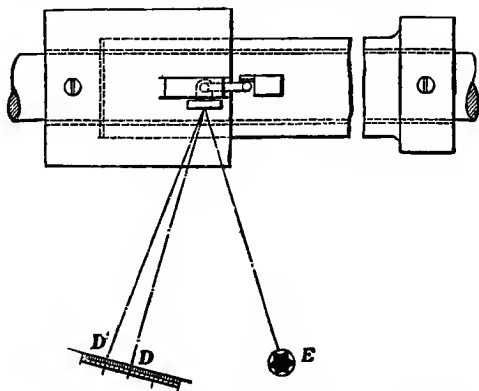
FIG. 100.   Hopkinson-Thring Torsion Meter (plan).

reflected on the scale D and appears as a line on the scale once in each revolution of the shaft. Good results are obtained with only a foot of distance between the collars. It should be noticed that what is measured is the torque at the instant, in each revolution, when the reflected beam passes the screen. If the torque varies throughout the revolution, as it will do in a shaft driven by reciprocating engines, a mean should be found by using three or more mirrors spaced regularly round the shaft[1].

176. Trials of an engine under various amounts of load. Although some engines are required to work always under the same or nearly the same conditions as to load, more commonly the load is liable to variation, and it may be as important to examine the performance under light loads as to make trials at full power. In electric light stations, for example, much of the work is done with a comparatively light load on the engine and the efficiency under these conditions is a matter of great practical interest. To be complete a trial should include a series of tests made at various grades of output from full power down to the extreme when the engine merely drives itself without doing external work. The papers by Mr Willans which were referred to in Chapter VII. contain many examples of trials which are complete in this sense. Such results may be represented graphically by drawing a curve in which the ordinates are the number of lbs. of steam consumed per horse-power-hour, with the rate of output in horse-power for abscissæ.

Two curves of this kind are shown in fig. 101, relating to two series of tests by Willans of one of his compound high-speed single-acting engines, using a condenser[2]. In one set of trials the ratio of expansion was 4·8 and the points through which the curve is drawn were determined by testing the consumption under various values of the initial steam pressure, ranging from 135 lbs. per sq. inch (absolute) down to 43 lbs. The other curve refers to a similar series of trials in which the ratio of expansion was 10.

Another useful way of showing the performance at all powers is to plot the whole quantity of steam consumed per hour in relation to the horse-power. Curves of this kind were first used

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[1] See *Engineering*, June 14, 1907 : *Trans. Inst. Nav. Arch.*, March, 1910.

[2] *Min. Proc. Inst. C. E.*, vol. cxiv. 1893.

by Willans and are often called Willans Lines : examples of them are given in fig. 102 relating to the same two sets of trials as fig. 101. In each set of trials the adjustment of the power was
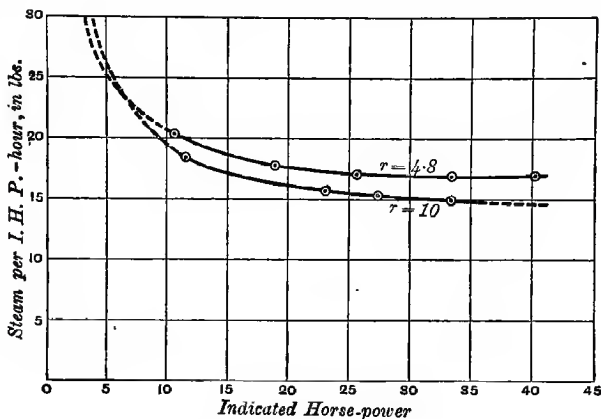
Fig. 101.

accomplished by varying the initial pressure of the steam, the cut-off remaining constant throughout the set, and the speed of the engine also remained constant. Under these conditions Willans found that the curve of total steam consumption in relation to power (fig. 102) was sensibly a straight line. With
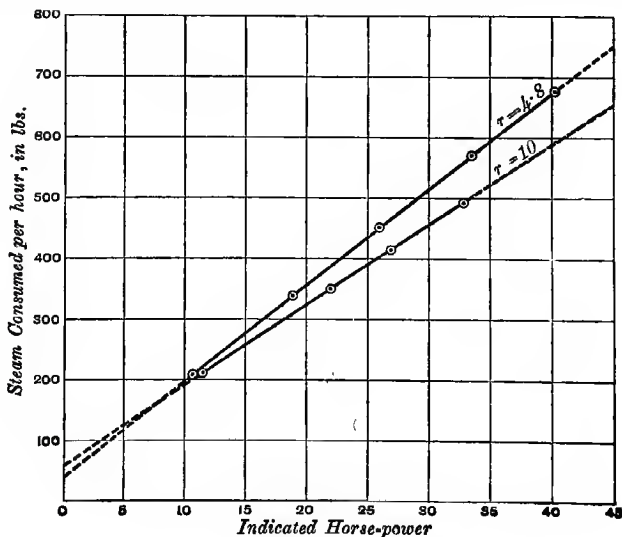
Fig. 102.

variable cut-off and constant pressure, on the other hand, the Willans line as found in his trials was slightly curved, having a steeper gradient at high powers than at low powers.

The same types of diagram are useful in representing the consumption of steam in relation to brake horse-power, pump horse-power, electrical horse-power, etc. They exhibit clearly under what condition the maximum of efficiency will be reached, and also what the performance will be under the less favourable conditions that may have to be submitted to in practice.

When the Willans line is a straight line the whole consumption of steam at any load may be regarded as made up of two parts—the constant unproductive consumption that takes place without doing work in the cylinder and a further consumption that is simply proportional to the indicated power. The whole consumption is equal to

$$a\,(i + b),$$

where i is the number of horse-power and a is the ratio at which steam is taken per horse-power after the unproductive supply ab has been furnished.

The same remark holds good in relation to a Willans line
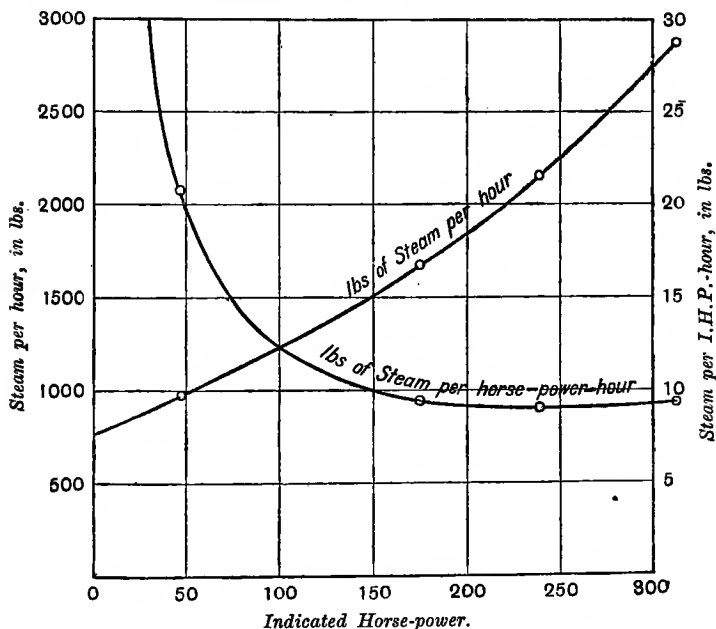
FIG. 103. Trials of a Schmidt Engine.

drawn with the external or brake horse-power as the base; ab then represents the steam that is used in making the engine drive itself; and b is what may be called the "idle-work," a quantity which is somewhat greater than the indicated work done in overcoming engine friction.

A comparison of the Willans lines relating to indicated and brake power respectively serves to show how far the work spent on engine friction remains constant at high powers and at low. In general it may be expected that this quantity will be greater at high powers since the forces at the joints of the mechanism become on the whole increased.

Two further examples of Willans lines are given in figs. 103 and 104, for tests made by the writer. Fig. 103 relates to a test of a Schmidt engine of 300 horse-power, using highly superheated steam. The following are particulars of the trials.

Indicated horse-power	312	239	175	47
Steam per hour, lbs.	2925	2150	1670	975
Steam per i.h.p.-hour, lbs.	9·37	9·00	9·54	20·7
Boiler pressure (by gauge) in lbs. per sq. inch	141	141	142	140
Temperature of steam on leaving superheater, ° C.	425	420	430	420
Vacuum, inches	27½	28	28	29

In this case the line showing the total quantity of steam used per hour is distinctly curved, the slope becoming less steep as the power is reduced. A second line is drawn to show the steam used per horse-power-hour. It passes a minimum at about 250 horse-power, the consumption then being only 9 lbs. per indicated horse-power-hour. This result has been already referred to in Chap. VII., as an example of the remarkably small steam consumption attainable through the use of superheating, even in small engines.

The other example is for a test made in 1900 of a Parsons steam turbine employed to drive a dynamo—an early large Parsons turbine. The effective output was measured electrically, and the steam consumption is accordingly given with reference to the load in kilowatts. Throughout all the trials, including the

no-load trial, the turbine was driving an auxiliary dynamo or exciter, but the output of the exciter is not included as part of the effective load. The steam was slightly superheated, as will be seen from the following particulars of the trials.

Effective load in kilowatts	1435	928	507	0
Steam per hour, lbs.	26200	17890	11570	3460
Steam per kilowatt hour	18·25	19·28	22·84	—
Pressure at stop-valve by gauge (lbs. per sq. inch)	152	158	155	145
Temperature of steam at stop-valve, ° C.	234	236	236	235
Vacuum, inches (Barometer 29·4)	27·0	27·8	28·3	28·4

A surface-condenser was used, and the steam consumption was found by delivering the condensed water into a pair of calibrated tanks. The curves in fig. 104, plotted to a kilowatt base, show respectively the total steam per hour, and the steam per kilowatt hour. Under the highest load the consumption is 18·25 lbs. per
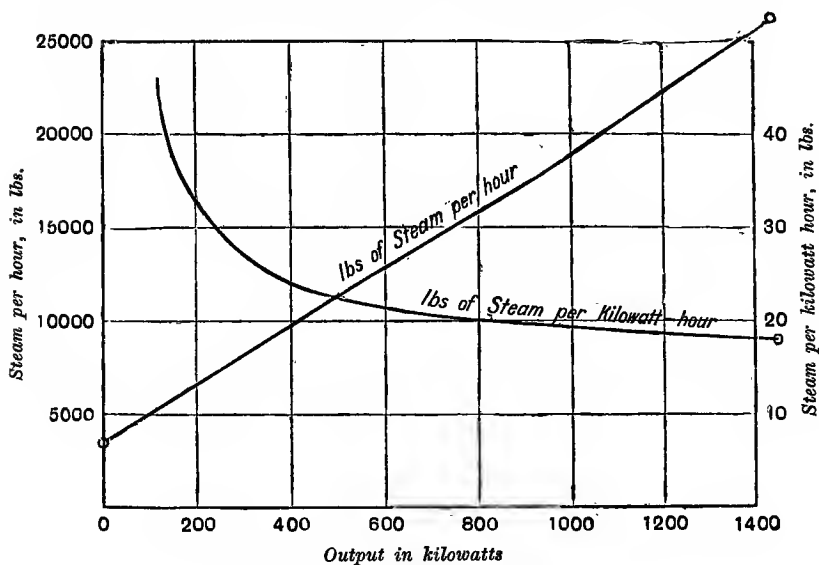
Fɪɢ. 104. Trials of a Parsons Turbine.

kilowatt hour. A piston-engine driving a dynamo usually converts about 84 per cent. of its indicated power into effective electrical power. Such an engine would therefore have to consume only $18\cdot25 \times \frac{746}{1000} \times \frac{84}{100}$ or 11·4 lbs. of steam per indicated horse-power-hour to put its performance on the same level with that of the turbine in this test, assuming it to be supplied with steam of the same quality. Good as this result is it has been much surpassed in more recent trials made with larger turbines.

177. Results of trials. It remains to cite a few results of trials made with good engines under conditions likely to give a highly efficient performance, by way of supplementing figures which have already been quoted.

In trials of a triple expansion horizontal mill engine built by Sulzer of Winterthur[1] developing 1850 horse-power, the mean steam consumption was 11·46 lbs. per horse-power-hour, the pressure of supply being 148 lbs. per square inch, without super-heating. The coal consumed per horse-power-hour was 1·30 lbs.

A triple expansion Leavitt pumping engine at the Chestnut Hill Station, Boston, Mass., developing 575 horse-power, supplied with saturated steam at 175 lbs. per square inch by gauge, con-sumed 11·2 lbs. of steam and 1·15 lbs. of coal per horse-power-hour.

In both of these cases each cylinder had four separately actuated valves, an arrangement which makes for efficiency. In the Leavitt engine the condenser pressure was 1·6 lbs. per square inch. By reference to the Mollier diagram the heat drop from 190 lbs. of the absolute pressure of supply to 1·6 lbs. (13·36 to 0·11 kg. per sq. cm.) is 174 units. The thermal equivalent of one horse-power-hour is 1414 units, and hence each lb. of steam does an amount of indicated work equivalent to $\frac{1414}{11\cdot2}$ or 126 units, which is $72\frac{1}{2}$ per cent. of the heat drop.

In the following table the results of a number of trials are briefly stated, for single cylinder, two-stage compound, and three-stage compound engines. In all these cases there was no super-heating. The best results are obtained in horizontal engines with separate admission and exaust valves, with jackets containing steam at boiler pressure on the ends and barrels of the cylinders, and with tubular reheaters in the intermediate receivers.

If the thermodynamic efficiency of these trials be worked out

[1] *Zeitschrift des Vereines deutscher Ingenieure,* vol. XL.

it will be found to range from 0·11 in the single cylinder engine tested by Mr Mair-Rumley, to about 0·21 in the most favourable case.

Results of Trials of Condensing Engines using Saturated Steam.

Type of Engine	Absolute boiler pressure lbs. per sq. inch	Steam used per hour per I.H.P. lbs.
Single-cyl. beam pumping engine (Mair-Rumley)	59	21·3
Single-cyl. Corliss engine (Hill)	111	19·4
Single-cyl. Sulzer engine (Linde)	105	19·0
Two-cyl. compound beam pumping engine (Mair-Rumley)	76	14·8
Two-cyl. compound pumping engine (Leavitt). (Mean of two trials.)	114	14·05
Two-cyl. compound high-speed single-acting engine (Willans)	180 (about)	14·26
Two-cyl. compond slow-running engine with jackets (M. Longridge)[1]	148	12·82
Two-cyl. compound Leavitt pumping engine of 643 horse-power, with jackets, and tubular reheater in the intermediate receiver	155	12·2
Triple marine engine of S.S. "Iona" (Kennedy[2])	180	13·35
Triple experimental engine (O. Reynolds[3])	207	12·68
Triple high-speed single-acting engine (Willans)	190 (about)	12·74
Triple Sulzer engine of 200 horse-power (Schröter)	171	12·2
Triple Sulzer engine of 1850 horse-power	163	11·46
Triple Leavitt pumping engine with jackets and tubular reheaters	190	11·2

[1] Chief Engineer's Report of the Engine, Boiler, and Employers' Liability Insurance Co., 1895.

[2] See Reports of Marine Engine Trials by a Committee of the Institution of Mechanical Engineers, from 1889. The trials of the "Iona" were reported in April 1891.

[3] Osborne Reynolds, *Min. Proc. Inst. C. E.* Dec. 1889.

178. Mechanical Efficiency of the Engine. By this phrase is meant the ratio of the brake horse-power to the indicated horse-power of the engine. Comparisons of the two powers show that in favourable cases the mechanical efficiency is about 0·85; in other words, some 15 per cent. of the indicated work is ineffective, being spent on friction within the engine. Occasionally the mechanical efficiency may approach 0·9; in general, however, it is a good deal less. When a steam-engine is directly employed to drive a dynamo, the comparison is often made between the electrical output and the indicated work : in that case the efficiency of the dynamo is of course involved as well as that of the engine as a mechanism. Similarly in dealing with pumping engines, the comparison is usually made between the indicated work and the work usefully applied by the pump—this latter being determined by the volume and pressure of the fluid delivered. Here again the mechanical efficiency of the pump and of the engine are both involved.

In trials of good pumping engines the output of the pump represents 84 to 85 per cent. of the indicated work done by the steam. In Professor Reynolds's tests of a triple-expansion laboratory engine, where the engine was loaded by means of brakes, the brake horse-power was 82 per cent. of the indicated power in the most favourable instances.

With a given engine running at a given speed the work expended in driving the engine itself is usually a nearly constant quantity, whether much or little effective work is being taken off. Hence the mechanical efficiency is reduced when the load on the engine is lightened.

179. Curve of Expansion to be assumed in estimating the probable indicated horse-power of steam-engines.

Largely as the exchanges of heat between the working substance and the cylinder affect the consumption of steam, it will be seen by examining indicator diagrams that their influence on the form of the expansion curve is but slight. In practical cases the curve is never very different from a rectangular hyperbola. The simple supposition that the pressure during expansion varies inversely with the volume will answer sufficiently well in designing engines, for the purpose of estimating the probable power to be exerted by an engine of given size, when the speed, the initial pressure, the back pressure and the ratio of expansion are assigned.

If there were no clearance, if the full pressure of supply, p_1, were maintained during the admission, if the cut-off and release were perfectly sharp, if the expansion continued to the very end of the stroke, and if there were a uniform back pressure p_b, without compression, then the assumption that the curve of expansion may be treated as a common hyperbola would make the mean effective pressure equal to

$$\frac{p_1\,(1 + \log_e r)}{r} - p_b,$$

where r is the ratio of expansion, namely, the ratio which the whole volume of the stroke bears to the volume that is swept through up to the point of cut-off. To show this, we may express the area of the indicator diagram as

$$p_1 v_1 + \int_{v_1}^{v_2} p\,dv - p_b v_2$$

where v_1 and v_2 are the volumes at cut-off and release respectively, so that $r = \dfrac{v_2}{v_1}$.

Since by assumption pv at any point in the expansion $= p_1 v_1$, the area is

$$p_1 v_1 \left(1 + \int_{v_1}^{v_2} \frac{dv}{v}\right) - p_b v_2$$
$$= p_1 v_1\,(1 + \log_e r) - p_b v_2.$$

The mean effective pressure is found by dividing the area by v_2, whence the formula given above is obtained. The same formula would be applicable to compound expansion, r being interpreted as the final ratio, if the further condition were satisfied that there should be no loss of pressure during the transfer of the steam from one cylinder to the next.

In practice, of course, these conditions are not fulfilled, and the general result is to make the actual mean effective pressure p_m less than the pressure calculated in this manner, in a proportion which is sometimes stated by the use of a coefficient e, thus:

$$p_m = e\left\{\frac{p_1\,(1 + \log_e r)}{r} - p_b\right\}.$$

The diagram factor, as Professor Unwin has called e, is a number less than unity, which may be estimated as a matter of experience from the results given by other engines of like types, working under more or less similar conditions.

CHAPTER X.

COMPOUND EXPANSION.

180. Woolf Engines. An engine is said to be " compound "
when the expansion of steam is begun in one cylinder and con-
tinued in another. The steam may be made to pass directly
from one cylinder to the next, or it may pass from the first
cylinder into an intermediate chamber, called a "receiver," from
which the second cylinder draws its supply. An advantage of the
latter plan is that it does not require the reception of steam by
the second cylinder to be simultaneous with the rejection of steam
by the first. This allows the cranks to be set at any angle : it
also allows the distribution of the expansion between the two
cylinders to be readily adjusted. Compound engines are now
rarely used with immediate transfer of steam from one cylinder
to the other. The connecting passages between the cylinders
have a certain capacity and form a receiver even in cases where
no special chamber is added for the purpose.

The original form of compound engine invented by Hornblower
and revived by Woolf had no receiver. Steam passed directly
from the high- to the low-pressure cylinder, entering one as fast
as it was exhausted from the other. This arrangement is possible
only when the high- and low-pressure pistons begin and end their
strokes together, that is to say, when their movements either
coincide in phase or differ by half a revolution. This was the
case in many of the early compound engines, which were beam
engines with a high-pressure cylinder and a low-pressure cylinder
standing side by side, the pistons rising and falling together.
Engines whose high- and low-pressure cylinders are placed side
by side, and act either on the same crank or on cranks set at 180°
apart, may also discharge steam directly from one to the other
cylinder. This kind of working is also possible in what are called

tandem engines, which have the high- and low-pressure cylinders in one line, with one piston-rod common to both pistons. The name "Woolf engine" is now sometimes applied to those compound engines which discharge steam directly from the high- to the low-pressure cylinder without the use of an intermediate receiver.

181. Receiver engine. An intermediate receiver becomes necessary when the phases of the pistons in a compound engine do not agree. With two cranks at right angles, for example, a portion of the discharge from the high-pressure cylinder occurs at a time when the low-pressure cylinder cannot properly receive steam. The receiver is in some cases an independent vessel connected to the cylinders by steam-pipes; very often, however, a sufficient amount of receiver volume is afforded by the steam-chests containing the valves, together with the steam-pipe which connects the cylinders. In marine engines for example the receiver consists simply of the steam-chests and connecting pipe, the capacity which these passages necessarily have in order that they may let the steam pass readily being sufficient for the amount of storage that is required. The receiver, when it is a distinct vessel, is frequently jacketed.

Even when the phases of the pistons do agree, as in tandem engines, it is generally preferred to use a receiver rather than transfer the steam direct from one cylinder to the other. This is partly because of the necessary capacity of the connecting passages and partly because the use of a receiver makes it easy to adjust the proportion of work done by each cylinder.

When a receiver is used the low-pressure cylinder, as well as the high-pressure cylinder, has a definite point of cut-off. During part of the stroke the receiver is storing steam, and the steam which has been admitted to the low-pressure cylinder is expanding independently there. By varying the position in the low-pressure stroke at which cut-off takes place the distribution of work between the two cylinders may be adjusted at will.

As a rule matters are regulated so as to make both cylinders do equal, or nearly equal, amounts of work. If the pistons act on separate cranks this has the effect of giving the same value to the mean turning moment for both cranks. The adjustment of the work will be discussed in § 183, below.

E.   20

Another adjustment which is sometimes aimed at is to make the range of temperature equal in both. In general, when the division of work is equal, the parts into which the whole temperature range is divided are nearly equal also.

182. Drop in the Receiver. Compound diagrams. Wherever a receiver is used, care must be taken that there is no large amount of unresisted expansion into it; in other words, the pressure in the receiver should not be greatly below that in the high-pressure cylinder at the moment of release. Any drop in the steam-pressure as the steam passes from the high-pressure cylinder to the receiver will show itself in an indicator diagram by a sudden fall at the end of the high-pressure expansion. This "drop" is, from the thermodynamic point of view, irreversible, and therefore wasteful. Practically some small amount of drop is desirable for the same reasons which make a rather incomplete expansion preferable to complete expansion in the working of a single cylinder. Drop can be avoided or reduced to any desired extent, as we shall presently see, by selecting a proper point of cut-off in the low-pressure cylinder. When there is no drop the expansion that occurs in a compound engine has precisely the same effect in doing work as the same amount of expansion would have in a simple engine, provided the law of expansion be the same in both and the waste of energy which occurs by the friction of ports and passages in the transfer of steam from one to the other cylinder be negligible. The work done in either case depends merely on the relation of pressure to volume throughout the process : and so long as that relation is unchanged it is a matter of indifference whether the expansion be performed in one vessel or in more than one. It has, however, been explained in Chapter VII. that in practice a compound engine has a thermodynamic advantage over a simple engine using the same pressure and the same expansion, inasmuch as it reduces the exchange of heat between the working substance and the cylinder walls and so makes the process of expansion more nearly adiabatic. The compound engine has also a mechanical advantage which will be referred to in § 187, below.

The ultimate ratio of expansion in any compound engine is the ratio of the volume of the low-pressure cylinder to the volume of steam present in the high-pressure cylinder at the point of cut-off.

Fig. 105 illustrates the combined action of the two cylinders in a hypothetical compound engine of the Woolf type, in which for simplicity the effect of clearance is neglected and also the loss of pressure which the steam undergoes in transfer from one to the
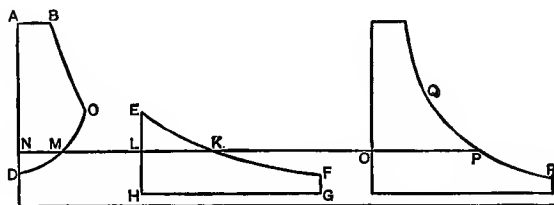
Fig. 105. Compound Diagrams: Woolf type.

other cylinder. *ABCD* is the indicator diagram of the high-pressure cylinder. The exhaust line *CD* shows a falling pressure in consequence of the increase of volume which the steam is then undergoing through the advance of the low-pressure piston. *EFGH* is the diagram of the low-pressure cylinder, and is drawn alongside of the other for convenience in the construction which follows. It has no point of cut-off; its admission line is the continuous curve of expansion *EF*, at each point of which the pressure is the same as at the corresponding point in the high-pressure exhaust line *CD*. At any point *K*, the actual volume of the steam is $KL + MN$. By drawing *OP* equal to $KL + MN$, so that *OP* represents the whole volume, and repeating the same construction at other points of the diagram, we may set out the curve *QPR*, the upper part of which is identical with *BC*, and so complete a single diagram which exhibits the equivalent expansion in a single cylinder. The area of the figure so drawn is equal to the sum of the areas of the high-pressure and low-pressure diagrams.

In a tandem compound engine of the receiver type the diagrams resemble those shown in fig. 106. During *CD* (which corresponds to *FG*) expansion is taking place into the large or low-pressure cylinder. *D* and *G* mark the point of cut-off in the large cylinder, after which *GH* shows the independent expansion of the steam now shut within the large cylinder, and *DE* shows the compression of steam by continued discharge from the small cylinder into the receiver. At the end of the stroke the receiver pressure is *OE*, and this must be the same as the pressure at *C*, if

20—2

there is to be no "drop." In the diagram sketched it is assumed that there is none. The case of "drop" would be illustrated if we were to cut off the corner at C by a vertical line drawn from
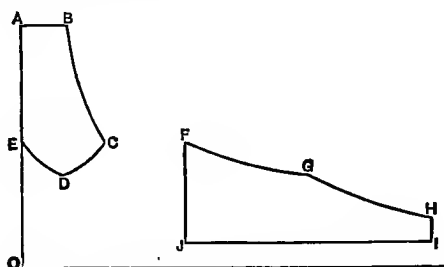
FIG. 106.   Compound Diagrams : Receiver type.

some earlier point in BC to meet the curve CD; this would of course also imply a shortened high-pressure stroke. Diagrams of a similar kind may be sketched without difficulty for the case of a receiver engine with any assigned phase-relation between the pistons.

It may be noticed in passing that an intermediate receiver has the thermodynamic advantage that it reduces the range of temperature in the high-pressure cylinder, and so helps to prevent initial condensation of the steam. This will be made obvious by a comparison of fig. 95 and fig. 96. The lowest temperature reached in the high-pressure cylinder is that corresponding to the pressure at D, and is materially higher in fig. 96 than in fig. 95.

183.   Adjustment of the division of work between the cylinders, and of the drop.   Graphic method. By making the cut-off take place earlier in the large cylinder we increase the mean pressure in the receiver; the work done in the small cylinder is consequently diminished. The work done in the large cylinder is correspondingly increased, for the total work (depending as it does almost wholly on the initial pressure and the total ratio of expansion) is unaffected or scarcely affected by the change. Hence we have the apparently anomalous result that a shorter admission to the low-pressure cylinder causes it to do a larger share of the whole work.

Further, the same adjustment—namely, hastening the cut-off in the low-pressure cylinder—serves, in case there is "drop," to

remove it, for it raises the pressure in the receiver. By selecting suitable values of the ratio of cylinder volumes to one another and to the volume of the receiver, and also by choosing a proper point for the low-pressure cut-off, it is possible to secure absence of drop along with equality in the division of the work between the two cylinders.

To determine beforehand that point of cut-off in the low-pressure cylinder which will prevent drop when the ratio of cylinder and receiver volumes is assigned is a problem most easily solved, or approximately solved, by a graphic process. The process
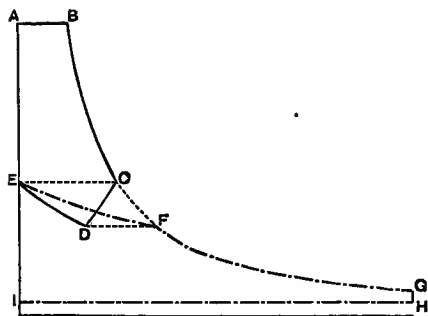
FIG. 107.
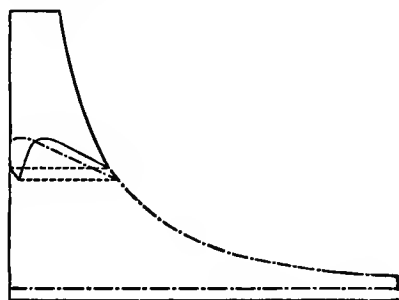
FIG. 108.

FIGS. 107 and 108. Determination of the point of Cut-off in the low-pressure cylinder of a compound engine.

consists in drawing the curve of pressure during admission to the low-pressure cylinder until it meets the curve of expansion which is common to both cylinders[1]. In fig. 107 (where for the sake of simplicity the effects of clearance are neglected) AB represents the admission line and BC the expansion line in the small cylinder. Release occurs at C, and from C to D steam is being taken by the large cylinder. D corresponds to the cut-off in the large cylinder, which is the point to be found. From D to E steam is being compressed into the receiver. If it is desired to avoid drop the receiver pressure at E is to be the same as the pressure at C. E is therefore known, and may be employed as the starting-point in drawing a curve EF which is the admission line of the low-pressure diagram $EFGHI$. This line is drawn by considering at each point in the low-pressure piston's stroke

[1] See a paper by Prof. R. H. Smith, "On the Cut-off in the Large Cylinder of Compound Engines," *The Engineer*, November 27, 1885.

what is then the whole volume of the steam. The place at which
EF intersects the continuous expansion curve BCG determines
the point of cut-off which will give no drop. The sketch (fig. 107)
refers to the case of a tandem receiver engine; but the process
may also be applied to an engine with any assumed phase-relation
between the cranks. Fig. 108 shows a pair of theoretical indicator
diagrams determined in the same way for an engine with cranks
at right angles, the low-pressure crank leading. In these examples
the volume of the receiver has been taken equal to the volume of
the high-pressure cylinder. With a larger receiver the variations
of pressure during the back stroke of the high-pressure piston
would be less conspicuous. In using the graphic method any
form may be assigned to the curve of expansion. Generally this
curve may be treated without serious inaccuracy as a common
hyperbola, in which the pressure varies inversely as the volume.
The construction may obviously be applied to triple and quad-
ruple expansion engines. For an accurate solution it would be
necessary to take the effect of clearance into account and also to
allow for some loss of pressure in the passage from one vessel to
another. The figures given here omit these complications, and
treat the expansion as hyperbolic.

184. Algebraic Method. When this simple relation be-
tween pressure and volume is assumed, it is not difficult to find
algebraically the low-pressure cut-off which will give no drop,
with assigned ratios of cylinder and receiver volumes. Taking
the simplest case—that of a tandem engine, or of an engine with
parallel cylinders whose pistons move together or in opposition—
we may proceed thus. Since the point of cut-off to be determined
depends on volume-ratios we may for brevity treat the volume of
the small cylinder as unity. Let R be the volume-ratio of the
receiver to the small cylinder, and L the volume-ratio of the large
to the small cylinder. Let x be the required fraction of the stroke
at which cut-off is to occur in the large cylinder; and let p be the
pressure at release from the small cylinder. If there is to be no
drop, p is also the pressure in the receiver at the beginning of
admission to the large cylinder. During that admission the
volume changes from $1 + R$ to $1 - x + R + xL$, and the pressure
at cut-off is therefore $\dfrac{p(1 + R)}{1 - x + R + xL}$. The steam that remains

is now compressed into the receiver, from volume $1 - x + R$ to volume R. Its pressure therefore rises to

$$\frac{p(1+R)}{1-x+R+xL} \cdot \frac{(1-x+R)}{R},$$

and this, by assumption, is to be equal to p. We therefore have

$$(1+R)(1-x+R) = R(1-x+R+xL),$$

whence
$$x = \frac{R+1}{RL+1}.$$

Thus, with $R = 1$ and $L = 3$, cut-off should occur in the large cylinder at half-stroke (which is the case illustrated by the diagram of fig. 106); with a greater cylinder ratio the cut-off in the large cylinder should be earlier, as it is, for instance, in fig. 107.

The above expression may be written $x = \dfrac{1 + \dfrac{1}{R}}{L + \dfrac{1}{R}}$, from which

it will be seen that when the receiver is very large x approximates in value to $\dfrac{1}{L}$.

A similar calculation[1] for a compound engine whose cranks are at right angles, and in which cut-off occurs in the large cylinder before half-stroke, shows that the condition of no drop is secured when

$$2R(xL - 1) = 1 - 2\sqrt{x(1-x)}.$$

In some compound engines a pair of high-pressure cylinders discharge into a common receiver; in some a pair of low-pressure cylinders are fed from a receiver which takes steam from one high-pressure cylinder, or in some instances from two. With these arrangements the pressure in the receiver may be kept much more nearly constant than is possible with the ordinary two-cylinder type. Occasionally compound engines work without any mechanical connection between the cranks, and the pressure within the receiver then depends not only on the adjustment of the points of cut-off but also on the relative frequency of stroke

[1] Examples of calculations dealing with particular arrangements of two- and three-cylinder compound engines will be found in an Appendix to Sennett and Oram's treatise on *The Marine Steam-Engine.*

of the pistons. An example of this was furnished in the non-coupled compound engines designed by the late Mr F. W. Webb which were used for some years on the London and North Western Railway.

185. Ratio of Cylinder Volumes. The size of the low-pressure cylinder in a compound engine is fixed by reference to the power the engine is intended to develop, when the speed, the boiler pressure, and the total ratio of expansion are given. But the size of the high-pressure cylinder remains a matter of choice when all these things are settled. Say that the total ratio of expansion is to be r; we may choose any ratio L less than r for the volume-ratio of the large to the small cylinder. It will then be necessary to make the cut-off in the small cylinder happen at a fraction of the stroke equal to $\dfrac{L}{r}$ in order that the final volume of the steam, when it fills the whole of the large cylinder, may be r times its initial volume up to the point of cut-off in the small cylinder. Thus an earlier or later adjustment of the cut-off in the high-pressure cylinder will allow the whole ratio of expansion to take whatever value may be wanted, no matter what be the ratio of the cylinder volumes.

Again, as we have seen above, by varying the cut-off in the large cylinder we can adjust matters so that equal amounts of work are done in both cylinders, irrespective of their sizes.

But it is only when a suitable ratio of volumes has been selected that this adjustment to equalise the work will also secure a reasonable absence of "drop"—or that an adjustment of the low-pressure cut-off to avoid drop will not too seriously disturb the balance of work.

This consideration serves to fix in a general way the proper proportion of the volumes. No hard and fast rule is followed; an exact balance in the work is not essential[1], and a complete absence of drop is not even desirable. The same practical considerations which make it undesirable in a simple engine to have complete

[1] It is sometimes preferable to divide the work unequally. In a four-cylinder marine engine, when the cranks are set at the best angles to give dynamical balance among the forces and couples due to the acceleration of the moving parts (see Chapter XIII.) the most uniform aggregate turning moment on the propeller-shaft is secured with an unequal distribution of the work among the cylinders, as was shown by Lorenz.

expansion apply in regard to compound engines: unless there is some little drop the last part of the stroke is ineffective.

Another consideration enters into the question. In some engines, especially marine engines, it is a point of importance to avoid having an early cut-off in any of the cylinders, partly to avoid unnecessarily severe stresses in the mechanism and partly to allow the valves to be of the simplest kind. This may lead to the existence of more drop than would otherwise be permissible. In practice the choice of volume-ratios is to some extent a compromise between conditions that are more or less incompatible, and, as might be expected, a good deal of variety is found.

In a two-stage or two-cylinder compound condensing engine, for instance, using steam of 80 or 90 lbs. pressure the large cylinder may have from three to four times the volume of the small cylinder. The steam in this case should expand about 12 times; if a ratio of 3 to 1 be chosen the conditions of equal work and very little drop will be secured by putting the cut-off at something like one-fourth of the stroke in the high-pressure cylinder and at about one-sixth of the stroke in the low-pressure cylinder. An example will be found in the indicator diagrams given below in fig. 110. On the other hand, if the high-pressure cylinder have only one-fourth of the volume of the other, a later cut-off will serve. The suitable ratio of volumes depends on the boiler pressure; thus if it is $3\frac{1}{2}$ with 70 lbs. it may be as much as $4\frac{1}{2}$ with 100 lbs.

In three-stage or triple-expansion engines of the mercantile marine, where the boiler pressure is rarely less than 150 lbs. and not often more than 180 lbs., the third cylinder has usually from 6 to 7 times the capacity of the first cylinder, and the second cylinder has usually from 2·3 to 2·6 times that of the first. In land engines of this type, where an earlier cut-off may be resorted to without inconvenience, the first cylinder may be rather larger: its capacity ranges from about one-fifth to one-sixth that of the low-pressure cylinder. In naval practice the boiler pressure is often higher, 250 lbs. per square inch being not unusual, but much of the steaming has to be done at reduced powers. The capacity of the third cylinder is, as a rule, from 5 to 6 that of the first and the capacity of the second cylinder from 2·2 to 2·4 that of the first.

In four-stage or quadruple expansion engines the proportion

of volumes in the successive cylinders is usually something like
1 : 2 : 4 : 8, though in some examples the fourth cylinder may
have 8½ or even 9 times the volume of the first, the boiler
pressure ranging generally from 200 to 250 lbs. per square inch[1].

In many large compound engines, especially in marine practice,
the last stage of the expansion is carried out in two cylinders
instead of one, the steam being divided between the two.   In
this way the use of a single excessively large cylinder is avoided,
and the arrangement often facilitates balancing, as will be shown
in Chapter XIII.   A very usual type of marine engine is one
with three-stage or triple expansion, but with four cylinders, two
of which jointly serve to carry out the last stage of the compound
action.

**186.   Advantage of Compound Expansion in the eco-
nomical use of High-Pressure Steam**.   The thermodynamic
advantage of compound expansion has already been pointed out.
It allows high-pressure steam to be used without the excessive
waste which would occur if a high grade of expansion were
attempted in a single cylinder.   So long as the boiler pressure
does not much exceed 100 lbs. this advantage is sufficiently
secured by dividing the expansion into two stages: accordingly
the ordinary compound engine with two stages of expansion is
used with pressures up to 100 lbs. but seldom with higher
pressures.   Beyond this triple expansion becomes in general
advisable if the full benefit of the higher pressure is to be
secured.   But when the expansion is divided into three stages
it becomes advantageous to use a pressure considerably higher
than the limit we have just named: thus with triple engines
a pressure of 160 to 180 lbs. is usual.   Intermediate pressures,
of say 120 or 130 lbs., are not often found: they are too high
to suit the two-cylinder compound engine and too low to let
triple expansion give its best effects.   Quadruple expansion has
little if any advantage when the pressure is under 200 lbs.;
up to this pressure and perhaps beyond it the thermodynamic
benefit of a fourth stage is scarcely sufficient to justify the
mechanical complication it involves.   In naval practice the use

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[1] Particulars of the cylinder dimensions in a large number of marine engines of
various classes will be found in Dr G. Bauer's work on *Marine Engines and Boilers*
(Translation edited by L. S. Robertson, 1905).

of water-tube boilers has raised the pressure of steam to 250 and even in a few cases to 300 lbs., but except in isolated instances warships do not use quadruple expansion. In the mercantile marine it has found employment to a considerable extent.

187. Mechanical advantage of Compound Expansion. Uniformity of Effort in a Compound Engine. A simple engine using high-pressure steam with an early cut-off has the drawback, from the mechanical point of view, that the thrust of the steam on the piston during the early part of the stroke is very great in comparison with the mean thrust. The initial pressure of the steam acts on the full area of a piston whose size is determined by reference to the mean pressure. The piston and connecting rod, the framing and other parts of the machine must be made strong enough for this relatively great initial thrust, also there is much wear and tear at joints, and for steady motion a large fly-wheel becomes necessary.

The compound engine avoids the extreme thrust and pull which would have to be borne by the piston-rod of a single-cylinder engine working at the same power with the same initial pressure and the same ratio of expansion. If all the expansion took place in the low-pressure cylinder, the piston at the beginning of the stroke would be exposed to a thrust greater even than the sum of the thrusts on the two pistons of a compound engine of equal power. Thus in the tandem engine of fig. 105 the greatest sum of the thrusts will be found to amount to less than two-thirds of the thrust which the large piston would be subjected to if the engine were simple. The mean thrust throughout the stroke is of course not affected by compounding; only the range of variation in the thrust is reduced. The effort on the crank-pin is consequently made more uniform, the strength of the parts may be reduced, and the friction and wear at joints lessened. Thus even in a tandem compound engine there is mechanically some advantage, and the benefit of compounding in this respect is obviously much greater when the cylinders are placed side by side, instead of tandem, and work on cranks at right angles. As a set-off to its advantage in giving a more uniform effort, the compound engine has the drawback of requiring more working parts than a simple engine with one cylinder. But in many instances—as in marine engines—two cranks and two cylinders are in any case almost indispensable, to

give a tolerably uniform effort and to get over the dead-points without the aid of a heavy fly-wheel; and the comparison should then be made between a pair of simple cylinders and a pair of compounded cylinders. Another point in favour of the compound engine is that, although the whole ratio of expansion is great, there need not be a very early cut-off in either cylinder; hence the common slide-valve, which is unsuited to give an early cut-off, may be used in place of a more complex arrangement. The mechanical advantages of compound working were recognized sooner than its thermodynamic economy, and did much to bring it into favour before, indeed, the practice had grown up of using steam high enough in pressure to make compounding very distinctly advantageous on the thermodynamic side.

Again, apart from its improved economy in fuel consumption, the mechanical merits of the triple engine have contributed much to bring it quickly to the position it now holds in marine practice. The advantage of three cranks over two in giving uniform effort and comparatively little friction and wear is conspicuous.

It also gives a better balance as regards the effects of inertia, and in this respect a great further gain may be secured by using four cranks, whether with quadruple expansion or, as is more usual, with triple expansion and a division of the third stage between two cylinders.
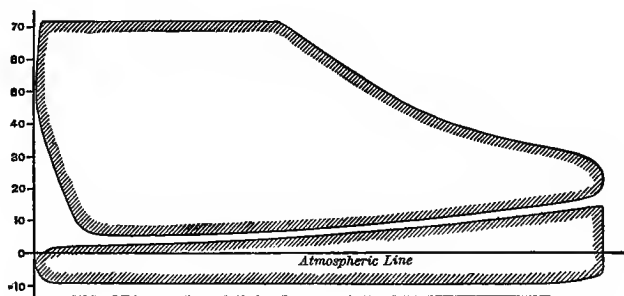
Fig. 109. Indicator diagrams of a Woolf Engine.

188. Examples of Indicator Diagrams from Compound Engines. Fig. 109 shows a pair of diagrams from the two cylinders of a Woolf engine, in which the steam passes as directly as possible from the small to the large cylinder. Both pistons have

the same length of stroke. The diagrams are drawn to the same scale of stroke and therefore to different scales of volume, and the low-pressure diagram is turned round so that it may fit into the space below the high-pressure diagram. There is some drop at the high-pressure release, and frictional losses in the passages cause the admission line of the large cylinder to lie slightly lower than the exhaust line of the small cylinder. The transfer of steam goes on throughout nearly the whole of the back stroke until compression begins in the small cylinder. The steam then present in the large cylinder continues expanding for the small part of the stroke that is left until the point of release is reached.
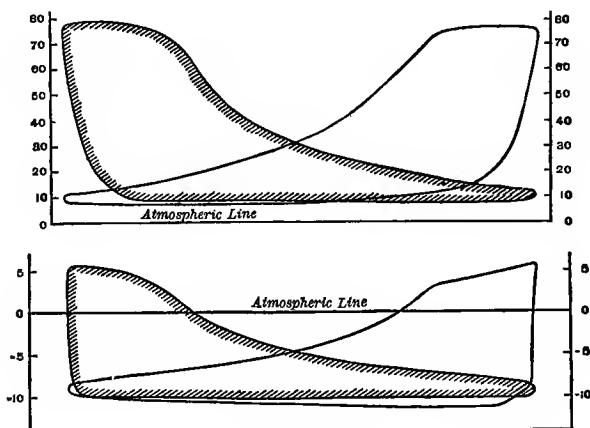
Fɪɢ. 110.

An example of compound diagrams for an engine of the receiver type has already been given in figs. 91 and 92, Chap. IX. The receiver in that engine was unusually large, which accounts for the nearly level line drawn during the back stroke of the small piston. Another example is given in fig. 110, which shows the diagrams of a tandem receiver engine with cylinders 30 and 52 inches in diameter and 6 ft. stroke (volume-ratio 1 to 3), taking steam at an initial pressure of 80 lbs. above the atmosphere. With this proportion of volumes and with the somewhat early cut-off shown by the diagrams there is a complete absence of any objectionable drop and a nearly equal division of work between the cylinders. Expansion valves (see Chap. XI.) were used to produce this early cut-off. The exhaust line of the small cylinder dips in the middle, as in fig. 106 or 107, but much less, for here the

receiver is more capacious. When the cranks are set at right angles this line rises towards the middle, as fig. 108 indicates.

Fig. 111 shows a set of triple expansion diagrams, from trials
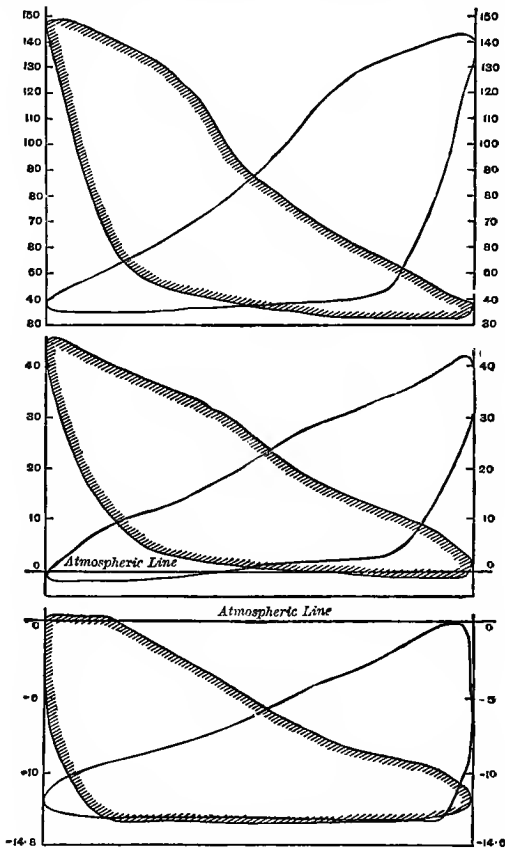
FIG. 111. Indicator diagrams of Triple expansion Engine.

of the steamship "Iona," by a Committee of the Institution of Mechanical Engineers. The cylinder diameters were 21·9 in., 34 in. and 57 in., giving a volume-ratio of $1 : 2·4 : 6·8$, and the stroke was 39 in. The engines made 61 revolutions per minute and developed 208 I.H.P. in the first cylinder, 217 in the second, and 220 in the third, with a consumption of 13·35 lbs. of steam per I.H.P.-hour. A simple slide-valve was used on each cylinder[1].

[1] Report of Research Committee on Marine Engine Trials, *Proc. Inst. Mech. Eng.*, April 1891.

189. Combination of the Indicator Diagrams in Compound Expansion.   The indicator diagrams of a compound engine may be combined in such a way that the pressures and volumes in the several cylinders are displayed in proper relation to one another, by the use of a single scale of pressures and a single scale of volumes.   Some care, however, is necessary in the interpretation of such combined diagrams, and the construction to be adopted will depend on the use that is aimed at.

A common practice is to set out each diagram from the line of
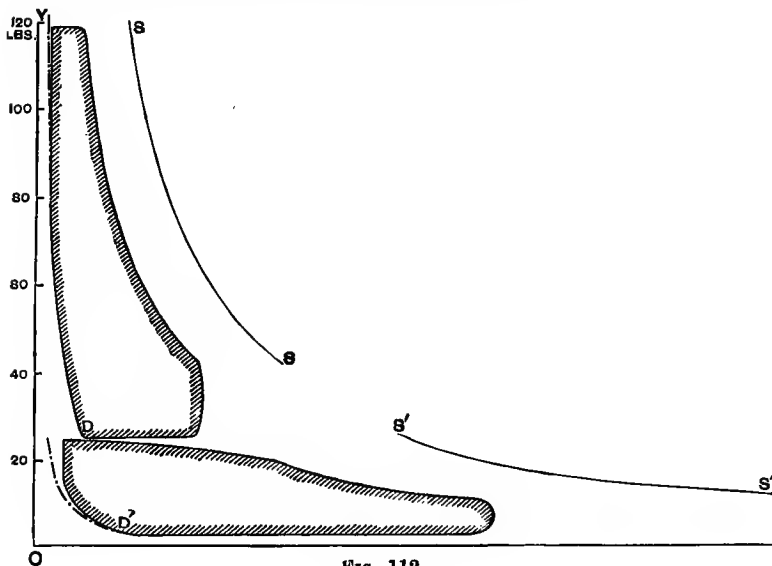
Fig. 112.

no volume through a distance which represents the clearance in the corresponding cylinder.   This is illustrated in fig. 112, which has been drawn to exhibit in combination the diagrams already shown in figs. 91 and 92, § 164.   Each of the two diagrams in fig. 112 is a mean for the two sides of the piston, and the distance of each from the line OY is the mean clearance in the corresponding cylinder.   Diagrams drawn in this way are not without their uses, but it must be remembered that the amount of substance which is taking part in the expansion is different in the two parts of the combination, and consequently a single adiabatic curve or a single saturation curve cannot properly be drawn to apply to both.   The line SS is the saturation curve

for the first stage of expansion, and the line $S'S'$ for the second stage. In this example the cylinder feed per single stroke was 0·0498 lb., and the cushion steam was 0·0074 lb. in the small cylinder and 0·0022 lb. in the large cylinder. The saturation curve SS is accordingly drawn for 0·0572 lb. and $S'S'$ for 0·052 lb.

The amount of the substance present in the cylinder is in general different in the successive stages because of differences in the amount of cushion steam in the several cylinders: the cylinder feed is the same throughout. If therefore we modify the diagram in such a way as to eliminate the cushion steam, leaving the cylinder feed only, we may draw a single saturation curve which will serve for the whole of the expansion.
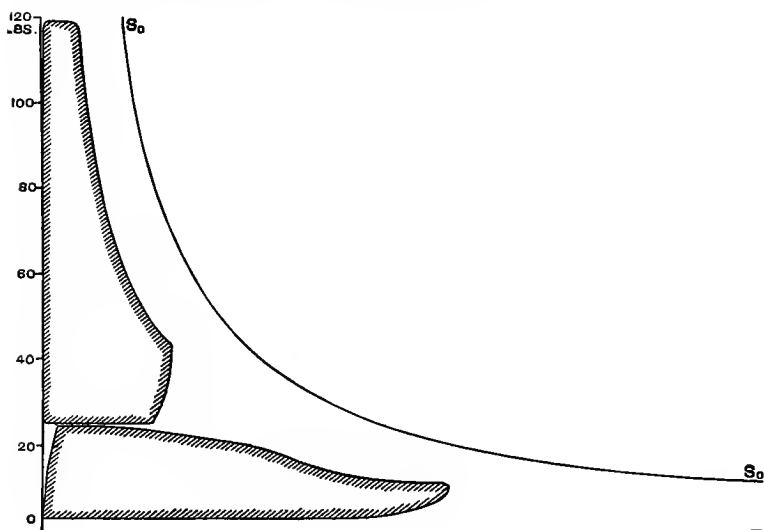
Fig. 113.

This is done in fig. 113, which represents the same pair of diagrams, transformed by the following device. From points D, D' (fig. 112) taken at the places where compression has begun and the exhaust is complete, saturation curves are drawn for the cushion steam in the respective cylinders. These curves are indicated by broken lines in the figure: the one that relates to the small cylinder is scarcely distinguishable from the compression curve of the indicator diagram. The diagrams are then redrawn as in fig. 113, using horizontal distances from these curves as abscissæ. This is

equivalent to subtracting from the actual volumes throughout the diagram a quantity which represents the volume the cushion steam would occupy if it were saturated at all pressures. The result is that the area of the diagram remains unaltered: its area is still a true measure of the work. But a single saturation curve $S_0 S_0$ may now be drawn—namely, for a quantity of steam equal to the cylinder feed—which will apply equally to both (or all) stages of the compound expansion. The horizontal distance at any pressure between the expansion curve in fig. 113 and the saturation curve $S_0 S_0$ is the same as the horizontal distance at that pressure between the expansion curve in fig. 112 and its corresponding saturation curve. It still represents the volume which has disappeared by

Cub. Ft. per lb.
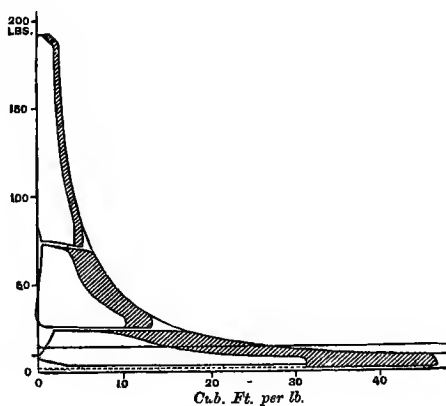
Fig. 114.

condensation and leakage or what is often called the "missing quantity." The chief advantage of this construction is that it makes a single saturation curve possible, and so allows the changes in the amount of water present to be readily exhibited as the steam passes through the whole course of its expansion.

This will be apparent from figs. 114 and 115, which are copied from Professor Osborne Reynolds' account of trials of a triple-expansion engine[1]. Here the cushion steam has been eliminated in the manner just described and a single saturation curve has been drawn for the cylinder feed. The horizontal width of shaded space between the actual expansion curves and this line measures the missing quantity at each stage in the expansion. Fig. 114

[1] *Min. Proc. Inst. C. E.* vol. xc.

refers to a test made without steam in the steam jackets, and
fig. 115 to a test when all the jackets were supplied with steam
at the full boiler pressure of 190 lbs.   The drying influence of
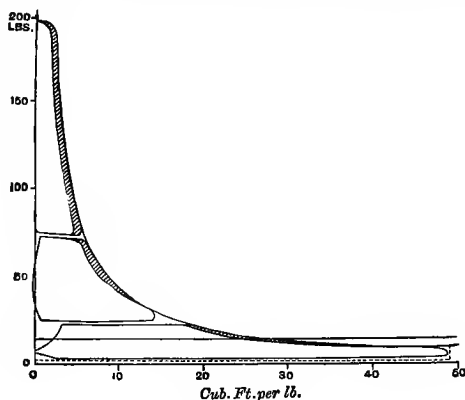
Fig. 115.

the jacket is conspicuous: in fig. 115 there is scarcely any con-
densation in the third cylinder.

These diagrams relate to an engine built for experimental use
in which the three pistons could move independently, at different
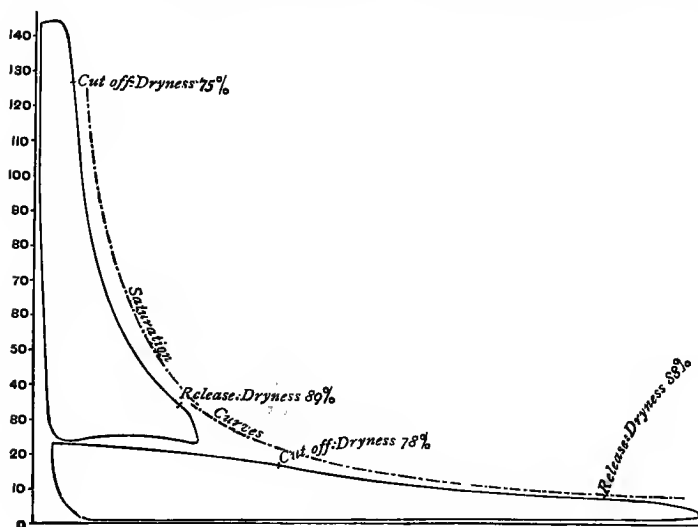
Fig. 116.

speeds, and the speeds were in fact different. Hence to prepare the diagrams for combination a further device was employed: the common scale of length of the diagrams was chosen so that the volumes represented in each are reckoned per lb. of cylinder feed. The scale of volume is accordingly divided in the figures to show cubic feet per lb. of water passing through the engine. This is a method of graduation which might be followed with advantage even in ordinary cases, where it is not rendered necessary by the pistons having independent speeds, for it facilitates comparison between various trials.

An additional example of compounded indicator diagrams is given in fig. 116, which represents in a combined form the diagrams taken in a trial by Mr Longridge of a two-cylinder compound engine where slow running and efficient jacketing made the amount of condensation in the cylinders considerably less than in the example of fig. 112. The diagrams are set out separately as in that figure, with the clearance appropriate to each, and the two corresponding saturation curves are drawn. Here the cylinders were 17 and 34 inches in diameter and the stroke was 5 ft. The mean clearance volume was 0·26 cubic foot in the high-pressure cylinder, and 0·84 cubic foot in the low. Figures are given for the dryness at the points of admission and release which involve the assumption that there is no leakage, in other words that the missing quantity is entirely due to the presence of water in the steam.

CHAPTER XI.

VALVES AND VALVE-GEARS.

190. The Slide-Valve. In early steam-engines the distribution of steam was effected by means of conical lift-valves, rising and falling on conical seats, and worked by tappets from a rod which hung from the beam. The slide-valve, the invention of which is credited to Murdoch, an assistant of Watt, came into general use with the introduction of locomotives, and is now
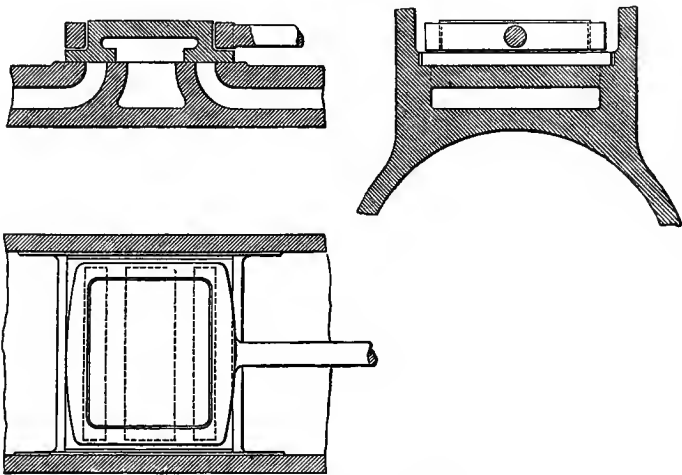
Fɪɢ. 117. Common Slide-Valve.

employed, in one or other of many forms, in the great majority of engines.

The common or locomotive slide-valve is illustrated in fig. 117, which shows a sectional side and end elevation and a plan. The seat, or surface on which the valve slides, is a plane surface formed on or fixed to one side of the cylinder, with three "ports" or openings, which extend across the greater part of the cylinder's

width. The ports are shown in the plan by dotted lines. The central opening is the exhaust-port through which the steam escapes; the others, or steam-ports, which are narrower, lead to the two ends of the cylinder respectively. The valve is a box-shaped cover which slides upon the seat, and the whole is enclosed in a chamber called the valve-chest, to which steam from the boiler is admitted. The valve is pulled backwards and forwards across the ports by means of a valve-rod which passes out of the valve-chest through a steam-tight stuffing-box. The valve is attached to the valve-rod, not rigidly but in such a way that while it has no longitudinal freedom to slide along the rod it is free to take a close bearing on the seat, under the pressure exerted by the steam on its back. In its middle position the valve covers both steam-ports completely, but when it is moved a sufficient distance to either side of the middle position, it allows fresh steam to enter one end of the cylinder from the valve-chest, and allows the steam which has done its work to escape from the other end of the cylinder through the cavity of the valve into the exhaust-port. The valve-rod is generally moved by an eccentric on the engine-shaft, which is mechanically equivalent to a crank whose radius is equal to the eccentricity, or distance of the centre of the shaft from the centre of the eccentric disc or sheave. The sheave is encircled by a strap to which the eccentric-rod is fixed, and the rod is connected by a pin-joint to the valve-rod outside of the valve-chest. The eccentric-rod is generally so long that the motion of the valve is sensibly the same as that which it would receive were the rod infinitely long. Thus if a circle (fig. 118) be drawn to represent the path of the eccentric-centre during a revolution of the engine, and a perpendicular PM be drawn from any point P on a diameter AB, the distance CM is the displacement of the valve from its middle position at the time when the eccentric-centre is at P. AB is the whole travel of the valve.
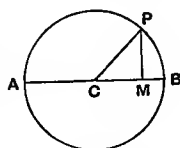
FIG. 118.

191. Lap, Lead, and Angular Advance. If the valve were formed so that when in its middle position it did not overlap the steam-ports (fig. 119), any movement to the right or the left would admit steam, and the admission would continue until the

valve had returned to its middle position, or, in other words, for
half a revolution of the engine.   Such a valve would not serve for
expansive working; it would admit steam to one end of the
cylinder during all the stroke, and at the same time would exhaust
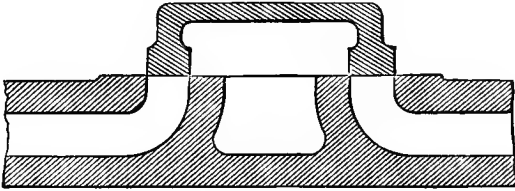
Fig. 119.   Slide-Valve without Lap.

steam from the other end during all the stroke.   As regards the
relative position of the crank and eccentric it would have to be set
so that its middle position was coincident in point of time with
the extreme position of the piston; in other words, the eccentric
radius would have to be set 90° in advance of the crank.

To make expansive working possible the valve must be able to
keep the cylinder ports closed during some part of the stroke.
For this purpose it must have what is called *lap*, that is to say its
edges must project beyond the ports as in fig. 120, where *e* is the
outside lap or *steam lap* and *i* is the *inside lap* or *exhaust lap*.
Admission of steam to either end of the cylinder now begins only
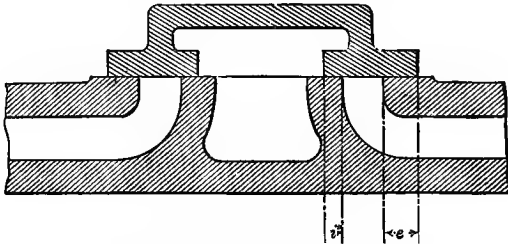
Fig. 120.   Slide-Valve with Lap.

when the displacement of the valve from its middle position is
equal to the outside lap, and continues only until the valve returns
to the same distance from its middle position.   Further, exhaust
begins only when the valve has moved past the middle position by
a distance equal to the inside lap and continues until the valve
has again returned to this distance from its middle position.   Thus

let a circle (fig. 121) be drawn to represent the path of the eccentric-centre, on a diameter fg which is the whole travel of the valve, let om be set off equal to the outside lap e and on equal to the inside lap i, and let perpendiculars amb and cnd be drawn at these distances from the centre. The points a, b, c and d then mark the positions of the eccentric-centre at which the four events of admission, cut-off, release and compression respectively occur for one end of the cylinder. As to the other end the four events are determined in the same way by setting off the corresponding
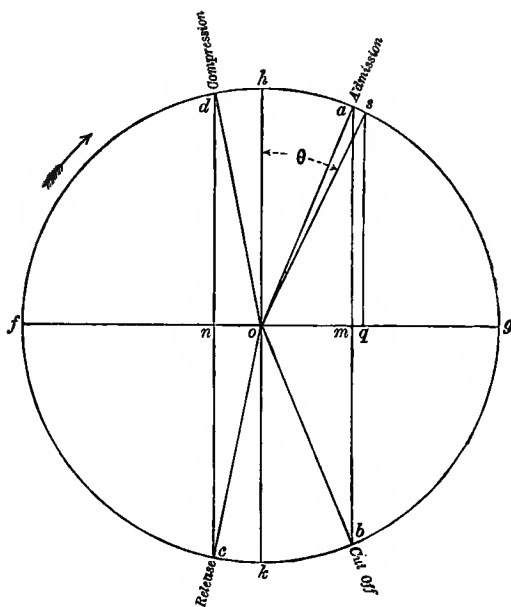
Fig. 121.

outside lap to the left of o and the inside lap to the right of o. The laps may or may not be equal for the two ends of the cylinder. For the sake of clearness we may for the present confine our attention to one of the two ends. Of the whole revolution the part from a to b is the arc of admission; in other words, the port is open to steam while the shaft turns through an angle equal to aob. Similarly bc is the arc of expansion, cd that of exhaust and da that of compression.

The relation of these events to the piston's position is still undefined. If the eccentric were set in advance of the crank by

an angle equal to *foa*, the valve would be just beginning to open
as the piston stroke begins. It is, however, desirable, in order to
allow the steam free entry, that the valve should be already some
way open when the piston stroke begins, and hence the eccentric
is set at a rather greater angular distance in advance of the crank.
Thus if the angular position of the eccentric be *os* while the crank
is at the dead-point (on the line *of*) the valve is already open by
the distance *mq*, which is called the *lead*. The angle θ by which
the whole angle between the crank and the eccentric exceeds a
right angle is called the *angular advance*, this being the angle by
which the eccentric is set in advance of the position it would hold
if the primitive arrangement without lap were adopted. The
lap *e*, the lead *l*, the angular advance θ, and the half-travel or
throw of the eccentric *r* are connected by the equation

$$e + l = r \sin \theta.$$

An effect of lead is to cause *preadmission*, that is to say, the lead
allows steam to enter before the back stroke is quite completed,
and this increases the mechanical effect of the compression in
"cushioning" the piston during the reversal of its motion.

The greatest amount by which the valve is ever open during
the admission of steam is the distance *mg*. The width of the
steam port is made at least equal to this distance, and is often
greater in order that the wider opening *nf* which occurs during
exhaust may be taken advantage of.

**192. Graphic method of examining the distribution of
steam given by a slide-valve.** Let the circle *APB* (fig. 122)
represent the path of the crank-pin about the centre *O*, the stroke
being *AB*. When the crank is at any point *P* the position of the
piston may be found by projecting the point *P* on *AB* by drawing
a circular arc *PD* with the length of the connecting-rod *PQ* as

Fig. 122.

radius and the cross-head Q as centre. Then DO represents the
displacement of the piston from its middle position, and AD and
DB represent its distance from the two ends of the stroke. Another
construction equivalent to this is to draw through O the arc OM
with the length of the connecting-rod as radius, and draw PM
parallel to AB. PM, being equal to DO, measures the displace-
ment of the piston from its position at mid-stroke. In speaking of
the two ends of the cylinder we shall distinguish the one nearer
the crank as the front end and the other as the back end. The
stroke towards the crank may be called the in-stroke and the
other the out-stroke, as marked in fig. 122.

To find the position of the piston at each of the four events we
have to make a construction which is equivalent to transferring

Fig. 123.

from fig. 121 the four positions of the crank which correspond to
the positions *a, b, c, d* of the eccentric. This is most readily done
by drawing a single circle (fig. 123) to represent the motion of the
crank-pin on one scale and the motion of the eccentric-centre on
another scale. Taking the diameter *AB* to represent the piston
stroke, draw another diameter *hk* to represent the line *hk* of
fig. 121 turned back through an angle of $90° + \theta$, so that the angle
AOh (fig. 123) is equal to θ. Draw *ab* and *cd* parallel to this line
at distances from it equal to the outside and inside laps respect-
ively. The effect is that each of the points *a, b, c* and *d* is turned
back, in fig. 123, through an angle equal to $90° + \theta$ as compared
with its position in fig. 121. Consequently these points in fig. 123
show the positions which the crank has at the four events. And
the corresponding positions of the piston may be found by
projecting the points *a, b, c, d* on *AB* by means of circular arcs.
This is shown in fig. 123, and the indicator diagram is also
sketched by reference to the positions projected on *AB*.

The following is an equivalent and rather more convenient
construction. Let a circle (fig. 124) be drawn as before to
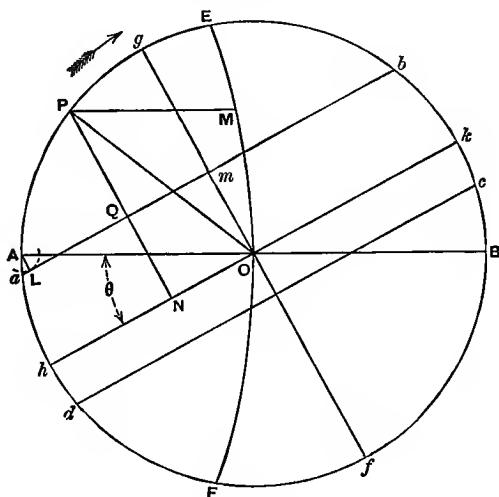
Fig. 124.

represent the eccentric's motion on one scale and the crank's on
another, and let *AB* be the piston stroke. Draw *hk* as before so
that the angle $AOh = \theta$, the angular advance. Taking a centre

on *OA* produced, draw the arc *EOF* through the centre *O* with radius equal to the length of the connecting-rod. Then when the crank has any position *OP* the displacement of the valve from its middle position is *PN* (drawn perpendicular to *hk*) and the displacement of the piston from mid-stroke is *PM*. Also, if *ab* and *cd* be drawn as before at distances from *hk* equal to the laps, the four events happen at *a*, *b*, *c*, and *d*, and *PQ* is the extent to which the valve is open when the crank is at *P*. Similarly *AL* is the extent to which the valve is open at the beginning of the stroke, that is the *lead*. The port has its maximum opening when the crank is at *Og* during admission and at *Of* during exhaust, unless its width is so small that it has become completely uncovered with a smaller displacement of the valve.

The diagram shown in fig. 124, which is a modified form of one due to Reuleaux, may readily be applied to determine the characteristics which a slide-valve must have to give a stated distribution of steam. Suppose for instance that the travel of the valve, the lead, and the position of cut-off are assigned. Having marked *b*, the position of the crank-pin at the given point of cut-off in relation to the stroke *AB*, draw a circle with centre *A* and radius *AL* equal to the lead. Then draw a line through *b* tangent to this circle. This will be the line *ba* of the diagram. Its inclination to *BA* determines the angular advance, and a perpendicular on it from *O* gives *Om*, which is the outside lap. The inside lap becomes determinate when either the point of release *c* or that of compression *d* is assigned, and it is found by drawing a line through *c* or *d* parallel to *ab*, and measuring the distance of this line from *O*.

193. Inequality of the distribution on the two sides of the piston. So far we have dealt only with the events corresponding to one end of the cylinder, namely (in the diagram) the back end. This has been done only to avoid complicating the diagram with too many lines. In fig. 125 the construction of fig. 124 is repeated with the outside-lap lines *ab* and *a'b'* drawn for both ends, and also the inside-lap lines *cd* and *c'd'*, and the corresponding events are marked. The construction lines relating to the front end of the cylinder are distinguished by being dotted and their reference letters are accented. The laps have been taken equal for the two ends, and an obvious result is that the cut-off is considerably later at the back than at the front.

Compare bm with $b'm'$, these being the distances by which the piston has passed mid-stroke when the cut-off occurs at the back and front respectively. This want of symmetry, proceeding as it does from the obliquity of the connecting-rod, is slight when the rod is many times longer than the crank but becomes important when the rod is short. In the sketches the length of the rod is supposed to be three times that of the crank.

This inequality may be remedied by making the outside laps unequal, giving less lap to the front end of the valve. When this is done, however, the amounts of lead (which are equal with equal
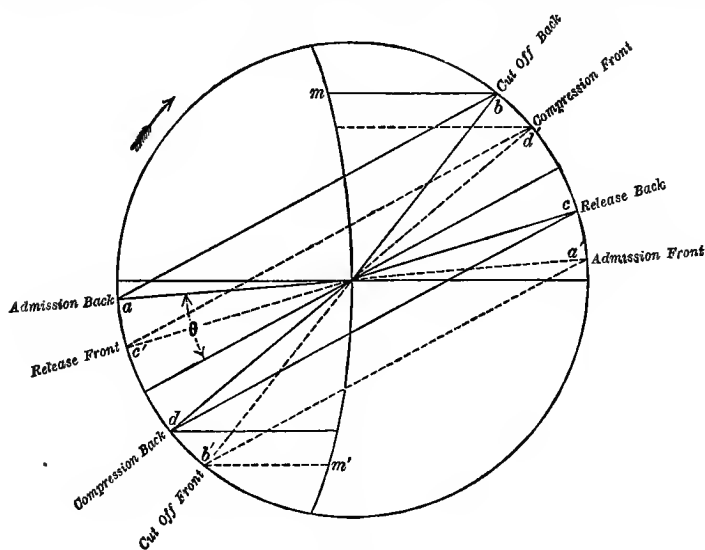
Fig. 125.

laps) become unequal. But in general it is better to sacrifice equality of lead and to secure at least approximate symmetry in the positions of the two points of cut-off, when the admission of steam is controlled by a simple slide-valve. When a separate expansion valve is used (§§ 207–208) the cut-off is determined by it, and not by the main slide-valve, and in that case the amounts of lead may properly be made equal. In certain cases a somewhat unequal distribution of steam is to be preferred, as in the ordinary vertical marine engine, where the work done by the steam against the front or bottom end of the piston is partly spent in raising the piston and rods and consequently should be greater than

the work done against the back or top end which is supplemented by the descent of these heavy weights.

In cases where the eccentric-rod is itself so short that its obliquity should be taken account of, this is readily done in Reuleaux's diagram (fig. 124 or 125) by using circular arcs in place of the straight lines *ab, hk, cd,* these arcs being described with a radius which represents the length of the eccentric-rod on the same scale as that on which the diameter AB represents the travel of the valve, from centres on Of produced beyond f. Except in rare cases it leads to no appreciable error to treat the eccentric-rod as infinitely long.

Fig. 126 illustrates how a symmetrical distribution is secured by reducing the outside lap at the front end. There *ab* is
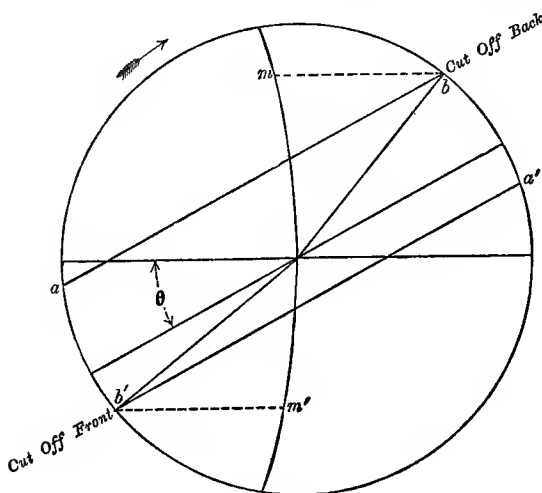
Fig. 126.

the outside-lap line for the back end and *a'b'* is the corresponding line for the front end. These lines are drawn so that the cut-off occurs at the same percentage of the stroke at both ends: *bm* and *b'm'* are equal. The inside laps may also be adjusted in the same way to give equal amounts of compression on both strokes (or, alternatively, to give symmetrical points of release). The amounts of lead, of course, are no longer equal: the lead at the front end has been considerably increased by the reduction of the lap.

194. Zeuner's Valve Diagram. A graphic construction much used practically in slide-valve investigations is the ingenious diagram published by Dr G. Zeuner in the *Civilingenieur* in 1856[1]. On the line AB (fig. 127), which represents the travel of the valve, let a pair of circles (called valve-circles) be drawn,
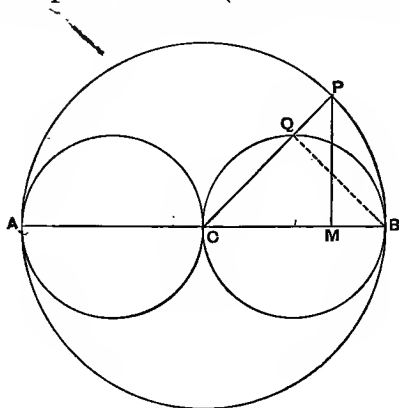
FIG. 127.

each with diameter equal to the half-travel. If a radius CP be drawn in the direction of the eccentric centre at any instant, it is cut by one of the circles at a point Q such that CQ represents the corresponding displacement of the valve from its middle position. That this is so will be seen by drawing PM and joining QB, when it is obvious that the triangles CPM and CBQ are equal in all respects and $CQ = CM$, which is the displacement of the valve. The line AB with the circles on it may now be turned back through an angle of $90° + θ$ ($θ$ being the angular advance), so that the valve-circles take the position shown to a larger scale in fig. 128. This makes the direction of CQP (the eccentric) coincide on the paper with the simultaneous direction of the crank, and hence to find the displacement of the valve at any position of the crank we have only to draw the line CQP in fig. 128 parallel to the direction which the crank has at the instant under consideration, when CQ represents the displacement of the valve to the scale on which the diameter of each valve-circle represents the half-travel of the valve. CL is the valve's displacement at the beginning of the stroke indicated by the arrow. Draw circular arcs

[1] Zeuner, *Treatise on Valve-Gears*, transl. by M. Müller, 1868.

EF and *IJ* with *C* as centre and with radii equal to the outside lap and the inside lap respectively. *CE* is the position of the crank at which preadmission occurs. The lead is *LM*. The greatest steam opening during admission is *GB*, and the greatest opening to exhaust is the whole width of the port, namely *KH*.
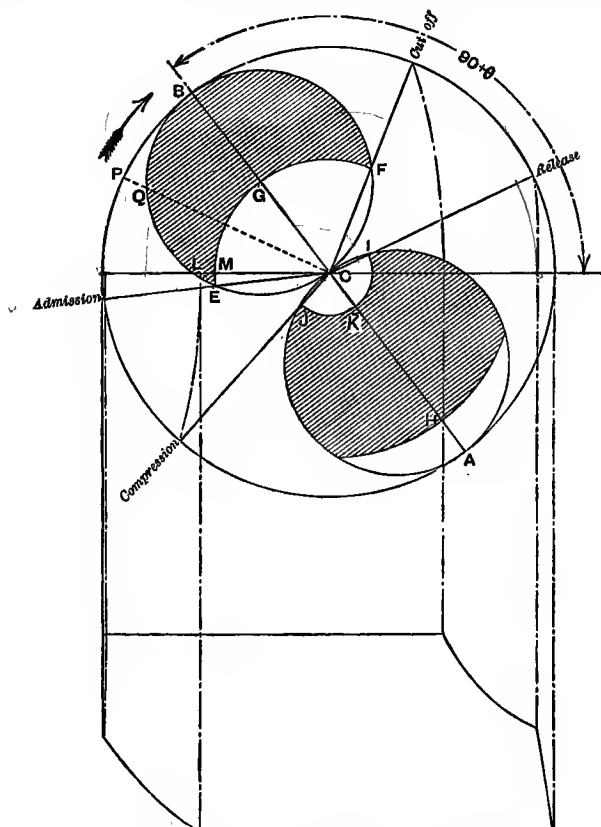
FIG. 128.   Zeuner's Slide-Valve Diagram.

Intercepts on the radii within the shaded areas give the steam and exhaust openings for any angular positions of the crank. The cut-off occurs when the crank has the direction *CF*. *CI* is the position of the crank at release, and *CJ* marks the end of the exhaust, or the beginning of compression.

In the diagram given in fig. 128 radii drawn from *C* mark the angular positions of the crank, and their intercepts by the valve-

circles determine the corresponding displacements of the valve. It remains to find the corresponding displacements of the piston. For this Zeuner employs a supplementary graphic construction, shown in fig. 129. Here ab or a_1b_1 represents the connecting-rod, and bc or b_1c the crank. With centre c and radius ac a circle ap is drawn, and with centre b and radius ab another circle aq. Then for any position of the crank, as cb_1, the intercept pq between the circles is equal to aa_1, and is therefore the distance by which the
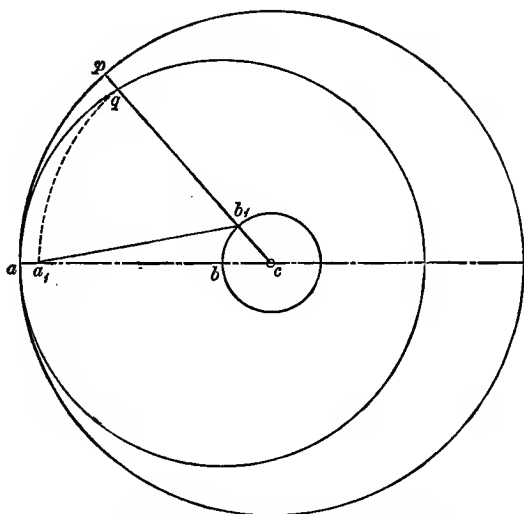
Fig. 129. Zeuner's construction to find the Displacement of the Piston.

piston has moved from the extreme position which it had at the beginning of the stroke. In practice this diagram may be combined with that of fig. 128, by drawing both about the same centre and using different scales for valve and piston travel. A radial line drawn from the centre parallel to the crank in any position then shows the valve's displacement from its middle position by the intercept CQ of fig. 128, and the simultaneous displacement of the piston from the beginning of its motion by the intercept pq of fig. 129. As an alternative to this the piston's displacement may be found in Zeuner's diagram by the construction used in Reuleaux's, which was described in connection with figs. 122—125.

As an example of the application of Zeuner's diagram we may take the same problem as before, namely, to find the outside lap

and angular advance when the point of cut-off and the lead for the corresponding side of the piston are assigned, as well as the travel of the valve. The solution is shown in fig. 130. On the base-line XX' mark the point M to represent the required point of cut-off and project this on the circle XPX' to find CP, which is the angular position of the crank at cut-off. With X as centre draw

Fig. 130.

a circle with radius XN to represent the given lead. From P draw PN tangent to this circle. Then CGB drawn perpendicular to PN defines the diameter of the valve-circle. The angle $X'CB$ is the angular advance, plus 90°, and CG is the required outside lap.

So far as the simpler problems of the slide-valve are concerned Zeuner's diagram has no marked advantage over Reuleaux's. It is however more readily applicable to cases where the events of the stroke depend on the movements of more than one eccentric, as, for instance, in the Meyer expansion gear to be presently described.

195. Bilgram's Valve Diagram. This is an alternative construction shown in fig. 132 and arrived at as follows. In fig. 131 let AED be a circle representing the path of the eccentric-centre E, and let C be the angular position of the crank when the eccentric is at E, so that COE is $90° + \theta$. Let OD be drawn making the angle AOD equal to the angular advance θ. Draw

DN perpendicular to the direction of the crank *OC*. Then the triangles *ODN* and *OEM* are equal, and *DN*, being equal to *OM*, measures the displacement of the valve from its middle position when the crank is at *C*.
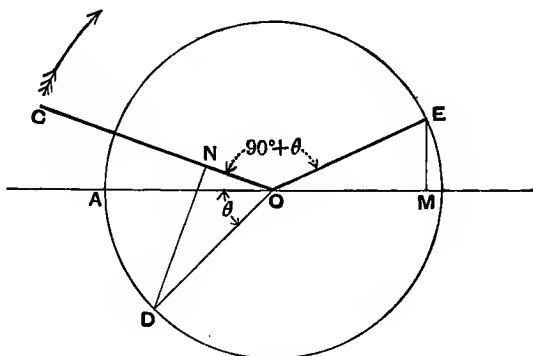
Fɪɢ. 131.

Hence the position of the crank when admission takes place is determined by the fact that the perpendicular from *D* on *OC* must then be equal to the outside lap: in other words, *OC* will then be tangent to a circle described about *D* as centre with the outside lap as radius. Further, when the crank travels round so that *CO* produced is tangent to the same circle on the other side it has reached the position of cut-off. Similarly, if a second circle be drawn about *D* with the inside lap as radius, the positions in which *OC* is tangent to it on its two sides are those of compression and release.

The complete construction, for one end of the cylinder, is shown in fig. 132. There *AOD* is the angular advance, *OD* the half-travel of the valve, *DP* the outside lap, and *DQ* the inside lap. C_1, C_2, C_3, C_4 are the positions of the crank at admission, cut-off, release, and compression, respectively. *OL*, being the half-travel minus the outside lap, is the greatest opening of the port to steam. *DF*, drawn perpendicular to *AO*, is the displacement of the valve when the piston is at the dead-point *A*, and *FG*, the intercept on this line between *AO* and a line *GH* drawn parallel to *AO* touching the outside-lap circle, is the lead.

In illustration of the use of this diagram we may take two problems. (1) Given the travel of the valve, the position of the crank

at cut-off, and the lead; to find the lap and the angular advance. Taking a base AO to represent the direction of motion of the piston, draw OC_2 parallel to the crank at cut-off, and produce it through O towards P. Draw a line GH parallel to AO and at a distance from it equal to the given lead. Bisect the angle GHP. The centre of the lap circle is on this bisecting line, and its distance from H is found by sweeping an arc with centre O and radius OD equal to the half travel. When D is found the circle is drawn touching GH and PH, and its radius gives the required lap. By joining D with O we have the angle AOD, which gives the angular advance.
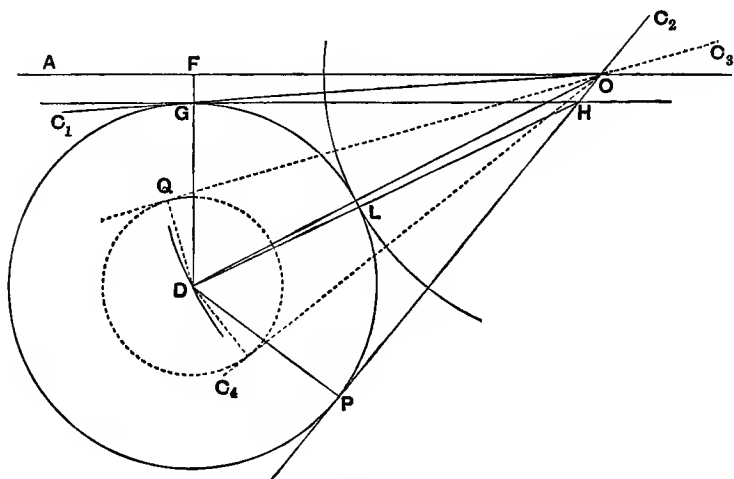
Fig. 132.

(2) Given the position of the crank at cut-off, the lead, and the greatest opening of the port to steam; to find the lap and the angular advance, and the travel of the valve. Draw OC_2 as before to represent the position of the crank at cut-off and produce it through O. Draw GH parallel to AO at a distance below it equal to the lead: with centre O and radius OL equal to the given greatest opening of the port draw an arc L. Then find a centre D such that a circle may be drawn touching this arc and the lines GH and OP: this is most conveniently done by trial. The radius of the circle gives the lap and AOD gives the angular advance as before.

22—2

196. Oval Diagram. A diagram is sometimes drawn which represents by a single curve the simultaneous displacements of the piston and the valve. When the position of the valve has been determined at various phases of the piston's stroke, whether by Reuleaux's or Zeuner's or any other method, a curve is drawn having for ordinates the displacement of the valve, on a base AB (fig. 133) which is the stroke of the piston, the scale of the ordinates being suitably exaggerated to prevent the curve from being
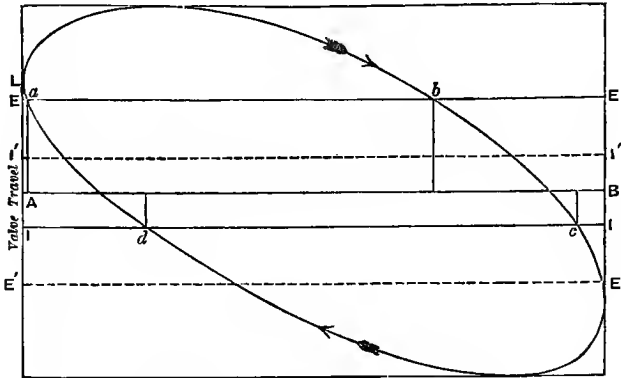
Fig. 133. Oval Diagram for the Slide-Valve.

inconveniently flat on account of the comparatively small amplitude of the valve's motion. This gives a species of oval figure resembling an ellipse, but somewhat distorted through the influence of the connecting-rod's obliquity. To find the events of the distribution, lines EE and II are drawn above and below the base at distances from it equal to the outside and inside laps respectively; their points of intersection with the curve at a, b, c and d mark the four events for the corresponding end of the cylinder. For the other end the outside-lap line $E'E'$ is to be drawn below the base and the inside-lap line $I'I'$ above it. The distance of the curve beyond the outside-lap line shows at any stage in the stroke the extent to which the steam port is then open. The lead, which is EL, is not well defined in this form of graphic construction.

197. Wave-form Diagram. A much more useful diagram is obtained by drawing (preferably on section paper) separate curves to represent the displacements of piston and valve respectively, each in relation to the angle turned through by the

crank-shaft, rectangular coordinates being used to represent the
crank-angle and the displacement. Taking a base (fig. 134) the
length of which represents the angle turned through in one
revolution, let the curve *ABCD* be drawn to represent by its ordi-
nates the displacement of the piston from mid-stroke, for all
positions of the crank. Similarly let a curve *EFIJ* be drawn with
ordinates which are (on any conveniently exaggerated scale) the
displacements of the valve. Owing to the angle between the
crank and the eccentric the phase of this curve is $90° + \theta$ in
advance of the other: in other words, the valve attains its maxi-
mum displacement at a point on the base-line $90° + \theta$ earlier than
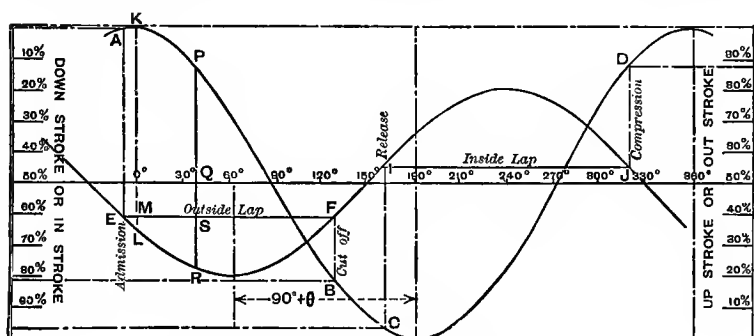
FIG. 134. Wave-form Diagram for the Slide-Valve.

(or to the left of) the point at which the piston attains its maxi-
mum displacement towards the same side. In drawing fig. 134 an
angular advance of 30° has been assumed, which makes the total
displacement of the valve curve to the left correspond to 120°.
When questions have to be considered regarding the effect of
varying the angular advance, one or other of the curves should be
drawn on tracing paper in order that it may readily be slipped
over the other into the position that will correspond to any desired
angle.

Let any line *PQR* be drawn perpendicular to the base-line to
intersect the piston curve in *P* and the valve curve in *R*. The
displacement of the piston is then *PQ* and that of the valve is (on
another scale) *QR*. The position of the piston in its stroke is
found by projecting *P* upon the end line of the diagram (to the
left) where a scale is marked to show percentages of the stroke.
If *EF* be drawn parallel to the base and at a distance below it equal

to the outside lap, SR, which is the excess of the valve's displacement
beyond the lap QS, gives the steam opening at the same phase of
the stroke.   Admission begins at E, and the corresponding position
of the piston is found by projecting E upon the piston curve at A
and then projecting A upon the scale at the side.   The vertical
distance from K to A shows the amount of preadmission.   At K,
the dead-point of the crank, the valve is open to the extent LM;
in other words, LM is the lead.   Cut-off occurs at F, and the
corresponding position of the piston is found by projecting F upon
the piston curve at B, and then projecting B upon the scale at the
side.   In the same way the positions of the piston at release and
compression correspond to the points I and J on the valve curve
when the line IJ is drawn at a distance above the base equal to
the inside lap.   All these events relate to one side of the piston;
to obtain the events for the other side the outside-lap line has to
be drawn above the base and the inside-lap line below it, and the
points found on the piston curve are to be projected upon the scale
which is set out on the right-hand side of the diagram in fig. 134.
The inequality of lap and lead which is needed to give a symmetrical
distribution, and other such problems of design, may be studied by
help of this diagram with great ease and clearness[1].   Another
example of its use will be given below in connection with separate
expansion valves (§ 208).

The ordinates of these curves may be found either by graphic
construction or by calculation.   As to the valve curve, the length
of the eccentric-rod is generally so great that its influence may
be neglected, and in that case the formula

$$y' = r' \cos \alpha'$$

may be used, r' being the eccentricity or the half-travel of the
valve and α' being the angle through which the eccentric has
turned from the position that corresponds to the maximum dis-
placement of the valve.

[1] The writer is indebted to Professor O. Reynolds for drawing his attention to
the advantages of the construction illustrated in fig. 134.   As a means of solving
slide-valve problems it is in several ways superior to the methods more generally
used by draughtsmen.   The labour of drawing the curves is considerable; but a
set of curves drawn once for all, for various ratios of crank to connecting-rod, will
allow any particular case of the simple slide-valve to be dealt with very readily.
Examples of the wave-form diagram as applied to link motions and other valve-
gears will be found in Professor Dalby's treatise on *Valves and Valve Gear
Mechanisms* (1906).

In the piston curve the influence of the rod is usually considerable. Let r be the effective length of the crank AP (fig. 135) and l that of the connecting-rod BP; when the crank has turned
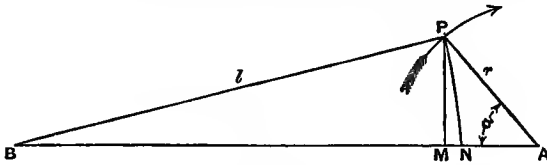
FIG. 135.

through any angle α from the dead-point the displacement of the piston from its middle position is

$$y = AN = AM + MB - l$$
$$= r\cos\alpha + \sqrt{l^2 - r^2\sin^2\alpha} - l,$$

or, writing μ for the ratio of the length of the connecting-rod to that of the crank,

$$y = r(\cos\alpha + \sqrt{\mu^2 - \sin^2\alpha} - \mu).$$

This is always less than $r\cos\alpha$, but approximates closely to that when μ is very great. An expression of the same form is of course applicable to the displacement of the valve and should be used when the eccentric-rod is so short as to require its length to be taken into account. The angles α (for the crank) and α' (for the valve) are connected by the equation $\alpha' = \alpha + 90° + \theta$ where θ is the angular advance as before.

198. Reversing Gear. The Link-motion. In locomotives, marine engines, winding engines, traction engines and some other types it is necessary to make provision for reversing the direction in which the engine runs. A primitive way of doing this is to shift the eccentric of the slide-valve round upon the shaft until it takes relatively to the crank the angular position proper to the reversed motion. The eccentric must stand in advance of the crank by an angle equal to $90° + \theta$, and if its position be CE (fig. 136) while the crank is at CK the engine will run in the direction of the arrow A. To set the engine in gear to run in the
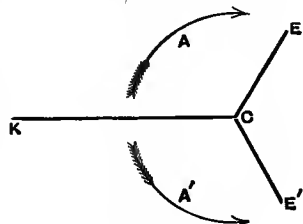
FIG. 136.

opposite direction it is only necessary to shift the eccentric into the position *CE'*, when it will still be in advance of the crank by the proper angle, the direction of motion now being that shown by the arrow *A'*. In some of the older engines this was substantially the actual method of reversal. The valve-rod was temporarily disengaged from the eccentric and the valve was moved by hand in such a way as to make the engine begin to turn backwards. It was allowed to turn until the crank had moved back through an angle equal to *ECE'*, the eccentric meanwhile remaining at rest, and the valve-rod was then re-engaged. To allow the eccentric to remain at rest while the crank turned back through the required angle, the eccentric sheave instead of being keyed to the shaft fitted loosely on it and was driven by means of a spur fixed to the shaft which abutted on one or other of two stops or shoulders projecting from the sheave. Consequently when the engine-shaft began to turn backwards the eccentric sheave did not at once follow it, until it had turned through an angle corresponding to the distance between the two stops. This device of the loose eccentric is not entirely obsolete[1], but nearly all modern engines which require reversing gear use either the *link-motion* or one of the forms of *radial gear* to be presently described.

In the link-motion there are two eccentrics keyed to the shaft in positions which correspond to *CE* and *CE'* in fig. 136, and the ends of their rods are connected to the ends of a link which gives its name to the contrivance. In Stephenson's link-motion— the earliest and still the most usual form—the link is a slotted bar or pair of bars forming a circular arc with radius equal or nearly equal to the length of the eccentric-rods (fig. 137), and capable of being shifted up and down by means of a pendulum rod to which it is jointed either at one end or at the middle of the link. This suspension by a pendulum rod also allows the link to move sideways as the eccentrics revolve.

The valve-rod ends in a block which slides within the link, and when the link is placed so that this block is nearly in line with the forward eccentric-rod (*R*, fig. 137) the valve moves in

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[1] It has been applied for instance in modern times to the low-pressure valve of some compound locomotives. In this case there is no need to disengage the rod for the engine is made to begin running backwards by the action of the high-pressure cylinders.

nearly the same way as if it were driven directly by a single eccentric. This is the position in "full forward gear." In "full backward gear," on the other hand, the link is pulled up until the block is nearly in line with the backward eccentric-rod R'. The
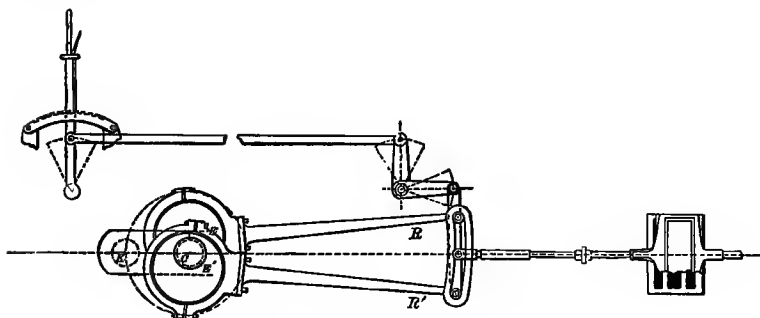
FIG. 137. Stephenson's Link-motion.

link-motion thus gives a ready means of reversing the engine,—but it does more than this. By setting the link in an intermediate position the valve receives a motion nearly the same as that which would be given by an eccentric of shorter throw and of greater angular advance, and the effect is to give a distribution of
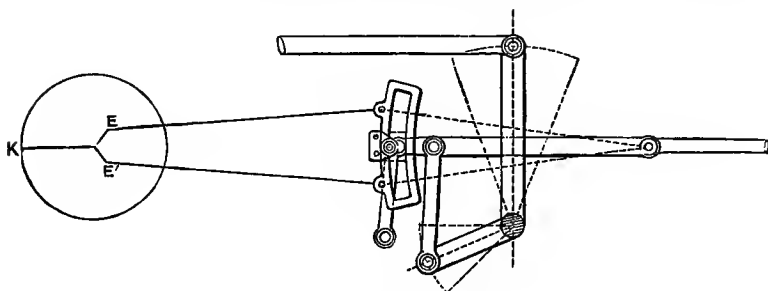
FIG. 138. Gooch's Link-motion.

steam in which the cut-off is earlier than in full gear, and the expansion and compression are greater. Hence the mechanism also serves to adjust the amount of work done in the cylinder to the demand which may at any time be made upon the engine. In mid gear, which is the position sketched in the diagram, the steam distribution is such that scarcely any work is done in the cylinder. The movement of the link is effected by a hand lever, or by a screw, or (in large engines) by an auxiliary steam-engine.

A usual arrangement of hand lever, sketched in fig. 137, has given rise to the phrase "notching up," to describe the setting of the link to give a greater degree of expansion, by bringing it nearer to mid gear. The eccentric-rods are sometimes crossed instead of being " open " as shown in the sketch.

In Gooch's link-motion (fig. 138) the link is not moved up in shifting from forward to backward gear, but a radius rod between the valve-rod and the link (which is curved to suit the length of this radius rod) is raised or lowered—a plan which has the advantage that the lead is the same in all gears. In Allan's straight
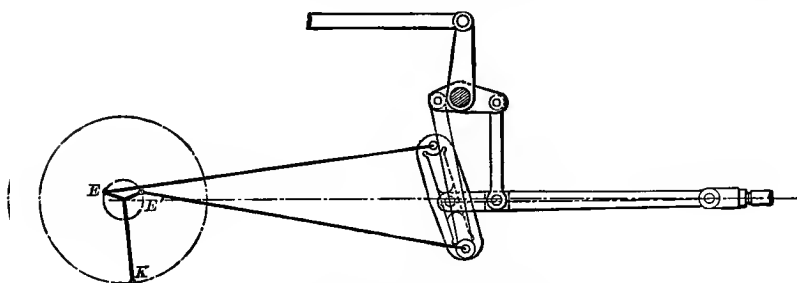
Fɪɢ. 139.   Allan's Link-motion.

link-motion (fig. 139) the change of gear is effected partly by shifting the link and partly by shifting a radius rod. This allows the link to be straight and has also the advantage that the weight of the link with its eccentric-rod on the one hand, and the radius rod on the other, can be arranged to balance one another when a suitable proportion is given to the two arms of the short beam from which the pendulum rods hang. Compared with the Stephenson motion it requires much less room for the play of the link between its extreme positions. The Allan motion has been fitted to many locomotives.

199. Graphic Solution of the Link-motion. The movement of a valve driven by a link-motion may be fully and exactly investigated by drawing with the aid of a template the positions of the centre line of the link corresponding to a number of successive positions of the crank. Thus, in fig. 140, two circular arcs passing through e and e' are drawn with E and E' as centres and the eccentric-rods as radii. These arcs are loci of two known points of the link, and a third locus is the circle a in which the

point of suspension must lie. By placing on the paper a template of the link, with these three points marked on it, the position of the link is readily found, and by repeating the process for other positions of the eccentrics a diagram of positions (fig. 140) is drawn for the assigned state of the gear. A line AB drawn across this diagram in the path of the valve's travel determines the displacements of the valve, and enables the wave-form diagram to be
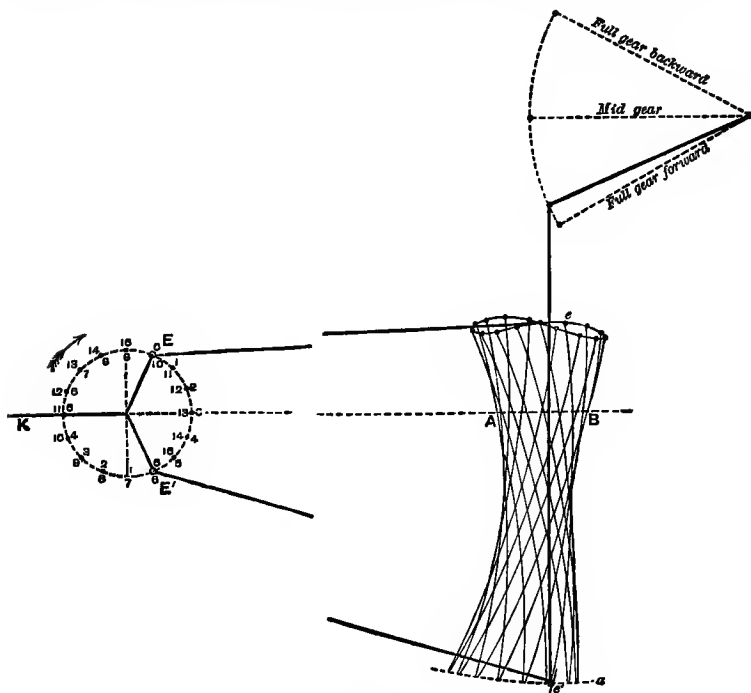
Fig. 140.

drawn as in fig. 134, or alternatively the oval diagram as in fig. 133. The example refers to Stephenson's link-motion in nearly full forward gear; with obvious modification the same method may be used in the analysis of Gooch's or Allan's motion. The same diagram serves to determine the amount of sliding motion of the block in the link. In a well-designed gear this sliding is reduced to a minimum for that position of the gear in which the engine runs most usually. In marine engines the suspension-rod is generally connected to the link at that end of the link which is next the forward eccentric, in order to reduce this

sliding as much as possible when the engine is running in its
normal condition, namely, in forward gear.

200.   Equivalent Eccentric. A less laborious, but less
accurate, solution of link-motion problems is reached by the use
of what is called the equivalent eccentric—an imaginary single
eccentric, which would give the valve nearly the same motion as
it gets from the link under the joint action of the two actual
eccentrics. The following rule for finding the equivalent eccentric,
in any state of gear, is due to Mr MacFarlane Gray:—

Connect the eccentric centres E and E' (fig. 141) by a circular
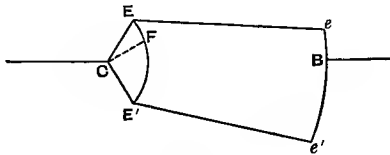
FIG. 141.

arc whose radius $= \dfrac{EE' \times \text{length of eccentric-rod}}{2 \times ee'}$, ee' being the
length of the link from rod to rod. Then, if the block is at
any point B, take EF such that $EF : EE' :: eB : ee'$. CF then
represents the equivalent eccentric both in radius and in angular
position. If the rods of the link-motion are crossed instead
of open,—an arrangement seldom used,—the arc EFE' is to
be drawn convex towards C. Once the equivalent eccentric
has been found the movement of the valve may of course be
determined by Zeuner's or any of the other methods already
described. The method of the equivalent eccentric should not be
taken as giving more than a first approximation to the actual
motion; for anything like a complete study of a link-motion the
graphic method of § 199 or the use of a model is to be preferred[1].

**201.   Resolution of the Valve Displacement into two
Components.** It is instructive in this connection to think of the
action of a single eccentric in driving a slide-valve as made up of
two components, due to two imaginary eccentrics one of which is
set opposite to the crank and the other at right angles to it. Thus

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[1] See Professor Dalby's treatise on *Valves and Valve Gear Mechanisms*.

in fig. 142 the actual eccentric CE gives the valve a motion which is the same as that which would be got by compounding the motions of two eccentrics CM and CN. Hence in the wave-form diagram we might draw separate curves for the two component eccentrics CM and CN, and by adding their ordinates we should get a curve which is the curve given by the actual eccentric CE. The characteristic, then, which must be possessed by any mechanism for
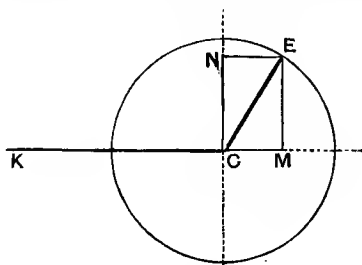
FIG. 142.

driving a slide-valve is that the displacement of the valve must have what we may call for brevity a component M, which differs in phase from the motion of the crank by 180°, and a component N which differs in phase from the motion of the crank by 90°. Further the maximum amount of the M component, namely the length of the imaginary eccentric CM or the value of CM when the crank is in the initial position sketched, is equal to the outside lap *plus* the lead.

Apply this idea to a reversing gear such as a link-motion driven by two eccentrics E and E', fig. 143, and we see that in shifting from forward to backward gear the M component is not altered, but the N component is first reduced, passes through a zero value, and then becomes negative.

In the mid-gear position the distribution of steam is that which would be given by the M component alone, and we approach full gear, in either direction, by introducing more and more of the N component. The amount of steam admission increases with the increase of the N component until full forward gear or full backward gear is reached.

What is wanted therefore in any form of reversing gear is a mechanism which will allow the valve to keep, unchanged, a component of reciprocating motion due to M directly opposite to the

motion of the piston, and will superpose on that another component due to N differing 90° from it in phase, which can be varied in magnitude and reversed in sign. The link driven by two eccentrics CE and CE' in any of the three arrangements which have been described offers one means of solving the problem: but there are other means in some of which only one actual eccentric is used and in some none at all. These go by
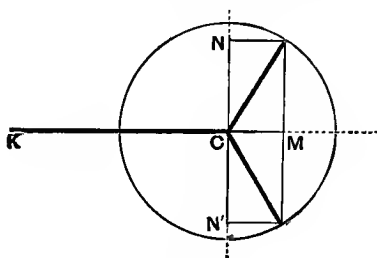
Fig. 143.

the general name of "radial gears." Like the link-motion they supply a means of varying the output of power as well as reversing the direction of running. The action of these gears becomes readily intelligible when it is recognized that in every one of them the function of the mechanism is to combine a motion due to a constant component eccentric M, at 180° from the crank, with that due to a component N at 90° from the crank which can change continuously from positive to negative through zero.

202. Radial Gears. Hackworth's. Of these Radial Gears the oldest is Hackworth's, which was patented in 1859, and has been the parent of several others. It dispenses with one of the two eccentrics which are used in an ordinary link-motion, but retains an eccentric which is set directly opposite the crank and gives the valve the M component of the motion. Referring to the skeleton diagram, fig. 144, E is the eccentric, set at 180° from the crank, which drives an eccentric-rod EQ, the mean position of which is perpendicular to the travel of the valve. This rod ends in a block Q, which slides on a fixed inclined guide-bar or link, and the valve-rod receives its motion through a valve connecting-rod from an intermediate point P of the eccentric-rod, the locus of which is an ellipse. To reverse the gear the guide in which Q

moves is tilted over to the position shown by the dotted lines, and intermediate inclinations give various degrees of expansion without altering the lead. When the guide is set at mid gear, or pointing directly towards the crank centre, the valve receives a motion which consists of the M component alone. Looking on Q as a fulcrum we see that P is then giving to the valve a motion which is the same, but on a reduced scale, as would be given by the direct action of the eccentric E. Under these conditions there is no N component. But by inclining the guide the N component is introduced. The up and down sliding movement of the block Q, which takes place with a phase differing by 90° from the crank,
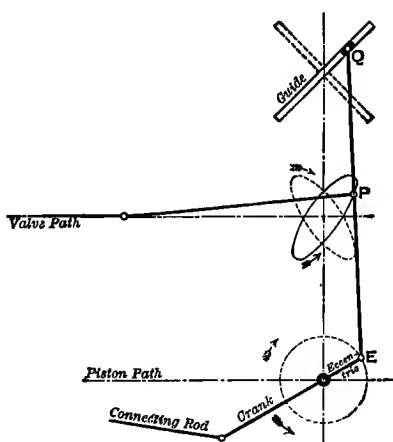
Fig. 144. Hackworth's Radial Valve-Gear.

does not contribute to the motion of the valve so long as the guide is not inclined, but does contribute as soon as the guide begins to be inclined, for the sliding motion of Q then has a lateral component, which is shared by P to an extent depending on the proportion of lengths PE and QE. This is distinct from the lateral motion which P receives in the former case from the eccentric alone, and differs in phase from it by 90°. Thus with the guide inclined we have the two necessary components M and N, the amplitude of the M component being constant while that of the N component ranges from a positive to a negative maximum according to the inclination of the guide.

An objection to this arrangement of Hackworth's gear is the wear of the sliding block Q and its guide. In a modified form

this objection is obviated with some loss of symmetry in the valve's motion by constraining the motion of the point Q, not by a sliding-guide as in fig. 144, but by a suspension-link, which makes the path of Q a circular arc instead of a straight line. To reverse the gear the centre of suspension R of this link is thrown over to the position R' (fig. 145). The same figure also serves to illustrate another modification of the Hackworth gear, namely, the placing of P beyond Q, with no angle between the crank and the eccentric; but P may be between Q and the crank (as in fig. 144), in which case the eccentric is
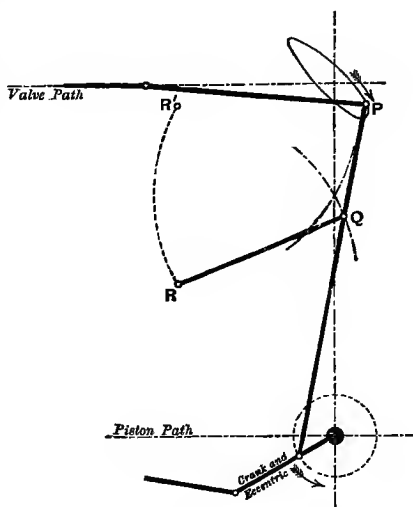
Fig. 145.

set at 180° from the crank. When P is beyond Q the angle between eccentric and crank is zero, for the eccentric-rod oscillating on a lever about the fulcrum Q itself introduces the 180° difference of phase which is required in the component M. The radius of the eccentric and the ratio of the distances to P and Q are taken such that the maximum value of the M component is equal to the lap plus the lead. By arranging the centre about which the guide-bar (fig. 144) rocks so that it coincides with the position of Q when the crank is at the dead-point we secure the condition that the lead does not alter during the shifting of the gear.

The Hackworth gear has been extensively applied to marine engines, in forms designed by Marshall, Bremme, and others. Fig. 146 illustrates an arrangement by Mr Marshall[1] for a marine engine of the usual vertical type. K is the crank turning about C, V the valve-rod, E the eccentric centre, EPQ the eccentric-rod, and RQ the suspension-rod, which rocks about a centre R. To reverse the gear R is shifted to R' by the screw gear shown at the side.

[1] *Proc. Inst. Mech. Eng.* 1880.
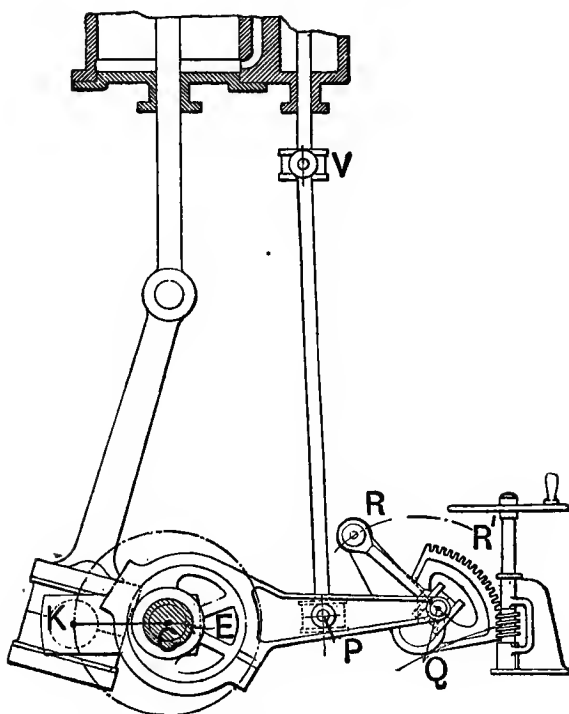
Fig. 146.   Marshall's form of Hackworth Gear.

203.   The Walschaert or Heusinger-Waldegg Gear.   In this gear, which is so largely used on continental locomotives as to be their normal type of reversing gear, one eccentric is dispensed with.   The M component is derived from the cross-head, through a lever which serves to reduce the motion and also to change its phase by 180°.   The N component is obtained from an eccentric set at 90° from the crank in a manner which will be understood by reference to the skeleton diagram, fig. 147.   There A is a projection fixed to and moving with the cross-head.   It is connected to the valve-spindle V by the rod AB and the lever BV, which is pivoted at D on another rod DG.   Regarding D for the moment as a fulcrum, it will be seen that this system gives to V a component of motion proportional to that of the cross-head, but on a reduced scale, and with its sign reversed.   These are the proper conditions for the component M.   To provide the component N there is an eccentric CF, at 90° from the crank CK, the rod of which rocks a link HI

FIG. 147. Walschaert or Heusinger-Waldegg Gear.

FIG. 148.

about a fixed centre O. D is connected to the link by the rod DG, which has a sliding block at G capable of sliding over the whole length of the link, up to I. To reverse the gear this block is shifted to I, on the opposite side of the fixed centre O about which the link oscillates. The rod DG is shifted by means of the rod J which is connected to the notching up lever.

It will be obvious that the eccentric F provides the N component by giving to D and to V a movement in phase with the eccentric itself. When the block G is shifted to O, which is the position for mid gear, this component disappears, and when the block is at I it is negative: in other words the component is then such as would be given by an eccentric opposite CF, and the gear is in the position for running backwards. To secure constant lead during change of gear the link is curved to a radius equal to DG, so that, if the crank is kept at its dead point, D remains at rest while the gear is shifted. An example of the Walschaert gear applied to a locomotive is given in fig. 148 [1].

204. Joy's Gear. Another type of radial gear is Joy's, which has been considerably used both in locomotives and marine engines. In Joy's gear no eccentric is required, but it presents points of relationship to Hackworth's. The rod SQP corresponding to the eccentric-rod in Hackworth's gear, which supplies the M component of the valve's displacement, receives its motion from a point in the connecting-rod through the linkage OTU shown in fig. 149, and is either suspended by a rod whose suspension centre
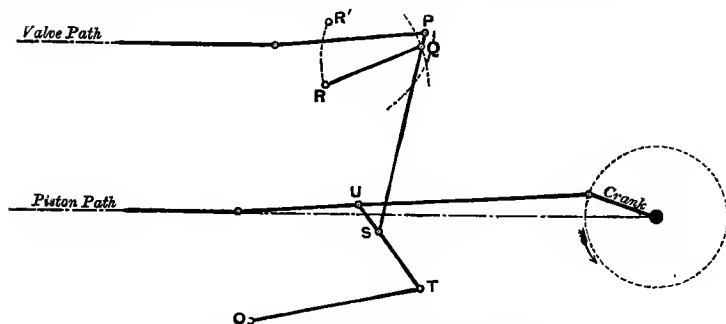
Fig. 149. Diagram of Joy's Valve-Gear.

[1] In practice the pin H by which the eccentric rod is connected to the rocking link is put some little distance behind the centre line of the link, nearer C, to correct for inequalities in the motion arising from the shortness of the rods, etc.

R is thrown over to R' to reverse the motion, or constrained, as in the original form of the Hackworth gear, by a slot-guide whose inclination is reversed. The rod SQP takes its motion from the connecting-rod not directly but from a point S in the link UT. O is a fixed centre. Fig. 150 shows Joy's gear as applied to a locomotive. A slot-guide E is used, and it is curved to allow for
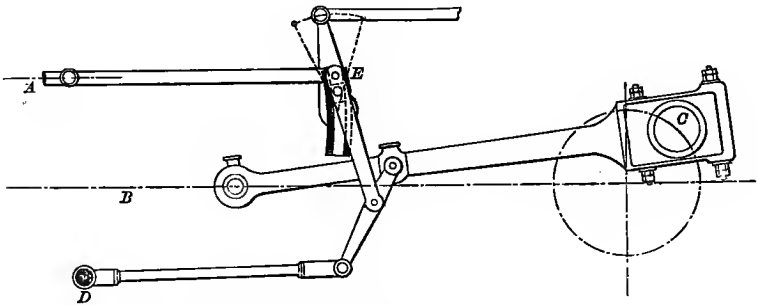
FIG. 150. Joy's Gear as applied to a Locomotive.

the obliquity of the valve connecting-rod AE. C is the crank-pin, B the line of the piston's motion, and D a fixed centre. We may regard the Joy gear as supplying the M component in much the same way as the Walschaert gear supplies it. If the gear is set in the mid position the M component alone is operative. Though the motion is taken from the connecting-rod the effect (for that position) is the same as if it were taken from the cross-head, as in Walschaert. But by taking it from the connecting-rod instead Joy gets a vertical sliding of the block in the link E (fig. 150) resembling the sliding motion in Hackworth, and consequently is able to introduce the N component by inclining the link.

It might be thought possible to take the motion of the rod SQP direct from the connecting-rod, by putting S at U, but it will be found that this would cause the valve to have a far from symmetrical motion about the mean position, on account of the obliquity of the main connecting-rod.

Reviewing these various radial gears, we see that Hackworth retains one eccentric, getting the M component from it almost directly, and getting also the N component from it indirectly by the device of an inclined sliding guide or its equivalent. Walschaert also retains one eccentric, but gets only the N component from it, through the medium of a link which serves to

vary the amount and reverse the direction of the motion so taken. He gets the M component almost directly from the cross-head, using a lever only to reduce the motion and also to change its phase by 180°. Joy uses a point on the connecting-rod to supply both components, the M component more or less directly, as in Walschaert, and the N component indirectly as in Hackworth by the same device of an inclined sliding guide or its equivalent.

205. Reversing by Interchange of Steam and Exhaust. Steam Steering Engines. An engine fitted with the most primitive type of slide-valve, namely, a valve without lap or lead, and therefore with its eccentric set without angular advance, will run backwards if we reverse the usual connections of the steam and exhaust pipes, by supplying steam to the central cavity of the valve, and allowing exhaust to take place from the outside of the valve. We have only to open what is usually the exhaust side to steam, and what is usually the steam side to exhaust. This can be done by a separate reversing valve on the steam and exhaust pipes, which may itself conveniently take the form of a slide-valve. When this valve is moved over by hand the engine has its direction of running reversed. The arrangement does not admit of expansive working, owing to the absence of lap: it gives admission of steam throughout the whole stroke. But its simplicity makes it useful in certain cases where economy of steam is comparatively unimportant.

A notable instance is in steam steering engines. There the function of the engine is to form a mechanical relay between the helmsman and the rudder, making it possible for the movement of the steering wheel to be effected with scarcely any force, while the rudder, which requires much force, is moved to a corresponding extent. A movement of the wheel in either direction opens a reversing valve of the kind just spoken of, and sets the steering engine in motion in the corresponding direction. This turns the rudder, but the turning has to stop when the rudder has gone over to the proper angle, depending on the extent to which the helmsman has turned his wheel. Accordingly the steering engine is fitted with what is called a "hunting gear," which is a device for automatically stopping it after the proper amount of movement has been effected. There are many forms of such gear, but all act on the general principle that the reversing valve, which has

been shifted by the movement of the steering wheel, is brought back to its central position by the action of the engine itself, though the helmsman continues to hold his wheel over. The result is that when the engine has forced the rudder over to the proper angle it stops, and makes no further movement until the helmsman alters the position of the wheel. When he does so the steering engine at once responds, turning one way or the other according as the helmsman gives the ship more helm or less. Thus the movements of the rudder are kept in complete correspondence with those of the wheel, and the resistance to be overcome by the steersman is no more than the friction of a small valve.

The hunting gear often consists of a screwed sleeve or nut on the spindle of the reversing valve. This nut is geared to the steering engine, so that it revolves so long as the engine is running. It gradually works the spindle back after it has received, from the steering wheel, a movement which has set the engine in motion, and the engine consequently stops after displacing the rudder to a corresponding extent.

206. Operating Reversing Gears. In small engines fitted with a link-motion or similar reversing gear the gear may be shifted by hand either directly by a lever or with the aid of a screw or worm, but in large engines steam power is employed to do the work.

In Brown's reversing apparatus there are two cylinders arranged in tandem with a single piston-rod common to both, the movement of which pushes over the gear. One of the two is a steam cylinder to either end of which steam can be admitted, while the other end is put in communication with the exhaust. The other cylinder is a hydraulic brake, the function of which is to prevent the movement of the piston-rod from being jerky. A control valve, operated by hand, admits steam to the steam cylinder of the apparatus, and is fitted with a hunting device, of the kind mentioned in the preceding paragraph, with the effect that whatever amount of movement is given to the hand control is accurately reproduced in the movement of the steam piston of the apparatus, which is brought to rest in a position corresponding to the position of the hand control, any further supply of steam being then cut off by the action of the hunting device.

Reversing gears are frequently operated by what is called an "all round" power mechanism. The weigh-bar which moves the link is driven from a pin on a worm wheel which may be completely revolved. A half revolution of the wheel shifts the gear from full gear ahead to full gear astern; the next half revolution shifts it back to full gear ahead. The wheel is revolved by a tangent screw which may be turned by a hand wheel or by a small auxiliary steam-engine. By making the wheel revolve continuously the gear may go on being rapidly shifted from full gear ahead to full gear astern and back again, an arrangement which is found convenient by marine engineers when waiting orders from the bridge or when warming the engines before getting under way.

207. Separate Expansion-valves. When the distribution of steam is effected by the slide-valve alone the arc of the crank's motion during which compression occurs is equal to the arc during which expansion occurs, and for this reason the slide-valve would give an excessive amount of compression if it were made to cut off the supply of steam earlier than about half-stroke. Hence, when an early cut-off is wanted it is necessary either to use an entirely different means of regulating the distribution of steam, or to supplement the slide-valve by another valve,—called an expansion-valve, and usually driven by a separate eccentric,—whose function is to effect the cut-off, the other events being determined as usual by the slide-valve. Such expansion-valves belong generally to one or other of two types. In one, which is much the less common, the expansion-valve cuts off the supply of steam to the chest in which the main valve works. This may be done by means of a disk or double-beat valve (§ 214), or by means of a slide-valve working on a fixed seat (furnished with one or more
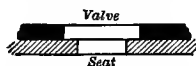
FIG. 151.
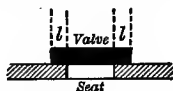
FIG. 152.

ports) which forms the back or side of the main valve-chest. Valves of this last type are usually made in the "gridiron" or many-ported form to combine large steam-opening with small travel. Expansion-valves working on a fixed seat may be arranged so that the ports are either fully open (fig. 151) or closed (fig. 152)

when the valve is in its middle position. In the latter case the expansion-valve eccentric is set in line with or opposite to the crank, if the engine is to run in either direction with the same grade of expansion. Cut-off then occurs when the crank is at P (fig. 153), the expansion eccentric being at P', the shaft having
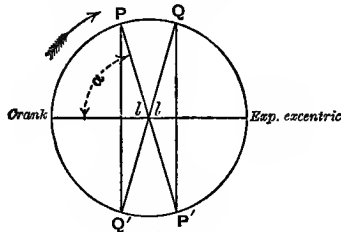
FIG. 153.

turned through an angle α from the beginning of the stroke. This is because the valve is then within a distance equal to l (fig. 152) of its middle position. The expansion-valve reopens when the crank is at Q, and the main slide-valve must therefore have enough lap to cut off earlier than $180° - \alpha$ from the beginning of the stroke, in order to prevent a second admission of steam to the cylinder. In the example shown in fig. 153 the expansion eccentric is set $180°$ in advance of the crank, which is a usual arrangement when the engine is provided with reversing gear, since it makes the cut-off happen at the same place in the stroke for both directions of running. If this condition need not be fulfilled, the expansion eccentric may have a somewhat different angular position, and in this way a more rapid travel at the instant of cut-off may be secured for one direction of running.

Since the separate expansion-valve of fig. 151 or 152 acts by cutting off the supply of steam from the steam-chest, but not directly from the cylinder, it does not prevent the steam which is stored in the chest from continuing to enter the cylinder until the main slide itself closes the admission port. When the cut-off by the expansion-valve is early and the steam-chest is capacious this affects the action materially.

208. Meyer's Expansion-valve. The other and much commoner type of expansion-valve is known as Meyer's. It consists of a pair of plates sliding on the back of the main

slide-valve, which is provided with through ports which these plates open and close. Fig. 154 shows one form of this type. Here it is the relative motion of the pair of plates forming the
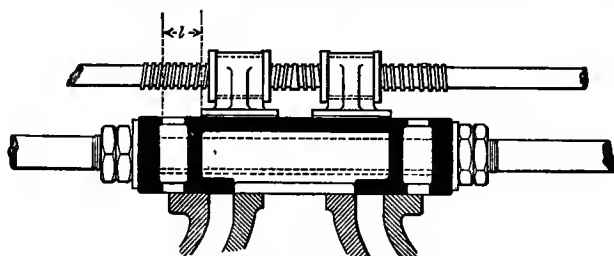
FIG. 154. Meyer Expansion-valve.

expansion-valve with respect to the main valve that has to be considered. If r_a and r_b (fig. 155) are the eccentrics working the main and expansion-valves respectively, then CR drawn equal and parallel to ME is the *resultant* eccentric which determines the motion of the expansion-valve relatively to the main valve. Cut-off occurs at Q, when the shaft has turned through an angle α, which brings the resultant eccentric into the direction CQ and makes the relative displacement of the two valves
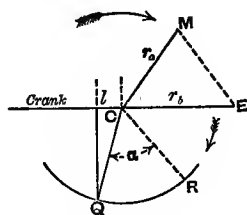
FIG. 155.

equal to the distance l. Another form of this valve (corresponding to the fixed-seat form shown in fig. 152) cuts off steam at the inside edges of the expansion-slides. With the form shown in fig. 154 the expansion eccentric will be set at 180° from the crank if the engine is to run in both directions with the same grade of expansion; otherwise a somewhat different angle may often be chosen with advantage, as giving a sharper cut-off.

The action of Meyer's valve may be conveniently examined by the help either of Zeuner's diagram or of the wave-form diagram of § 197. Taking Zeuner's diagram first and assuming, for greater generality, that the expansion eccentric is not set just opposite the crank, let the circles I. and II. (fig. 156) be drawn to show, as in fig. 128, by the lengths of their chords through C the amount of absolute displacement of the expansion-slide and main slide respectively each from its middle position, when the crank is in the angular position corresponding to the direction of the chord.

In drawing these circles the angle XCM is set out to represent the whole angle by which the main valve eccentric is ahead of the crank, as in fig. 128, and the angle XCE (also measured against the direction of the arrow) represents the angle by which the expansion eccentric is ahead of the crank, CM and CE being diameters of the circles II. and I. respectively. Then if any chord CQP be drawn from C to meet both circles the distance PQ,
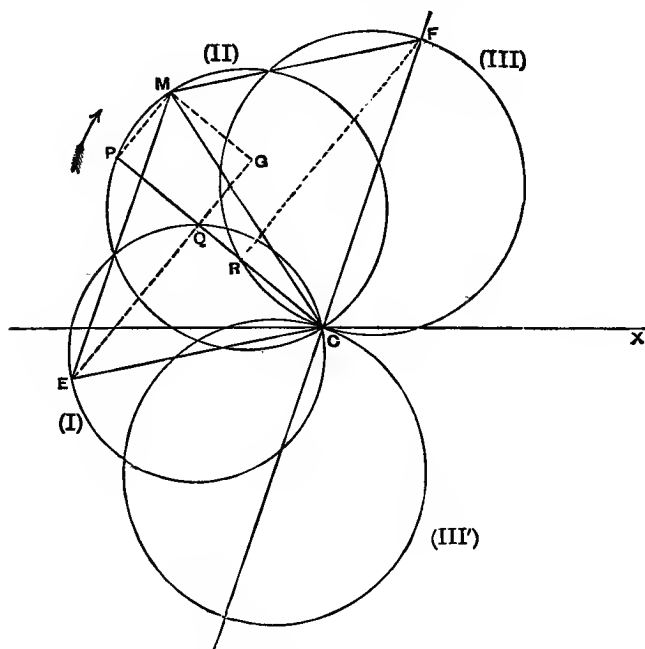
Fig. 156. Application of Zeuner's Diagram to the Meyer Expansion-valve.

which is the difference between CP the absolute displacement of the main valve and CQ the absolute displacement of the expansion-valve, measures the relative displacement of one with respect to the other. This distance PQ is equal to the chord CR of a third circle (III.) drawn with CF equal and parallel to EM as its diameter. To prove this make the supplementary construction shown by the dotted lines. Then since the angles at P, Q, R and G are right angles, being angles in semicircles, PQ equals MG in the parallelogram PG, and RC equals MG in the equal triangles EMG and FCR. Hence PQ equals RC. A similar construction

of course applies to the return stroke, for which the circle showing the resultant motion is III'.

Thus by drawing the circles III. and III'. we at once determine, by the length of their chords through C, the relative displacement of the two valves for all positions of the crank. Cut-off, on the part of the expansion-valve, occurs when the crank is in such a position that the chord CR is equal to l (fig. 154), which is the amount of relative displacement that suffices to close the steam passage through the main slide. The expansion-valve reopens when the chord is again diminished to this value, towards the end of the stroke, and care must be taken that the main slide has enough outside lap to close the steam port leading into the cylinder before this stage in the revolution has been reached.

When the expansion-valve is furnished with a means of varying l, as in fig. 154, where right- and left-handed screws enable the blocks which make up the expansion-valve to be brought nearer together or separated by turning the spindle, the point of cut-off may be made to take place early or late, the limit of earliness being imposed by the condition that l must not be reduced below the amount which will give a fair steam opening, and the limit of lateness being imposed by the consideration that the main slide itself becomes closed at a position determined by its own outside lap. The events of release, compression and admission, depending as they do on the main slide-valve alone, are found by drawing lap arcs on the main-valve circle I. in the same manner as in earlier examples of Zeuner's diagram.

The wave-form diagram of § 197 gives an excellent means of studying the action of Meyer's valve. Three distinct curves having been drawn for the piston, main valve and expansion-valve respectively, showing the displacement of each in relation to the angle turned through by the crank-shaft, they are to be superposed as in fig. 157 (using tracing paper as before) with the proper differences of angular position set out by distances measured along the base-line between the points at which the maximum displacement towards one side occurs in each. Both valve curves must have the same scale. Then the relative displacement of the valves is everywhere shown by the vertical distance between the main-valve curve and the expansion-valve curve. Cut-off is made to occur at any desired place in the

motion by making the quantity l of fig. 154 equal to the distance
found by measurement between the two valve curves at the
corresponding point of the base-line. Thus in fig. 157, if it is
wished to cut off steam when the piston has travelled $25°/_\circ$ of its
stroke, the corresponding point P is found by projection from the
scale at the side, PR is drawn and the intercept TR is measured:
this determines the proper length of the "lap" l. With a smaller
"lap" the cut-off comes earlier, and in the particular example
shown in the figure the admission may be reduced to $10°/_\circ$ of the
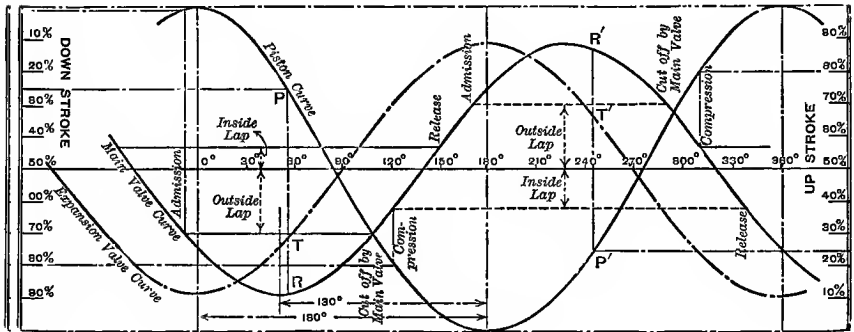
Fig. 157.

stroke, or even less, by reducing l. The diagram[1] relates to a
case in which the expansion eccentric is set at $180°$ in advance
of the crank and the main-valve eccentric at $130°$ (making $\theta = 40°$).
Both eccentrics have the same throw, giving a travel of 1.55 inches
to each valve. The main valve has an outside lap of 0.4 inch
on both sides: this gives equal amounts of lead, namely 0.1 inch,
but would make the cut-off unequal on the two sides, namely at
$70°/_\circ$ of the in-stroke or down-stroke and at $62°/_\circ$ of the out-stroke
or up-stroke, if the cut-off depended on the main valve. Since
the cut-off is accomplished earlier, by means of the expansion-
valve, this inequality does not matter. The inside laps of the
main valve are made unequal so that they give the same com-
pression on both sides, namely by stopping the exhaust at $80°/_\circ$
in each back-stroke; their values, found by projection from points
at $80°/_\circ$ on the stroke scale, are 0.24 inch on the front or bottom
side and 0.14 inch on the back or top side.

[1] Drawn by Prof. Dalby for a small vertical experimental engine in the
Engineering Laboratory at Cambridge.

By measuring distances such as TR between the two valve curves it will be seen that equal cut-off on the two sides can only be secured by having different values of l at the two ends of the valve. Thus TR is 0·33 inch and $T'R'$, which also corresponds to a 25 °/$_o$ cut-off, is 0·42 inch, or nearly one-tenth of an inch more. A constant difference between the values of l at the two ends is in fact preserved in Meyer's gear while the values of l are varied, and in this case the diagram shows that a constant difference of about one-tenth of an inch suffices to keep the points of cut-off practically symmetrical from say 10 °/$_o$ to 35 °/$_o$ of the stroke. When the cut-off is to be later than 35 °/$_o$ equality can only be preserved by reducing slightly the difference between the values of l. The difference between them can be varied in practice by shifting the expansion-valve bodily towards or from its eccentric, provided the valve-spindle in the eccentric-rod be furnished with a screw coupling or other device which permits its length to be altered.

The alteration of the expansion by varying the "lap" l is accomplished in the ordinary form of Meyer's valve in a way which will be evident on reference to fig. 154. The valve-rod has right- and left-handed screws on it working in nuts which control the longitudinal positions on the rod of the two blocks that make up the expansion-valve. Hence by rotating the rod the blocks are made to approach or recede from each other, thus increasing or reducing the lap l at each end, but leaving any difference between the laps at the two ends unchanged. Matters are generally arranged so that this adjustment can be made while the engine is running, by means of a sleeve and hand-wheel which are usually fitted on a prolongation of the valve-rod through the back end of the steam-chest. The cut-off may also be varied by altering the travel of the expansion-valve, instead of its lap. In some examples of the Meyer gear the expansion is varied automatically to suit the varying load upon the engine, the governor being connected to the expansion-valve in such a way that either the lap or more commonly the travel is varied in response to variation in the speed. When the travel is to be altered a link, oscillating about a fixed centre, is interposed between the valve-rod and the eccentric-rod, and by sliding the end of the eccentric-rod up or down in the link, the link is made to act as a lever of variable length.

In a modified form of this valve, known as Rider's, the

expansion-valve is a species of piston working in a cylindrical hole bored out of the main valve. The steam passages terminate in a pair of oblique slots within this hole, and the front and back edges of the piston-shaped expansion-valve are also cut obliquely, with the result that when the valve is turned about its axis its edges approach or recede from the oblique slots which form the steam ports. This turning can be effected by the governor.

209. Forms of Slide-valves. Double-ported valve. Trick valve. In designing a slide-valve the breadth of the steam ports in the direction of the valve's motion is determined with reference to the volume of the exhaust steam to be discharged in a given time, the area of the ports being generally such that the mean velocity of the steam during discharge is less than 100 feet per second. The travel is made great enough to keep the cylinder port fully open during the greater part of the exhaust; for this purpose it is $2\frac{1}{2}$ or 3 times the breadth of the port. To facilitate the exit of steam the inside lap is always small, and is often wanting or even *negative*, especially in engines which are designed to run at a high speed. During admission the steam port is rarely quite uncovered when the valve reaches the end of its travel, particularly if the outside lap is large and the travel moderate. Large travel has the advantage of giving freer ingress and egress of steam, with more sharply-defined cut-off, compression, and release, but this advantage is secured at the cost of more work spent in moving the valve and more wear of the faces[1]. To lessen the necessary travel without reducing the area of steam ports, double-ported valves are often used, and occasionally there are even three ports at each end. An example of a double-ported valve is shown in fig. 159. Fig. 158 shows the Trick valve, a device which accomplishes the same purpose by giving simultaneous admission in two ways; steam enters directly past the outer edge, as in an ordinary slide-valve and at the same instant an opening at the other end
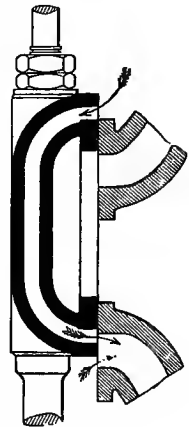
Fig. 158.
Trick Valve.

[1] For an experimental investigation of the friction of locomotive slide-valves see a paper by Mr J. A. F. Aspinall, *Min. Proc. Inst. C. E.* Vol. xcv. 1888.

of the valve is uncovered by passing beyond the edge of the raised seat on which the valve works. This gives a supplementary admission, to the same cylinder port, through a passage cast in the back of the valve itself.
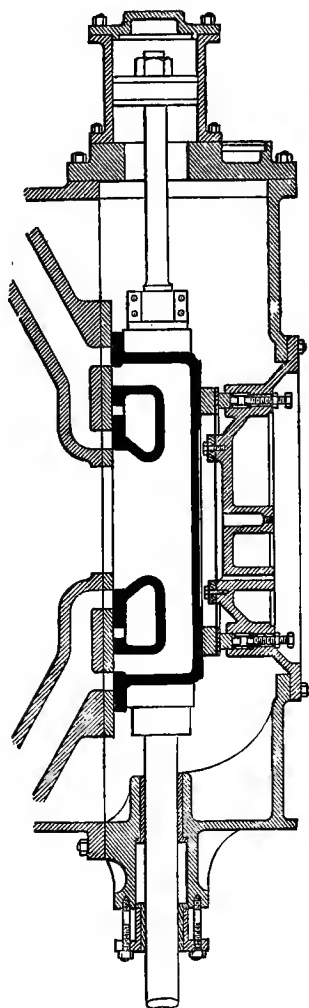
FIG. 159. Double-ported Valve with balance-piston and relief-frame.
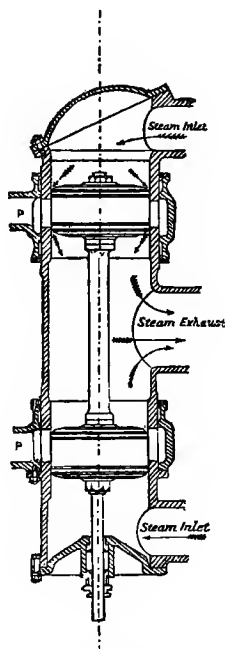
FIG. 160. Piston Slide-Valve.

210. Balance-Piston. Incidentally, fig. 159 illustrates an arrangement that is usual in heavy slide-valves whose travel is

vertical—the *balance-piston*, which is pressed up by steam on its lower side and so supports the weight of the valve, valve-rod, and connected parts of the mechanism. This weight may be as much as two or three tons in a very large engine. The space above the piston in the balance cylinder is kept in communication with the condenser.

211. Relief-Frames. To relieve the pressure of the valve on the seat, large slide-valves are generally fitted with what is called a *relief-frame*, which excludes steam from the greater part of the back of the valve. In a common form of relief-frame a ring fits steam-tight into a recess in the cover of the steam-chest, and is pressed by springs against the back of the valve, which is planed smooth to slide under the ring. Another plan is to fit the ring into a recess on the back of the valve, and let it slide on the inside of the steam-chest cover. Steam is in either case excluded from the space within the ring, any steam that leaks in being allowed to escape to the condenser (or to the intermediate receiver when the arrangement is fitted to the high-pressure cylinder of a compound engine). A flexible diaphragm is sometimes used to make a steam-tight partition between the back of the relief-frame and the cover of the valve-chest, and in that case the frame may take the form of a rectangular casting with a planed face, which remains at rest while the valve, the back of which is also planed, slides beneath it. Fig. 159 gives an example of a relief-ring fitted on the back of a large double-ported slide-valve for a marine engine.

212. Piston-Valves. The pressure of valves on cylinder faces is still more completely obviated by making the back of the valve similar to its face, and causing the back to slide in contact with the valve-chest cover, which has recesses corresponding to the cylinder ports and communicating with them. This arrangement is most perfectly carried out in the *piston slide-valves* now very largely used in the high-pressure cylinders of marine engines. The piston slide-valve may be described as a slide-valve in which the valve face is curved to form a complete cylinder, round whose whole circumference the ports extend. The pistons are often packed like ordinary cylinder pistons by split metallic rings, and the ports are crossed here and there by diagonal bars to keep the

rings from springing out as the valve moves over them. In some cases the packing rings are omitted, and in other cases uncut floating rings are used instead of split rings. Fig. 160 shows one
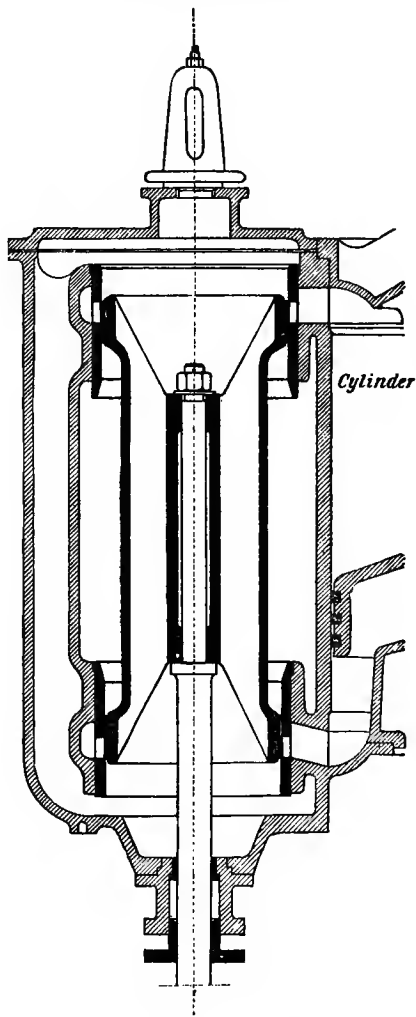
Fig. 161. Piston Slide-Valve.

form of piston-valve. *PP* are the cylinder ports, and the supply of steam reaches the valve through two distinct inlets at the top and bottom. In another and more common form of piston-valve

the rod connecting the two pistons is hollow and forms a communi-cation between the steam chambers above and below the valve, thus making one steam inlet suffice. An example of a hollow piston-valve for a marine engine is shown in Fig. 161. Steam is admitted to the outer casing and passes freely through the interior of the valve. The exhaust pipe (which does not appear in the diagram) opens on the inner casing, in the space between the two pistons of the valve.

213. Rocking Valves. The slide-valve sometimes takes the form of a rocking cylinder (fig. 162), the motion of which is produced by the turning of a spindle through the required arc, the valve being a block loosely set on a squared part of the spindle so that it is constrained to turn but at the same time is free to take a close bearing against the cylindrical port-face.
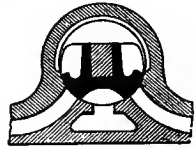
Fig. 162. Rocking Slide-Valve.

In some engines the distribution of steam, for each cylinder, is effected by two rocking valves, as in fig. 163, one for each end of
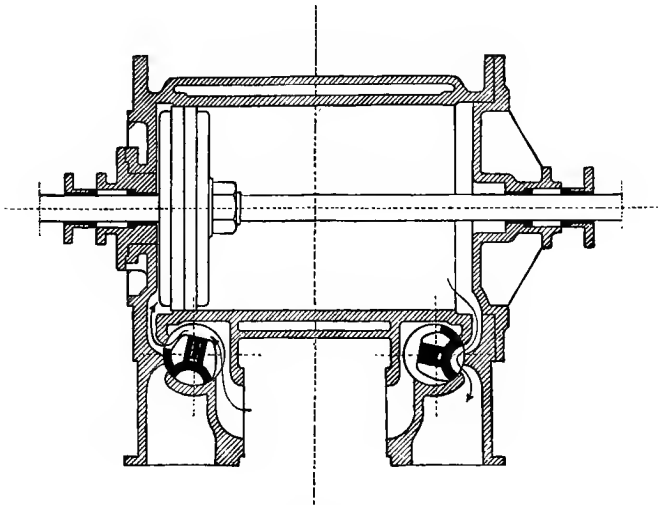
Fig. 163.

the cylinder. Each valve opens the corresponding cylinder port alternately to steam and to exhaust, and has a lap which closes the port during expansion and compression.

In another arrangement four rocking valves are used, namely a separate steam and exhaust valve for each end of the cylinder, as in fig. 164, where one end of the cylinder is shown with the
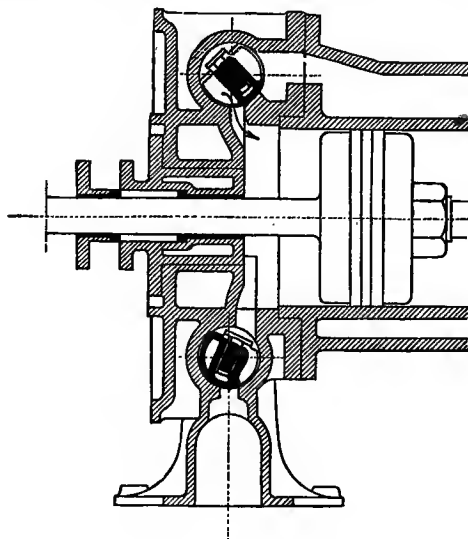
IG. 164.

steam-valve above and the exhaust-valve below. Valves of this type are called Corliss valves and are frequently found in connection with the Corliss and other forms of trip-gear, which will be mentioned in the next chapter. The characteristic of such gear is that after the steam admission valve has been opened to the extent required for admission it is disconnected from the eccentric by a trip device which allows it to close suddenly under the action of a spring and thus produces a sharp cut-off.

214.   Double-beat Lift valves. In many stationary engines *lift* or *mushroom* valves are used, worked by tappets, cams, or eccentrics. These valves open by being raised from their seats, and close by dropping back. Lift valves are generally of the double-beat type (fig. 165), in which equilibrium is secured or rather approximated to by the use of two conical faces of nearly the same size, which open or close together. Thus a valve of large area may be opened against a heavy steam pressure with but little exertion of force.

In many large horizontal stationary engines four double-beat

valves control the action; two on top of the cylinder providing for the admission and two underneath providing for the exhaust.
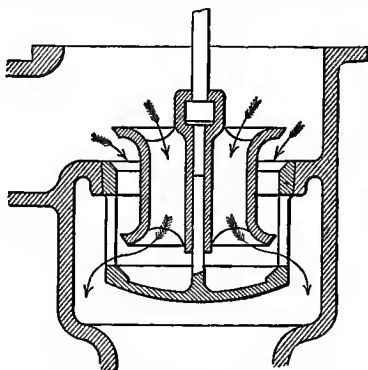
Fig. 165.   Double-Beat Lift-Valve.

The steam-valves in such cases are usually operated by a form of trip-gear. Fig. 166 shows one end of a cylinder with steam
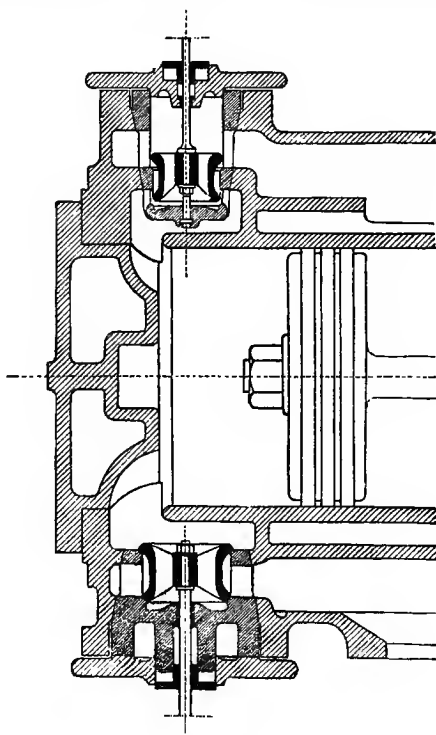
Fig. 166.

and exhaust valves of this type.   The arrangement, like that
of fig. 164, has the advantage of making it possible to have very
small clearance spaces, and also of perfectly draining the cylinder.
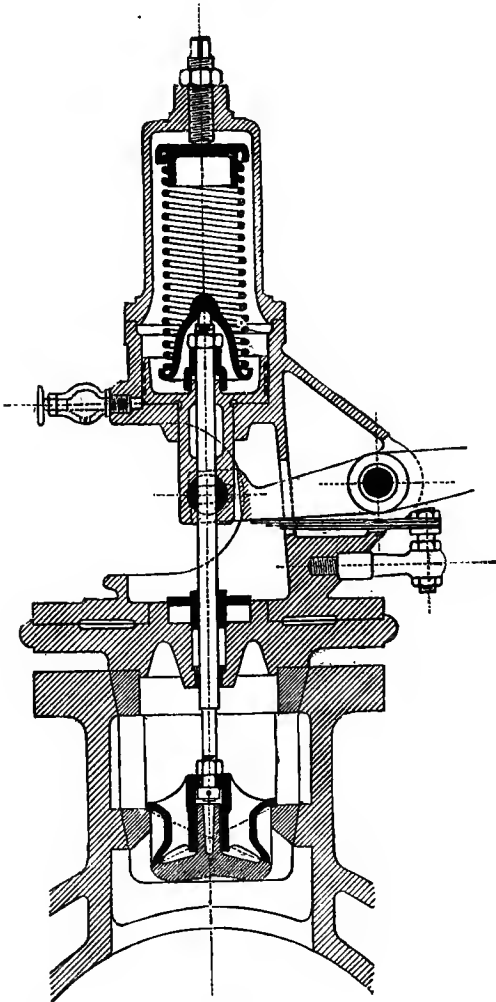
FIG. 167.

Fig. 167 shows a steam-valve of this class with the spring for
closing it after it has been opened and released by a trip-gear.
The spring, which is at the top of the valve-rod, is compressed by
lifting the valve. This also lifts a piston in a dash-pot beneath

the spring, admitting air to a space below. When the valve is released this air has to be expelled through a small orifice at the side, the size of which is regulated by a small screw plug. This prevents the spring from operating too violently, and by adjusting the orifice the valve may be made to come softly to its seat.

The Sulzer engines, already referred to in Chapter VII. as showing excellent economy of steam, give one out of many examples that might be cited from Continental practice, in which
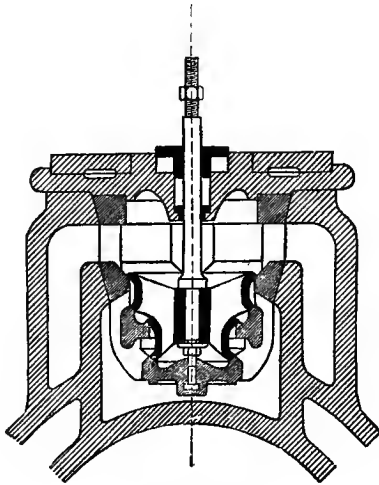
Fig. 168.

the admission and exhaust are controlled by lift valves of the double-beat type. In some of Messrs Sulzer Brothers' designs the valves are four-seated, with the effect that a large opening is secured for a small lift. A four-seated valve is shown in fig. 168.

215. The Cornish Cataract. In the Cornish pumping engine, which retains the single action of Watt's early engine, three double-beat valves are used, as steam-valve, equilibrium-valve, and exhaust-valve respectively. These are closed by tappets on a rod moving with the beam, but are opened by means of a device called a cataract, which acts as follows. The cataract is a small pump with a weighted plunger, discharging

fluid through a stop-cock which can be adjusted by hand when it is desired to alter the speed of the engine. The weighted plunger is raised by a rod which hangs from the beam, but is free in its descent, so that it comes down at a rate depending on the extent to which the stop-cock is opened. When it comes down a certain way it opens the steam and exhaust valves by liberating catches which hold them closed; the " out-door " stroke then begins and admission continues until the steam-valve is closed; this is done directly by the motion of the beam, which also, at a later point in the stroke, closes the exhaust. Then the equilibrium-valve is opened, and the " in-door " stroke takes place, during which the plunger of the cataract is raised. When it is completed, the piston pauses until the cataract allows the steam-valve to open and the next " out-door " stroke then begins. By applying a cataract to the equilibrium-valve also, a pause is introduced at the end of the " out-door " stroke. Pauses have the advantage of giving the pump time to fill and of allowing the pump-valves to settle in their seats without shock.

CHAPTER XII.

GOVERNING.

216. Methods of regulating the work done in a Steam-engine. To make an engine run steadily an almost continuous process of adjustment must go on, by which the amount of work done by the steam in the cylinder is adapted to the amount of external work demanded of the engine. Even in cases where the demand for work is sensibly uniform, fluctuations in boiler pressure still make regulation necessary. Generally the process of governing aims at regularity of speed; occasionally, however, it is some other condition of running that is to be maintained constant, as when an engine driving a dynamo-electric machine is governed by an electric regulator to give a constant difference of potential between the brushes—a condition which often requires the engine to run rather faster when it is giving a greater output.

The ordinary methods of regulating are either (a) to alter the pressure at which steam is admitted by opening or closing more or less a throttle-valve between the boiler and the engine, or (b) to alter the volume of steam admitted to the cylinder by varying the point of cut-off. The former plan was introduced by Watt, and is still common, especially in small engines. The second plan of regulating is in general to be preferred, especially when the engine is subject to large variations of load, and it is usually followed in stationary engines of the larger types.

217. Automatic regulation by Centrifugal speed Governors. Watt's Conical Pendulum Governor. Within certain limits regulation by either plan can be effected by hand, but for the finer adjustment of speed some form of automatic

governor is necessary. Speed governors are commonly of the *centrifugal* type: a pair of masses revolving about a shaft or spindle which is driven by the engine are kept from flying out by a certain controlling force. When an increase of speed occurs this controlling force is no longer able to keep the masses revolving in their former path; they move out until the controlling force is sufficiently increased, and in moving out they act on the regulator of the engine, which may be a throttle-valve or some form of automatic gear by which

FIG. 169. Watt's Governor.

the cut-off is varied. In the conical pendulum governor of Watt (fig. 169) the revolving masses are balls attached to a vertical spindle by rods, and the controlling force is furnished by the weight of the balls, which, in receding from the spindle, are obliged to rise. When the speed exceeds or falls short of its normal value they move out or in, and so raise or lower a collar C which is in connection with the throttle-valve through a lever. The suspension-rods may be hung from the ends of a T-piece attached to the revolving spindle instead of being pivoted in the axis as in fig. 169, and in some cases they cross each other and the spindle itself (see figs. 174–176).
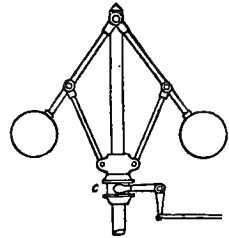
In many modern high-speed engines the masses which form the governor revolve about the shaft of the engine itself, in a vertical plane, and the controlling force is furnished by means of springs which oppose their movement away from the axis of rotation.

218. Loaded and Spring Governors. In a modified form of Watt's governor, known as Porter's, or the *loaded* governor, the balls revolve, as in Watt's form, about a vertical axis, but the tendency which they have to fly out is resisted not only by their own weight but also by a supplementary controlling force which is furnished by a weight resting on the sliding collar (fig. 170). This device is equivalent to increasing the *weight* of the balls without altering their *mass*. In other governors the controlling force instead of being due to gravity only is wholly or partly produced by springs. Fig. 171 shows a governor by Messrs Tangye in which the balls are controlled partly by their own weight and partly by a spring, the
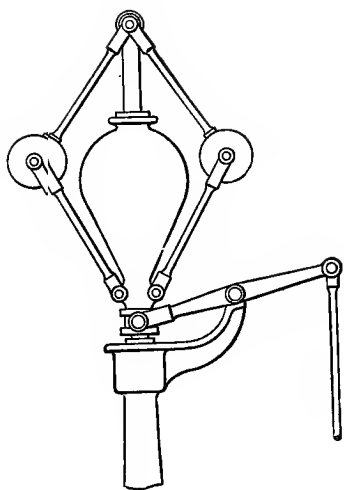
Fɪɢ. 170.   Loaded Governor.
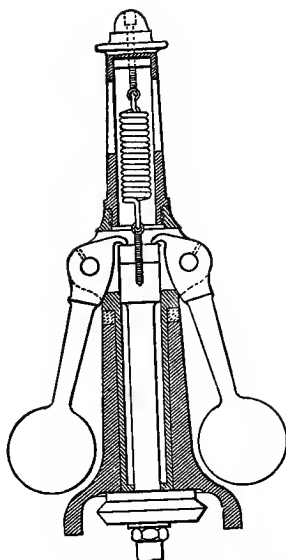
Fɪɢ. 171.   Spring Governor (Tangye).
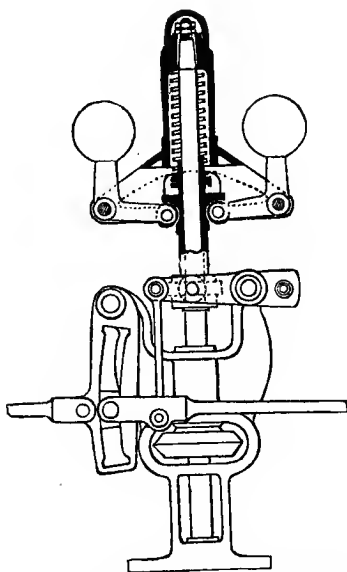
Fɪɢ. 172.   Spring Governor (W. Hartnell).

tension of which is regulated by turning the cap at the top. Another example of a governor with spring control is shown in fig. 172. There the balls move in a sensibly horizontal line and consequently their weight contributes nothing towards resisting the tendency to fly out. A certain amount of additional control, however, is supplied by the weight of the sliding collar and the parts which rise with it, unless these are counterbalanced. In this example the governor regulates the volume of steam admitted per stroke by altering the position of a sliding block in a rocking link, which determines the travel of an expansion-valve.

219. Shaft Governors. This name is given to the type, common in high-speed engines, where the masses revolve in a vertical plane about the engine-shaft, the control being furnished by springs. Shaft governors are sometimes arranged within the fly-wheel of the engine. They may be arranged to act on a throttle-valve or to alter the point of cut-off. An example of
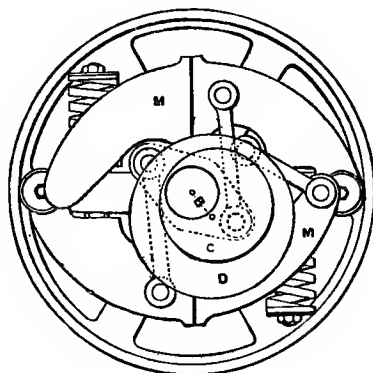
Fɪɢ. 173. Governor of Armington and Sims Engine.

a shaft governor is shown in fig. 173, which is the governor of the Armington and Sims engine. Here the governor produces automatic variations of the cut-off by acting on the main slide-valve of the engine (there being no separate expansion-valve). The displacement of the revolving masses M, M changes both the throw and the angular advance of the eccentric, thereby effecting a change in the steam supply similar to that produced by "notching up" a link-motion. The eccentricity B is altered by the relative displacement of two parts C, D into which the eccentric sheave is

divided. This relative displacement not only changes the length of B but gives it more or less of angular advance.

220. Controlling force. In whatever way the tendency of the balls to fly out be resisted, whether through their own weight or through a supplementary load or through springs, it is convenient to treat the control as equivalent to a certain force F acting on each ball in the direction of the radius towards the axis of revolution. We shall call this the *controlling force*. The value of F varies, in a given governor, when the position of the balls changes. If it were not for friction, the controlling force for any position of the balls could be found experimentally by applying a spring-balance to each ball, with the governor at rest, and noting the force required to hold the ball in the assigned position when this force was applied directly away from the axis of the governor. Owing to friction such an experiment would give two extreme values of the force, for if the pull on the spring-balance were increased the ball would not move further out until the pull became equal to $F + f$, where f denotes the force due to friction. And if the effect of friction in resisting the return of the ball were the same as its effect in resisting the displacement outwards, the pull of the spring-balance might be reduced to $F - f$ before the ball would begin to move in. A mean of these extremes would give the true controlling force in cases where the influence of friction remained unchanged.

When the governor is running the influence of friction is in general less than when it is at rest; but the effect still is to make the actual force which the ball experiences, pulling it towards the spindle, greater or less than F according as the ball is on the point of moving out or moving in.

221. Condition of Equilibrium. Once the controlling force F is known for each position the speed at which the governor must revolve in order to make the balls take up any assigned position is readily calculated. If M be the mass of the ball (in lbs.), n the number of revolutions per second and r the radius (in feet) of the path in which the balls revolve, equilibrium will be maintained when the speed is such that the controlling force and the "centrifugal force" are equal, that is to say when

$$F = 4\pi^2 n^2 r M,$$

F, the controlling force, being expressed in poundals. Hence the speed corresponding to the assigned configuration of the governor is defined by the equation

$$n = \frac{1}{2\pi} \sqrt{\frac{F}{Mr}}.$$

For the present, friction is left out of account; its influence on the speed will be considered immediately.

222. Condition of Stability. When a governor is running at a steady speed, imagine the balls to be slightly displaced from the position proper to that speed, by temporarily applying some displacing force. Then, assuming the speed to remain constant, let the balls be left to themselves. If the governor is *stable* they will tend to recover their original position. The temporary displacement caused the controlling force to be changed by an amount δF, while the radius of the balls' path was changed by δr. The "centrifugal force" was temporarily changed from $4\pi^2 n^2 r M$ to $4\pi^2 n^2 (r + \delta r) M$ while the controlling force was changed from F to $F + \delta F$. The condition of stability requires that the change of controlling force should be greater than the change of centrifugal force, so that there may be a resultant force tending to bring the balls back.

Thus in order that the governor should be stable δF must be greater than $4\pi^2 n^2 M \delta r$, and this will be the case only if

$$\frac{\delta F}{F} \text{ is greater than } \frac{\delta r}{r}.$$

That is to say the controlling force must increase more rapidly than in simple proportion to r as the balls move outwards.

When a stable governor is running in a given configuration the effect of any small increase or decrease of speed is (friction apart) to make the balls go out or come in by a finite amount so that they reach a new position of equilibrium corresponding to the new speed. As r increases a higher and higher speed is required to preserve equilibrium in each new position of the balls.

If F varied just proportionally to r, the speed would be constant for all values of r. This state of things would correspond to neutral equilibrium on the part of the governor: its consequences are considered more particularly in § 225 below.

223. Equilibrium of the Conical Pendulum Governor. Height of the Governor. When the governor is a simple
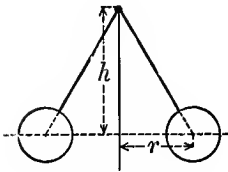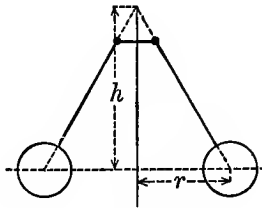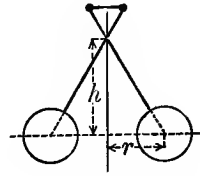
FIG. 174.   FIG. 175.   FIG. 176.

conical pendulum controlled by gravity only and without load other than the weight of the balls themselves—a condition never quite realised in practice, since the weight of the sliding collar and its attached parts always applies some extra load which adds to the controlling force—F, the controlling force, is the resultant of F_2 the tension in the suspending rod and F_1 or Mg the weight of the ball. The triangle of forces is sketched in fig. 177. This applies whichever of the three forms shown in figs. 174, 175, and 176 is given to the governor. Let the *height* of the pendulum governor, that is the vertical distance from the plane of rotation of the balls to the point where the axis of the suspending rod (produced if necessary) cuts the axis of the spindle, be called h. Then, using absolute units for the forces,

FIG. 177.

$$F : Mg :: r : h,$$

from which

$$F = \frac{Mgr}{h}, \text{ and } n = \frac{1}{2\pi}\sqrt{\frac{g}{h}}.$$

The condition of stability requires that the speed n should increase as the balls move out, and hence h must diminish when r increases if a governor of this class is to be stable. This will obviously happen if the form be that of fig. 174 or fig. 175. With the crossed-rod form of fig. 176 the height h will increase when the balls rise only if the points of suspension are not far from the axis. By placing them at a particular distance from the axis, h may be kept very nearly constant: in other words,
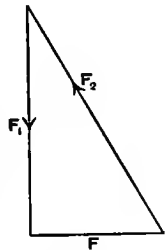
this governor may be arranged to have nearly neutral equilibrium, so that a very small change in the speed n may be associated with a large change in the position of the balls. When the centres of suspension are put further from the axis the mechanism sketched in fig. 176 becomes unstable, and is then unfit to serve as a governor.

224. Equilibrium of Loaded Governor. The results obtained in the last paragraph are readily adapted to the case of a governor of the Porter type (fig. 170). Let M' be the amount of the extra load, per ball (in general M' is one-half the total extra load), and let q be the velocity ratio of the vertical movement of the load to the vertical movement of the ball—a quantity which is easily found by calculation or graphically when the form of the governor is given. Then each ball, in being displaced outwards, has not merely to raise its own weight but has to raise what is equivalent to an additional weight equal to q times the weight of M'. The effect of the load is therefore to increase the controlling force F from $\dfrac{Mgr}{h}$, in poundals, to $\dfrac{(M+qM')gr}{h}$. But the condition of equilibrium still is that F should be equal to $4\pi^2 n^2 r M$. Hence the speed n at which the governor must now turn to maintain any assigned height h is

$$n = \frac{1}{2\pi}\sqrt{\frac{(M+qM')g}{Mh}}.$$

Compared with the simple or unloaded form, this governor requires a higher speed in the proportion of $\sqrt{M+qM'}$ to $\sqrt{M}$.

In the ordinary construction of the Porter governor the four links form a parallelogram, and consequently the vertical movement of the load borne by the sliding collar is twice that of the balls, or $q = 2$. And as the whole load is divided between two balls, each ball virtually has its weight, but not its mass, increased by an amount equal to the whole weight of the central load.

Another way of considering the equilibrium of the Porter governor may be mentioned. Let the mass of each ball be M and let that of the load be $2M'$ as before. The load, the weight of which is $2M'g$ in poundals, is borne by the tensions in the two lower rods (fig. 178). By drawing the triangle abc (fig. 179) which is the diagram of forces for the load, and in which ab or

F_3 is the weight of the load, we find the value of F_4 and F'_4 which are the tensions in the lower rods. Then draw bd or F_1 to represent the weight of one ball (namely Mg), and draw the horizontal line de to meet a line ce drawn from c parallel to the
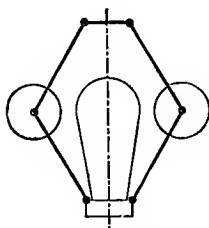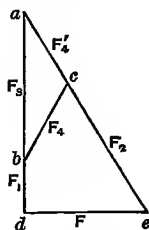
FIG. 178.   FIG. 179.

direction of the upper rod. The figure $ecbd$ is the polygon of forces acting on the ball, ed being the resultant controlling force F. The speed at which the governor will run is determined by the condition that $4\pi^2 n^2 rM$ is to be equal to this force ed. In the usual case of parallel rods, ace is one straight line and then

$$ed = ad \tan a, \quad \text{or} \quad F = (M + 2M') g \tan a,$$

where a is the inclination of the rods to the vertical. Since $\tan a = \dfrac{r}{h}$ this expression agrees with the one given above.

225. Sensibility in a Governor. Isochronism. Any change of speed in a governor tends to produce a change in the position of the balls, and if the governor itself and the regulating mechanism connected with it were free from friction only one position of the governor would be possible for any one speed, provided the condition of stability were complied with. If therefore the supply of steam depends on the position taken up by the governor balls a stable governor does not maintain a strictly constant speed in the engine it controls. Whenever the boiler pressure or the demand for work changes a certain amount of displacement of the balls is necessary to increase or reduce the steam supply, and the balls can retain their new position only by virtue of continuing to turn slower or faster than before. The maximum change of speed which can occur under the control of the governor is that which will make the balls move from one to

the other extremity of their range—namely, from the position which allows the full supply of steam to the position which completely checks the supply. Of course if the engine is overloaded by giving it too much external resistance to overcome, the speed may be further reduced after the governor has done all that it can do to let steam in freely, but the variation of speed for which the governor is responsible is only that which makes the change from no steam to full steam. When a small variation of speed suffices to do this the governor is said to be sensitive, its sensibility being measured by the reciprocal of the ratio which this variation of speed bears to the mean speed.

The more stable a governor is the less sensitive is it; on the other hand when the equilibrium is neutral the sensibility is indefinitely great. The controlling force F then varies as r, and hence n is constant (§ 222) at whatever distance from the axis the balls revolve. In other words, the balls are in equilibrium at one speed and only at one (except for friction), and the least variation from this speed suffices to send them to one extremity or the other of their range. A governor having this quality is said to be isochronous. Friction makes the condition of strict isochronism impossible, but many governors are made nearly isochronous by arranging them so that, as the balls are displaced, the controlling force increases only a little more rapidly than r.

226. Isochronism in the Gravity Governor. Parabolic Governor. An ideal frictionless governor, in which the controlling force is furnished by gravity, can be made isochronous if the balls instead of being hung by rods from fixed points are constrained to move in a parabolic path, as in fig. 180, where the

Fig. 180

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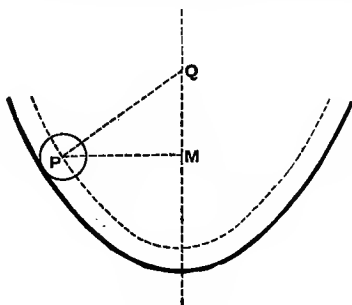
cup or channel which holds the ball is so shaped that the locus of
the centre of the ball, shown by a dotted curve, is a parabola. The
pressure of the ball against the cup is equivalent to the tension
of an imaginary suspension-rod PQ; and it is a property of the
parabola that the sub-normal QM, which represents h, is constant
wherever P be taken along the curve.   Hence a ball supported in
this way would remain in equilibrium at one particular speed of
rotation on the part of the cup, but would fly up to the rim of the
cup if the speed were ever so little increased, and would sink to
the foot if the speed were ever so little reduced.
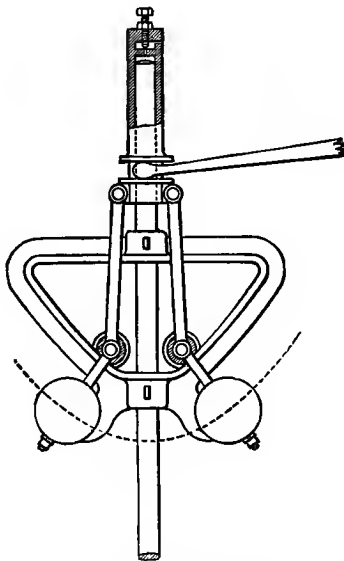
FIG. 181. Parabolic Governor.

Fig. 181 shows an actual form of parabolic governor[1].   An
important feature is the air-cylinder at the top, forming a dash-
pot, which is furnished with a small adjustable orifice through
which air is driven out or in as the balls rise or fall.   The function
of this is to check the tendency which the balls have to fly
violently in or out when the speed drops below or rises above the
normal value.

[1] From Mr J. Head's paper on "A Steam-engine Governor," *Proc. Inst. Mech.
Eng.* 1871.

227. Approximate Isochronism in Pendulum Governors.

A useful approximation to the condition of isochronism can be reached in the conical pendulum governor by using crossed rods with the centres of suspension at a suitable distance from the axis. If each centre of suspension were so placed as to be at the centre of curvature of a parabolic arc which coincided, at the position corresponding to the normal speed, with the actual circular curve along which the balls rise and fall, the governor would be sensibly isochronous at that speed. By taking points a little nearer the axis for the two centres of suspension a margin of stability, always necessary in practice, is secured, but the governor
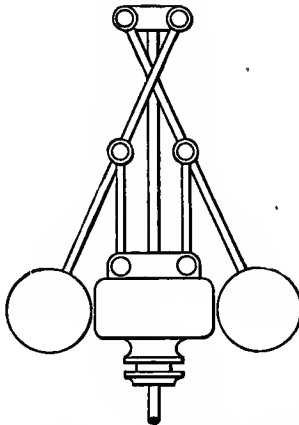
Fɪɢ. 182. Loaded Governor with Crossed Rods.

is left nearly enough isochronous to be very sensitive. This crossed-rod type of governor, which is due to Farcot, is often met with in a loaded form. An example is given in fig. 182. Loading a governor (whether the rods are crossed or open) need not affect the sensibility; it makes a higher speed necessary, but the proportion of the fluctuation of speed to the mean speed is not changed, provided the links are arranged in such a way that the vertical velocity-ratio of the load and the balls does not alter as the balls rise.

Another approximately isochronous form of gravity governor is Pröll's (fig. 183), which is interesting as exemplifying a different method of reducing the stability of the pendulum type. Let the ball be supported not at the joint between the links as in the

25—2

ordinary Porter governor but at the end of an arm projecting upwards and rigidly connected to the lower link. By a proper choice of the length of this arm the controlling force may be made as nearly proportional to the radius as may be desired.
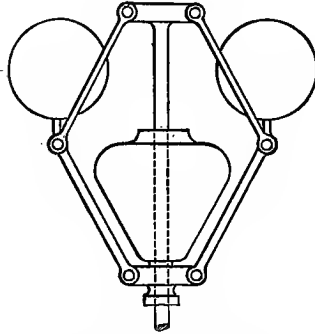
Fig. 183.   Pröll's Governor.

Pendulum governors of the stable class are occasionally loaded indirectly, the weight which forms the load being applied at some point in the lever by which the governor is connected with the valve. This allows the load and therefore the speed to be adjusted : further, by applying the load at the end of a cranked arm in the lever in such a way that it becomes less effective when the balls go out, the system can be made approximately isochronous.

228. Governors with spring control. Adjustment of sensitiveness.   When springs furnish the controlling force, in whole or part, as in the governors shown in figs. 171 and 172, their tension is generally adjustable. This gives a convenient means of altering the speed ; at the same time it affects the sensitiveness of the governor. In spring governors which are constructed so that the radial displacement of the balls produces a proportional change in the tension of the spring, the condition of isochronism can be approached, as nearly as may be wished, by giving the spring a suitable amount of initial tension. Thus in Hartnell's apparatus, fig. 172, where the balls move in a nearly horizontal direction and gravity has almost nothing to do with the control, the governor can be made isochronous by screwing down the spring so that the initial force exerted by the spring (before the balls are displaced) is to the increase of this force by the displacement of

the balls, as the initial radius of the ball's path is to the increase of that radius by the displacement. This makes F vary proportionally to r, and therefore (§ 222) requires no change in n as the balls move out. Any greater initial tension would make the governor unstable, and a less tension is in fact necessary, in order that the sensitiveness may not be impracticably great.

229. Determination of the Controlling Force. Whatever be the method of control, by weights or springs or both, the controlling force F may generally be calculated for any assumed position of the balls. The simple pendulum governor both unloaded and loaded as in fig. 178 has already been considered. A case such as that of fig. 171 or fig. 172 presents no difficulty when the

stiffness and initial tension of the spring are given. Slightly less simple cases of the loaded governor present themselves when the balls are not placed at the joints between the upper pair of links and the lower pair which carry the load. Let the ball M (fig. 184) be fixed on the upper link AB, at any place either beyond B or between A and B. First find F_1, the stress in BC, from the consideration that BC and its twin link on the other side are in simple tension and support between

Fig. 184.

them the load. Then the forces acting on ABM, namely F_1, the tension in BC, F_2 which is the weight of the ball, and F which is the force to be determined, are in equi-

librium, and hence F is readily found by taking moments about A. We here treat F as the equilibrant instead of the resultant of the forces which are actually applied, for the sake of bringing the system into static equilibrium.

When the ball is carried by the link BC or by a piece rigidly connected to it as in Pröll's governor we may proceed thus (fig. 185):— The forces concerned in the equilibrium of the rigid piece CBM are (1) F_3 the half

Fig. 185.

weight of the load acting at C, which is the vertical component of the pull at the joint C, (2) the horizontal component F_4 of the
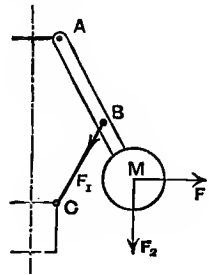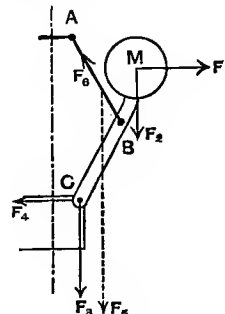
pull at the joint C, (3) the tension F_6 in the link AB, (4) the weight of M, or F_2, and finally (5) the force F which is to be determined. The resultant of F_3 and F_4 no longer acts along BC for there is a bending moment on the piece CBM. Compound F_2 and F_3 into a single force F_5. Since F_4 and F are horizontal, this vertical force F_5 must be wholly balanced by the vertical component of the stress in AB. Hence F_6 is found by drawing a right-angled triangle (fig. 186) with a line parallel to AB as hypotenuse and with a vertical side equal to F_5.
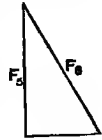
FIG. 186.

Having found F_6 we are in a position to take moments about C in order to find F, which is now the only unknown force not acting through C.

230. Influence of Friction. Power of the Governor. We may express the influence of friction on the behaviour of a governor by treating it as equivalent to a force with some limiting value f, acting radially on each ball, in the same direction with the controlling force F when the balls are moving out and in the opposite direction when they are moving in. This makes the whole controlling force $F + f$ in the former case and $F - f$ in the latter. Let n be the speed proper to the force F alone, then if there were no friction any increase of speed above n would begin to alter the configuration, making the balls move out; but in consequence of friction this does not happen until the speed has increased by some finite amount Δn such that

$$n + \Delta n = \frac{1}{2\pi}\sqrt{\frac{F+f}{Mr}}.$$

Similarly, should the speed fall below the normal speed n proper to any configuration, friction prevents the balls from beginning to move in until the reduction of speed $\Delta'n$ is such that

$$n - \Delta'n = \frac{1}{2\pi}\sqrt{\frac{F-f}{Mr}}.$$

Hence in consequence of friction the speed may alter as much as Δn above and $\Delta'n$ below the normal speed n, while the position of the balls remains unchanged. From the above equations, if f be small relatively to F, as it always should be in practice, so that

Δn may be small in comparison with n, Δn and $\Delta' n$ are nearly equal, and we have, approximately,

$$\frac{\Delta n}{n} = \frac{f}{2F}.$$

This variation of speed due to friction is independent of whatever further variation of speed the governor may allow in consequence of its equilibrium being stable (§ 225), and would of course be experienced even with a governor which except for friction was isochronous.

To keep the effects of friction within moderate limits it is essential that F should be great in comparison with f. The frictional resistance f proceeds partly from the joints of the governor itself but mainly from the throttle-valve spindle or from the expansion gear the position of which the governor has to regulate. A *powerful* governor, namely a governor with a large amount of controlling force F, is therefore required when any considerable amount of frictional resistance in the valve or gearing is to be overcome. With simple pendulum governors, the only way to secure power in this sense is to make the balls large. Loaded governors have the advantage that great power may be secured with comparatively small revolving masses. The quality of powerfulness in a governor is increased whenever the controlling force is increased, whether by gravity loading or by the use of springs. From another point of view, the loaded governor (with the same revolving masses) is more powerful because it runs at a higher speed; but this is just because its controlling force F is greater. A governor in which the control is given by springs, such as a shaft governor, may be made very powerful without the use of large masses, by using stiff springs and a high speed.

231. Curves of Controlling Force. The consideration of sensitiveness and powerfulness in governors generally is greatly elucidated by using a graphic method, suggested by W. Hartnell[1], of exhibiting the controlling force. Having found the controlling force F for various positions of the balls, let a curve P_1P_2 (fig. 187) be drawn in which abscissæ represent r the radius of the balls' path and ordinates represent F. To find the configuration proper to any assigned speed n draw a line OS at such an inclination that

[1] *Proc. Inst. Mech. Eng.* 1882.

$\tan SOX = 4\pi^2 n^2 M$, due regard being had to the scales of F and r.
When the base OX is taken equal to unity on the scale used in
plotting r, the value of SX is equal to $4\pi^2 n^2 M$ on the scale used
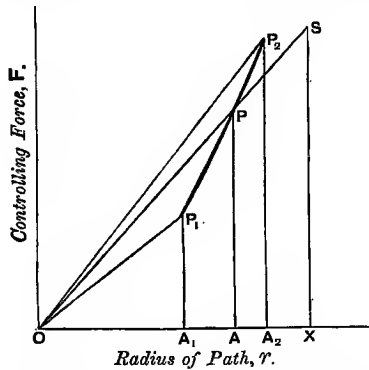
FIG. 187.   Curve of Controlling Force.

in plotting F.   Let P be the point in which this line cuts the
curve of F.   Then since

$$F = PA = OA \tan POA = 4\pi^2 n^2 r M,$$

it follows that the point of intersection P determines the radius
OA at which the governor will run when the speed is n.   Similarly
the tangent of the angle which is made with the base by any other
line drawn from O to meet the curve, such as OP_1 or OP_2, is
proportional to the square of the speed at the corresponding path-
radius OA_1 or OA_2.   Thus if OA_1 be set off to represent the least
and OA_2 the greatest radius, corresponding to the positions giving
full steam and no steam respectively, the inclinations of the lines
OP_1 and OP_2 determine the whole range through which the speed
will alter in consequence of the stability of the governor (apart
from any effect of friction).

Further, if a pair of additional curves $Q_1 Q_2$ and $R_1 R_2$ be drawn
as in fig. 188 to represent the values of $F + f$ and $F - f$ respectively,
in relation to r, the diagram shows the additional changes of speed
that are due to friction.   The lowest possible speed is then de-
termined by the inclination of the line OR_1, the highest by that
of the line OQ_2.   Thus the lower limit of speed, when there is
full admission of steam, is

$$\frac{1}{2\pi} \sqrt{\frac{A_1 R_1}{M \cdot OA_1}}$$

and the higher limit of speed, when the admission is reduced to nil, is

$$\frac{1}{2\pi}\sqrt{\frac{A_2 Q_2}{M \cdot O A_2}}.$$

Again, the whole work done in altering the configuration of the governor, while the balls move out from A_1 to A_2 would be (for each ball) equal to the area $A_1 P_1 P_2 A_2$ if there were no friction to be overcome: actually it is the area $A_1 Q_1 Q_2 A_2$. And as the ball comes in from A_2 to A_1 the part of the stored energy which is recovered is measured by the area $A_2 R_2 R_1 A_1$, the rest having been spent on friction. The area $P_1 Q_1 Q_2 P_2$ is the work spent against friction while the governor is closing the throttle-valve or shifting the expansion-gear from full steam to no steam.
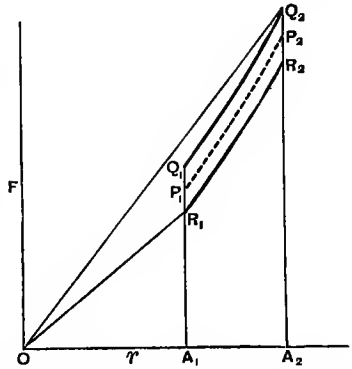
Fig. 188. Curves of Controlling Force, taking friction into account.

The powerfulness of the governor is measured in a definite manner by the area $A_1 P_1 P_2 A_2$, namely, the work stored and restored (save for friction) as the governor balls open or close throughout their range. In order that friction should cause no very serious irregularity in speed this area must be many times greater than the area $P_1 Q_1 Q_2 P_2$ or $P_1 P_2 R_2 R_1$. These last areas are equal if the friction f has the same value in closing as in opening the valve (as we have assumed above), but the construction shown in fig. 188 is evidently applicable whether f has or has not the same value during the rise and fall of the balls.

Again, the governor is stable provided the inclination of the curve to the axis OX be greater than the inclination of a line drawn from O to meet the curve at any point within the range of possible positions. Thus in fig. 187 the curve shows the governor to be stable because any line OP is less steep than the inclination of the curve itself at P. This is the condition of stability stated in § 222, namely, that the controlling force must increase more rapidly than the radius. A strictly isochronous governor would

have for its curve of F and r a straight line passing, when produced, through O. If this condition were fulfilled by the line without friction P_1P_2, the line with friction Q_1Q_2, which lies above P_1P_2 at a more or less constant distance from it, would in general be less steep than a line from O drawn to meet it, which would mean that friction would make the otherwise neutral governor *unstable*. This is one reason why the isochronous governor is impracticable. The governor of fig. 188 is stable notwithstanding friction.

232. Hunting. Apart from the reason just stated it is indispensable to give a governor some margin of stability, especially when any change of speed takes some time to affect the supply of steam. An over-sensitive governor is liable to produce in the engine which it governs a state of forced oscillation called *hunting*. Several reasons contribute to produce this effect. When an alteration of speed begins to be felt, however readily the governor alters its form the engine's response is more or less delayed. The action of the regulator does not immediately take full effect upon the speed in consequence of the energy that is stored within the engine itself, not only in its moving parts but also in the steam that has passed the regulator and is still doing work in the engine. If the governor acts by closing a throttle-valve, the engine has still a capacious valve-chest on which to draw for steam. If it acts by changing the cut-off, its opportunity has passed if the cut-off has already occurred, and the control only begins in the next stroke. This lagging of effect is specially felt in compound engines, where that portion of the steam which is already in the engine continues to do its work for nearly a whole revolution after passing beyond the governor's control. The result of this storage of energy in an engine whose governor is too nearly isochronous is that whenever the demand for power suddenly falls the speed rises so much as to force the governor into a position of over-control, such that the supply of steam is no longer adequate to meet even the reduced demand for power. Then the speed slackens, and the same kind of excessive regulation is repeated in the opposite direction. A state of forced oscillation is consequently set up. The tendency to hunt depends upon the fact that the rate at which steam does work is not immediately controlled when the load on the engine varies, but that there is

a time-lag between any variation in the load and the proper corresponding variation in the action of the steam. A similar time-lag with a consequent tendency on the part of the engine to hunt may proceed from another cause, which is independent of the storage of steam between the regulating valve and the engine piston. A sensitive governor, especially when it is of the relay type described below (§ 236), may take some time to come to its new position when the load is suddenly reduced. The governor begins to close the throttle-valve or to hasten the cut-off. But this takes time, and meanwhile the supply of steam is excessive and spends itself on the fly-wheel of the engine, giving it increased speed. By the time the supply is adjusted the speed has risen beyond its normal value, and a stage is reached when the regulating mechanism is carried too far and the supply is too much reduced. Thus a condition of forced oscillation may be set up even in cases where there is no storage of steam. The tendency is especially noticeable in engines with heavy fly-wheels running under light loads: under a heavier load the same engine may govern well without hunting. Again, hunting may be caused by the friction of the governor and of the regulating mechanism. Friction prevents the governor and regulator from beginning to change its position until the speed has changed by a finite amount, and when once the movement begins it goes beyond the point proper for steady control. The effect is aggravated by the momentum which the governor balls acquire in being displaced. Oscillations of the governor due to its own inertia are often prevented by introducing a *viscous* resistance to the displacement of the governor, which prevents the displacement from occurring too suddenly, without affecting the ultimate position of equilibrium. For this purpose many governors are furnished with a *dash-pot*, which is a hydraulic or pneumatic brake, consisting of a piston connected to the governor, working loosely in a cylinder which is filled with oil or with air. An instance of the use of a dash-pot has already been mentioned in speaking of the parabolic governor of fig. 181.

233. Throttle-valve and automatic Expansion-gear. The throttle-valve, as introduced by Watt, was originally a disk turning on a transverse axis across the centre of the steam-pipe. It is now usually a double-beat valve (§ 214) or a piston-valve.

When regulation is effected by varying the cut-off, and an expansion-valve of the slide-valve type is used, the governor generally acts by changing the travel of that valve. Fig. 172 illustrates one usual mode of doing this, by giving the expansion-valve its motion from an eccentric-rod through a link the throw of which is varied by the displacement of the governor balls. In some forms of automatic expansion gear the governor acts upon the lap of the expansion-valve. In others it acts by shifting the expansion eccentric round upon the shaft and so changing its angular advance. In others, again, it acts on an ordinary slide-valve through some form of link-motion or in such a way as has just been described.

234.   Trip-gear for governing by varying the expansion. In large stationary engines the most usual plan of automatically regulating the expansion is to employ some form of trip-gear, the earliest type of which was introduced in 1849 by G. H. Corliss of Providence, U.S. In this system the valves which admit steam are distinct from the exhaust-valves. The latter are opened and closed by a reciprocating piece which takes its motion from an eccentric. The former are opened by a reciprocating piece, but are closed by springing back when released by a trip- or trigger-action. The trip occurs earlier or later in the piston's stroke according to the position of the governor. The admission-valve is opened by the reciprocating piece with equal rapidity whether the cut-off is going to be early or late. It remains wide open during the admission, and then, when the trip-action comes into play, it closes suddenly. The indicator diagram of an engine fitted with trip-gear consequently has a nearly horizontal admission-line and a sharply defined cut-off. In the original trip-gear engines of Corliss the valves were cylindrical plates turning in hollow cylindrical seats which extend across the width of the cylinder, of the rocking type mentioned in § 213 as Corliss valves, and this type is still found in some trip-gear engines. Often, however, the admission-valves are of the double-beat type, and spring into their seats when the trip-gear acts. Valves of this type are used in the highly efficient engines designed by Sulzer Brothers, who took up the manufacture of trip-gear engines in 1867 and have brought it to great perfection. Many forms of trip-gear have been devised by Corliss himself, and by others. One of these, the Spencer Inglis[1]

[1] *Proc. Inst. Mech. Eng.* 1868.

trip-gear, by Messrs Hick, Hargreaves & Co., is shown in figs. 189 and 190. A wrist-plate A, which turns on a pin on the outside of the cylinder, receives a motion of oscillation from an eccentric. It opens the cylindrical rocking valve B of the Corliss type, by pulling the link C, which consists of two parts, connected to each
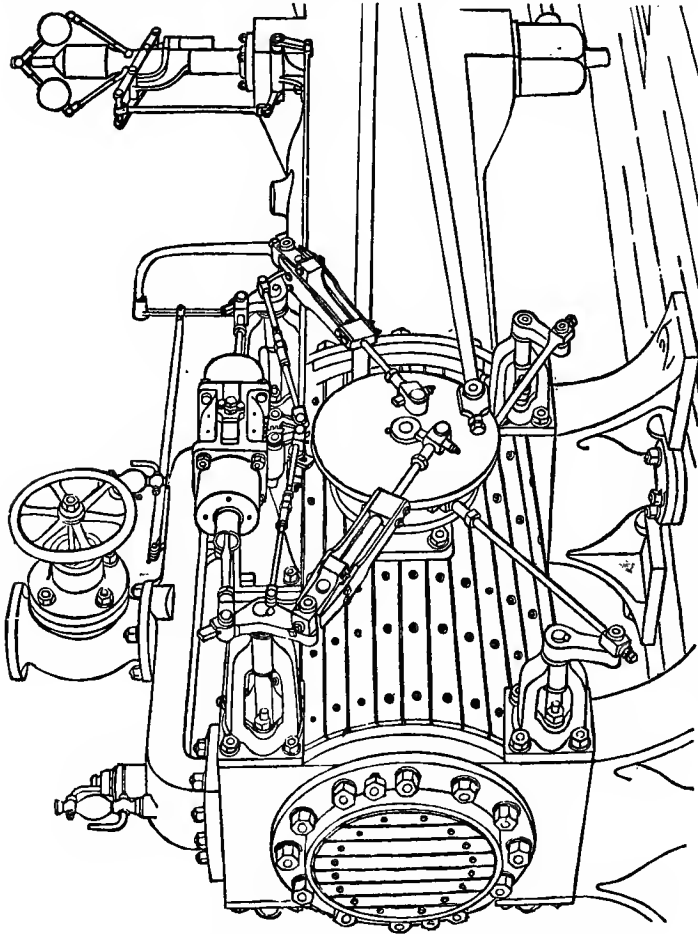
FIG. 189.   Engine Cylinder with Corliss Valve-gear.

other by a pair of spring clips a, a. Between the clips there is a rocking cam b, and as the link is pulled down this cam places itself more and more athwart the link, until at a certain point it forces the clips open. Then the upper part of the link springs back and allows the valve B to close by the action of a spring in

the dash-pot *D*. When the wrist-plate makes its return stroke the clips re-engage the upper portion of the link *C*, and things are ready for the next stroke. The rocking cam *b* has its position controlled by the governor through the rods *F* and *E* in such a way that when the speed of the engine increases it stands more athwart the link *C*, and therefore causes the clips to be released at an earlier point in the stroke. A precisely similar arrangement governs the admission of steam to the other end of the cylinder.
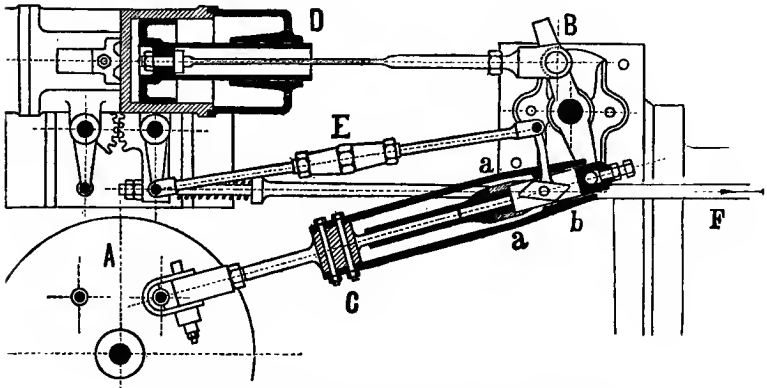
FIG. 190.   Corliss Valve-gear, Spencer Ingus form.

The exhaust-valves which are also rocking valves of the Corliss type, are situated on the bottom of the cylinder, at the ends, and take their motion from a separate wrist-plate which oscillates on the same pin with the plate *A*.

Besides securing a sharp cut-off, without wiredrawing of the steam, trip-gears have the advantage that the force to be exerted by the governor in changing the position of the trip or trigger is very slight in comparision with the forces that would be involved in moving the valve.

A modern example of Sulzer's gear is shown in fig. 191. A shaft, not shown in the figure, revolves alongside of the cylinder, giving motion to an eccentric which causes the rods *R* and *Q* to oscillate in the direction of their length. The top of *R* is pinned at *P* to a short radius rod which oscillates about the fixed centre *B*. On the same fixed centre there is a lever *A* by which the valve is opened. *A* is caused to open the valve by the downward movement of the rod *R*, through the action of the bell-crank

trigger-piece T. The angular position of T is controlled by Q, which in its turn is controlled by the governor. In each downward movement of R there is a rocking of the bell-crank T, and at
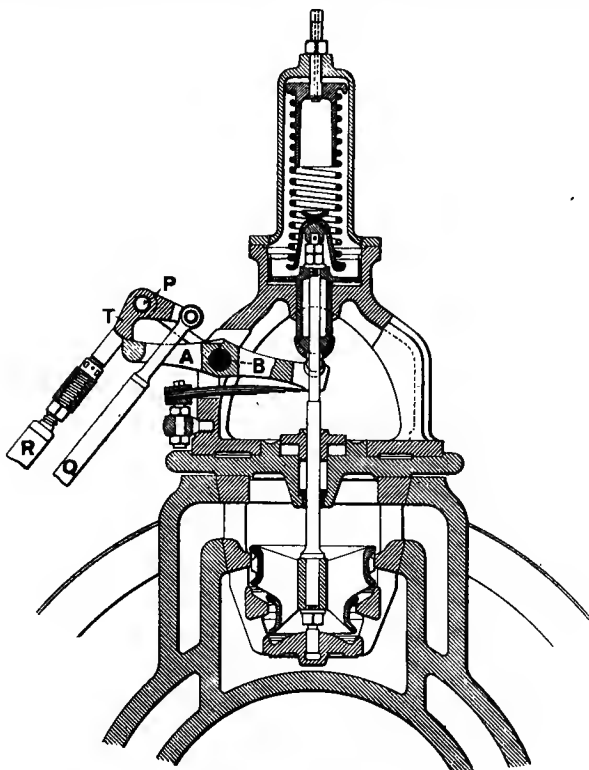
Fig. 191. Trip-gear (Sulzer).

the appropriate moment, determined by the governor, A escapes from the edge of the trigger and the valve is closed by the spring, at the bottom of which there is a dash-pot to make the valve settle gently on its seat.

Fig. 192 shows a compact form of trip-gear by Dr Pröll. A rocking-lever ab is made to oscillate on a fixed pin through its centre by a connection to the cross-head of the engine. When the end a rises, the bell-crank lever c engages the lever d, and when a is depressed the lever d is forced down and the valve e is opened to admit steam to one end of the cylinder. As a continues moving down a point is reached at which the edge of c slips past

the edge of d, and the valve is then forced to its seat by a spring in the dash-pot f.   The disengagement occurs early or late according to the position of a central fulcrum piece, on which the heel of the bell-crank c rests during the opening of the valve. The position of the fulcrum piece is determined by the governor,
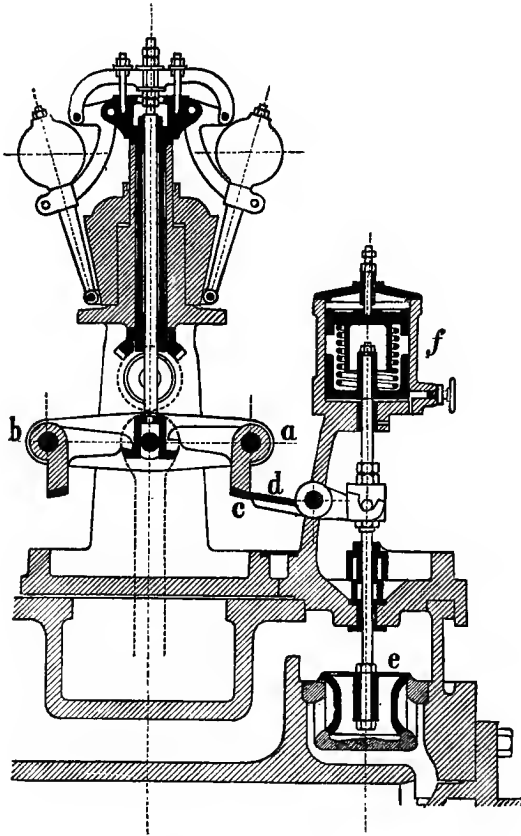
Fig. 192.   Pröll's Trip-gear.

which is of the kind already mentioned in § 227.   A similar action, occurring at the other end of the rocking bar ab, supplies steam to the other end of the cylinder.   In one form of Pröll's gear both ends of ab act on the same steam-valve, which is then a separate expansion-valve fixed on the back of a chest in which an ordinary slide-valve works.

235. Hit and miss Governors. A usual method of controlling the speed of gas-engines is to prevent any combustible gas from entering the cylinder when the speed is in excess of a certain limit. This is often done by means of a hit and miss arrangement, in which the gas-admission valve is operated by a reciprocating piece through an intermediate piece which is lifted out of the way by the governor when the speed-limit is exceeded, so that no opening of the valve takes place and no charge is taken in. The governor itself may be of any ordinary centrifugal type. In small engines the hit and miss is sometimes effected without the use of a centrifugal governor by giving the intermediate piece the form of a bell-crank, pivoted to the reciprocating piece, with a heavy end projecting at right angles to the direction in which reciprocation takes place. When the speed of reciprocation exceeds the desired limit the inertia of this heavy end causes the bell-crank to tilt into such a position that the end of the valve spindle is missed and the valve remains closed.

236. Disengagement Governors. With the ordinary form of centrifugal governor the position of the throttle-valve, or the expansion-link, or the Corliss trigger depends on the configuration of the governor, and is definite for each position of the balls. In disengagement governors, of which the governor A shown on the right-hand side in fig. 193 is an example, any reduction of speed below a certain value sets the regulating mechanism in motion, and the adjustment continues until the speed has been restored. This is done by means of the wheel c which comes into gear with a wheel on the end of the spindle a when the speed falls below a certain limit. Similarly a rise of speed above a certain limit sets the regulating mechanism in motion in the other direction by putting b in gear with a. If the spindle a is connected to the regulator so as to give more steam when it turns one way and less when it turns the other, the speed at which the engine will run in equilibrium must lie between narrow limits, since at any speed high enough to keep b in gear with a the supply of steam will go on being reduced, and at any speed low enough to bring c into gear with a the supply will go on being increased. This mode of governing, besides being sensibly isochronous, has the important advantage that the power of the governor is not limited by the controlling force on the balls, since the governor acts by applying

E.   26

a portion of the power that is being developed by the engine to the work of moving the regulator. It is rarely applied to steam-engines, mainly because its action is too slow. This defect has been ingeniously remedied in the supplementary governor of Mr W. Knowles, who has combined a disengagement governor with one of the ordinary type in the manner shown in fig. 193[1]. Here the spindle a, driven by the supplementary or disengagement governor A, acts by lengthening the rod d which connects the ordinary governor B with the regulator. It does this by turning
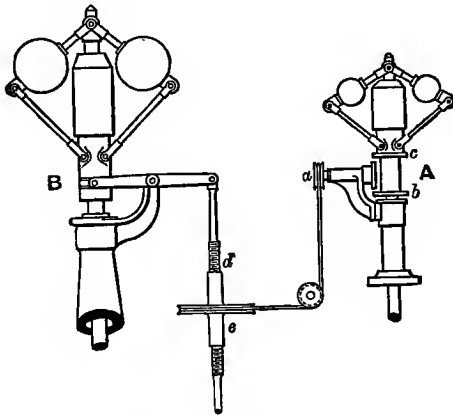
Fig. 193. Knowles's Supplementary Governor.

a coupling nut e which unites two parts of d, on which right- and left-handed screws are cut. Any sudden fluctuation in speed is immediately responded to by the ordinary governor. Any more or less permanent change of load or of steam-pressure gives the supplementary governor time to act. It goes on adjusting the supply until the normal speed is restored, thereby converting the control of the ordinary governor, which is stable, and therefore not isochronous, into a control which is isochronous as regards all fluctuations of long period. The power of the combination, however, is limited to that of the ordinary governor B.

237. Relay Governors. Other governors which deserve to be classed as disengagement governors are those in which the displacement of the governor affects the regulator, not directly by a mechanical connection, but by admitting steam or other fluid

[1] *Proc. Inst. Mech. Eng.* 1884.

into what may be called a relay cylinder, whose piston acts on the regulator. In order that a governor of this class should work without causing the engine to hunt, the piston and valve of the relay cylinder should be connected by what is termed differential gear, the effect of which is that for each displacement of the valve by the governor the piston moves through a distance proportional to the displacement of the valve. An example of differential gear is shown in fig. 194. Suppose that the rod a is connected with the governor so that it is raised by an acceleration of the engine's speed. The rod c which leads from the relay piston b to the regulator serves as a fulcrum, and the valve-rod d is consequently raised. This admits steam to the upper side of the piston and depresses the piston, which pulls down d with

Fig. 194. Differential Gear for Relay Governor.

it, since the end of a now serves as a fulcrum. Thus by the downward movement of the piston the valve is again restored to its middle position and the movement of the regulator then ceases until a new change of speed occurs. A somewhat similar differential contrivance is used in steam-steering engines to make the position of the rudder follow, step by step, every movement of the hand-wheel[1]; also, in the steam reversing gear which is applied to large marine engines, to make the position of the drag-link follow that of the hand-lever; and also in certain electrical governors[2]. The effect of adding a differential gear such as this to a relay governor or other disengagement governor is to convert it from the isochronous to the stable type.

The governing device applied by Parsons to his steam turbines (§ 146) gives an example of a relay governor of a somewhat different type, where the relay is employed to operate a steam piston which opens and closes the main valve. In that device the relay has a regular oscillation imposed upon it by the mechanism of the engine, whatever be the position of the governor, and the direct function of the governor is to determine how long the relay will admit steam to the piston which operates the main valve, in each of its regular periodic movements.

[1] See a paper by Mr J. MacFarlane Gray, *Proc. Inst. Mech. Eng.* 1867.
[2] Willans, *Min. Proc. Inst. C. E.*, Vol. LXXXI. p. 166.

238. Differential or dynamometric Governors.   Another group of governors is best exemplified by the " differential " governor of the late Sir W. Siemens[1] (fig. 195). A spindle *a* driven by the engine drives a piece *b* (whose rotation is resisted by a friction-brake) through the dynamometer coupling *c*, consisting of a nest of bevel-wheels and a lever *d* which is loaded, the weight of the load acting at right angles to the plane of the paper. So long as the speed remains constant the rate at which work is done on

Fig. 195.   Siemens's Governor.

the brake is constant and the lever *d* is steady.   If the speed increases, more power has to be communicated to *b*, partly to overcome the inertia and partly to meet the increased resistance of the brake, and the lever *d* is displaced.   The lever *d* works the throttle-valve or other regulator, either directly or by a steam relay.   The governor is isochronous when the force employed to hold *d* in position does not vary; if the control of *d* is arranged so that the force tending to hold it in position increases when *d* is displaced, the governor is stable.   A governor of this class may properly be called a dynamometric governor, since it regulates by endeavouring to keep constant the rate at which energy is transmitted to the piece *b*.   In one form of Siemens's governor[2] the friction-brake is replaced by a sort of centrifugal pump, consisting of a paraboloidal cup, open at the top and bottom, whose rotation causes a fluid to rise in it and escape over the rim when the speed is sufficiently great.   Any increase in the cup's speed augments largely the power required to turn it, and consequently affects the position of the piece which corresponds to *d*.   Siemens's governor is not itself used to any important extent, but the principle it embodies finds application in a number of other forms.

The " velometer " or marine-engine regulator of Messrs Durham and Churchill[3] is a governor of the same type.   In it the rotation of a piece corresponding to *b* is resisted by means of a fan revolving in a case containing a fluid, and the coupling piece which is the mechanical equivalent of *d* in fig. 195 acts on the

[1] *Proc. Inst. Mech. Eng.* 1853.

[2] *Proc. Inst. Mech. Eng.* 1866 ; or *Phil. Trans.* 1866.

[3] *Proc. Inst. Mech. Eng.* 1879.

throttle-valve, not directly but through a steam relay. In Silver's marine governor[1] the only friction-brake that is provided to resist the rotation of the piece which corresponds to b is a set of air-vanes. The inertia is, however, very great, and any acceleration of the engine's speed consequently displaces the dynamometer coupling, and so acts on the regulator in its effort to increase the speed of b.

Another example of the differential type is the Allen governor[2], which has a fan directly geared to the engine, revolving in a case containing a fluid. The case is also free to turn, except that it is held back by a weight or spring and is connected to the regulator. So long as the speed of the fan is constant, the moment required to keep the case from turning does not vary, and consequently the position of the regulator remains unchanged. When the fan turns faster the moment increases, and the case has to follow it (acting on the regulator) until the spring which holds the case from turning is sufficiently extended, or the weight raised. The term "dynamometric governor" is equally applicable to this form; the power required to drive the fan is regulated by an absorption-dynamometer in the case instead of by a transmission-dynamometer between the engine and the fan. In Napier's governor the case is fixed, and the reaction takes place between one turbine-fan which revolves with the engine and another close to it which is held from turning by a spring and is connected with the regulator.

239. Pump Governors. Pump governors form another group closely related to the differential or dynamometric type. An engine may have its speed regulated by working a small pump which supplies a chamber from which water or other fluid is allowed to escape by an orifice of constant size. When the engine quickens its speed the fluid is pumped in faster than it can escape, and the accumulation of the fluid in the chamber may be made to act on the regulator through a piston controlled by a spring or in other ways. This device has an obvious analogy to the cataract of the Cornish pumping engine (§ 215), which has, however, the some-what different purpose of introducing a regulated pause at the end of each stroke, or rather serves this purpose in addition to regulating the number of strokes per minute. The "differential

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[1] *Brit. Ass. Rep.* 1859, p. 123.   [2] *Proc. Inst. Mech. Eng.* 1893.

valve-gear" invented by Mr H. Davey, and successfully applied by
him to modern pumping engines, combines the functions of the
Cornish cataract with that of a hydraulic governor for regulating
the expansion[1]. In this gear, which is shown diagrammatically in
fig. 196, the valve-rod of the engine (a) receives its motion from a
lever b, one end of which (c) copies, on a reduced scale, the
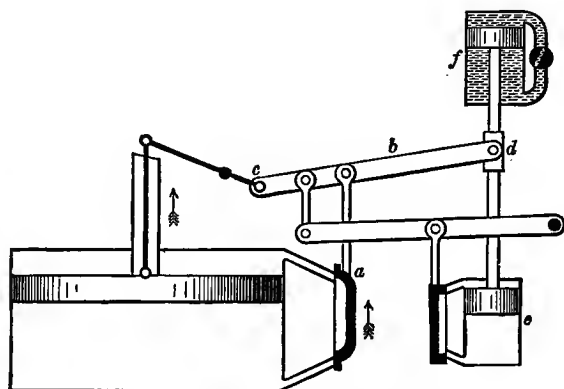
FIG. 196.   Davey's Differential Valve-Gear.

motion of the engine piston, while the other end (d), which forms
the fulcrum, has its position regulated by attachment to a sub-
sidiary piston-rod, which is driven by steam in a cylinder e, and
is forced to travel at a nearly uniform rate by a cataract f. The
point of cut-off is determined by the rate at which the main piston
overtakes the cataract piston, and consequently comes early with
light loads and late with heavy loads.

240.   Governing Marine Engines.   The governing of
marine engines is peculiarly difficult on account of the sudden
and violent fluctuations of load to which they are liable through
the alternate uncovering and submersion of the screw in a heavy
sea.   However rapidly the governor responds to increase of speed
by closing the throttle-valve, an excess of work is still done by
the steam in the valve-chest and in the high-pressure cylinder.
To check the racing which results from this, it has been proposed
to supplement the control which the throttle-valve in the steam-
pipe exercises by throttling the exhaust or by spoiling the

[1] *Proc. Inst. Mech. Eng.* 1874.

vacuum. With the same object Messrs Jenkins and Lee have given supplementary regulation by causing the governor to open a shunt-valve connecting the top with the bottom of the low-pressure cylinder, thus allowing a portion of the steam in it to pass the piston without doing work. In Dunlop's pneumatic governor[1] an attempt is made to anticipate the racing of the screw by causing the regulator to be acted on by the changes of pressure on a diaphragm which is connected by an air-pipe with an open vessel fixed under the stern of the ship. None of these devices has met with general adoption, and marine engines run, as a rule, without any automatic governor.

[1] *Proc. Inst. Mech. Eng.* 1879.

CHAPTER XIII.

DYNAMICS OF THE ENGINE.

241. Fluctuations of Speed during any single revolution: function of the Fly-wheel. Besides those variations of speed which occur from stroke to stroke, which it is the business of the governor to check, there are variations within each single stroke over which the governor exercises no control. These are due to the varying rate at which work is done on the crank-shaft during its revolution. To keep them within reasonable limits is the function of the fly-wheel. It acts by forming a reservoir of energy to be drawn upon during those parts of the revolution in which the work done on the shaft is less than the work done by the shaft, and to take up the surplus in those parts of the revolution in which the work done on the shaft is greater than the work done by it. To accomplish this alternate storing and restoring of energy the fly-wheel must undergo slight fluctuations of speed, whose range depends on the ratio which the alternate excess and defect of energy bears to the whole stock of energy the fly-wheel holds in virtue of its motion. The duty of the fly-wheel may be studied by drawing a *diagram of crank-effort*, which shows the work done on the crank in the same way that the indicator diagram shows the work done on the piston. The same diagram serves another useful purpose in determining the twisting and bending stress in the crank.

242. Diagram of Crank-Effort. The diagram of crank-effort is best drawn by representing, in a curve drawn with rectangular co-ordinates, the relation between the torque or moment which the connecting-rod exerts to turn the crank and the angle turned through by the crank. When the angle is expressed in circular measure, the area of the diagram is the work done on the

crank. Or instead of selecting the turning moment and the angle turned through as the two co-ordinates, we may take the tangential effort on the crank-pin as one co-ordinate, namely the force which is found when the thrust against the crank-pin is resolved along the tangent to the crank-pin's path, the other component being directed towards the centre of the crank-shaft and consequently exerting no turning moment. The linear motion of the crank-pin in its circular path is then taken as the other co-ordinate of the crank-effort diagram; and the area still represents the work done upon the crank.

Neglecting friction for the present, and supposing in the first place that the engine runs so slowly that the forces required for the acceleration of the moving masses are negligibly small, the moment of crank-effort is found by resolving the thrust P of the
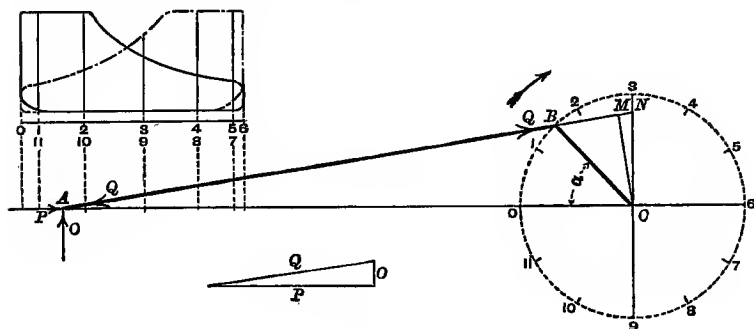
Fig. 197.

piston-rod into a component Q along the connecting-rod and a component O normal to the surface of the guide (fig. 197). The moment of crank-effort is

$$Q \cdot CM = P \cdot CN = Pr \sin \alpha \left(1 + \frac{r \cos \alpha}{\sqrt{l^2 - r^2 \sin^2 \alpha}}\right),$$

where CN is drawn perpendicular to the centre line or travel of the piston, r is the crank, l the connecting-rod, and α the angle ACB which the crank makes with the centre line. A graphic determination of CN is the most convenient in practice, unless the connecting-rod is so long that its obliquity is negligible, when the second term in the above expression vanishes. Fig. 198 shows the diagram of crank-effort determined in this way for an engine whose connecting-rod is $3\frac{1}{2}$ times the length of its crank, and in

which steam is cut off at about one-third of the stroke.   The thrust P is determined from the indicator diagrams of fig. 197 by taking the excess of the forward pressure on one side of the piston
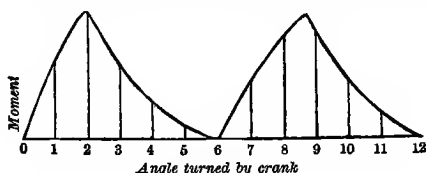
Fɪɢ. 198.   Diagram of Crank-Effort.

over the back pressure on the other side, and multiplying this effective pressure by the area of the piston.   The area of the diagram of crank-effort is the work done per revolution.

In the example for which this diagram is drawn it happens that there is very little compression of steam at the end of each back stroke, and consequently the forward pressure is greater than the back pressure throughout the whole of the stroke.   In many cases, however, the back pressure rises so much toward the end of the stroke that the resultant thrust on the piston opposes its motion, the diagram of resultant steam-

Fɪɢ. 199.

pressure taking a form such as that sketched in fig. 199, and consequently the ordinates in the corresponding part of the crank-effort diagram become negative.
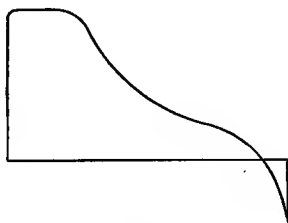
Another way of expressing the relation of the moment of crank-effort to the thrust P on the piston is to resolve the thrust Q along the rod into a component T in the direction of the tangent at B, and a component along BC.   The former alone exerts a moment on the shaft, and its moment is $T \cdot CB$.   To find T we have, by the principle of work, $T \cdot V_B = P \cdot V_O$, where V_B and V_O are the velocities of the crank-pin and piston respectively. Hence

$$T = \frac{P \cdot V_O}{V_B} = \frac{P \cdot IO}{IB},$$

where I is the instantaneous centre for the movement of the connecting-rod. I is found graphically by producing CB to meet

a perpendicular to OC from O.   Since $\dfrac{IO}{IB} = \dfrac{CN}{CB}$, the expression $T \cdot CB$ for the moment of crank-effort has the same value as that found before, namely $P \cdot CN$.

243.   Effect of Friction.   The friction of the piston in the cylinder and the piston-rod in the stuffing-box is easily allowed for, when its amount is known, by making a suitable deduction from P.   Friction at the guides, at the cross-head, and at the crank-pin has the effect of making the stress at each of these places to be inclined to the rubbing surfaces at an angle ϕ, the angle of repose, whose tangent is the coefficient of friction[1].   Hence the thrust O of the guide upon the cross-head instead of being normal to the surface of the guide, is inclined at the angle ϕ in the
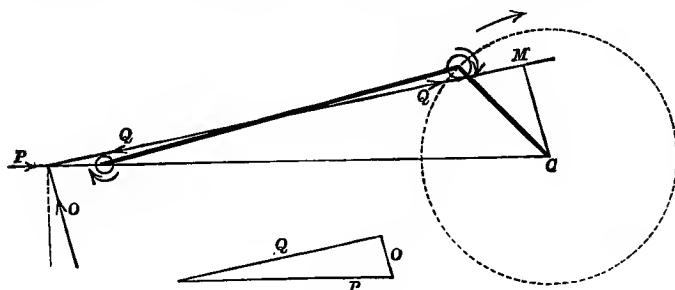
Fig. 200.

direction which resists the piston's motion (fig. 200); and the thrust along the connecting-rod, instead of passing through the centre of each pin, is displaced far enough to make an angle ϕ with the radius at the point where it meets the pin's surface.   To determine this displacement of the line of thrust let a "friction-circle" be drawn about the centre of each pin, namely a circle with radius equal to $p \sin \phi$, where p is the actual radius of the pin.   Any line drawn tangent to this circle will make the angle ϕ with the radius of the pin at the surface of the pin and will therefore satisfy the required condition as regards friction.   The thrust of the connecting-rod must be tangent to both circles; it must therefore be drawn as in fig. 200, so that it resists the rotation of

[1] This is what would take place if the friction at these surfaces were of the same kind as between dry solid bodies.   In fact however the presence of the oil, in a properly lubricated engine, makes the action so different that the construction given in the text cannot be held as applying in practical cases.

the pins relatively to the rod. The direction of rotation of the pins is shown by curved arrows in the figure, where the friction-circles are drawn to a greatly exaggerated scale. Finally, P (after allowing for the friction of piston-packing and stuffing-box) is resolved into O and Q, and then $Q \cdot CM$, the moment of Q on the shaft, is determined. This gives a diagram of crank-effort whose area is no longer equal to that of the indicator diagram. The difference, however, does not represent the whole work lost through friction in the mechanism, since the friction of the shaft itself, and of the valves and other parts of the engine which it drives, has still to be allowed for if the frictional efficiency of the engine as a whole is in question.

244. Effect of the inertia of the reciprocating pieces. The diagram of crank-effort is further modified when we take account of the inertia of the piston and connecting-rod, and the influence of inertia is generally much more important than that of friction. For the purpose of investigating the effects of the inertia of the reciprocating pieces, we may assume that the crank is revolving at a sensibly uniform rate of n turns per second. Let M be the mass of the piston, piston-rod, and cross-head in pounds, and a its acceleration at any instant in feet per second per second, the force required to accelerate it is $\dfrac{Ma}{g}$, in pounds-weight, and this is to be deducted in estimating the effective value of P. The effect is to reduce P during the first part of the stroke and to increase it towards the end, thereby compensating to some extent for the variation which P undergoes in consequence of an early cut-off. If the connecting-rod is so long that its obliquity may be neglected the piston has simple harmonic motion, and

$$a = -4\pi^2 n^2 r \cos \alpha,$$

when the crank has turned through any angle α from its dead point. More generally, whatever ratio the length l of the connecting-rod bears to that of the crank r,

$$a = -4\pi^2 n^2 r \left(\cos \alpha + \frac{r l^2 \cos 2\alpha + r^3 \sin^4 \alpha}{(l^2 - r^2 \sin^2 \alpha)^{\frac{3}{2}}} \right)^{*}.$$

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* To prove this, let θ be the angle BAC of fig. 197; then

$$\theta = \sin^{-1}\left(\frac{r \sin \alpha}{l}\right).$$

The effect is to make, on the diagram of P, a correction of the character shown in fig. 201 where the broken line cd refers to the case of an indefinitely long connecting-rod and the full line aeb to the case of a connecting-rod $3\frac{1}{2}$ times the length of the crank. In a vertical engine the weight of the piston and piston-rod is to be added to or subtracted from P

The form of the inertia line aeb of fig. 201 may be determined
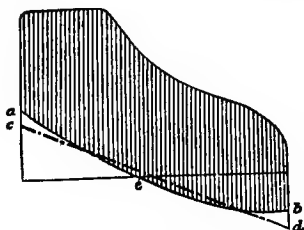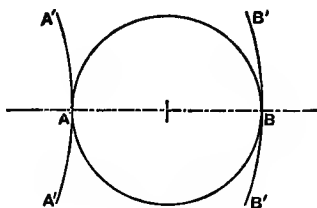
FIG. 201.   FIG. 202.

much more shortly and with sufficient accuracy for any graphic application by finding the points a and e and b as follows, and then sketching a smooth curve through these three points. The position of the point e in the stroke is found from the fact that since the acceleration is then zero the velocity of the piston is a maximum: this happens when the crank and connecting-rod are nearly at right angles (see next paragraph). The acceleration at a is the centrifugal acceleration due to the sum of the curvatures

Hence,
$$\frac{d\theta}{dt} = \frac{r\cos a}{\sqrt{l^2 - r^2\sin^2 a}} \qquad \frac{da}{dt} = \frac{2\pi n r \cos a}{\sqrt{l^2 - r^2\sin^2 a}}.$$

Differentiating again, and remembering that $\frac{d^2\theta}{dt^2} = 0$ since the rotation is assumed to be sensibly uniform, we obtain

$$\frac{d^2\theta}{dt^2} = \frac{-r(l^2-r^2)\sin a}{(l^2-r^2\sin^2 a)^{\frac{3}{2}}}\left(\frac{da}{dt}\right)^2 = \frac{-4\pi^2 n^2 r (l^2-r^2)\sin a}{(l^2-r^2\sin^2 a)^{\frac{3}{2}}}.$$

Again, writing x for AC (fig. 197),

$$x = r\cos a + l\cos\theta,$$

$$\frac{dx}{dt} = -r\sin a\frac{da}{dt} - l\cos\theta\frac{d\theta}{dt},$$

and
$$a = \frac{d^2x}{dt^2} = -r\cos a\left(\frac{da}{dt}\right)^2 - l\cos\theta\left(\frac{d\theta}{dt}\right)^2 - l\sin\theta\frac{d^2\theta}{dt^2}.$$

Substituting the values found above for $\frac{da}{dt}$, $\frac{d\theta}{dt}$ and $\frac{d^2\theta}{dt^2}$, and putting $r\sin a$ for $l\cos\theta$ and $\sqrt{l^2 - r^2\sin^2 a}$ for $l\cos\theta$, this gives the expression in the text,

$$a = -4\pi^2 n^2 r\left(\cos a + \frac{rl^2\cos 2a + r^3\sin^4 a}{(l^2-r^2\sin^2 a)^{\frac{3}{2}}}\right).$$

of the path of the crank-pin and of the arc AA' struck with l for radius (fig. 202). Similarly the acceleration at b (fig. 201) is due to the difference of these curvatures. Hence at a the acceleration is $\dfrac{v^2}{r} + \dfrac{v^2}{l}$ where v is the velocity of the crank-pin, and at b it is $\dfrac{v^2}{r} - \dfrac{v^2}{l}$. Substituting $2\pi n r$ for v in these expressions the acceleration of the piston is found to be

$$4\pi^2 n^2 r \left(1 + \frac{r}{l}\right) \text{ and } 4\pi^2 n^2 r \left(1 - \frac{r}{l}\right)$$

at a and b respectively.

245. Graphic construction for finding the acceleration of the Piston. A graphic method of finding the acceleration of the piston at any point in the stroke is shown in fig. 203. Produce AP to N and describe a circle with centre P and radius PN. Bisect the connecting-rod in E, and with E as centre and
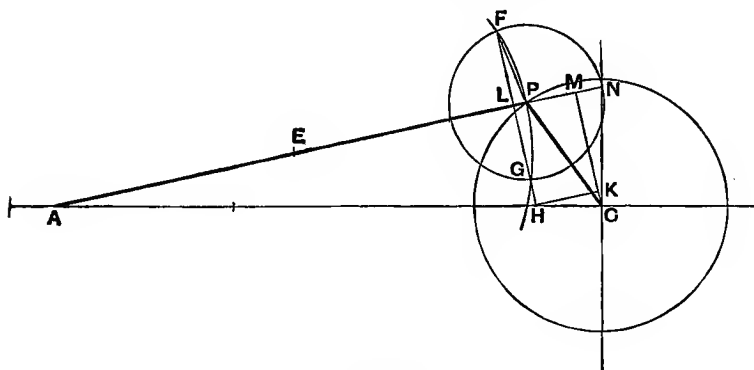
Fɪɢ. 203.

EP as radius draw a circular arc cutting the first circle in F and G. Join FG and produce it when necessary to cut the line AC in H. Then the length HC, when multiplied by the square of the angular velocity of the crank, gives the acceleration of the piston. In other words, if the length CP represents the radial acceleration of the crank-pin, HC represents the acceleration of the piston[1].

To prove this, draw CM perpendicular to PN and HK parallel

[1] J. F. Klein, *Journal of the Franklin Institute*, Vol. cxxxɪɪ. 1891. See also Kirsch, *Zeitschrift des Vereines deutscher Ingenieure*, 1890, p. 1320. Another construction, by Mr G. T. Bennett, is described by Professor Dalby in *The Balancing of Engines*, 2nd Ed. p. 257.

to LM. Find also I the instantaneous centre of the connecting-rod by producing CP through P to meet a line drawn at A perpendicular to AC. The triangle AIP (not shown in the diagram) is similar to NPC and $\dfrac{IP}{CP} = \dfrac{AP}{PN}$. For brevity we shall write ω for the angular velocity of the crank and ω' for the corresponding angular velocity of the connecting-rod. If v is the velocity of the crank-pin, $\omega = \dfrac{v}{CP}$ and $\omega' = \dfrac{v}{IP}$. Hence $\omega' = \omega \cdot \dfrac{CP}{IP} = \omega \cdot \dfrac{PN}{AP}$. The motion of the rod may be regarded as made up of (1) a translation with velocity v in the direction of the tangent to the crank-pin circle at P, and (2) an angular movement about P with angular velocity ω'. The acceleration of P along PC is $\omega^2 CP$. Resolve this into components along the rod and perpendicular to it. The component along the rod is $\omega^2 PM$, and this is also the acceleration of A in the direction AP, so far as A receives acceleration in consequence of the rod's movement of translation. The acceleration of A due to rotation of the rod about P is $\omega'^2 AP = \dfrac{\omega^2 \cdot PN^2}{AP} = \dfrac{\omega^2 \cdot PF'^2}{AP} = \omega^2 LP$, since $\dfrac{LP}{PF} = \dfrac{PF}{AP}$. This acceleration is in the direction AP.

The total acceleration of A in the direction AP is therefore $\omega^2 (LP + PM) = \omega^2 LM$, the other component of acceleration being perpendicular to AP.

Hence a, the acceleration of A along AC, is

$$\omega^2 LM \cdot \frac{HC}{HK} = \omega^2 HC,$$

which was to be proved.

Further, the component acceleration of A in the direction perpendicular to AP is $\omega^2 CK$. But this is made up of (1) a component of the general acceleration of the rod $\omega^2 CP$ due to its translation, namely $\omega^2 CM$, and (2) the acceleration due to rotation about P with angular velocity ω', namely $AP \cdot \dfrac{d^2\theta}{dt^2}$.

Hence $\omega^2 CK = \omega^2 CM - AP \cdot \dfrac{d^2\theta}{dt^2}.$

Or $AP \cdot \dfrac{d^2\theta}{dt^2} = \omega^2 LH*.$

---

* The author owes this extension to Prof. Dunkerley. Another construction, by Rittershaus, for graphically determining the acceleration of the piston will be found in Unwin's *Elements of Machine Design*, Vol. II. p. 72.

Collecting the results we have:—

The angular velocity of the connecting-rod varies as PN, being equal to $\dfrac{\omega \cdot PN}{AP}$.

The angular acceleration of the connecting-rod varies as LH, being equal to $\dfrac{\omega^2 LH}{AP}$.

The acceleration of the piston varies as CH, being equal to $\omega^2 CH$.

246. Position of the crank for which the piston has no acceleration. By means of this construction, or otherwise, the position of the crank may be found for which the acceleration of the piston is zero and its velocity a maximum. This happens in the diagram (fig. 203) when H coincides with C. The corresponding crank-angle α is given by the cubic equation in $\sin^2 \alpha$,

$$\sin^6 \alpha - n^2 \sin^4 \alpha - n^4 \sin^2 \alpha + n^4 = 0,$$

where n is the ratio of the length of the connecting-rod to that of the crank[1]. The following table gives values of the angle for various values of n, and also values of the angle at which the connecting-rod is tangent to the crank-pin circle.

Ratio of connecting-rod to crank	Angle from the dead point at which the velocity of the piston is a maximum	Angle at which the connecting-rod is perpendicular to the crank
2	67° 42′	63° 26′
3	73° 11′	71° 34′
4	76° 43′	75° 58′
5	79° 7′	78° 41′
6	80° 48′	80° 32′
7	82° 2′	81° 52′
8	82° 59′	82° 52′
9	83° 44′	83° 40′
10	84° 20′	84° 17′

On comparing the two it will be seen that the rough approximation made by taking as the position of no acceleration the place where the rod is tangent to the crank-pin circle introduces no serious error in ordinary cases.

[1] See Papers in *Min. Proc. Inst. C. E.* Vol. cxxiv. *et seq.* by Prof. Hill, Prof. Unwin, and Mr G. A. Burls. The above cubic equation in $\sin^2 \alpha$ for determining the position of maximum velocity will be found in Prof. Minchin's *Uniplanar Kinematics* (1882), p. 48.

247. Inertia of the Connecting-rod. The treatment of the inertia of the connecting-rod presents more difficulty than that of the piston. A fair approximation to the real effect is often arrived at by supposing part of the whole mass of the rod to be gathered at the cross-head, forming an addition to the mass which has simply a reciprocating motion, and the remainder to be gathered at the crank-pin, forming an addition to the rotating mass of the fly-wheel.

To obtain an exact solution the motion of the rod may be analysed as consisting of translation with the velocity of the cross-head, combined with rotation about the cross-head as centre. By means of this analysis, the force required for the acceleration of the rod is determined as the resultant of three components, namely, F_1, the force required for the linear acceleration a (which is the same as that of the piston); F_2, the force required to cause angular acceleration about the cross-head; and F_3, the force towards the centre of rotation, which depends on the angular
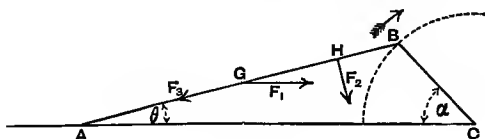
Fig. 204.

velocity, and is equal and opposite to the so-called centrifugal force. Let θ as before be the angle BAC (fig. 204), so that $\dfrac{d\theta}{dt}$ is the angular velocity of the rod about A, and $\dfrac{d^2\theta}{dt^2}$ is its angular acceleration, and let M' be the mass of the rod. Then, using gravitational units,

$$F_1 = \frac{M'a}{g},$$

and acts through the centre of gravity G, parallel to AC;

$$F_2 = \frac{M'(AG)}{g}\frac{d^2\theta}{dt^2}.$$

and acts at right angles to the rod through the centre of percussion H;

$$F_3 = \frac{M'(AG)}{g}\left(\frac{d\theta}{dt}\right)^2,$$

and acts along the rod towards A.

E. 27

The values of a and of $\dfrac{d\theta}{dt}$ and $\dfrac{d^2\theta}{dt^2}$ in relation to the crank-angle α have already been given, in the foot-note to § 244.

If now we imagine the directions of the forces F_1, F_2, F_3 to be reversed, these reversed forces will, when taken along with the weight of the rod, equilibrate the external forces applied to the rod at A and B. To draw the diagram of forces, refer to the joints A and B each of these reversed forces and also the weight. Then treat the rod as if it were a member in a frame, loaded at the joints and exerting simple thrust along its length. At A all the forces are known in direction but two of them are unknown in magnitude. These are found by drawing the polygon of forces for A, and then the polygon of forces for B gives the magnitude and direction of the force on the crank-pin.

248. Treatment of Inertia and Friction together. When in addition to the inertia of the rod, the friction at the cross-head and crank-pin is to be taken account of, the whole group of forces acting on the rod may be considered as follows. Compound forces equal and opposite to F_1, F_2, and F_3 into a single
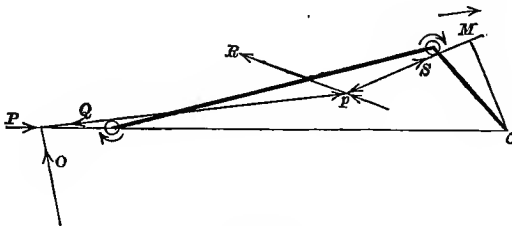
FIG. 205.

force R (fig. 205), which may be called the resultant resistance to acceleration of the connecting-rod. If the weight of the rod is to be considered, let it also be taken as a component in reckoning R. Then the rod may in any position be regarded as in equilibrium under the action of the forces Q, R and S, where Q and S are the forces exerted on it by the cross-head and crank-pin respectively. These three forces meet in a point p in the line of action of R, which point is to be found by trial, the condition being that in the diagram of forces, fig. 206, after the

FIG. 206.

triangle POQ has been drawn, and the force R set out, the force-line S shall be parallel to a line drawn from p tangent to the friction-circle of the crank-pin, as shown in fig. 205. When this condition has been satisfied by trial, the value of S, which is the thrust on the crank-pin, is determined, and then $S . \overline{CM}$ is the moment of crank-effort. This method is due to Fleeming Jenkin, who applied it with great generality to the investigation of mechanism in two important papers[1], the second of which deals in detail with the dynamics of the steam-engine. Fig. 207, taken from that paper, shows the diagram of crank-effort in a horizontal direct-acting engine,—the full line with friction, and the dotted line without friction,—the inertia of the piston and connecting-rod
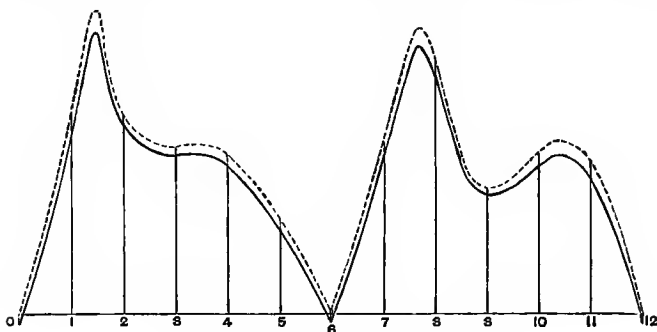
FIG. 207. Crank-Effort Diagram.

being taken account of, as well as the weight of the latter. It exhibits well the influence which the inertia of the reciprocating parts exerts to equalize the crank-effort in the case of an early cut-off. The cut-off is supposed to occur pretty sharply at about one-sixth of the stroke. The engine considered is of practical proportions, and makes four turns per second; and the initial steam-pressure is 50 lb. per square inch. It appears from the diagram that, with a slightly higher speed, or with heavier rods, a better approach to uniformity in the crank-effort might be secured, especially as regards the stroke towards the crank, which comes first in the diagram; on the other hand, by unduly increasing the mass of the reciprocating pieces or their speed the inequality due to expansion would be over-corrected and a new inequality would come in.

[1] *Transactions of the Royal Society of Edinburgh*, Vol. XXVIII. p. 1 and p. 703.

In drawing crank-effort diagrams it is scarcely necessary in practice to take account of the friction of the guide and of the pins, indeed the treatment of friction in the way here indicated is, for the reason already given, to be regarded as little more than an academic exercise. But the inertia of the piston, piston-rod and connecting-rod is of the utmost importance, especially in high-speed engines. The graphic method which is exhibited in figs. 205 and 206 of finding S, the thrust on the crank-pin, after R, the resistance to acceleration of the connecting-rod, has been determined, may of course be as readily applied when friction is neglected as when it is taken into account. Though for the sake of greater generality the influence of friction at the joints has been included, it must not be supposed to be a practically important factor in the problem. Neglecting friction, the lines of action of the forces P, O, and Q meet in the centre of the cross-head pin, and O is perpendicular to P. The line of action of S passes through the centre of the crank-pin. The forces P and R are known. By trial a point p is found, in the line of action of R, such that the forces Q and S in the force-diagram (corresponding to fig. 206) shall be parallel to lines joining p with the cross-head centre and the crank-pin centre respectively.

249. Forms of Crank-Effort Diagrams. When two or more cranks act on the same shaft a complete diagram, showing the resulting turning moment, may be drawn by combining the separate diagrams of crank-effort for the several cranks. An ex-
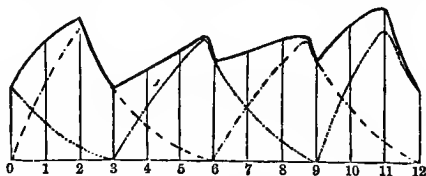
Fig. 208.   Crank-Effort Diagram for Two Cranks.

ample is shown in fig. 208, where the dotted lines are the separate diagrams for two cranks set at right angles to each other and the full line is the combined diagram. It is obvious that the inequalities of crank-effort are vastly reduced by using two cranks instead of one, and with three cranks the effort becomes still more uniform. An illustration of this is given in fig. 209, which also

exemplifies the circular form in which the diagram of crank-effort is sometimes drawn. In this construction lines proportional to the moment are set off radially from a circular line which represents the zero of moment. The figure is one drawn by Kirk for a triple-expansion marine engine with three cranks at 120° from each other. The curves show the resulting crank-effort, as determined from actual indicator diagrams and as affected by the inertia of the reciprocating parts.   They are drawn for various numbers of revolutions per minute, which are indicated by the distinguishing
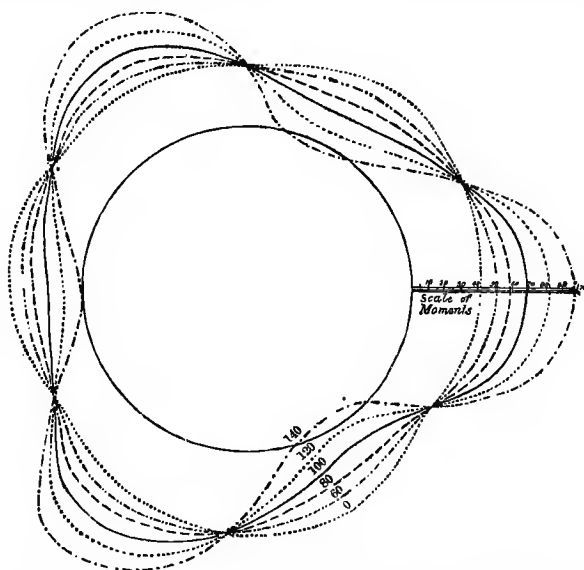
FIG. 209.   Circular Diagram of Crank-Effort for a Three-Cylinder Engine.

numbers, the line marked 0 referring to an indefinitely slow motion.

An opposite extreme to the nearly uniform crank-effort that may be obtained by the use of three or more cranks is found when we have a single-cylinder explosive gas- or oil-engine using the "Otto" cycle, in which under the most favourable conditions the whole effective action on the crank takes place only in one single stroke out of two revolutions (or four strokes), two of the other three strokes being idle, and the third being that in which the explosive mixture is compressed before ignition (see Chapter XVII.). The student will find it an interesting exercise to draw a crank-

effort diagram for such a case, extending the diagram over two revolutions to get a complete cyclic period, and then to apply the method described below of determining the size of fly-wheel which is necessary to prevent the speed from fluctuating beyond assigned limits. In the case of a gas-engine, however, it is not practically necessary to take into account the inertia of the reciprocating parts in order to find the amount of energy that has to be alternately absorbed and given out by the fly-wheel. That is readily determined, from the indicator diagram, by comparing the work done by the gas on the piston during the single effective stroke, and the work done by the piston on the gas during the compression stroke, with the mean amount of work got from the engine during the four strokes which make up the cycle.

250. Fluctuation of Speed in relation to the Energy of the Fly-wheel. The extent to which the fly-wheel has to act as a reservoir of energy in any engine is found by comparing the diagram of effort exerted on the crank-shaft by the piston or pistons with a similar diagram drawn to show the effort exerted by the crank-shaft throughout the revolution, in overcoming the resistance of the mechanism which it drives as well as the resistance due to its own friction. Like the driving effort, this resistance may be expressed as a torque or moment, or (dividing the
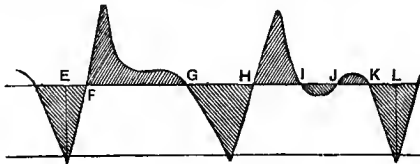
Fig. 210.

moment by the radius of the crank) we may state the equivalent resistance referred to the crank-pin as a force acting always tangent to the crank-pin's path. In general, except in dealing with direct-acting pumping and blowing engines, or engines which are compressing air or other gases, the resistance may be taken as having a constant moment on the shaft, and the diagram of effort exerted by the crank-shaft is then a straight line, as $EFGHIJKL$ in fig. 210. At F, G, H, I, J, and K the rate at which work is being done on and by the shaft is the same;

hence at these points the fly-wheel is neither gaining nor losing speed. The shaded area above FG is an excess of work done on the crank, and raises the speed of the fly-wheel from a minimum at F to a maximum at G. From G to H the fly-wheel supplies the defect of energy shown by the shaded area below GH, which represents the amount by which the demand for work exceeds the supply; the speed of the wheel again reaches a minimum at H, and again a maximum at I. The excesses and defects balance in each revolution if the engine is making a constant number of turns per second. In what follows it is assumed that the greatest excess or defect is only a small fraction of the whole energy stored up by the fly-wheel in virtue of its revolution, and consequently that the variations in speed are small in comparison with the mean speed. In practice the dimensions and speed of the fly-wheel are chosen so that this is the case: indeed the chief object of the investigation is to find what amount of energy must be given to the wheel in order that the variations in speed may not exceed a narrow prescribed range.

Let ΔE be the greatest single amount of energy that the fly-wheel has to give out or absorb, which is determined by measuring the shaded areas of the diagram and selecting the greatest of these areas; and let ω_1 and ω_2 be the maximum and minimum values of the wheel's angular velocity, which occur at the extremes of the period during which it is storing or supplying the energy ΔE. The mean angular velocity of the wheel ω_0 will be sensibly equal to $\frac{1}{2}(\omega_1 + \omega_2)$ if the range through which the speed varies is moderate. Let E_0 be the energy of the fly-wheel at this mean speed. Then
$$E_0 = \tfrac{1}{2}I\omega_0^2,$$

where I is the moment of inertia of the fly-wheel. Also
$$\Delta E = \frac{I(\omega_1^2 - \omega_2^2)}{2} = I\omega_0(\omega_1 - \omega_2) = 2E_0\frac{(\omega_1 - \omega_2)}{\omega_0}.$$

The quantity $\dfrac{\omega_1 - \omega_2}{\omega_0}$, which we may write q, is the ratio of the extreme range of speed to the mean speed, and measures the degree of unsteadiness which the fly-wheel leaves uncorrected. If the problem be to design a fly-wheel which will keep q down to an assigned limit, the energy of the wheel must be such that
$$E_0 = \frac{\Delta E}{2q}.$$

The periodic fluctuations of speed which are due to the limited capacity the fly-wheel has for storing energy may be examined experimentally by means of the familiar chronographic device of causing a vibrator, such as a tuning-fork electrically maintained in vibration, to scribe its oscillations on a surface which moves with the fly-wheel shaft. A sheet of smoked paper clasping the shaft itself forms a convenient surface, on which the fork draws an undulating line by means of a bristle or light pointed spring attached to one of its prongs. The fork should be mounted on a carrier such as the slide-rest of a lathe so that it may be kept moving slowly in a direction parallel to the axis of the shaft, in order that the records of successive revolutions may be traced on fresh portions of the smoked surface[1].

251. Influence of the Inertia of the Reciprocating parts on the Stresses to be borne. In the design of high-speed engines the stresses which arise from the movements of the reciprocating parts have to be taken account of in determining the proper form and dimensions, from the point of view of strength. Thus in designing, say, the forked end of the connecting-rod, which embraces the cross-head, we have to estimate the greatest thrust which will be exerted on it through the cross-head pin. This thrust will in general be divided equally between the two limbs of the fork and will produce a bending moment tending to open the fork. The thrust exerted on the connecting-rod through the cross-head pin is found by compounding the force of the steam on the piston with the force required to accelerate the mass of the piston, piston-rod, and cross-head. In a vertical engine the weight of these parts also enters into the account. The mass whose inertia comes into question here is the whole moving mass between the joint under consideration and the steam. Similarly in considering the thrust at the crank-pin we have to take account of all the reciprocating mass between that joint and the steam: this will include, in addition to the mass of the piston, piston-rod, and cross-head, a proportion of the mass of the connecting-rod. For the purpose of estimating the effect of the inertia of the connecting-rod at the two ends of the stroke, where the forces in question have their greatest values,

[1] For examples of the use of this method of finding q see Mr H. B. Ransome's paper, *Min. Proc. Inst. C. E.* Vol. xoviii., or the *Society of Arts Report on Trials of Motors for Electric Lighting* (1889).

we may regard the rod as dynamically equivalent to two masses, one concentrated at the crank-pin, and therefore contributing nothing to the forces due to reciprocation, and the other concentrated at the cross-head, and therefore forming part of the reciprocating mass.   In this reckoning the whole mass of the rod is to be divided into two parts, inversely proportional to the distance of the two ends from the centre of gravity of the rod, and these are to be regarded as concentrated at the ends.

252.   Reversal of thrust at the joints.   Let the diagram of resultant steam thrust upon the piston be represented by the line *SS* as in fig. 211 for the two successive strokes of a revolution,
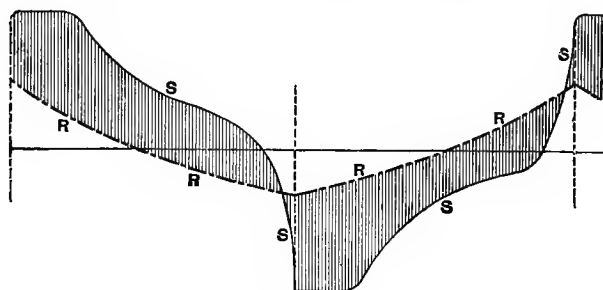
Fɪɢ. 211.

the line being drawn in such a way as to show that the steam is pushing the piston towards the crank when it lies above the base, and is pulling the piston away from the crank when it lies below the base.   Let the line *RR* represent in the same way the forces that are used up in producing the acceleration of the reciprocating pieces.   Then the points at which the steam curve *SS* crosses the inertia curve *RR* mark the places at which the direction of thrust at the bearings become reversed.   If in drawing *RR* the mass of the piston, piston-rod and cross-head only is taken account of the intersection of the two curves will show at what places the thrust changes its sign at the cross-head pin.   But if a suitable proportion of the mass of the connecting-rod also has been added in calculating the forces represented by this curve, the points where *SS* crosses *RR* will relate to the reversal of thrust on the bearing surfaces of the crank-pin.   Two inertia lines may be drawn, one referring to the masses between the steam and the cross-head pin, the other to the whole reciprocating mass, up to the crank-

pin.   Since the bearings are necessarily somewhat loose to admit of lubrication and free turning of the pins in their brasses, a sudden reversal of the thrust from pull to push at either joint will give rise to a knock.   To prevent an engine from knocking badly the clearance at the bearings is of course to be kept as small as possible, and the form of the thrust-diagram (fig. 211) should be such that when the steam and inertia curves cross each other the change from positive to negative in the distances intercepted between them shall be gradual.

253.   Prevention of reversal of the thrust in single-acting engines.   In some forms of high-speed single-acting engines this change is entirely avoided, and in that case the bearings may be left slack.   In the Willans engine for example the back is the active end and the piston and connecting-rod are kept in compression throughout the revolution.   During the stroke towards the crank this is their natural state, except when the speed is so great as to make the point a of fig. 201 rise above the steam thrust line.   But during the out-stroke there is nothing happening in the cylinder, except a little compression towards the end of the stroke, to provide the force that is required to reduce the velocity of the reciprocating pieces after the point of maximum velocity (near mid-stroke) has been passed.   Hence unless special provision for this force were made the connecting-rod would be pulling instead of pushing the crank-pin during the later portion of the out-stroke.   In the Willans engine the special provision consists in an air-cylinder, the piston of which is arranged tandem with the steam piston (or steam pistons, in the case of a tandem compound engine of this single-acting class).   The air in this cylinder begins to be compressed early in the out-stroke and becomes more and more compressed to the end, the energy which is expended in compressing it being given out again during the in-stroke or effective stroke of the engine.   The force exerted by the compressed air (along with that exerted by the steam during the up-stroke) is arranged to be always in excess of the force that is required for the (negative) acceleration of the pistons and rods, and hence the thrust both at the cross-head and at the crank-pin is continuously a push, never a pull[1].

[1] See Discussion on High-Speed Motors, *Min. Proc. Inst. C. E.* Vol. LXXXIII. 1885.

An example will help to make this clear. Let *dd*, fig. 212, be the indicator diagram of a single-acting vertical engine to which steam is admitted on the top of the piston only. It is required to find what amount of air-compression on the part of an air-buffer piston will serve to keep the thrust on the crank-pin from changing its sign at any point in the revolution. The line *aa* is the atmospheric datum line, representing the constant pressure which acts below the piston of the steam cylinder. The line *ii* represents the forces due to the inertia of the whole reciprocating mass which is carried by the crank-pin—namely, the piston and piston-rod of the steam cylinder and of the air-
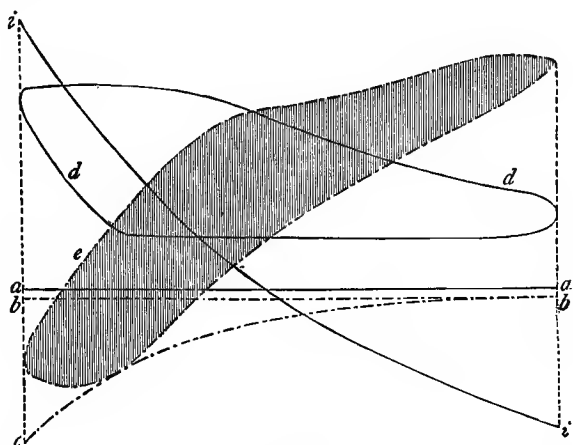
Fig. 212.

buffer and also the connecting-rod. The forces due to inertia are represented per square inch of piston area, to the same scale as the steam-pressures. Let the diagram *ee* be drawn to compound the forces due to inertia with those due to steam-pressure. In other words, let its ordinates above or below the datum line *aa* be the excess of the ordinates of *dd* above those of *ii*. So long as the figure *ee* lies above the datum line *aa*, the steam-pressure pushing the piston down exceeds the force necessary for acceleration, and consequently there is push, not pull, at the crank-pin. But when the figure *ee* comes below the line *aa* the force required for acceleration exceeds the force exerted by the steam. We must however take account of the weight of the reciprocating pieces, which assists the steam-pressure. This is readily done by shifting

the datum line down to *bb*, the distance *ab* representing the weight, expressed in lbs. per square inch of piston area.   The forces which are left to be balanced by the compression of air in the air-buffer are represented by the projection of *ee* below *bb*. Any such compression line for the air cylinder as *cc*, touching or lying wholly below the projecting part of *ee*, will therefore serve to prevent the force at the crank-pin from ever changing from a push into a pull.   In practice, in the Willans engine, there are generally two, or three, steam pistons arranged tandem on each piston-rod, and the steam diagram to be used in the foregoing construction would be a diagram representing the sum of the pressures on the two, or three, pistons.

Again, in a high-speed single-acting vertical engine which was made by Messrs Mather and Platt steam is admitted to the under
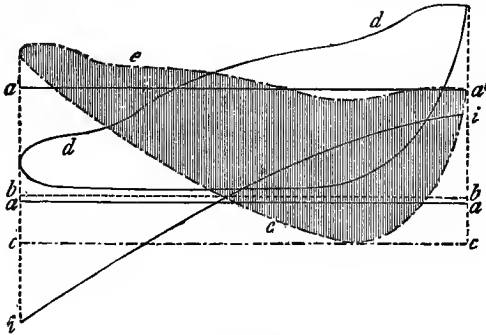
FIG. 213.

side of each piston and the rods are kept in tension instead of compression.   In this case the connecting-rod should always pull against the crank-pin, and never push.   This is effected by adding a balance-piston at the top of the cylinder which is continuously exposed to the full pressure of the steam on its lower side.   The case is illustrated in fig. 213.   There *dd* is the indicator diagram and *ii* the inertia line as before.   The datum line *aa* is not the atmospheric line *a′a′* but a line showing the pressure in the space above the piston, which space is connected with the condenser through the exhaust pipe.   As before, the indicator diagram *dd* is compounded with the inertia line to give the figure *ee*: the datum line is shifted, in this case, *up* to *bb*, to allow for the weight of the reciprocating parts, which now hinders instead of helping

the steam to keep the rod in tension. Then the projection of *ee* below *bb* shows what has had to be provided for on the part of the steam balance piston, and the pressure which the steam exerts on it must be at least equal to the height *bc* in order to prevent a reversal of force at the crank-pin.

254. Balancing. An important matter in the kinetics of the steam-engine is the balance of its working parts. A machine is said to be perfectly balanced when the relative movements of its parts have no tendency to make it vibrate as a whole. In other words, perfect balance implies that the reactions of those forces that are required for the acceleration of the parts should neutralize each other in every phase of the motion, so that no resultant reaction is ever felt by the bed-plate of the machine. A perfectly balanced machine would be self-contained as regards the stresses between the parts and would run steadily without foundations. Actual machines rarely do more than approximate to this condition. The question of balance is especially important in marine engines and in locomotives, but even in stationary engines it often requires attention, for a want of balance may give rise to serious nuisance through the vibrations which it sets up in the neighbouring ground.

The inertia effects in an engine are due in part to masses which simply revolve, such as cranks or eccentric sheaves, and in part to masses which reciprocate, such as pistons or valves. For the complete consideration of balance we have to deal with these separately.

255. Balance of Revolving Masses. Consider first the conditions which will give balance in a system of masses *revolving* about an axis, such as a number of cranks on the same shaft. It is convenient for this purpose to take as a plane of reference a plane situated anywhere along the axis and normal to it. Let the masses be M_1, M_2, M_3, etc.; let r_1, r_2, r_3, etc. be the respective radii at which they revolve, and let x_1, x_2, x_3, etc. be their respective distances from the assumed plane of reference. Calling the angular velocity of rotation ω, the centrifugal forces exerted by the masses, radially away from the axis and, in each case, in the plane in which the mass revolves are $M_1\omega^2 r_1$, $M_2\omega^2 r_2$, $M_3\omega^2 r_3$, and so on. Each of these forces is equivalent to an equal and parallel

force acting in the plane of reference, together with a couple the moment of which is equal to the value of the force multiplied by the distance x through which it has been shifted to bring it into that plane. Thus the system as a whole is equivalent to a group of forces in the plane of reference, namely,

$$M_1\omega^2 r_1, \quad M_2\omega^2 r_2, \quad M_3\omega^2 r_3, \text{ etc.,}$$

together with a group of couples

$$M_1\omega^2 r_1 x_1, \quad M_2\omega^2 r_2 x_2, \quad M_3\omega^2 r_3 x_3, \text{ etc.}$$

To secure balance the resultant of each of these two groups must vanish. In dealing with the proportions between the forces or couples we may omit ω^2 which enters as a constant factor into all. Hence if we draw, in the plane of reference, a system of vectors $M_1 r_1$, $M_2 r_2$, $M_3 r_3$, etc. in the directions of the several forces at any instant, lines equal and parallel to these vectors must, if the system is in balance, form a closed polygon; and further, if we draw, to represent the couples, a system of vectors $M_1 r_1 x_1$, $M_2 r_2 x_2$, $M_3 r_3 x_3$, etc., lines equal and parallel to them must also form a closed polygon. These vectors, which represent the moments of the couples, are to be drawn in the direction of the axes of the several couples: or, what will do equally well, they may be drawn respectively parallel to the radii r_1, r_2, r_3 at which the masses for the moment are revolving[1].

We therefore have, as the conditions of balance, the requirement that two polygons shall close, one for the forces and one for the moments of the couples which are introduced in transferring the several forces to the plane of reference. The balance is imperfect if there is either a resultant force or a resultant couple.

It follows from these considerations that if two masses are to balance they must revolve in the same plane, and that if three masses are to balance they either must lie in one plane of revolution or, if they are in different planes one (the middle one) must be 180° in advance of the other two: in other words, they must all three lie either in a plane of revolution, or in a plane which contains the axis of the shaft.

[1] This graphic method of investigating the conditions of balance appears to have been first given by Mr W. W. Taylor (*Journal of the American Society of Naval Engineers*, 1891). It was developed by Professor Dalby, whose book on *The Balancing of Engines* should be referred to for a full exposition of the general subject of balancing.

Four masses can be arranged to balance in an indefinite number of ways, by suitable selection of the relative values of Mr, of the crank-angles, and of the distances apart. This, as will be seen presently, forms a very important practical problem, especially with reference to the design of marine engines. With four masses to be considered, revolving at defined distances from the axis, there are in all nine independent variables, namely, the ratio of Mr for the second, third and fourth masses to the first, the crank-angles made by the second, third and fourth cranks respectively with the first, and the three distances apart of the cranks. It should be noted that the balance depends on the ratio of the masses, or rather of Mr, and not on the absolute values. Generally, if we have n masses the total number of independent variables is

$$3 (n-1),$$

made up of $n-1$ values of Mr, $n-1$ crank-angles, and $n-1$ distances from crank to crank along the shaft.

The two conditions for balance are that the Mr polygon shall close and the Mrx polygon shall close. The closing line in each polygon is determined by the lines already drawn. Each closing line fixes two independent variables. Hence the number of the independent variables which may be assigned beforehand, as data of the problem, is short of the whole number by four. In other words,

$$3 (n-1) - 4$$

is the largest number of variables which may be fixed beforehand without making the problem of attaining a balance insoluble.

256. Four-crank system. We may fix five of the quantities to begin with, and then by drawing the two polygons the other four quantities are found. We may, for example, have as data two masses at given radii, with the angle between their cranks, and the axial distances between the planes of all four masses. This means one ratio of Mr, one angle, and three distances: five quantities in all. By drawing the polygons we find the other two values of Mr and the other two crank-angles which are required to complete the determination of the system. The problem presents itself in this form in the balancing of locomotives when masses treated as revolving at the two cranks are to be balanced

by two other masses in the planes of the wheels. Or, to take a case arising in the design of a four-cylinder marine engine, we may fix to begin with the four planes and three of the masses (namely, three distances and two ratios of Mr), leaving the fourth mass and all the crank-angles to be found.

In the solution of any such problem it is a convenient artifice to take as the plane of reference the plane in which one of the unknown masses revolves. This eliminates one value of Mrx from the polygon of moments. That polygon is then drawn, and with the information afforded by the closing line the force polygon may next be drawn, which completes the solution.
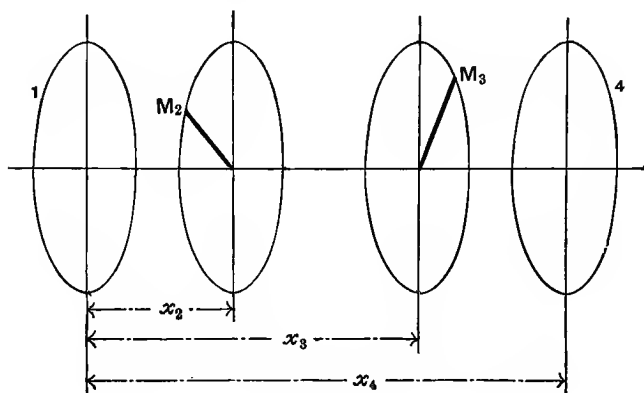
Fig. 214.

As an example, say that given masses M_2 and M_3 (fig. 214) are revolving on given cranks, and are to be balanced by masses at planes 1 and 4, the distances x_2, x_3, x_4 being given. Take (1) as plane of reference. Calculate $M_2r_2x_2$ and $M_3r_3x_3$. Since x_1 is zero, $M_1r_1x_1$ vanishes. Draw for the polygon of moments a triangle with $M_2r_2x_2$ and $M_3r_3x_3$ as sides and complete it by a third line. This line represents $M_4r_4x_4$ and since x_4 is known it fixes M_4r_4 both in magnitude and in angular position. Next take the known values of M_2r_2, M_3r_3 and M_4r_4 and with them draw the force polygon, which is a four-sided figure, the closing line of which is M_1r_1. Hence M_1r_1 is also fixed in magnitude and direction. No difficulty will be found in applying a generally similar method to other cases[1] or in extending it to a larger number of cranks.

[1] For numerical examples see Prof. Dalby's book already cited.

257. Balance of Reciprocating Masses. It is easy to extend these considerations to the important practical case of reciprocating masses such as the pistons of a steam-engine. To simplify the problem we shall as a first approximation treat the pistons as having sensibly simple-harmonic motion—a condition which is more and more closely approximated to the longer the connecting-rods are relatively to the cranks. A reciprocating mass such as a piston cannot be completely balanced by a revolving mass, or by any combination of masses revolving on one shaft, for the force due to acceleration of each reciprocating mass acts only in the line of its motion and therefore has an invariable direction, while the forces due to revolving masses change their direction continually. But a system of reciprocating masses may be arranged to secure a complete state of balance amongst themselves, by an application of the same method as has been applied in the foregoing paragraph. Any reciprocating mass M exerts on the frame of the engine a force Ma, where a is the acceleration, and since the motion is treated as simply harmonic, a is $\omega^2 r \cos \theta$ where θ is the angle between the crank and the centre line. Thus the disturbing force, which acts along the line of motion, is $Mr\omega^2 \cos \theta$. This is equal to one component of the force $Mr\omega^2$ which would be exerted by the mass at radius r if it were revolving instead of reciprocating, namely, the component in the direction of the stroke, the other component being taken at right angles to the stroke. In other words, if we imagine an equal revolving mass, concentrated at the crank-pin, to be substituted for the actual reciprocating mass, and resolve its centrifugal force along and across the line of stroke, the component along the line of stroke represents the disturbing force which the reciprocating mass in fact produces. Hence when we have a number of masses reciprocating in parallel lines, as for instance in the ordinary form of marine engine, if we substitute for each an imaginary revolving mass the conditions which would secure balance among the imaginary revolving masses would also secure balance among the reciprocating masses of the actual machine. These conditions, as we have seen, are that the two polygons should close which are drawn with their sides equal to Mr and to Mrx respectively. All that has been said regarding the balance of revolving masses, the number of independent variables, and the number of these variables which may be assigned as data of the problem, applies

equally to reciprocating masses, when these are considered as a system by themselves. It must however be understood that in an engine where there are real revolving masses to be considered as well as reciprocating masses, a state of complete balance can be secured only by making the revolving masses form a balanced system among themselves and the reciprocating masses also form a balanced system among themselves, the two systems being dealt with separately.

In such calculations the connecting-rod is to be regarded as belonging in part to the revolving masses and in part to the reciprocating masses, a portion of it being treated as concentrated at the crank-pin and the remainder at the cross-head. These parts are to be taken (as in § 251) in the inverse ratio of the distances of crank-pin and cross-head respectively from the mass-centre of the rod.

258. Four-Crank Marine Engine. Yarrow-Schlick-Tweedy System. These principles have received particular application in marine engines, where a want of balance, especially in high-speed engines, may cause much trouble by setting up vibrations of the hull. The consequences are specially serious when the speed of the engines is such that the impulses happen to agree in frequency with the free vibrations of which some part of the elastic structure of the ship is susceptible. Under these conditions the vibrations may become very large, and if the synchronism is approximate they may increase to a maximum, subside and increase again periodically.

In the usual form of marine engine the cylinders are arranged vertically over the crank-shaft, with their centre lines in a fore-and-aft plane. The three-cylinder type which became common on the introduction of triple expansion does not admit of balance except by setting the cranks at 180° from one another, a plan which would be impracticable on account of the difficulty of starting and the non-uniformity of the crank-effort. Taking the three-cylinder engine, Mr Yarrow showed in 1892 that its reciprocating masses could be effectively balanced by adding two others, which he called "bob-weights[1]," making five masses in all. In 1894 Mr Schlick[2] pointed out that by using four cylinders the

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[1] A. F. Yarrow, *Transactions of the Institute of Naval Architects*, 1892.

[2] Otto Schlick, *ibid.* 1894.

condition of balance might be secured in a practical manner by a suitable choice of crank-angles and relative masses. This arrangement has found extensive application in large merchant vessels and war-ships under the name of the Yarrow-Schlick-Tweedy system. The solution in any particular case may conveniently be arrived at by the graphic method already described, in which a plane of reference is taken through one of the cranks, and the two polygons, one of moments and the other of forces, are successively drawn. In general the distances along the shaft are assumed, and the masses for three of the pistons, which may in the first instance be calculated from the dimensions which, for other reasons, the pistons should have: with these data the fourth mass and the crank-angles are found. If necessary the masses of one or more of the pistons can be modified in the design by adding material for the purpose, until the solution gives suitable values both for them and for the angles. The designer may start by assuming values for the angles between three of the cranks, instead of three masses.

For a complete treatment of the practical problem the valves as well as the pistons have to be taken into account. For each valve there are in general two cranks, namely, the two eccentric sheaves of the link-motion. The engine is treated as running in full forward gear, that being the state for which balancing is most important, and the mass of the valve and valve-spindle accordingly reciprocates with the ahead eccentric. In a four-cylinder engine the consideration of the valve-gear brings the number of reciprocating masses up to twelve, but a good approximation may be obtained by substituting for each pair of eccentrics a single imaginary resultant eccentric[1].

259. Three-crank Engine. Balance of Forces but not of Couples. If the conditions are such that the force polygon closes but the moment polygon does not, there is no resultant force tending to move the engine as a whole in any one direction, but there is a resultant couple tending to make it rock or pitch on its bed. Take for example a three-cylinder vertical marine engine with cranks $120°$ apart, equal reciprocating masses M_1, M_2, M_3,

[1] For a numerical example of the solution taking valve-gears into account see Professor Dalby's book already cited. See also his paper on "A comparison of Five Types of Engines, etc." *Trans. of the Inst. of Naval Architects*, 1902.

equal strokes, and equal distances along the shaft (fig. 215).
Here the force polygon is an equilateral triangle, and there is no
resultant force on the engine as a whole. There is left however
an unbalanced couple which tends to make the engine pitch in
the fore-and-aft line AB. The closer the spacing of the cranks
the smaller this couple will be.
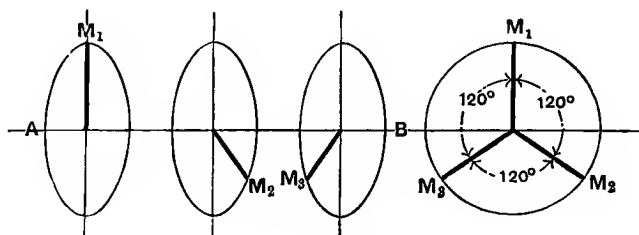
Fɪɢ. 215.

260. Balanced Six-crank and Five-crank Engines.
Taking the engine of fig. 215 let it be duplicated by the addition
of three more cylinders but with cranks in the reversed order,
namely, M_4, M_5, and M_6 in fig. 216, the second set being like
the first but forming a mirror-image of them about a transverse
plane through the centre C. The second set, considered by
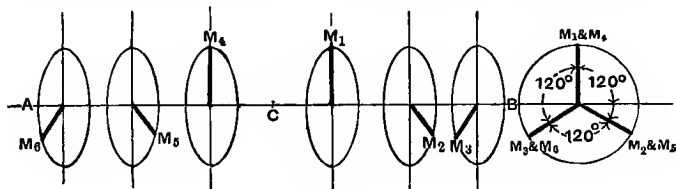
Fɪɢ. 216.

themselves, will have no resultant force, but will have a result-
ant couple equal and opposite to that of the first set. Hence
the couples neutralize each other, and this six-crank engine is
completely balanced as regards both forces and couples.

From this it is an easy step to derive a completely balanced
five-crank engine. We have only to suppose the two sets brought
close together so that the crank of M_4 coincides with that of M_1.
The reciprocating mass for that crank must then be equal to
$M_1 + M_4$: the other masses remain unaltered. This gives the

five-crank engine of fig. 217, which is also in a state of complete balance, having neither resultant force nor resultant couple.
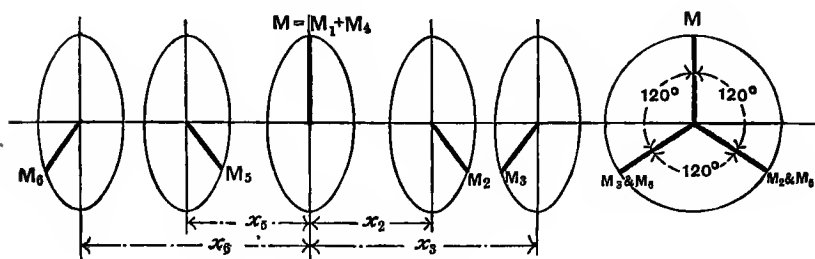
FIG. 217.

It is not essential that the masses should be equal and the pitches of the cranks equal, provided the following relations hold

$$M \text{ or } M_1 + M_4 = M_2 + M_5 = M_3 + M_6,$$

$$M_2 x_2 = M_5 x_5, \quad M_3 x_3 = M_6 x_6.$$

Hence it is sufficient that the masses M_2, M_3, M_5 and M_6 should bear the following ratios to the central mass M,

$$M_2 = \frac{M x_5}{x_2 + x_5}, \quad M_3 = \frac{M x_6}{x_3 + x_6}, \quad M_5 = \frac{M x_2}{x_2 + x_5}, \quad M_6 = \frac{M x_3}{x_3 + x_6}.$$

The pitches of the cranks can therefore be chosen arbitrarily, and masses determined from them which will form a balanced system, as regards both forces and couples.

It is assumed throughout these statements about balancing that the engine bed is sufficiently rigid to admit of no distortion, otherwise we could not regard the geometrical balance of forces and moments as resulting in no disturbance of the ground on which the engine stands, or of the hull of the ship.

261. Use of revolving masses to produce partial balance of reciprocating masses. Although a reciprocating mass cannot be correctly balanced by revolving masses it is often useful to neutralize, wholly or partially, the longitudinal forces on the engine-frame due to a reciprocating mass by applying revolving balance masses, though a necessary effect of this is to introduce a new disturbance by giving rise to transverse forces.

The simplest case is that of a single-cylinder engine. Here we have a reciprocating mass, consisting of the piston, piston-rod, cross-head, and a portion of the connecting-rod: we have also a revolving mass consisting of the crank and the remainder of the connecting-rod. The revolving mass is readily balanced by prolonging the two crank-cheeks towards the opposite side from the crank and adding sufficient mass there. This leaves unbalanced a longitudinal force only, due to the reciprocating mass. If now we increase the balance-weights on the crank-cheeks, on the opposite side from the crank, we may reduce the longitudinal force to any desired extent. But the extra masses so added exert transverse forces when they are out of the line of stroke. Thus, taking for example a horizontal engine, we may neutralize, in whole or in part, the forces which make the frame tend to move in a horizontal plane, but we do this only by introducing forces which tend alternately to raise it up from and push it down upon its foundations. Similarly in an engine with two or more cranks we may neutralize to any degree the longitudinal forces and couples by adding revolving masses, but only at the expense of introducing transverse forces and couples. A partial balance of longitudinal forces in this way may give the best compromise in practical cases.

262. Balancing Locomotives. Such a compromise is in fact carried out in the locomotive, where we have, in general, two horizontal cylinders, coupled by cranks at right angles. The practical problem is to add such revolving masses as will, on the whole, make the resulting disturbance as little objectionable as possible. If the reciprocating masses were left wholly unbalanced the forces due to their acceleration would not only cause violent variations in the tractive force but, what is still more serious, would apply a couple tending to make the leading wheels of the engine sway from side to side, and this effect might in extreme cases cause the engine to leave the rails. On the other hand, if the longitudinal forces were wholly balanced by means of revolving masses, large unbalanced vertical forces would arise causing variation of pressure between each wheel and the rail to which the name of "hammer-blow" is given. The name is scarcely appropriate, for the pressure varies not suddenly but gradually from a minimum to a maximum and back in each

revolution of the wheel. But in extreme cases (not permissible in practice) the centrifugal force may exceed the total weight borne by the wheel, and in that case the wheel will actually lift when the force is acting upwards and will return to the rail with a blow. Large variations of vertical pressure are objectionable in any event because they may cause slipping between wheel and rail by reducing the frictional adhesion.

The usual practical compromise is to make the revolving masses balance two-thirds of the forces due to reciprocating masses. The revolving masses introduced for this purpose are placed in the wheels. In determining their amounts the masses of the coupling-rods, which also revolve, have to be taken into consideration. The problem is readily solved on the lines already explained, treating the whole as a system of revolving masses, two-thirds of each reciprocating mass being supposed concentrated at the corresponding crank[1].

263. Balancing an engine with one horizontal and one vertical cylinder. An interesting special case is met with in a type of stationary engine where a horizontal and a vertical
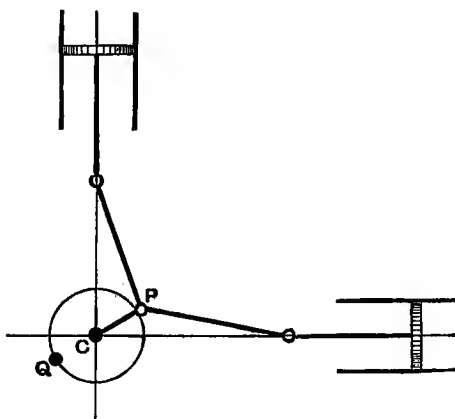
Fɪɢ. 218.

cylinder are combined as in fig. 218, both driving on one crank P. Treating the motion of each piston as simple harmonic, and making the pistons have equal mass, a complete balance may

[1] See W. E. Dalby, " On the Balancing of Locomotives," *Min. Proc. Inst. Mech. Eng.* Nov. 1901.

be secured by adding revolving balance masses opposite the crank, at Q. For the force due to a mass at Q, in any position, may be resolved into a horizontal and a vertical component, and these balance the respective pistons. The possibilities of balance which this type of coupled engine offers have been somewhat overlooked, and in stations where it has been erected vibrations have occurred. To prevent them it is only necessary to make the masses of the two reciprocating systems equal, and add a revolving mass at Q equivalent to the mass of either[1].

264. Secondary balancing. In what has been said about the balance of reciprocating parts we have treated them as having simple harmonic motion, ignoring the influence of shortness in the connecting-rod. In this way the conditions of what may be called *primary* balance have been arrived at. It remains to refer briefly to the forces that arise in consequence of the shortness of the rod, and to the conditions under which a further balance may be attained, among these forces. To this refinement of the problem the name *secondary balancing* is applied.

On the assumption of simple harmonic motion the force along the line of the stroke due to a reciprocating mass M is $M\omega^2 r \cos \theta$, where θ is the angle between the crank and the direction of the stroke. This would apply to a real piston if the connecting-rod were indefinitely long. At the two ends of the stroke, when the force is at its maximum, the value is $M\omega^2 r$.

With a rod of finite length l the force is (by § 244)

$$M\omega^2 r \left(1 + \frac{r}{l} \right)$$

at the end A of the stroke furthest from the crank-shaft, and

$$M\omega^2 r \left(1 - \frac{r}{l} \right)$$

at the end B of the stroke nearest the crank-shaft. Comparing this with the force $M\omega^2 r$ which would be found if the motion were simple harmonic we see that the actual force is obtained by adding $\left(\dfrac{rM}{l} \right) \omega^2 r$ at one end and subtracting the same quantity at the other.

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[1] Engines of this class, but not balanced in the manner here explained, working at the East Greenwich Electric Power Station have been found to disturb instruments at the Royal Observatory, half a mile away.

Hence, so far as the ends of the stroke are concerned, the actual effect is the same as if a mass equal to $\dfrac{rM}{l}$ were introduced, reciprocating with a simple harmonic motion due to a crank of the same radius but of twice the angular velocity, in addition to the mass M reciprocating with simple harmonic motion. This imaginary crank is to be thought of as coinciding with the real crank on the near dead-point, that is to say, when $\theta = 0°$. The imaginary mass $\dfrac{rM}{l}$ is then exerting a force in the same direction as the real mass. When the real crank has made half a revolution the imaginary crank has come round again to the near dead-point, and the force due to the reciprocation of the imaginary mass $\dfrac{rM}{l}$ is then opposed to the direction of the force due to M. Thus when $\theta = 0°$ the term $\dfrac{rM}{l}\,\omega^2 r$ is to be added, and when $\theta = 180°$ it is to be subtracted, which is the proper correction for the shortness of the rod.

Now suppose there is an engine comprising a system of reciprocating masses in "primary" balance, that is to say, properly balanced for simple harmonic reciprocation, but with connecting-rods which make their motion differ sensibly from the true simple harmonic form. Think of each as having associated with it an imaginary crank, agreeing in phase when at the near dead-point, revolving with double the angular velocity, and driving (also with simple harmonic motion) an imaginary reciprocating mass equal to $\dfrac{rM}{l}$, where l is the actual length of the connecting-rod. Let these imaginary masses be balanced amongst themselves: this secures the condition of "secondary" balance.

When it is attempted to apply these principles to the design of a four-crank engine, with the object of securing secondary as well as primary balance, it will be found impossible to satisfy the conditions completely. But the four-crank engine can be arranged in such a way that besides being balanced as regards primary forces and primary couples it is also balanced as regards secondary forces though not as regards secondary couples. To do this the engine is generally made symmetrical, as in fig. 219, so that $x_1 = x_3$, $x_2 = x_4$, $M_3 = M_1$, $M_4 = M_2$, $\alpha_3 = \alpha_1$, $\alpha_2 = \alpha_4$, and under these conditions balance of secondary forces (but not of

secondary couples), along with complete primary balance, is secured by a suitable choice of the ratios of mass $\frac{M_1}{M_2}$, of distance $\frac{x_1}{x_2}$, and of angles[1].

In designing a four-cylinder engine on this basis it is usual
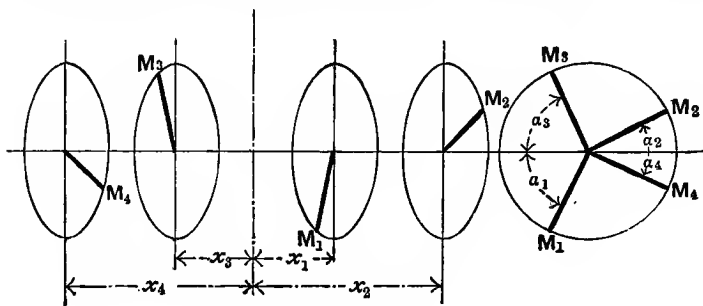
FIG. 219.

to place the two relatively heavy pistons over the central cranks (1) and (3), with the lighter pistons over (2) and (4), and to make the distances between (1) and (2) and between (3) and (4) smaller than that between (1) and (3).

In the practical solution of this problem the valve-masses have to be taken into account, which leads to some modification of the
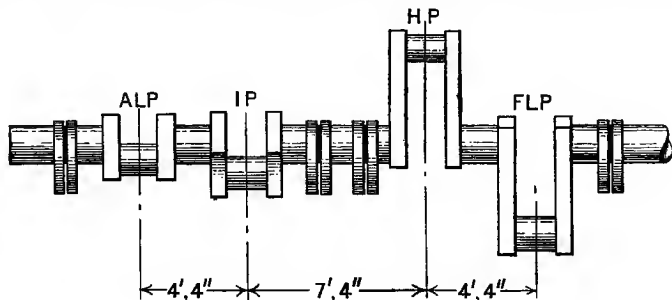
FIG. 220.

crank-angles. An example given by Dalby[2] of a balanced four-crank engine is shown in figs. 220-222, along with the resulting unbalanced forces and moments. Here a first approximation is reached by designing a symmetrical arrangement for the four

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[1] O. Schlick, "On Balancing Steam Engines," *Trans. Inst. Naval Architects*, 1900.

[2] W. E. Dalby, *Trans. Inst. Naval Architects*, 1902.
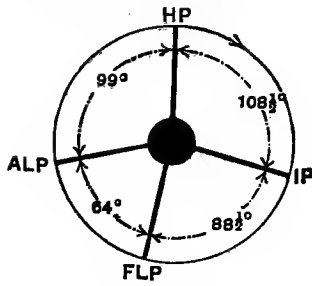
Fig. 221.
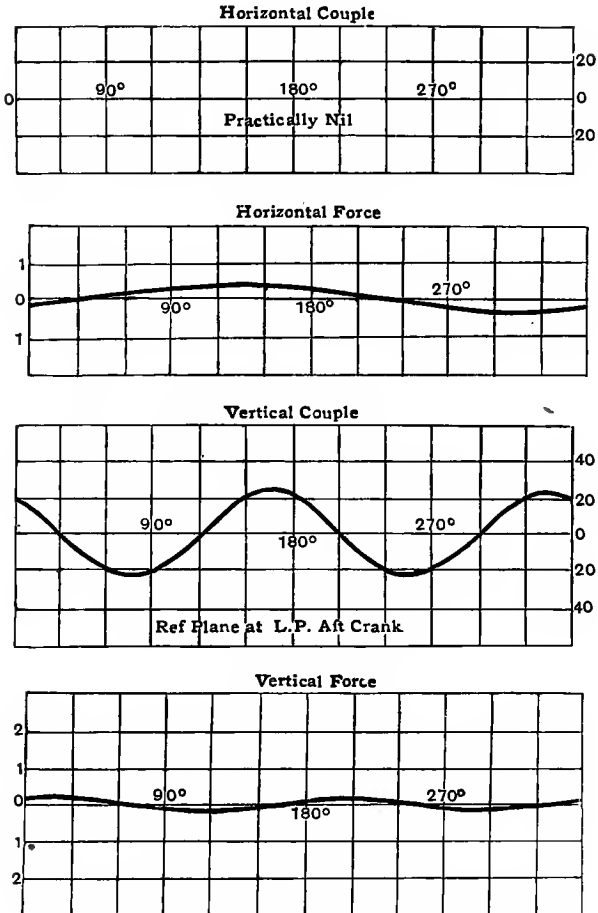
Fig. 222.

main masses only, and the design is then adjusted so as to include the valves. As the revolving masses are not separately balanced they are included with the reciprocating masses in the reckoning, and this has the effect of leaving a small unbalanced horizontal force. The vertical couple is the only serious part of the system that is left unbalanced, and as the diagram shows it is a secondary couple, for its period is half that of the revolution. It is instructive to compare, as Dalby does, the force and couple diagrams of a well-balanced engine such as this with those of other types in which balancing considerations have been more or less overlooked.

265. Application of Fourier's Series. The disturbances due to the shortness of the connecting-rod may usefully be treated in another manner, by analysing them into a series of simple harmonic terms[1]. The acceleration of the piston at any instant in its motion may be expressed in the form

$$\omega^2 r \left(\cos \theta + A \cos 2\theta - B \cos 4\theta + C \cos 6\theta \ldots \right),$$

where A, B, C, etc. are coefficients whose magnitude depends on the ratio of the length of the crank to that of the connecting-rod[2].

Thus a complete dynamical equivalent for the real reciprocation of the mass M is arrived at by superposing the effects of, first, an equal mass moving with simple harmonic motion with the actual period of the piston, then a smaller mass equal to A times M reciprocating with twice the frequency, then a still smaller mass equal to B times M with four times the frequency and so on, all these imaginary masses having simple harmonic motion with a stroke equal to that of the real piston. From this point of view we may state the condition for perfect balance in any system of actual pistons by saying that when for each piston this series of imaginary pistons is substituted, every one of the subordinate imaginary systems must be in balance within itself. That is to say, not only must the primary system balance but the system of

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[1] See a paper by J. H. Macalpine, *Engineering*, Oct. 22, 1897.

[2] Their values are:—

$$A = \frac{r}{l} + \frac{r^3}{4l^3} + \frac{15r^5}{128l^5} + \ldots,$$

$$B = \frac{r^3}{4l^3} + \frac{3r^5}{16l^5} + \ldots,$$

$$C = \frac{9r^5}{128l^5} + \ldots.$$

imaginary pistons whose masses are A times M_1, M_2, and so on, all of which reciprocate with double frequency, must balance in itself, and so must each of the systems with higher frequency. Speaking generally, it is the primary and secondary systems only that are important in practice.

With certain arrangements of engines it is possible to satisfy the condition of obtaining secondary as well as primary balance.

Take for instance the six-crank engine of fig. 216, where the crank-angles are 120°. We have already seen that this is in complete primary balance, as regards both forces and couples. Now think of the imaginary masses, forming the second group, which move with double frequency. When each real crank is at the near dead-point the corresponding imaginary crank is also at that dead-point, and when the real crank has moved through 120° the imaginary one has moved through 240° and so on. Hence the imaginary masses, forming the second system, also form a system with crank-angles of 120° and are in balance amongst themselves, both as regards forces and couples. This conclusion also applies to the five-crank engine of fig. 217. In both of these cases the conditions which secure primary balance secure secondary balance as well. The forces and couples due to the $A \cos 2\theta$ term are in balance, but those due to the $B \cos 4\theta$ term are not.

CHAPTER XIV.

THE PRODUCTION OF STEAM.—BOILERS.

266. Heating Surface. In the transfer of energy from fuel to steam two stages may be distinguished. First, the potential energy of combustion is transformed into actual heat, which shows itself in the raised temperature of the furnace gases; and, second, the heat of the furnace gases passes by conduction through the *heating surface* into the water of the boiler. The furnace gases serve as a vehicle for the conveyance of heat from the furnace or fire-box where it is generated to the various parts of the heating surface, some of which may be a long way from the actual seat of combustion. The heating surface is made up of the surface of the furnace or combustion-chamber, so far as that is brought into contact with the water, and of the flues or tubes through which the hot gases pass on their way to the chimney. The effectiveness of any portion of the heating surface depends not only on the difference in temperature between the gases and the water in the neighbourhood, but on the rapidity with which circulation takes place in the gases and in the water. Differences in specific conductivity and in thickness of metal are almost without effect on the result because the chief resistance to the flow of heat arises from the difficulty of getting heat to pass from the gas to the metal. On the one side, the steam that is generated on the surface opposes the conduction of heat and must give place to unevaporated water before much more heat can be taken in, and, on the other side, the gas which has parted with its heat to the metal must escape and be replaced by hot gas in order that the transfer of heat may continue. It is the circulation of the substances on either side, far more than the conductivity of the plate itself, that determines the rate at which heat will pass through a boiler shell.

267. Influence of the velocity of the gases on the effectiveness of the Heating Surface. It was pointed out by Osborne Reynolds[1] in 1874 that the rate at which heat is given off by a hot gas to any colder surface over which it flows depends primarily on the velocity of flow. If the gas were stagnant or nearly stagnant it would part with heat very slowly, for the portion of it in immediate contact with the metal surface would become chilled and would form a protective layer of very feeble conductivity, with the result that nearly all the temperature gradient would be expended across this layer. The passage of heat across the layer would be determined by gaseous diffusion between the coldest parts, close to the metal, and the hotter parts at greater distances from the metal, and diffusion would be competent to produce only a very gradual transfer of heat. But if the gas is moving past the surface at a sufficient speed to give rise to turbulence in the motion (and a very moderate speed will do that) we may regard this layer as being continuously formed and continuously broken down by eddies which stir it up, making a mixture between its contents and the hot gases of the warm stream. The faster the stream moves the more effective will this mixing or scrubbing action be, with the result that the rate at which the gases part with heat to the metal surface will be correspondingly increased.

Consequently the effectiveness of any part of the heating surface in a boiler depends, most materially, on the rapidity of movement of the hot gases over it. The faster the gases move the more heat they give up, and within wide limits the heat transmitted through a given surface is not far from proportional to the velocity of the gases[2].

[1] O. Reynolds, "On the Extent and Action of the Heating Surface for Steam Boilers," *Proc. Lit. and Phil. Soc. of Manchester*, 1874; reprinted in Prof. Perry's *Steam-Engine*, Chap. xxxiii. See also J. T. Nicolson, "On the Laws of Heat Transmission in Steam Boilers," *Proc. of the Junior Institution of Engineers*, Jan. 1909.

[2] Even under ordinary conditions of conduction of heat through the plates of a boiler, by far the greater part of the whole temperature difference between the furnace gases and the water is used up in overcoming the resistance of the chilled gaseous layer in contact with the plate. Dr Nicolson (*loc. cit.*) shows as a simple deduction from the known thermal conductivity of steel and the known rate of evaporation in a boiler that in ordinary cases the side of the plate next the gases is only about 5° C. hotter than the side next the water, although the temperature of the gases, a little way from the plate, is many hundreds of degrees higher.

The same considerations apply in the transfer of heat to, or from, a liquid[1]. It is important to have a scrubbing action on the water side, as well as on the gas side, of the heating surface, not only to facilitate the escape of the steam as it forms but also to promote rapid conduction from the metal to the water. This is supplied, more or less, by convection currents in the water, and by the circulation which the escaping steam itself sets up, but there are wide differences in different boilers in this respect. A mere statement of the number of square feet of heating surface conveys very little as an indication of a boiler's evaporative capability, and a given area may be ten-fold more effective in one situation or in one design than in another through rapid circulation of the hot gases and of the water, as well as through higher temperature in the gases themselves.

An illustration of these principles is afforded when a boiler is *forced,* that is to say when, by artificially increasing the draught, we compel a larger amount of coal to be burnt in a given time with the result that a larger amount of steam is generated, the area of the heating surface of course remaining unchanged. The rate at which heat passes across the surface is augmented by the higher velocity of the gases, and, in a less degree, by the higher velocity of circulation of the water. Each square foot of heating surface becomes much more effective, and the efficiency, which is the proportion of heat transferred to heat generated, is not in general reduced much if at all. The gases, at the end of their course over the heating surface, have given up nearly the same proportion of the heat they have received, as is shown by the fact that their temperature is little if at all higher than when the boiler works without forcing.

268. Additional Heating Surface obtained by the use of a Feed-water Heater. As the gases traverse the flues or tubes their temperature falls, until they finally escape at a temperature which is necessarily somewhat higher than that of the water to which they have been yielding up their heat. The temperature of the gases, however, need not be higher than or even as high as the temperature of the steam, for after ceasing to be in contact with

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[1] See experiments and theoretical discussion by Dr T. E. Stanton, "On the passage of heat between metal surfaces and liquids in contact with them," *Phil. Trans. Roy. Soc.* 1897.

the boiler proper the gases may continue to give up heat to a *feed-water heater*, which is a set of pipes through which the comparatively cold feed-water passes on its way to the boiler. The feed-water heater virtually forms an extension of the heating surface, with the advantage that it may be more effective for the transfer of heat than an equivalent extension of the boiler surface proper would be, on account of the lower temperature of the contents; and it allows the initial temperature of the feed-water, instead of the temperature of the steam, to form the lower limit to which the temperature of the gases might conceivably be allowed to fall. Conduction however would become so slow if the temperature of the gases approached this limit that in practice they are always considerably hotter. Even after passing a feed-water heater, the gases rarely have a temperature less than 200° C. When the draught through the fire is maintained by means of a chimney there is this independent reason for allowing the gases to escape at a relatively high temperature that the draught depends on the contents of the chimney being lighter than the air outside, and this lightness is secured by their being considerably hotter than the atmospheric air.

269. Draught. The furnace gases are made up of the products of combustion along with a quantity of air of dilution which passes through the furnace without undergoing chemical change. For the complete combustion of each pound of coal about 12 pounds of air are required to furnish the necessary oxygen, and usually about 12 pounds more have to enter as air of dilution. The greater part of this air enters through the grate, between the fire-bars on which the burning fuel rests, but some air has to be admitted above the fire to complete the burning of the combustible gases. This is specially necessary when fresh coal is thrown on the fire and volatile hydrocarbons are being given off. The furnace door has apertures to allow a small part of the air to pass through it, and these are often made adjustable in area.

A *natural* or *chimney* draught is one which is produced wholly by the lightness of the contents of the chimney. A *forced* draught is one in which other means are taken to produce a difference between the pressure of the air inside and outside of the furnace. A fan, for instance, may be used to force the draught,

either by extracting the gases from the flues or by blowing air into a closed room from which the furnace takes its supply. Or a jet of steam may be allowed to escape up the chimney, producing a partial vacuum there on the principle of the jet pump.

With a forced draught it is easy to produce much more difference in air pressure above and below the grate than can readily be produced by means of a chimney, and consequently to compel the entrance of a larger quantity of air through the fuel, with the result that a much larger quantity of coal can be burned per square foot of grate. A furnace using chimney draught does not as a rule burn more than 20 lbs. of coal per hour per square foot of grate, but with forced draught the combustion may go on at four or five times this rate and still be fairly perfect.

Further, when the draught is forced the combustion is intensified and localised, and it is found that a smaller proportion of air will suffice for dilution. Instead of the 24 lbs. or so of air which chimney draught requires per lb. of coal, 18 lbs. or less will serve. Hence with a forced draught the temperature of the furnace gases is higher, and as we have seen the effectiveness of the heating surface is increased on account of the more rapid movement of the gases. Again, since the proportion of air passing through the furnace is reduced by forcing the draught, the proportion of heat lost in the hot gases is also reduced, provided the action of the heating surface is such that they leave the flues at no higher temperature than before.

But the theoretical advantage of forced draught in respect of efficiency does not stop here. When the draught does not depend on the action of a chimney there is no need to let the escaping gases have any higher temperature than is imposed by the condition, already indicated, that they must be reasonably hotter than the temperature of the feed. With a chimney, on the other hand, as much heat is necessarily wasted as will keep the temperature of the escaping gases up to the comparatively high value necessary to maintain the draught. A chimney being an exceedingly inefficient form of heat-engine, the heat which is expended in maintaining its draught is vastly greater than the equivalent of the work that a fan would do in producing the same draught, or even than the heat that would have to be supplied to an engine employed in driving the fan.

In practical instances in which the draught is forced, namely,

in locomotives and in some marine and a few land engines, the
theoretical advantages of forced draught, in respect of efficiency,
are imperfectly realised. The draught is generally forced with
the object of increasing the power of a given boiler rather than
of securing a high efficiency. The motive is to burn more coal
per square foot of grate surface, and to evaporate more water in
a boiler of given weight. It is clear however that the most
efficient boiler would be one using a strong mechanically forced
draught, with a relatively small area of grate and with a heating
surface adapted both by its area and by the speed of the gases
over it to extract as much as possible of the heat, supplemented by
the use of a feed-water heater. Under these conditions not only
would the gases be cooled as far as possible before escaping, but
the proportion of air to coal would be as small as is consistent
with thorough combustion.

270. Sources of loss of Heat. Ordinarily about seven-
tenths and rarely more than eight-tenths of the potential energy
of the fuel are conveyed to the steam. The remaining two or three-
tenths are accounted for as follows:—(1) waste of fuel in the
solid state by dropping through the grate; (2) waste of fuel in the
gaseous and smoky state by imperfect combustion; (3) waste of
heat by external radiation and conduction; and (4) waste of heat
in the escaping gases due mainly to their high temperature, but
partly also to their containing as one of the products of combustion
a certain amount of water-vapour which passes off uncondensed.
Of these sources of waste the first is generally trifling and the
fourth is the most important. If we assume the air of dilution to
be 12 lbs. the whole quantity of gas escaping from the chimney
is 25 lbs. per lb. of coal burnt. The specific heat of this gas
is nearly the same as that of air, say 0·24 thermal unit. Hence
about 6 thermal units are lost, per lb. of fuel burnt, for every
degree by which the temperature of the escaping gas is allowed
to exceed the lowest attainable limit. It is not unusual to have
a chimney temperature as much as 250° or even 300° C. higher
than the limit which is imposed by the temperature of the feed-
water. This represents a more or less preventable loss of 1500 or
1800 thermal units (pound-degree-centigrade) per lb. of coal, or in
round numbers fully one-fifth of the whole energy of the coal.

With forced draught a great part of this loss might be avoided,

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by extracting more heat from the gases before they escape. At the best, however, some loss results from their high temperature, and it is important to keep their loss down by supplying no more air than is really necessary. Any unnecessary dilution of the gases means additional loss by increasing the quantity of the discharge relatively to the quantity of fuel that is consumed. It is obvious that to minimize the loss two conditions should be aimed at: the quantity of the furnace gases should be no more than is required for satisfactory combustion, and their temperature should be brought down to the lowest practicable figure by conduction of heat to the water before they are discharged.

271. Chimney Draught. In a chimney draught the "head" (usually stated in inches of water pressure) under which the current of air is kept up is equal to the amount by which the weight of a column of air in the chimney falls short of the weight of a corresponding column of outside air. Except for their excess of temperature the contents of the chimney would be heavier than the air outside, namely in the ratio of $n + 1$ to n, where n is the number of pounds of air which have taken up 1 lb. of fuel in passing through the furnace. The actual density of the gases is less than that of the air outside in the proportion $\dfrac{\tau}{\tau_0} \left(\dfrac{n+1}{n} \right)$ to 1, where τ and τ_0 are the absolute temperatures inside and outside respectively. The difference in actual density multiplied by the height of the chimney gives the effective head. This head is used up partly in setting the column of air in motion and partly in overcoming the resistance to its passage which is offered by the flue, by the chimney itself, and by the grate. With a forced draught and a short chimney the resistance of the grate is the most important of these items; with a tall chimney on the other hand the resistance of the chimney itself comes to be so considerable that an increase of height produces almost no increase of draught, and may even diminish the draught if the sectional area is at all reduced in the added part. Under such conditions also there is a limit in the extent to which the draught will be assisted by letting the chimney temperature remain high. In raising the temperature of the chimney gases a stage is reached at which the gain in head and consequently in velocity of current is more than counterbalanced by the diminution of density, and if the gases are

hotter than this the amount of gas passing through the chimney in a given time is actually reduced. In cases where the resistance is practically all met with after the gases have become heated— in other words, when the resistance of the grate is a very small part of the whole, the maximum draught is produced when the contents of the chimney have a density equal to half that of the air outside[1]. Assuming 24 lbs. of air to be admitted per lb. of fuel this condition is reached when the temperature in the chimney is about 315° C. or about the melting point of lead. When the resistance of the grate is a substantial part of the whole a rather higher temperature will make the draught a maximum. No advantage whatever is gained by making the temperature higher than corresponds to maximum draught, and on the score of thermal efficiency a lower temperature is of course to be preferred, as diminishing the heat lost in the escaping gases.

272. Coal and its Combustion. The chief constituents of coal are (1) carbon, (2) certain volatile hydrocarbons which may be distilled out by the application of heat along with a little free hydrogen, and (3) incombustible ash. In bituminous coal the hydrocarbons are present in considerable quantity, and when the coal begins to burn some of them exude in liquid form, tending to make it cake. Anthracite is a variety of coal in which they are almost entirely absent, and coals which are anthracitic are free from any tendency to cake. In the complete combustion of coal all the carbon, including that which is present in the hydrocarbons, should be converted into carbon dioxide (CO_2), and the hydrogen of the hydrocarbons is also completely oxidised, becoming water-vapour. Any appearance in the chimney gases either of carbon, in the form of smoke, or of carbon monoxide (CO) is evidence of incomplete combustion. Incomplete combustion may be due to an insufficient supply of air, but it often occurs, although the air supply is sufficient or even excessive, as a result of the gases having their temperature prematurely lowered, which generally occurs through their being brought into contact with surfaces which cool them below the point at which combustion will continue, before the combustion is complete. When fresh coal is thrown on the fire the volatile constituents are quickly distilled, and would escape unburnt or very imperfectly burnt if air for

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[1] See Rankine's *Steam-Engine*, p. 289.

their combustion were not admitted above the fire, and if care were not taken in the design of the furnace to provide a combustion-chamber of sufficient size. Incompleteness of combustion in the distillates from fresh coal is readily detected by the presence of smoke. But a smokeless fire is no proof that the air supply is rightly regulated. The supply may be excessive, which, as we have seen, results in serious waste of heat, or it may be insufficient, for the gases may contain carbon monoxide, which is an invisible gas. Important information may be obtained by testing the furnace gases for CO and also for CO_2. If CO is found the air supply is defective. The greater the quantity of CO_2 the more satisfactory is the combustion, provided there is neither CO nor smoke, for this means that the combustion is perfect with a small supply of air. Having measured the proportion of CO_2 it is easy to calculate what is the proportion of carbon in the whole weight of gas escaping to the chimney, and hence to infer what number of lbs. of air are being supplied per lb. of fuel. With suitable apparatus the test is readily applied, and instruments have been devised for giving a continuous indication of the proportion of CO_2 as a check on the efficiency of the firing.

The calorific value of coal varies somewhat widely. In some of the best kinds used for steam raising, such as " Nixon's navigation," it is about 8300 thermal units, in others of high quality it is 8000 units, but in coals containing much moisture or ash or both it may be only 7000 units or less. It is best determined by a laboratory experiment on a small scale, using a combustion calorimeter in which a sample of the coal previously ground to powder is electrically ignited and burnt in a closed vessel with a supply of oxygen to maintain the combustion, the vessel being enclosed in another containing a known weight of water, through which the burnt gases are allowed to escape. The rise in temperature of the containing water serves to measure the heat produced, after the calorimeter has been calibrated by observing the amount of temperature rise produced by the development within it of a known quantity of heat generated electrically.

In the process of burning which goes on in a furnace much the same considerations apply as have been mentioned in regard to the action of gases in giving up heat to a metal surface. On the surface of every fragment of ignited coal there must, if combustion

is to go on reasonably fast, be a scrubbing action which removes the inert layer consisting of the gaseous products of combustion and the nitrogen of the burnt air, and brings up fresh supplies of oxygen. This inert layer is continually being formed as combustion proceeds, and continually being removed through the action of the draught. Under a strong forced draught its removal takes place very rapidly: that is to say the average amount of inert layer becomes very small. Hence not only is combustion much stimulated but a smaller proportion of the whole supply of oxygen passes the fuel without being consumed. In other words, with a strong forced draught less air of dilution is required over and above the air whose oxygen actually enters into union with the fuel. Experience shows that this is so : with a natural draught, which is necessarily not very strong, the total quantity of air supplied has generally to be 24 or 25 lbs. per lb. of coal, but under favourable conditions with forced draught this may be reduced to 18 or even 16 lbs., without sacrificing completeness of combustion.

273. Cornish and Lancashire Boilers. In large stationary boilers the forms known as the "Cornish" and "Lancashire" are the most common. They are internally fired, that is to say the furnaces are enclosed within the water space of the boiler. The shell of these boilers is a long horizontal cylinder with flat ends, and within this, stretching from end to end within the water space, is a single large tube in the Cornish form and two parallel tubes in the Lancashire form, each tube containing a furnace at one end and communicating at the other end with external flues which are arranged to make nearly all the external surface of the shell below the water line act as part of the heating surface. The remainder of the heating surface is given by the large tube or tubes which contain the furnace, with the addition generally of several short cross tubes containing water, which traverse the main furnace tube at right angles to its length and not only serve the purpose of enlarging the heating surface, but promote circulation in the water, and strengthen the main tube. Fig. 223 shows a Cornish boiler in longitudinal section, and fig. 224 is a cross-section which shows the arrangement of the external flues. The furnace extends from the front up to the bridge of fire-brick *C*. In continuing their passage beyond this through the main tube or flue the hot gases come in contact with the cross tubes,
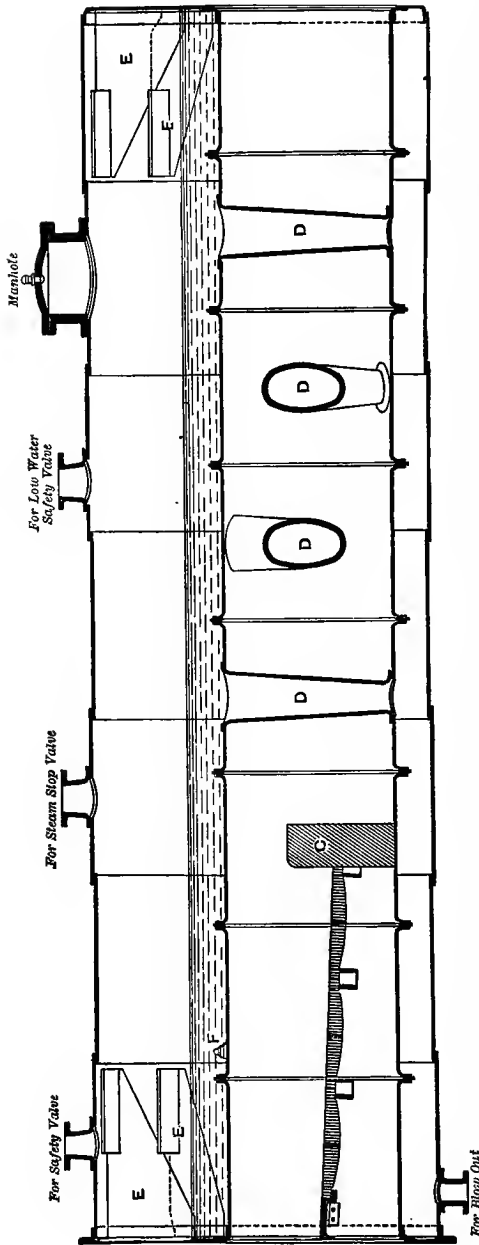
Fig. 223. Longitudinal section of Cornish Boiler. *C* bridge of fire-brick forming termination of the grate. *DD* Galloway tubes to stiffen the flue tube, to promote circulation of the water, and to increase the heating surface. *EE* gusset stays to support the flat ends. *F* fusible plug.

or Galloway tubes, DD, which have a somewhat conical form so that they may allow the steam formed in them to rise readily. At the end of the internal flue the gases are diverted downwards into the external flue B, and having traversed it towards the front of the boiler they are made to rise into the two side flues AA, by which they again pass to the back end and thence to the chimney. The form of the Lancashire boiler is essentially the same, except that there are two furnace tubes placed side by side, the diameter of the shell being larger. Fig. 225 is the cross-section of a Lancashire boiler. In a modified form of this boiler, introduced by Mr Galloway, the two furnace tubes unite beyond the bridge into one with a flat section, which is prevented from collapsing by having a number of Galloway tubes in it to act as stays.

The shell of a Lancashire boiler is commonly about 28 feet long, with a diameter of 7 feet, which allows each of the two furnace tubes to be 2 feet 9 inches wide. A boiler of this size, burning 20 lbs. or so of coal per hour per square foot of grate, will evaporate about 6000 lbs. of water per hour, or enough to yield, with an efficient condensing engine, from 300 to 400 indicated horse-power[1].

In boilers of this type the curvature of the cylindrical shell and furnace tubes enables them to resist the pressure of the steam : only the flat ends require to be stayed. This is done by means of gusset stays EE (fig. 223), which tie the end plate to the circumference of the shell, and often also by means of longitudinal stay-bolts stretching from end to end within the water space. The furnace flues are made up of a series of short welded lengths united by joints which give the whole tube stiffness to resist collapse, but leave it some freedom to bend when the top expands more than the bottom through the unequal action of the fire. To provide for unequal expansion is one of the most important points in the design of a boiler : when it is neglected a racking action occurs which induces leakage at the joints and tends to tear the plates. For this reason the furnace flues are attached only to the end plates and not to the cylindrical part of the shell, and the stays of the end plates are arranged to leave these plates some freedom to bulge out and in when the flues lengthen and contract.

[1] For particulars of the Lancashire boiler see a paper by Mr L. E. Fletcher, *Proc. Inst. Mech. Eng.* 1876.
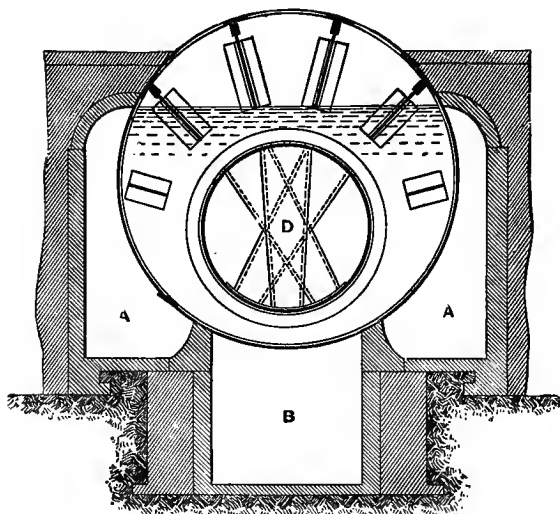
Fig. 224. Transverse section of Cornish Boiler.
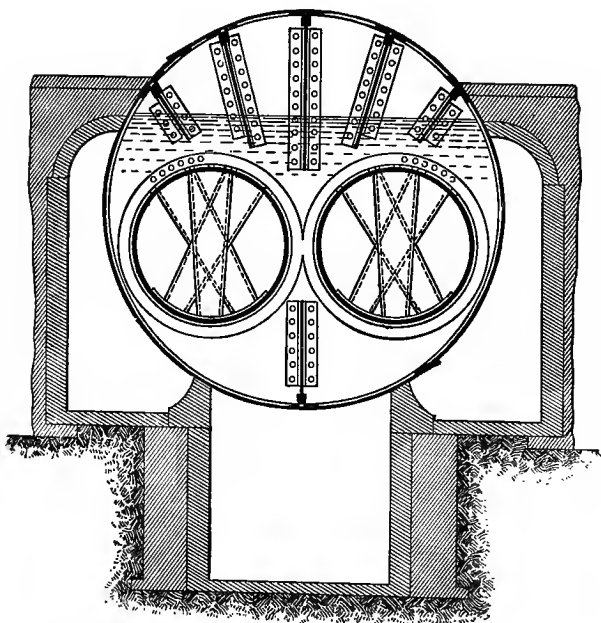
Fig. 225. Transverse section of Lancashire Boiler.

274. Boiler Mountings. The steam-dome, which used to be an ordinary feature in boilers of this type, is now generally omitted, and steam is taken direct from the steam space within the shell through a perforated "antipriming" pipe, from which it passes through a nozzle on the top of the boiler (fig. 223) to the stop-valve. The other openings on the top of the shell are the man-hole, on which a cover is bolted, and openings for two safety-valves. One of these valves is frequently of the dead-weight type, in which the force by which the valve is held closed is furnished by the direct action of a pile of weights: in many cases however springs and weighted levers are used. The second safety-valve is often arranged to form what is called a low-water safety-valve, being connected to a float in such a way that the valve will open if the water is allowed to sink below a safe level. At the bottom of the shell there is another nozzle for the blow-out cock, and in the front plate, below the furnace tubes, there is another man-hole. Feed-water is supplied by a pipe which enters through the front plate on one side near the top of the water and extends a good way in, distributing the water by holes throughout its length. A pipe at the same level on the other side serves to collect scum. On the top of each furnace is a fusible plug which melts if the furnace crown becomes overheated. On the front plate are a pair of glass gauge tubes showing the level of the water within and a pressure gauge of the Bourdon type. This important fitting consists of a metal tube, oval in section, which is bent into a nearly circular form. One end is closed and is free to move : the other is held fixed and is open to the steam. The pressure of the steam tends to make the oval section rounder and to straighten the tube. The free end accordingly moves through a small distance which is proportional to the excess of pressure within the tube above the atmospheric pressure to which its outer surface is exposed, and this movement is magnified by an index turning on a dial. Most of the fittings which have been mentioned are common to boilers of all types.

275. Multitubular Boilers. In several other forms of boiler an extensive heating surface is obtained by the use of a large number of small tubes through which the hot gases pass. This construction is followed in locomotive and marine boilers, and boilers of the typical locomotive and marine forms (to be

presently described) are, especially the former, frequently used with stationary engines. The multitubular construction is also applied in some instances to boilers of the ordinary cylindrical form by making a host of small tubes take the place of that part of the flue or flues which lies behind the bridge, or by using small tubes as channels through which the gases return from the back to the front after they have passed through the main flue. A favourite form, known as the *dry-back* boiler, consists of a short cylinder with flat ends, containing one or more cylindrical furnace tubes from which the gases pass at once into an external chamber behind the boiler. From this they return to the front through a group of many small horizontal return tubes placed at the side of or above the furnace tubes. Still another form of tubular boiler is an externally fired horizontal cylinder filled with return tubes extending from back to front: in this case also the boiler is a plain circular cylinder with flat ends. In all these forms the tubes are placed within the water space of the boiler. Except in some locomotives the tubes are commonly of iron, and a usual diameter is about 3 inches. They are usually secured in the tube-plates, in which they terminate by expanding the ends so as to make a steam-tight joint. They give so much heating surface that the outside surface of the shell need not be used, and hence in a tubular boiler the external flues are dispensed with which are a necessary feature of the Cornish or Lancashire type[1].

276. Locomotive Type. This is a multitubular boiler which, in its normal form, has a nearly rectangular fire-box at one end, surrounded by narrow water spaces, and a long cylindrical barrel extending horizontally to the other end, containing numerous horizontal tubes through which the gases pass from the fire-box to the smoke-box under the funnel. The heating surface is made up of the tubes and of the crown and sides of the fire-box. In most English locomotives the inner part of the fire-box, on which is the grate, is sufficiently narrow to be contained between the frames and wheels of the engine, but in America the need of locomotives of great power has led to the

[1] Numerous examples of these and other classes of boilers will be found described in Mr W. H. Fowler's *Practical Treatise on Steam Boilers and Supplementary Appliances.*

almost universal use of a wide fire-box extending over the frames and trailing wheels.

Figs. 226 and 227, which show an English locomotive boiler of a few years ago, will serve to illustrate the characteristic features of the locomotive type, although in recent years the augmented speeds and weights of trains have caused boilers of much larger power to become common. In this example the barrel is 10 feet long and a little more than 4 feet in diameter, and is made up of three rings of steel plates arranged telescopically. It contains 198 brass tubes, each $1\frac{7}{8}$ inches in external diameter. The front tube-plate in which the tubes terminate is of steel and is stayed to the back tube-plate by the tubes themselves, and the upper part of the front tube-plate above the tubes is also tied by longitudinal rods to the back end-plate. The fire-box is of copper and is nearly rectangular, with a horizontal grate. Round its sides, front, and back (except where the fire-door interrupts) is a water space about 3 inches wide, which narrows slightly towards the bottom. The flat sides of the fire-box are tied to the flat sides of the shell by copper stay-bolts, 4 inches apart, which are secured by screwing them into both plates and riveting over the ends. The crown of the fire-box is stiffened by a number of girders on the top, to which the plates are secured by short bolts. The girders are themselves hung from the top of the shell above them by slings which are secured to angle-irons riveted on the inside of the shell plates. A sloping bridge of fire-brick partially separates the upper part of the fire-box from the lower and prevents the flame from striking the tubes too directly. Under the grate is an ash-pan, to which the supply of air is controlled by a damper in front. The fire-door opens inwards, and can be set more or less open, to regulate the amount of air admitted above the fire. On top of the barrel is a steam-dome, from which the steam supply is taken through a pipe S traversing the forward part of the steam space and passing down to the valve-chest through the smoke-box. The stop-valve or "regulator" R is situated in the smoke-box, and is worked by a rod through the boiler from the cab at the back. Above the fire-box end of the shell are a pair of Ramsbottom safety-valves, V, V—two valves pressed down by a single spring attached to the middle of a cross bar, which is prolonged to form a hand lever by which the valves may be eased up to see

Fig. 227. Transverse section, half through fire-box and half through barrel.

Fig. 226. Locomotive Boiler, longitudinal section.

that they are free upon their seats. In front of the forward tube-plate is the smoke-box, containing a blast-pipe B by which the exhaust steam is used to produce a partial vacuum and so force a draught through the furnace.

Instead of stiffening the fire-box crown by the use of girder stays, the plan is sometimes followed of staying it directly to the shell above. The outer shell above the fire-box is often cylindrical, but the practice is becoming increasingly common of making it flat. This construction, which is known as the Belpaire fire-box, not only facilitates direct staying, but has the advantage of increasing the area of the water surface at the hottest part of the boiler, and so reducing the tendency for foam to form in the boiling. By making the height over the water sufficient (about 2 feet) it is found that dry steam may be obtained without the use of a steam-dome, and the dome is sometimes dispensed with.

Modern locomotive boilers have generally greater diameter of barrel, greater length between the tube-plates, more tubes with a much greater heating surface, and greater area of grate. Thus in the Great Western Railway example shown in fig. 228 there are 250 tubes of 2 inches external diameter and over 15 feet long[1], which with the fire-box give 2140 square feet of heating surface. The grate area is 27 square feet and the pressure is 225 lbs. It will be observed that the barrel tapers slightly, its diameter near the fire-box being as much as $5\frac{1}{2}$ feet. The fire-box is of the Belpaire type, with a flat top, 5 feet 9 inches wide in its upper part, but contracted to 4 feet at the bottom to get between the frames. This boiler supplies steam for an express engine which in ordinary running is exerting about 900 effective horse-power on the drawbar.

Fig. 229 illustrates a very large American locomotive boiler, of the Baltimore and Ohio Railroad, with a barrel over 7 feet in diameter, tubes nearly 21 feet long and a spreading fire-box no less than 9 feet wide over all, giving 72 square feet of grate area. There are 436 tubes of $2\frac{1}{4}$ inches diameter, giving in all about 5600 square feet of heating surface. This boiler supplies steam to two pairs of compounded cylinders, each 20 inches and 32 inches in diameter, with 32 inches stroke. The locomotive weighs

[1] For particulars see G. J. Churchward on "Large Locomotive Boilers," *Proc. Inst. Mech. Eng.* 1906. Figs. 228 and 229 are taken from Mr Churchward's paper.
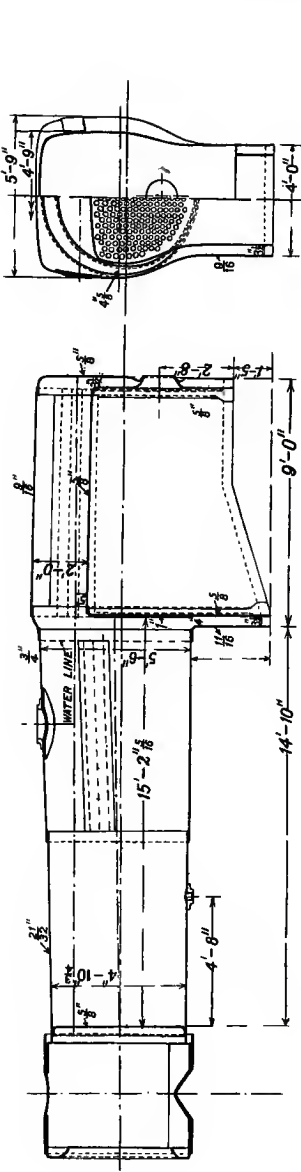
Fig. 228. Locomotive Boiler, Great Western Railway.
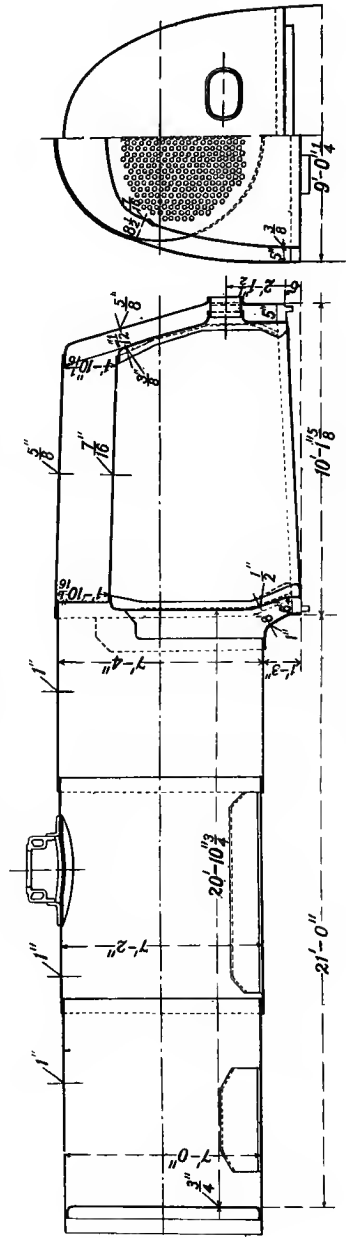
Fig. 229. Locomotive Boiler, Baltimore and Ohio Railroad.

144 tons, and the whole weight is taken on twelve coupled driving wheels.

The large spreading grates which characterise American practice are required not only on account of the large power that is wanted, but also to make it practicable to burn coal of comparatively poor quality. The spreading fire-box which is now universal came into use in the Wooton engines of the Philadelphia and Reading Railroad. In a few locomotive boilers water-tubes have been introduced in the fire-box : in some instances the fire-bridge is supported by water-tubes, and water-tubes are also used as grate-bars. This is done rather to promote circulation of the water than to give heating surface. The practice of American and English locomotive engineers differs somewhat as regards the materials of construction. American shells are of mild steel, English shells generally of mild steel but often of wrought-iron. In English practice the fire-boxes are of copper and the tubes of brass ; in America the fire-boxes are of mild steel and the tubes of wrought-iron. In England the fire-box stays are generally of copper or of bronze ; in America of wrought-iron.

The locomotive type of boiler is used for portable and semi-portable engines, and to a considerable extent for stationary engines of small and medium power. It has also been used in marine practice in cases where lightness is of special advantage, such as torpedo craft, but it has now given place on board ship to boilers of the water-tube type.

277. Multitubular Marine Boilers. Scotch Boiler. So long as marine engines used steam of a pressure less than about 35 lbs. per square inch the marine boiler was generally a box with flat sides, elaborately stayed, with a row of internal furnaces near the bottom opening into a spacious combustion-chamber enclosed within the boiler at the back, and a set of return tubes leading from the upper part of the chamber to the front of the boiler, where the products of combustion entered the uptake and passed off to the funnel. The use of higher pressures has made this form entirely obsolete. The ordinary boiler of the mercantile marine is now a short circular horizontal cylinder of steel, closed by flat plates at the ends, with internal furnaces in cylindrical flues, internal combustion-chambers, and return tubes above the flues. This type is often described as the Scotch boiler.

In one variety, called the double-ended boiler, there are furnaces at both ends of the shell, each pair leading to a combustion-chamber in the centre that is common to both, or to separate central chambers with a water space between them.

Figs. 230 and 231 show a double-ended marine boiler of this class, built for supplying steam at a pressure of 165 lbs. per square inch. At each end there are three furnaces in flues made of welded corrugated steel plates. The use of corrugated plates for flues, introduced by Mr Fox, makes thin flues able to resist collapse, and allows the flues to accommodate themselves easily to changes of temperature. One combustion-chamber is common to each pair of furnaces. It is strengthened on the top by girder stays and on the sides by stay-bolts to the neighbouring chamber and to the shell. The tubes are of iron, and a certain number of them are fitted with nuts so that they serve as stays between the tube-plate of the combustion-chamber and the front of the boiler. The upper part of the front plate is tied to the opposite end of the boiler by long stays extending from end to end above the combustion-chamber. The uptakes from both ends converge to the funnel base above the centre of the boiler's length. The boiler shown is one of a pair, which lie side by side in the vessel, the uptake at each end being common to both. Each boiler in this example has a steam-drum, which is a part generally omitted, and from it the steam-pipe leads to the engine ; it consists of a small cylindrical vessel, with flat ends tied together by a central stay. Short pipes connect the drum near each end with the steam space of the main shell. The shell in this example is $12\frac{1}{4}$ feet in diameter, and $16\frac{1}{2}$ feet long. The plates are of mild steel $1\frac{1}{8}$ inches thick round the shell and 1 inch in the ends, the corrugated flues are $\frac{1}{2}$ inch thick. There are 127 tubes at each end, 46 of which are stay-tubes. The tubes are of iron, $3\frac{1}{2}$ inches in external diameter. Above these are 18 longitudinal steel stay-rods extending from one end-plate to the others in the steam space. The crowns of the combustion-chambers are stiffened by girder stays, and their sides and bottom by short stay-bolts which tie them to one another and to the shell. In some cases the crowns are stayed by ties from the top of the shell, instead of by girder stays. Smaller tubes, of about $2\frac{1}{2}$ inches external diameter, are now more usual. Large Scotch boilers have as many as four furnaces. They may have as much as 6000 square feet of heating surface when
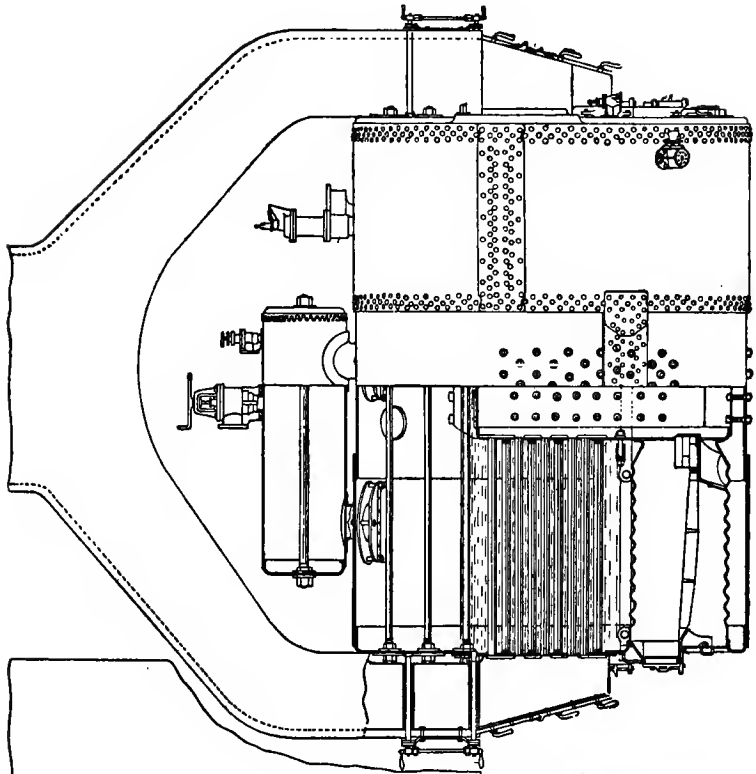
Fig. 231.
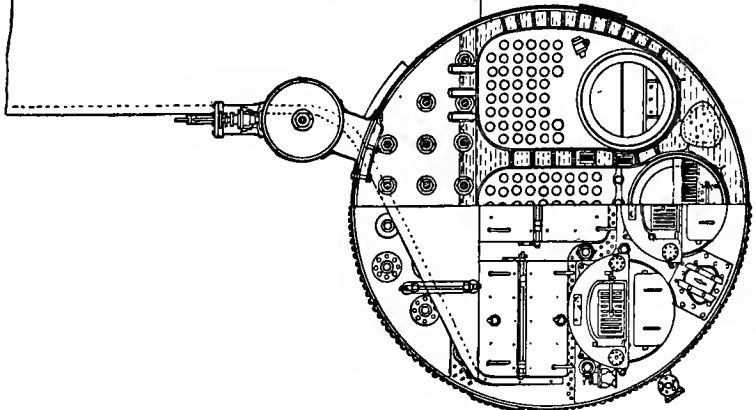
Fig. 230. Marine Boiler.

double-ended, or 3000 feet when single-ended. With natural draught and with pressures of 200 lbs. and over they develope from 10 to 12 horse-power per square foot of grate surface, the grate surface being about one-thirtieth of the heating surface.

The single-ended marine boiler is practically half a double-ended boiler. The furnace doors are at one end only, and the boiler terminates in a flat end-plate which leaves only a few inches of water space between it and the back of the combustion-chambers, the end plate and the back plate of each chamber being tied together across this space by short stay-bolts.

With but few exceptions the boilers used on shipboard are either of the Scotch type or of one of the water-tube types which have still to be mentioned. In merchant vessels the Scotch boiler is the usual form, but in warships, where saving in weight and rapidity in steam raising are specially important, the use of water-tube boilers has become almost universal.

278. Vertical Boilers. In the boilers which have been referred to the axis of the cylindrical shell is horizontal. But the cylinder may be turned up on end and the boiler take a vertical form, the grate of course remaining horizontal and forming the floor of a fire-box to which access is given by a door in the side of the cylindrical shell. Large vertical boilers are now uncommon, but the type is a very usual one for boilers of small power. It has the advantage of occupying little floor space, with the drawback that the free surface of the water from which the steam rises is comparatively small and consequently the steam rises with a higher velocity, which increases the risk of priming. There are many designs of vertical boilers, some with multitubular flues for the hot gases, and others with water-tubes. Fig. 232 shows an ordinary small vertical boiler with Galloway tubes across the upper part of the fire-box; and fig. 233 is another form, in which the water-tubes are curved channels which allow the water to circulate from the space round the sides of the fire-box to the space above the crown. In other forms of vertical boiler the heating surface is increased by water-tubes which hang from the crown of the fire-box and are closed at the lower end, circulation of water being maintained in them by means of a partition in the form of an inner tube inside of which water flows down to allow an upward movement of water and steam to be maintained

between the inner tube and the outer. Tubes of this kind, closed at one end and provided with an inner pipe to secure circulation of the water, are called Field tubes. A section of a Field tube is given in fig. 234, with arrows to indicate the manner in which the water circulates. A simple water-tube with a closed end forming a pocket would be quickly burnt out through being left dry if no provision were made for systematic circulation, but the inner tube of the Field combination makes the provision that is required.
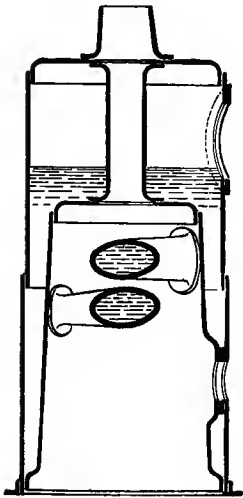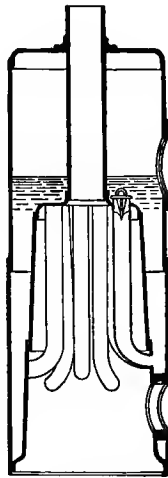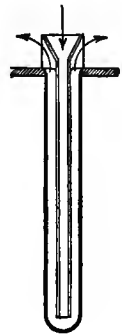
FIG. 232.   FIG. 233.   FIG. 234.

279. Water-tube Boilers. Many forms of boiler have been designed in which the firing is external, and the heating surface is made up of the outer surface of numerous tubes, through which a circulation of water is kept up in virtue of the differences in density between the hotter and colder portions of the water. In ordinary boilers the circulation is more or less casual: when a bubble of steam is detached from any part of the heating surface its place is taken by water which may come in from any side. In a properly designed water-tube boiler the circulation is systematic: water enters each of the tubes at one end and passes through in a continuous thin stream, becoming partly converted into steam as it goes. The tubes generally deliver into a separating vessel, from the upper part of which the steam-pipe takes its

supply, while water collects in the lower part to be returned by gravity to the lower end of the tubes.   Boilers of this type can be constructed so as to have, with their contents, a relatively small total weight in proportion to the rate at which they can make steam, which is a distinct merit in respect of marine and especially of naval use.   They also carry a relatively small weight of water, which makes them quick steam-raisers.   For erection in confined situations they have the advantage that they can be brought together in small pieces.   Further they are easily made strong enough to resist exceptionally high pressures owing to the absence of any large shell: an early tubular boiler, for instance, designed by Mr Loftus Perkins delivered steam at 500 lbs. per square inch. In many water-tube boilers the ordinary working pressure is 250 or 300 lbs.
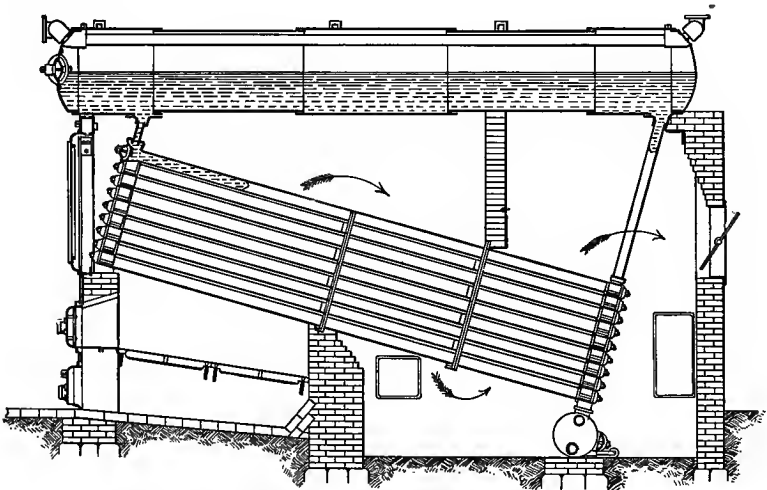
Fɪɢ. 235. Babcock and Wilcox Boiler.

280. Babcock and Wilcox Boiler. A successful example of this type, largely used both on land and on board ship, is the Babcock and Wilcox boiler, of which fig. 235 shows a form for use on land. The heating surface is almost wholly composed of a series of straight inclined tubes up which water circulates in parallel streams. These are joined at their ends by wrought-steel connecting boxes or "headers" to one another and also to a horizontal drum on the top in which the mixture of steam and water which rises from the tubes undergoes separation. At the

lowest point of the boiler is another drum for the collection of
sediment. The route taken by the hot gas is indicated by
arrows in the figure. When the boiler is in action a regular
circulation of water is established down from the drum to the
rear header, up the sloping tubes in which evaporation takes
place, and back by the front header to the drum.

For use on land this boiler generally has tubes about $4\frac{1}{2}$ inches
in diameter. In marine forms smaller tubes are used, a group of
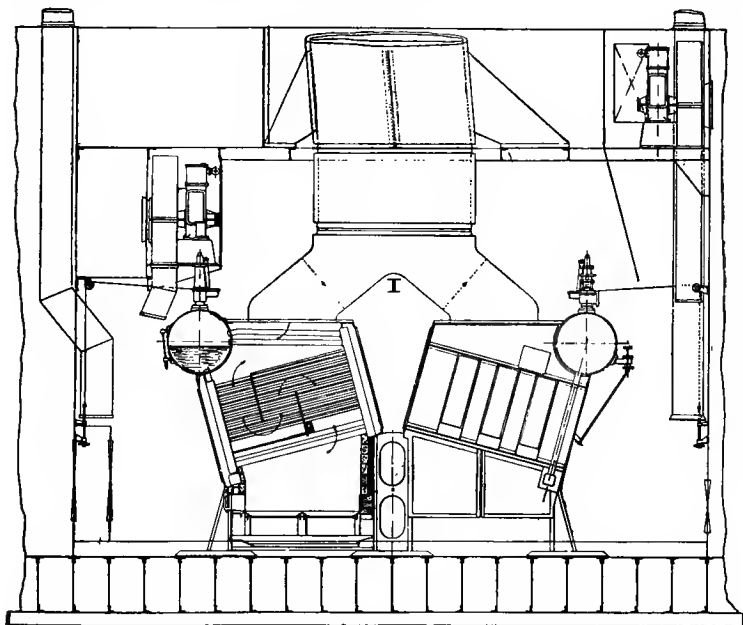
FIG. 236. Babcock and Wilcox Marine Boilers in place on board ship.

four tubes each about $1\frac{3}{4}$ inches in diameter taking the place of a
single large tube. The tubes in the marine type are commonly
set to slope up towards the back instead of down as in the land
form illustrated in fig. 235; the steam-drum extends trans-
versely across the top of the boiler and a group of nearly
horizontal tubes serves to connect it with the top ends of the
back headers. Fig. 236 is a longitudinal section through part
of a steam-ship showing a pair of Babcock and Wilcox boilers
of the marine type in place in the stokehold. The fire-doors
are under the lowest part of the sloping tubes.

281. Belleville Boiler. This is another water-tube type which was at one time extensively adopted in British as well as French warships. Since 1902, when it was unfavourably reported on by an Admiralty Committee, it has not been supplied to new ships of the Royal Navy. In the Belleville boiler the tubes are grouped in sets, each set forming a flattened helix through the whole length of which the water rises from the sediment chamber to the separating drum. The tubes are of steel, about $4\frac{1}{2}$ inches in diameter. They slope up with a gradient of about 1 in 25, alternately to left and right, forming a zigzag of straight lengths, which are made continuous by malleable cast-iron junction boxes. Each set or section has in all a length of about 150 feet through which the current of steam and water passes from end to end. Eight or more such sections stand side by side to make up the complete boiler. A non-return check-valve at the bottom assists in preventing the flow from taking place in the wrong direction. When the boiler begins to make steam the circulation occurs in a series of gusts, the check-valve closing while each gust makes its way up through the zigzag of tubes.

282. Stirling Boiler. In this boiler the heating surface is supplied almost wholly by water-tubes, the general direction of which is not far from vertical. The general arrangement is illustrated in fig. 237. The main tubes connect three steam drums above with a water drum below. At the back end the tubes stand in a nearly vertical position : at the front they slope steeply over the fire, providing a lofty combustion-chamber. They are grouped in three banks, with fire-brick baffles between, which make the hot gases take a zigzag course over the tubes. In addition to the main tubes there are short cross tubes, lying more or less horizontally, which connect the three steam drums with one another. The feed-water is delivered to the backmost steam drum and has first to go down the backmost bank of tubes, for there is no other exit below the water-line. The tubes are straight throughout the greater part of their length, but curve with easy bends towards each end so that they may enter the drum in a radial direction. The ends are expanded into the drum. The design is well adapted to save the structure from racking through changes of temperature. The tubes have a diameter of about $3\frac{1}{4}$ inches. Circulation takes place up the front bank of tubes and

also up the forward tubes of the second bank, and down the remainder. The backmost bank are in effect feed-water heaters rather than steam generators. The flat S-shaped bend which appears in the figure between the first and second bank of tubes is a superheater through which the steam passes on its way from the middle steam drum to the engine.
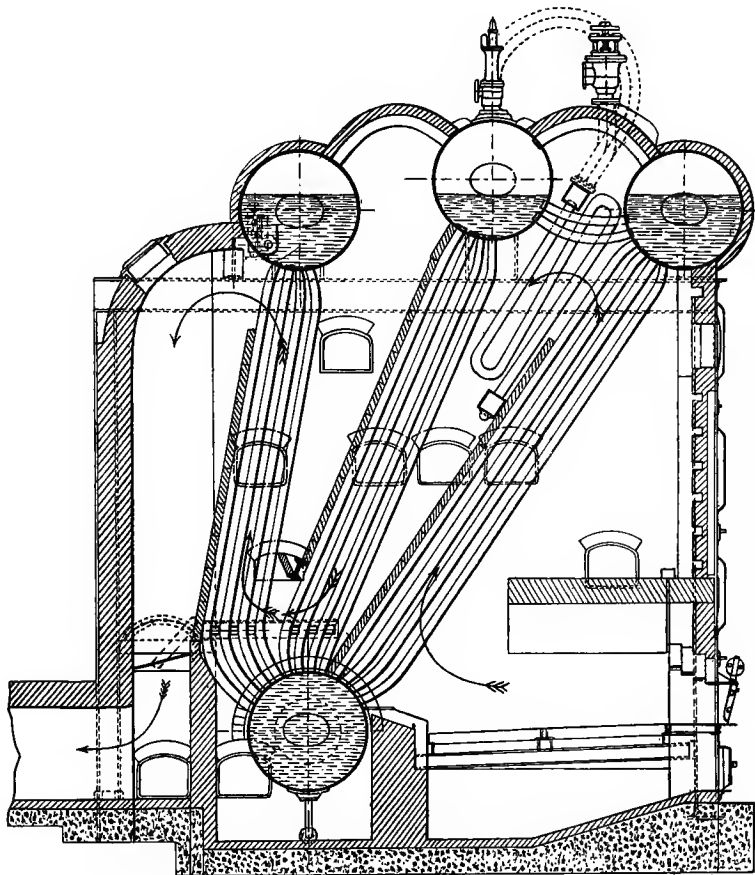
FIG. 237.   Stirling Boiler.

283. Niclausse and Dürr Boilers.   These are two forms much resembling one another, in both of which the heating surface is furnished by inclined " Field " tubes. The Niclausse boiler is illustrated in fig. 238, and fig. 239 is a section on a larger scale

through two tubes and headers.   The water comes down from the
drum at the top through the front header and is fed to each tube
through the inner supply pipe.   After evaporation the stream of
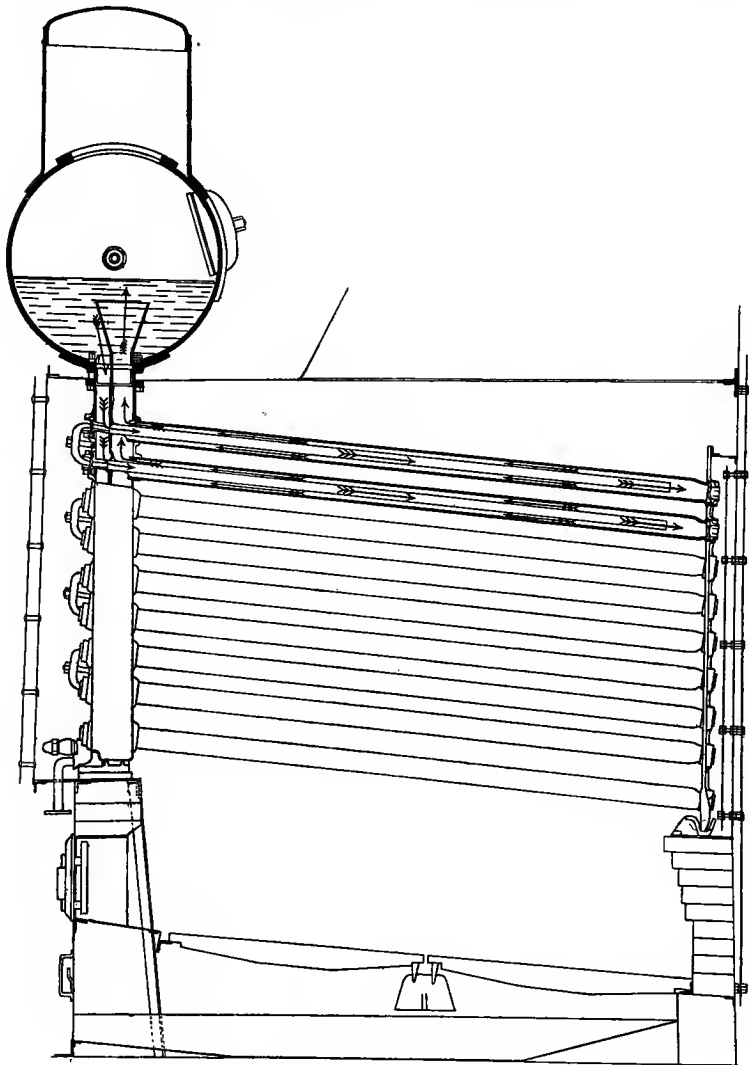
Fig. 238. Niclausse Boiler.

mixed steam and water rises to the drum through the somewhat
wider header nearer the tubes, which is separated by a partition
from the down-coming header.   The Niclausse boiler has been

extensively used in the French Navy as well as on land. The
Dürr, which is a German naval boiler, is generally similar but
differs in header details.
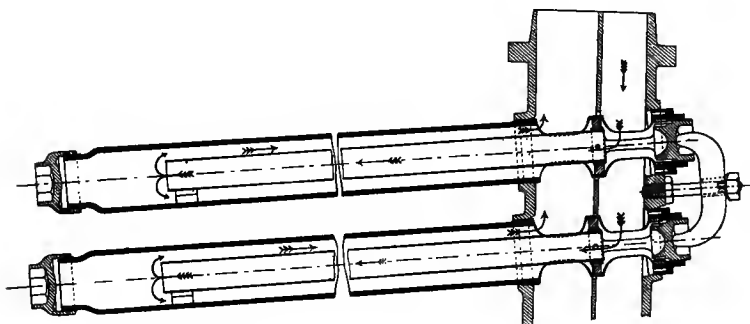
FIG. 239. Tubes of Niclausse Boiler.

284. Thornycroft Boilers. Special mention should be
made of the work of Mr J. I. Thornycroft in the development of
water-tube boilers using small tubes. One of his early forms[1],
installed in H.M.S. *Speedy* in 1885, is shown in fig. 240. Here
the heating surface is made up of tubes only an inch or an inch
and a half in diameter, which form an arch over the fire and after
bending out again terminate in the top of a separating drum
from which the water descends by a pair of external pipes to the
two drums which are seen at the base of the arch on either side.
One of these two "down-comers" is seen on the left side in
the illustration. A boiler of this class with 1837 square feet
of heating surface and 30 square feet of grate surface was
tested by Professor Kennedy under various degrees of forced
draught ranging up to a stokehold pressure of two inches of
water[2]. Under the highest pressure of air it made enough steam
to give 775 indicated horse-power in the engine ; the heat used in
making the steam was 67 per cent. of the whole energy of the fuel,
and nearly 70 lbs. of coal were burnt per hour per square foot of
grate. Analysis of the furnace gases showed that the supply of
air per lb. of coal was 17·2 lbs., and that about 9 per cent. of the
energy of the fuel was lost through imperfect combustion. In

[1] *Min. Proc. Inst. C. E.* Vol. xcix. p. 41.
[2] *Min. Proc. Inst. C. E.* Vol. xcix. p. 57.

another trial when the air pressure in the stokehold was equivalent to only half-an-inch of water the engine gave 450 indicated horse-power, and with practically the same supply of air per lb. of coal 78 per cent. of the energy was used in making steam and only
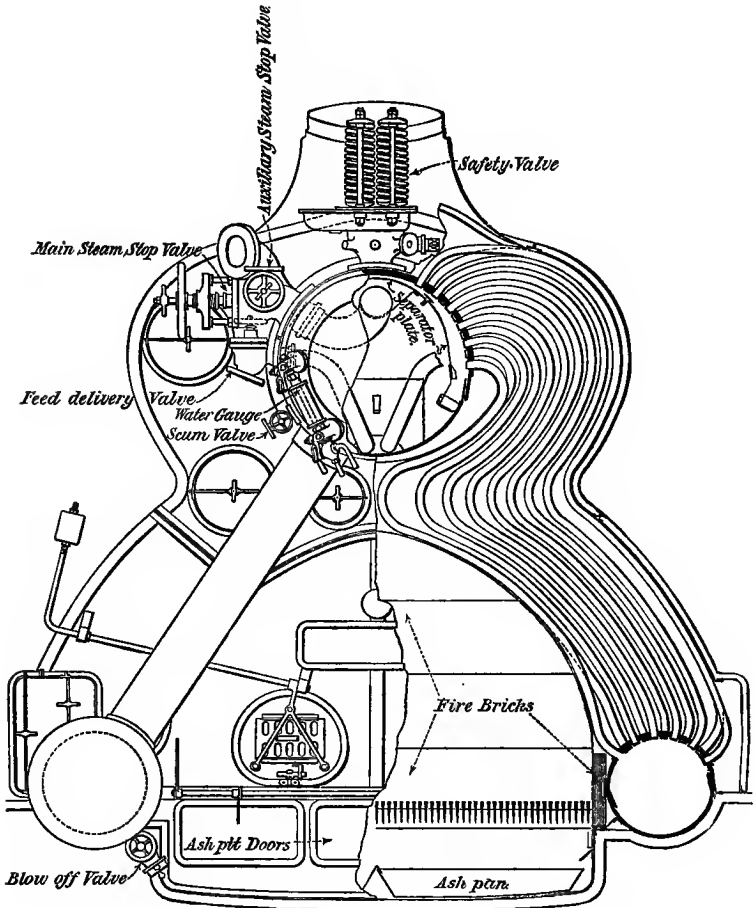
FIG. 240.   Thornycroft Boiler.

5 per cent. was lost through imperfect combustion.   These figures show that a boiler of this kind can make steam with great freedom and with but little reduction in efficiency even when the draught is strongly forced, while its efficiency at more ordinary

rates of output is remarkably high[1]. In this example the tubes discharge into the steam space of the upper drum, and this, while it favours the circulation and the separation of the steam, has the drawback that when the boiler is out of use part of the tube surface is exposed to moist air which makes it liable to corrode.
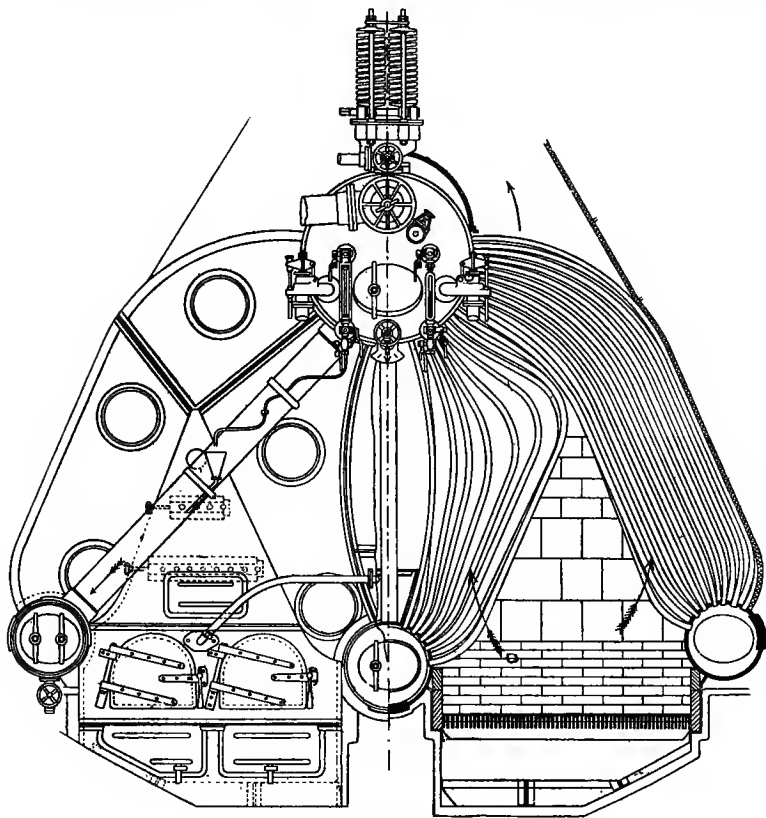
Fig. 241. Thornycroft-Schultz Boiler.

In more recent forms of the boiler this is avoided by giving the tubes a form which allows them to be kept full of water when the boiler is not in use, as in the Thornycroft-Schultz design of fig. 241, which also serves to illustrate a form in which there are

[1] For a discussion of water-tube boilers see the *Transactions* of the Institution of Naval Architects, 1894. Reference should particularly be made to the experiments of Professor Watkinson on model boilers with glass tubes, exhibiting the characteristic action in boilers of the Babcock, Belleville, Thornycroft, Yarrow, Niclausse, and other types (*Trans. Inst. Nav. Arch.* 1896).

three water-drums, the middle one separating two fire-grates. The Normand and Reed boilers are generally similar small-tube types, also with curved tubes.

285. Yarrow Boiler. In Mr Yarrow's form (fig. 242) the tubes are straight. They enter the steam-drum below the water-line, and the portions of the water-drums which form tube-plates
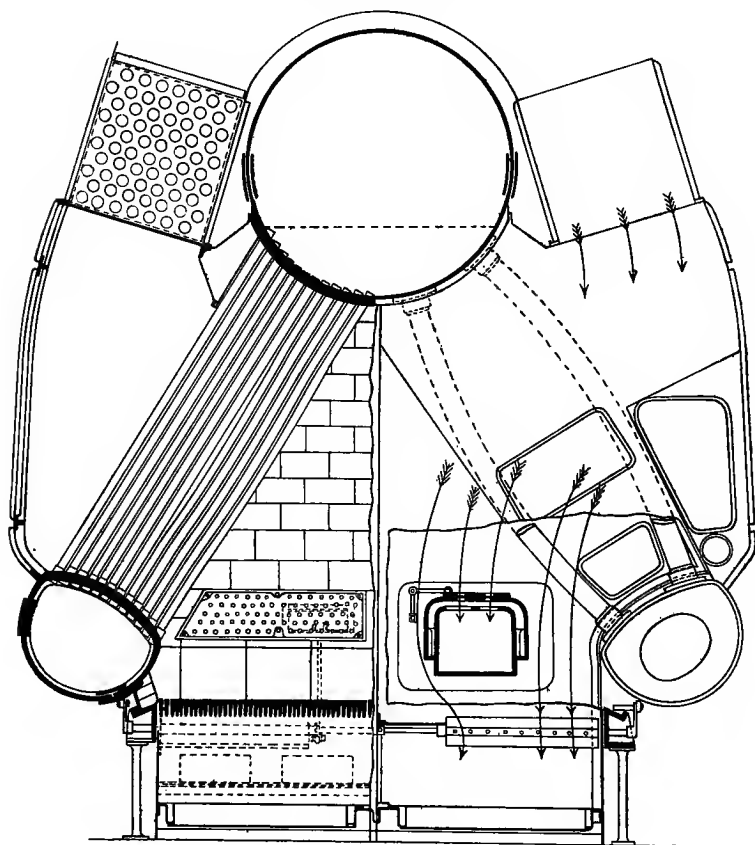
FIG. 242. Yarrow Boiler.

are considerably flattened to make the surface nearly normal to the direction of the tubes. Thick plates are used where the tubes enter, in both the steam and the water-drums. These boilers are built both with large and with small tubes and are very ex-

tensively used in battleships and large cruisers as well as in small craft. In some instances external down-comers are fitted, but in others there are none, and the circulation becomes established by some of the tubes acting as down-comers in the region less actively exposed to the fire. Large Yarrow boilers are made with grates of double length, fired from both ends. The figure shows a large-tube Yarrow boiler with external down-comers. Level with the steam-drum in this example are two sets of tubes running horizontally fore and aft through the uptake, the function of which is to heat the air which is on its way to the grate. In this case it happens that the draught is forced by Howden's system, using closed ashpits (see § 294 below) and the air-heaters are a part of that arrangement. It is much more usual, in naval practice, to force the draught by using a closed stokehold. The arrows in the figure indicate the course of the air from the heating tubes through the front casing to the ashpit and to the fire-door.

286. White-Forster Boiler. This is another water-tube boiler of the same class, resembling Yarrow's, but with a feature which is of considerable practical importance. What is distinctive in it is that all the tubes are slightly curved, and to the same radius (figs. 243 and 244). This allows them to be individually removed or replaced and facilitates the cleaning of the tubes in place, as will be seen by reference to the longitudinal section in which a tube and a cleaning brush are shown in position for insertion. There is an external down-comer at the back end.

287. Other Water-tube Boilers. Among many other designs of water-tube boilers one of the simplest is the Cahall vertical boiler, which is largely used in America. It consists of a group of long, nearly vertical straight tubes connecting an upper with a lower drum. The drums are short and stand vertically. The whole is enclosed in a vertical shaft of brickwork, and a central flue is fitted through the upper drum to allow the gases to escape after doing their work in the tubes. There is an external grate which delivers the gases to the boiler shaft just above the lower drum. Circulation is maintained by means of external down-comers from the water space of the steam-drum to the lower drum. The water-tubes are grouped in annular fashion with an ample central space in which there are baffle plates to divert the upward stream of hot gas into close contact with the tubes. The
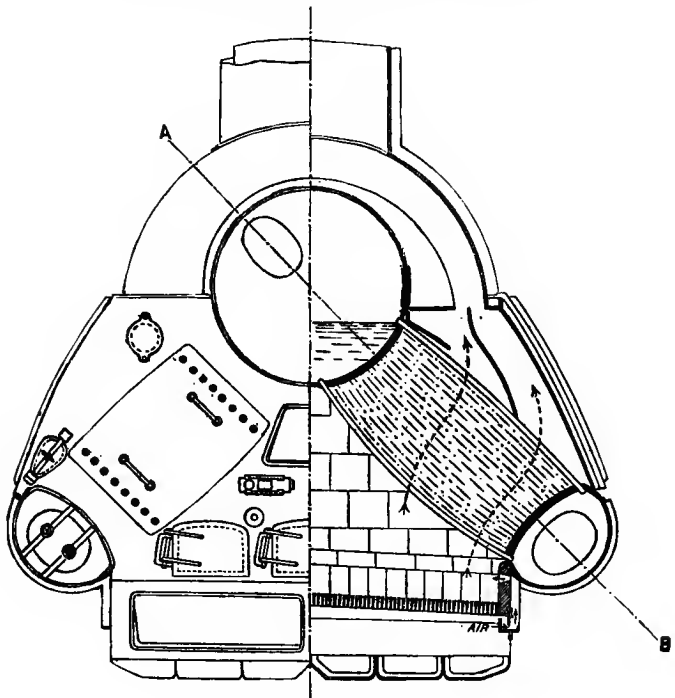
Fig. 243.   White-Forster Boiler.

SECTION AT A,B.
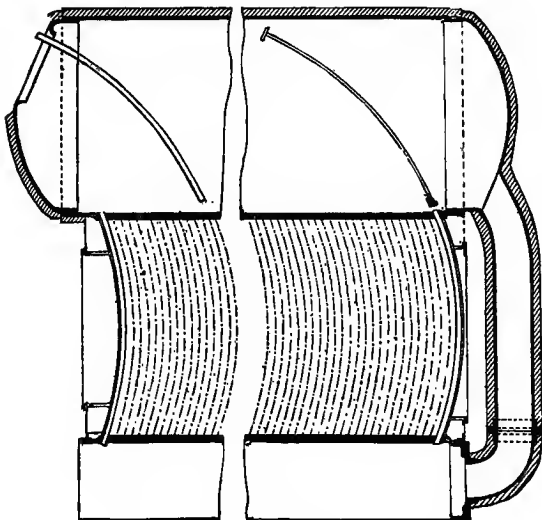
Fig. 244.

Climax boiler is an interesting vertical form of an entirely different pattern. It has a central core consisting of a vertical cylinder which constitutes the main water and steam space. Round this, and within an enclosing casing, is an annulus consisting of many tubes arranged in the form of spiral loops. Each tube has both its ends terminating in the central cylinder, one of them at a considerably higher level than the other. The grate is at the bottom and the hot gases pass up through the annular space which contains these tubes.

288. Supply of Water to Boilers. Feed-Pumps. Feed-water is forced into boilers either by a feed-pump or an injector. In some small steam plants the feed-pump is driven by the engine, but in most cases an auxiliary engine is used for the purpose. The most common forms of steam-pump for feeding boilers are direct acting: the steam-cylinder and pump-cylinder are in line with one another, a single piston-rod serving for both. Steam is admitted for the whole, or nearly the whole, of the stroke. Such pumps are often arranged to form a " duplex " combination by having two placed side by side, connected so that the piston of one actuates the valve of the other. In the Worthington duplex pump, for example, each of the two piston-rods is connected to an oscillating lever which gives motion to the slide-valve of the other cylinder. By this means steam is admitted to make one piston begin its stroke just as the stroke of the other is about to be completed, with the result that a smooth and continuous action is secured. In several other steam-pumps the arrangement is " simplex " and the valve receives its motion from tappets on the piston-rod. Generally this is done through a species of steam relay, the direct effect of the tappets being to move an auxiliary steam-valve which then causes the main steam-valve to be thrown over by admitting steam to one or other of its ends which are shaped as pistons to enable the action to take place. Blake's steam-pump is an example of this type, and another is Weir's, which is extensively used in the navy and the mercantile marine. In Weir's pump there is a regulating device by which the steam may either be cut off some time before the stroke is completed or admitted until the end. The adjustment of this secures silent running. The auxiliary valve slides on the back of the main valve, parallel to the piston, and the main valve moves trans-

E. 31

versely to this when it is thrown over by the action of the
steam.

289.   Injectors.   The injector, invented about 1858 by
Giffard, is a form of jet pump in which steam is employed to set a
stream of water into rapid motion and the momentum of the
steam is then utilized to overcome the pressure which opposes its
admission to the boiler.   It is used almost universally on loco-
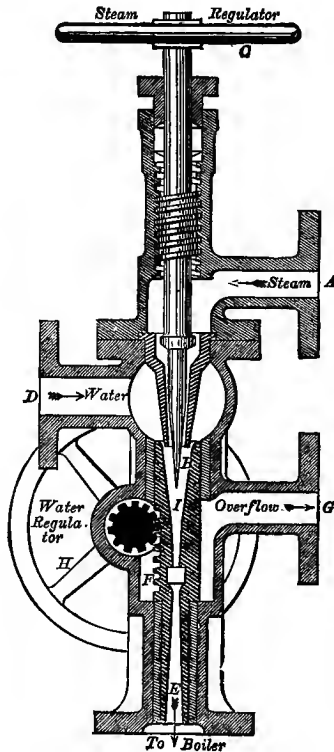
FIG. 245.   Giffard's Injector.

motives and largely on land boilers, where it often serves as an
auxiliary to a feed-pump.

To explain the action of the injector we may refer to fig. 245,
which illustrates an early form.   Steam enters at A and blows
through a nozzle at B, the amount of opening in which is regulated
in this example by means of a conical spindle which can be drawn

back by turning the hand-wheel C. When B is opened slightly the first effect is that the steam jet emerging from B drags the surrounding air along with it, producing a partial vacuum in the space round the nozzle. The feed-water is allowed to enter this space, through D. It mixes with the steam in the "combining" nozzle I, and condensation of the steam takes place. The momentum of the condensing jet is given up to the water, which consequently streams through I at a high velocity. Beyond the combining nozzle I is the delivery nozzle E, which is divergent. In this, owing to the enlargement of sectional area the stream loses velocity, and consequently, by a well-known hydrodynamical principle gains pressure, until at the end its pressure is higher than the pressure in the boiler. It can therefore pass into the boiler, and it does so through a non-return valve which opens to admit it. The orifice F which in this example is placed in the neck between the combining nozzle and the delivery nozzle leads to the overflow pipe G. Its function is to allow the injector to start into action by providing a means of escape for the steam and water until the stream has acquired enough momentum to force its way into the boiler. The action in this form of injector depends on a rather nice adjustment of the supply both of steam and of water. The supply of water is regulated by the handle H, which by moving the tube containing the nozzles I and E changes the annular opening through which the water enters, round the point of the steam nozzle.

In modern forms of the injector devices are used which to a great extent do away with the necessity of regulating by hand. In some forms the injectors are classified as *non-lifting*: that is to say they are designed for use in cases where the water does not have to be drawn up from a lower level and consequently the steam jet does not have to produce a partial vacuum before the water arrives to condense it. This allows the arrangements for steam admission to be simplified, for in a *lifting* injector the initial vacuum is generally obtained by using a finer jet of steam than is afterwards required when the injector is in full action. It is possible to make an injector lift water through as much as 20 feet, but usually there is a much shorter lift, or none at all. Again, the action of an injector is liable to be interrupted by mechanical shocks, air getting in, and so forth, and in such cases the older forms required that the steam should be shut off and

the action then restarted by hand. Many modern injectors are
arranged as *restarting*, that is to say they will restart auto-
matically, by means of a device which causes the overflow to open
freely when the injector fails to act and closes it more or less
completely when the action is resumed. Provision is also made
in some for an automatic adjustment of the opening through
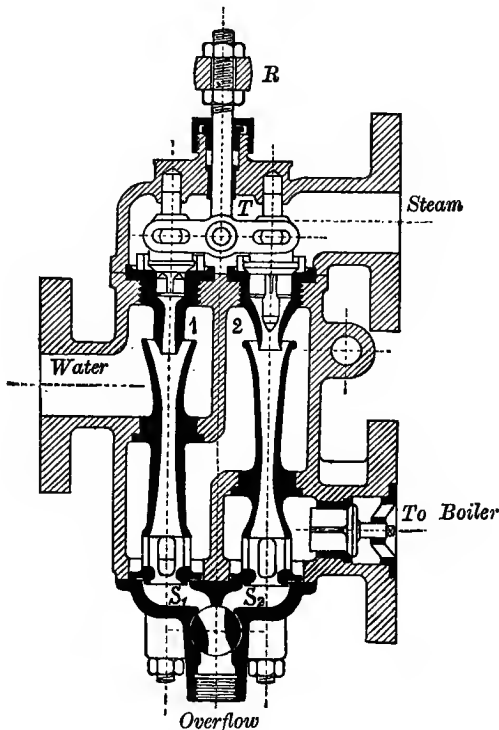
FIG. 246. Körting's Compound Injector.

which the feed-water enters, to suit variations in the pressure of
the steam.

In Körting's compound injector (fig. 246) the work is done in
two stages. There is first a "lifter," on the left, which brings in
the feed-water and delivers it at moderate pressure into a chamber
from which it passes into the "forcer" on the right, which sends
it into the boiler. Here, in each case, the combining nozzle and
delivery nozzle are in one piece, and provision for overflow comes

after the delivery nozzle.   Each steam nozzle is divergent—which, as we have seen in Chapter VIII., is the right form.   The steam nozzles are closed by valves at the top, which are lifted by turning a handle which raises the rod R and at the same time turns the cock at the bottom which allows overflow to take place first from S_1 and afterwards from S_2 as the injector is starting.   The rod R carries a loose cross piece which opens the valve on the left hand first, admitting steam to nozzle No. 1.   The valve on the right hand opens later, by a further movement of the handle, and admits steam to nozzle No. 2.   In the first movement, which sets No. 1 in action, there is free overflow for the left-hand injector. The jet of steam flows through freely, creating a sufficient vacuum to bring in the water, and when that appears at the overflow the handle is turned further, which opens the steam-valve of No. 2 and also closes the overflow S_1 of the left-hand injector or lifter. Finally, when the forcer (No. 2) has come into action a further movement of the handle in the same direction closes the overflow S_2 as well, and delivery to the boiler then goes on through a check-valve.   This combination of two injectors in series makes it possible not only to have a high lift, but also to deliver water to the boiler at a high temperature, owing to the relatively high pressure at which water is supplied to the "forcer."

A good example of an automatic restarting injector is Holden and Brooke's, shown in fig. 247.   In this example there is a device for adapting the instrument to work at any one of a wide range of possible boiler pressures.   For this purpose an adjusting handle is provided at the top, with a pointer which reads on a graduated circular rim on which various steam pressures are marked.   By turning this handle the amount of steam opening at A is adjusted simultaneously with the amount of water opening at B, for the steam nozzle between the two is raised or lowered through the action of a screw, in a manner which will be clear from a study of the figure.   When the steam nozzle is raised the steam opening is contracted, between the top of the steam nozzle and the conical point of the screw itself, which preserves a fixed level.   At the same time the water opening at B is increased.   This adapts the injector to work with steam at a higher pressure.   On the other hand, to set it for a lower pressure the handle is turned so as to increase the steam opening and by the same movement to decrease the water opening.   The handle being suitably set

beforehand steam may be turned on by a separate valve.   The free blow-through which is required to start the action (or to restart should it for any reason fail) is provided by having a gap in the combining nozzle with a hinged flap at *D* which opens to the overflow when steam is passing.   When the water arrives and condensation begins the partial vacuum produced in the combining nozzle causes this flap to close, leaving only the small permanent
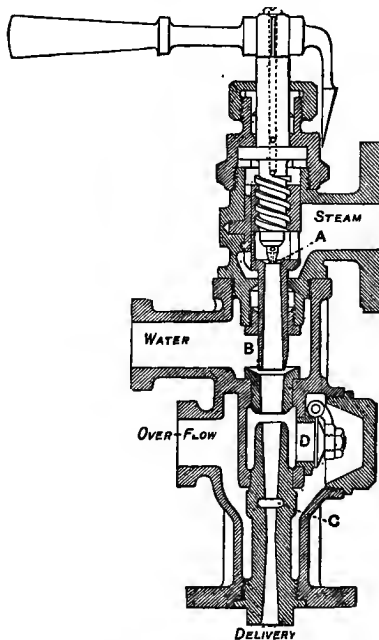
Fig. 247.   Holden and Brooke's Injector.

overflow opening *C*.   Another substantially equivalent device is to make one side of the combining nozzle itself in the form of a hinged flap, which opens to allow free escape of steam in the initial stage and then closes, contracting the nozzle to its proper form, when water begins to pass.   Another device is to have part of the combining nozzle slide axially in such a manner as to open a free way to the overflow until the stream of water is established, after which it returns to its normal place[1].

[1] For accounts of various forms of injector see *Proc. Inst. Mech. Eng.* 1860, 1866, 1884. Also *Practice and Theory of the Injector* by Strickland Kneass. (J. Wiley and Sons).

Considered as a heat-engine the injector is far from efficient, for the work done in the delivery of the water is something like one per cent. of the energy contained in the steam. But from another point of view its efficiency as a boiler feeder may be described as perfect, for (if there is no overflow) it returns to the boiler all the heat it takes from the boiler. It is in fact a feed-heater, as well as a feed-pump, and though it converts into work very little of the heat which it takes in none of the rejected heat is lost, for all goes to heat the feed.

The *exhaust-steam injector* works by steam from the exhaust of non-condensing engines, instead of using live steam from the boiler. The steam orifice is then larger in proportion to the other parts, the volume of the steam supply being greater. It affords a useful means of restoring to the boiler heat which would otherwise be wasted. Where delivery has to take place against a high boiler pressure a small supply of live steam may be required to aid the exhaust steam in working the injector, and in some cases there is a compound arrangement in which the exhaust steam drives an injector which delivers water to a second injector using live steam.

290. Feed-water heaters. The primary advantage in heating feed-water before it is delivered to a boiler is that in many cases this affords a means of economising, by saving what would otherwise be waste heat. Thus with a non-condensing engine, the feed-water may be heated by letting it take up part of the heat left in the exhaust steam. If there are non-condensing auxiliaries associated with a condensing engine-plant their exhaust steam can be utilized in a similar way to warm the main boiler-feed. Or again the chimney gases, after giving up as much heat to the boiler as it will receive from them may readily have a further portion of their heat extracted by bringing them into contact with pipes through which the colder feed-water is made to flow on its way to the boiler.

Apart from the question of economy of heat, there are incidental advantages in heating the feed-water. By bringing it to a temperature nearly as high as that of the steam we escape to a great extent the racking stresses, to which differences of temperature may give rise in the structure of the boiler. Further, heating the feed-water removes air, and in cases where the water

contains solid matter in solution of a kind less soluble in hot water than in cold it is advantageous that these should be deposited in an outside vessel by heating the water there, rather than on the surface of the boiler itself where they might form a scale difficult of removal. Accordingly, where other means of heating the feed are not available, it is sometimes useful to heat it even by taking live steam from the boiler for that purpose.

Green's "economizer" is a feed-water heater largely employed in stationary plants, where the water is heated by cooling the chimney gases to a temperature considerably lower than that at which they leave the heating-surface of the boiler. It consists of a stack of vertical pipes through which the feed-water is pumped on its way to the boilers. These are placed in the flue between the boilers and the chimney-shaft, and a by-pass is provided through which the hot gases can be diverted if necessary if for any reason the feed-heater is not in action. There is a tendency for soot to deposit on the outside of the pipes, and to prevent this scrapers are provided consisting of loose collars which are kept slowly moving up and down to clean the surfaces.

Any heater of this type is best arranged on the "contraflow" principle : that is to say the general direction in which the water passes through the heater should be opposite to the direction of flow of the gases which are giving up heat. Thus the water enters where the gases are coldest, and is drawn off where the gases are hottest, and the gases finally pass into the chimney after being in thermal contact with the water in its coldest state.

In Weir's feed-heater, which is applicable to compound engines with two or more stages of expansion, a portion of the steam is taken from the last receiver and is mixed with the cold feed-water.

291. Separation of Oil from feed-water and from exhaust steam. The condensed water which has passed through an engine generally contains some oil or grease in suspension, from mixture with the lubricant applied to the cylinder, piston-rod and valve-rod; and in cases where this occurs to such an extent as might cause trouble in the boiler when the condensed water is used for the feed, grease-filters are some-times employed through which the contaminated water is passed on its way back to the boiler. These generally consist of a

screen or bed of canvas, saw-dust, cocoa-nut fibre, or such material adapted to catch the grease and exposing a large area of filtering surface. They are usually arranged in pairs, so that one may be in use while the other is being cleaned, and a by-pass is provided in case of the filter becoming choked. Another method of extracting the oil is to treat the exhaust steam before condensation by passing it through an oil and water separator which is designed to remove all liquid particles, whether of oil or water, leaving only pure steam to pass on to the condenser. In Reid's form of separator the exhaust steam passes through a large central expanding nozzle into a chamber where its direction of motion is rapidly reversed : it then passes radially through an annulus, surrounding the nozzle, in which are stacked a large number of corrugated scrubbing plates on the sides of which the liquid particles collect which have not been already thrown off at the exit from the expanding nozzle. The liquid collected on the scrubbers drains to the bottom of the vessel where it is discharged along with the liquid thrown off at exit from the nozzle. The steam escapes by a pipe at the top, communicating with the outer circumference of the scrubber plates.

292. Evaporators. In marine boilers the fresh water which is required to make up the waste of working substance in the circulating system of boiler, engine and condenser, is obtained by an apparatus called an evaporator in which sea water is distilled. The heat is generally supplied by a coil of pipe into which steam is passed, and the steam used for this purpose is in many cases furnished by collecting the exhaust steam from auxiliary engines into a system of pipes forming what is called a closed exhaust. The evaporator also serves to supply fresh water for other purposes in the ship. Two evaporators are sometimes connected to form a compound system, the steam produced by evaporation in the first being used in turn to form the distilling agent of the second, in the tubes of which it is condensed.

293. Use of Zinc to prevent corrosion in boilers. To prevent corrosion in boilers it is usual to introduce blocks of zinc in metallic connection with the shell. These are set in the water space, preferably at places where corrosion has been found specially liable to occur. Their function is to set up a galvanic action,

in which zinc plays the part of the active element, and is dissolved while the metal of the shell is kept electro-negative.   Otherwise there would be a tendency for differences of electric quality between different parts of the shell to set up galvanic actions between the parts themselves, by which some parts, being positive to others, would be attacked.   The zinc raises the potential of the whole shell enough to bring it in all parts above that of the water. The zinc blocks have to be renewed from time to time, as they dissolve.   To make them act effectively care must be taken to keep them in good metallic contact with the shell.

294.   **Methods of forcing draught**.   The simplest but by no means the most economical way of forcing the draught is to let a jet of steam from the boiler discharge itself up the chimney.   It tends to carry the furnace gases with it and so to reduce the air pressure in the space where the jet escapes.   Allusion has already been made to the system which is universal in locomotive boilers of utilizing the exhaust steam from the engine as a means of forcing the draught.   In some cases a fan is fitted in the uptake of the flues, to assist mechanically in exhausting the gases.   Two other methods of mechanically forcing the draught have come into extensive use in marine practice.   One plan is to box in the stokehold and keep the air in it at a pressure which may correspond to as much as two or three inches of water, by the use of blowing fans.   This is known as the *closed stokehold* system and is common in warships.   In the other system, or *closed ashpit*, the stokehold is open, and air is supplied by a blowing fan to a reservoir formed by enclosing the ashpit and also to another reservoir from which air gets access to the grate above and through the fire-door.   In Mr Howden's closed ashpit arrangement the air is heated on its way to the reservoir by passing through tubes placed in a portion of the uptake traversed by the hot gases on their way to the chimney.   This method of restoring to the furnace what would otherwise be waste heat forms an interesting alternative to the method of restoring heat to the boiler by means of a feed-water heater.   It is an application to boiler furnaces of the regenerative principle alluded to in Chap. II.[1]

By either of these means the power of the boiler is increased

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[1] A description and discussion of these alternative methods of forcing draught will be found in papers read before the Institution of Naval Architects, April 1886.

in the ratio of 3 to 2, or even more, as compared with its power under chimney draught. The efficiency of the boiler is, in general, slightly but not very materially reduced by severe forcing. Arrangements are often made which allow the chimney draught to serve in ordinary steaming and the fan to be resorted to when an exceptional demand for power has to be met: in other cases the pressure at which air is supplied and consequently the rate of combustion of fuel on the grate is regulated by varying the speed of the fan. Forced draught enables a boiler to adapt itself to a wide range of power. An ordinary marine boiler burns 15 to 20 lbs. of coal per hour per square foot of grate with natural draught, and this is easily raised to 30 lbs. or more under forced draught. A locomotive using the steam blast will burn 70 or 80 lbs. per hour per square foot of grate, and in boilers of the locomotive type which have been employed in torpedo boats a consumption at the rate of 140 lbs. has been reached. In such extreme cases however the efficiency is low, for the combustion is not very perfect and the temperature of the escaping gases is high.

295. Mechanical Stoking. Many appliances have been devised for the mechanical supply of coal to boiler furnaces, to escape the labour of hand-stoking, and, incidentally, to secure a more uniform condition of the fire and to avoid the inrush of cold air which occurs when the fire door is opened for the admission of fuel. Several methods of mechanical stoking are effective for their purpose, and have found considerable application to stationary boiler plants. They are not used in marine work, nor, except very rarely, in locomotives. The most suitable cases for their application are those in which there is an extensive battery of boilers, and a mechanical conveyer can be used to carry the coal from the bunkers to the hopper of each furnace. By the combination of mechanical conveyance with mechanical stoking in a large installation, such as a power-supply station, much labour may be saved. So long as the coal has to be thrown by hand into the hopper there is little advantage in distributing it mechanically over the grate, but the case is different when it is worth while to install a conveying plant which makes the whole operation nearly automatic. The saving in wages may then exceed the charges due to capital cost and to wear and tear of mechanism working under trying conditions.

In some mechanical stokers the coal is deposited, from a hopper, on the front of the grate and is then slowly carried towards the back: in others it is thrown in at intervals by a sprinkling shovel which distributes it at once more or less uniformly over the fire.

To the first of these types belongs the Juckes chain grate, which is one of the earliest forms of mechanical stoker. The furnace bars are in short lengths, pinned to one another to form a chain or web which is made endless by returning over pulleys at the front and rear. The upper part of the endless web constitutes the grate, the lower part hangs in the ashpit, and the whole is kept in slow continuous motion, so that coal dropped from a hopper in front moves with the travelling grate towards a bridge piece at the rear. This catches the fuel remaining unburnt, and the excess drops over it into a chamber at the back of the ashpit. Owing to the burning of the coal the fire becomes thinner towards the rear, and this tends to let an excess of air enter through the rear portion of the grate. To remedy this the Coxe stoker has a series of air chambers below the bars supplied with air at graduated pressures, the chamber with highest pressure being in a region a little way from the front where the coal layer is completely ignited but still comparatively thick.

In other forms, such as Vicars's or Medrum's coking stoker, the coal on dropping from the hopper is pushed by plungers on to what is called a coking plate, which is a dead plate forming the front part of the furnace. Here the volatile constituents are driven off, and the coked fuel is then pushed in further to the fire-bars which have a reciprocating motion through a few inches in the direction of their length. In this motion the bars first move together towards the rear, and then they return towards the front singly or by the successive movement of two sets made up of alternate bars. This gradually works the charge on the grate inwards.

In the sprinkling stoker of Proctor charges of fuel from the hopper drop into a small chamber from which they are thrown on to the fire by the sudden release of a spring shovel. The shovel is drawn outwards against the tension of a stiff spring, by means of a revolving wheel which carries three tappets. As each of these passes a lever attached to the shovel the shovel jerks forward, scattering its charge on the fire; and the three tappets are so shaped that the three successive jerks are graduated to throw the

coal to various distances, and so ensure a fairly uniform thickness of fire. To keep the fire dressed and to move the clinker towards the bridge there is also an alternating motion of the fire-bars through a small distance, sets of alternate bars being periodically lifted and dropped a little as well as moved backward and forward. The Bennis stoker is a generally similar device.

The stokers which have been named admit of use with internally fired as well as externally fired boilers. With externally fired boilers there is more space for a mechanical grate, and other forms have been successfully developed, especially in America, where the use of externally fired boilers is comparatively common. An interesting type is represented by the Rooney stoker, in which the grate is made up of a series of T-shaped bars, set transversely and stepped so as to form an incline sloping downwards away from the fire-door at an angle of about 35°. The tops of the T's are nearly horizontal, but form a series of steps, each T being a little lower than the one in front. Coal is fed from a hopper on to the front of the grate, and to make it pass gradually down the slope the T-bars are all tilted periodically through a small angle by movement about an axis in the direction of the length of each bar. The ashes and clinkers which reach the bottom of the slope are collected on a small horizontal grate there which is hinged at the back, and this is dropped from time to time to discharge them into the ashpit. The Wilkinson stoker is another example of the inclined grate type. Its bars extend down the slope; they are notched to provide a stepped bed for the fire, and have a slight reciprocating motion in alternate groups to make the burning fuel travel downwards.

Still another type is the "underfeed" stoker, an American development of an old device. The coal is fed to a trough which forms a depression in the middle of the grate, and is pushed by a ram or screwed by a screw conveyer along this trough towards the rear. It becomes heaped up over the whole length of the trough and spreads laterally over the grate, which extends on either side.

Another old device, which has also been revived, is to pulverise the fuel before introducing it into the furnace. In Crampton's dust fuel furnace the coal was ground to powder and fed by rollers into a pipe from which it was blown into the furnace by an air-blast. This gave so intimate a mixture of fuel and air that the excess of air required for dilution was only one-fifth of the amount

required for combustion[1]. But the process of pulverising is expensive, and the particles of fuel are apt to be drawn up the chimney in a partially consumed state, causing both nuisance and waste.

296. Liquid Fuel. The use of liquid fuel for boilers began to acquire importance when petroleum was discovered in large quantity at Baku on the Caspian Sea. The "astatki" or heavy petroleum refuse which is left after distilling paraffin from the crude oil forms an exceedingly cheap fuel, with a calorific value from one-third to one-half greater than that of an equal weight of coal. It superseded coal in the steamers of the Caspian, and was largely employed for locomotives in the south-eastern part of Russia. In the arrangement for burning petroleum introduced in Russian locomotives by Mr T. Urquhart the flame impinges on a structure of fire-brick, built in the fire-box with numerous openings to allow the gases to diffuse themselves throughout the combustion-chamber. This guards against a too intense play of flame on the metallic surfaces, and at the same time the bricks serve as a reservoir of heat to rekindle the flame should the combustion be intermittent. In getting up steam an auxiliary boiler is used to supply the jet which serves to convert the oil into spray and to inject it along with air into the furnace[2]. Obvious advantages of liquid fuel are the ease with which the rate of combustion can be regulated to suit sudden changes in the demand for steam, also the nicety with which the supply of air taken in by the oil injector can be adjusted, and the continuity with which the fuel is supplied, its entrance requiring no opening of the fire-door with a consequent inrush of cold air.

Owing to the cost and danger of transporting oil in bulk its use as furnace fuel, although perfectly successful in the neighbourhood of the oil wells, was at first almost wholly restricted to certain districts. In England however Mr Holden has used with success not only Russian petroleum refuse but also the residuum from shale-oil distilleries, with tar and other heavy liquid refuse, as a substitute for coal in a number of the locomotives of the Great Eastern Railway. In some cases the oil is used alone; in others it is used in conjunction with coal. It is sprayed into the

[1] *Proc. Inst. Mech. Eng.* 1869.

[2] Urquhart, *Min. Proc. Inst. C. E.* 1884.

fire-box by steam injectors, mixing in the injector with air which has been previously heated by passing through a heater in the smoke-box.   Mr Aspinall has also used oil fuel in locomotives of the Liverpool Dock lines, mainly to reduce the risk of fire from sparks and to prevent smoke[1].

For general use in the furnaces of stationary engines or the mercantile marine the cost of oil fuel is prohibitive, and any extensive use of it would increase this objection, for the total supply of petroleum from all sources is reckoned to be only some six per cent. of the total supply of coal[2].   But for naval purposes its use is attended by great advantages which have led, in certain cases, to its adoption in warships as a substitute for, or more generally as a supplement to, coal fuel.   It is easily stored, conveyed, and delivered on board ship; and, even when full power is wanted, it can be handled by a small boiler-room staff.

In burning oil fuel it is "atomized" or reduced to a finely divided spray as it enters the furnace.   In some cases this is done by a steam blast, in others by a blast of compressed air, which strikes the oil as it trickles from the mouth of a supply pipe. But neither steam nor air is necessary for the purpose, and the oil may be effectively atomized by forcing it at high pressure through a small orifice, such as a sharp-edged hole in a plate. It is advantageous to give the oil a motion of rotation as it approaches the hole, and for this purpose the channel leading to the hole has the shape of a coarsely-pitched helix.   The pressure of delivery may range from 30 to 300 lbs. per square inch.   On leaving the hole the atomized oil spreads in the form of a divergent cone, made up of minute particles which expose a large surface for combustion.   Air for the combustion is supplied through an annular orifice surrounding the oil nozzle, and the entering stream of air may also with advantage receive a helical movement from suitable guides.   By these means effective combustion is secured with absence of smoke.

297. Superheaters. Superheaters generally consist of a coil or group of pipes through which the steam is passed while hot gases circulate outside.   In some cases the superheater takes its heat

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[1] Aspinall on Petroleum as Steam-Engine Fuel.   Institution of Civil Engineers, Conference, 1897.

[2] See *Report of United States Liquid Fuel Board,* 1904.

from the furnace which heats the boiler: in others it is independently fired, and where the plant is big enough to justify the cost of a separate fire for the purpose this allows superheating to be carried out in a more satisfactory manner. Independent firing admits of a better control of the temperature to which the steam is raised, especially when the demand for steam is liable to considerable variations.

When the main boiler-flue gases are used for superheating, the superheater must be set in a position such that the gases act on it before their temperature is so much lowered as to be ineffective. Thus in Cornish or Lancashire boilers the superheaters are generally placed at the after end of the boiler, where the gases, after traversing the internal flue or flues, have still to give up more heat to the boiler in the external flues. Provision has to be made for protecting the superheater tubes from being burnt while steam is being raised or at other times when little or no steam is passing through them. In some cases this is done by means of an arrangement permitting the superheater to be flooded by admitting water from the boiler, but a generally better plan is to have a by-pass to the superheater flue so that the hot gases may take another route when the superheater is not in use.

For moderate amounts of superheating the gases may still be hot enough to act on the superheater after they have completed their action on the heating surfaces of the boiler, without prejudice to economy, provided they afterwards pass an effective feed-water heater by which they are finally cooled to a temperature suitable for discharge to the chimney. Superheaters, like feed-water heaters, are best arranged on the principle of contra-flow referred to in § 290. The tubes of a superheater are frequently shaped in the form of a U or S to enable them to stand severe variations of temperature without racking. In applying superheating to locomotives Schmidt arranges for a division of the hot gas of the fire-box into two streams. One, which is the chief part, passes through the boiler tubes in the ordinary way, simply generating steam. The other passes through a few large tubes in the water space, giving up but little of its heat to the water but acting on a superheater made up of small pipes through which the steam passes on its way to the cylinders.

CHAPTER XV.

FORMS OF THE STEAM-ENGINE.

298. Terms used in classification. There is first a broad division between steam *turbines* which act by impulse or reaction or both, and *pressure engines* which act by the changes in volume of a chamber or chambers containing steam under pressure. Steam turbines having been already considered in Chapter VIII., it is pressure engines only with which we are concerned here. They are in general of the *reciprocating* or piston-and-cylinder type. Then there is a general distinction of *condensing* from *non-condensing* engines, with a subdivision of the condensing class into those which act by surface condensation and those which use jet condensers. Next there is the division into *simple* or non-compound and *compound* engines, with a further classification of the latter, according to the number of stages in the compounding, as double-, triple-, or quadruple-expansion engines. Again engines are *single-acting* or *double-acting* according as the steam acts on one side only or alternately on both sides of the piston. Some engines, such as steam-hammers and certain kinds of steam-pumps, are *non-rotative*, that is to say, the motion of the piston does work simply on a reciprocating piece; but generally an engine does work on a continuously revolving shaft, and is termed *rotative*. In most cases the crank-pin of the revolving shaft is connected directly with the piston-rod by a connecting-rod, and the engine is then said to be *direct-acting*; in other cases, of which the *beam engine* is the most important example, a lever resembling the beam of a balance is interposed between the piston and the connecting-rod leading to the crank. The same distinction applies to non-rotative pumping engines, in some of which the piston acts directly on the pump-rod, while in others it acts through a beam. The position of the cylinder is another element of classification, giving *horizontal, vertical,* and *inclined*

cylinder engines. Most vertical engines have the cylinder or
cylinders set above the crank-shaft, and these are sometimes
described as being of the *inverted cylinder* class. In *oscillating
cylinder engines* the connecting-rod is dispensed with ; the piston-
rod works on the crank-pin, and the cylinder oscillates on trunnions
to allow the piston-rod to follow the crank-pin round its circular
path. In *trunk engines* the piston-rod is dispensed with ; the con-
necting-rod extends as far as the piston, to which it is jointed, and
a trunk or tubular extension of the piston, through the cylinder
cover, gives room for the rod to oscillate. In *rotary* engines there
is no piston in the ordinary sense ; the steam does work on a
revolving piece, and the necessity is thus avoided of afterwards
converting reciprocating into rotary motion. Still another mode
of classification speaks of engines in reference to the conditions
under which they are at work, as *stationary, locomotive,* or *marine.*
Locomotive, marine and some kinds of stationary engines such as
those employed in heavy rolling mills belong to the *reversing*
class, having valve mechanism which enables them to run either
way. Other descriptive terms will be mentioned in the paragraphs
which follow.

299. Beam Engines. In the single-acting atmospheric
engine of Newcomen the beam was a necessary feature ; the use
of water-packing for the piston required that the piston should
move down in the working stroke, and a beam was needed to let
the counterpoise pull the piston up. Watt's improvements made
the beam no longer necessary ; and in one of the forms he designed
it was discarded—namely, in the form of pumping engine known
as the Bull engine, in which a vertical inverted cylinder stands
over and acts directly on the pump-rod. But the beam type was
generally retained by Watt, and for many years it was preferred by
the builders of engines. The early engines were designed to
pump water out of mines, and the beam was highly convenient for
pumping. The cylinder could stand on a firm foundation near the
mine shaft and the beam project over the shaft. Moreover the
beam was a convenient driver for the valve-rod, feed-pump rod and
air-pump rod of the engine itself, and it lent itself to the method
invented by Watt of guiding the end of the piston-rod to move in
a straight line, by what is called the " parallel motion." In a
modern direct-acting engine the end of the piston-rod is con-

strained to move in a straight line by means of plane surfaces forming guides, but in the early days of engine building accurate methods of shaping plane surfaces had not been developed, and it was easier to secure a good approximation to straight line motion by means of Watt's device. In modern practice the direct-acting type of engine has almost wholly displaced the beam type. The beam is still found as a feature of some pumping engines, but even for pumping the tendency is to use direct-acting forms.

The "parallel" motion is illustrated diagrammatically in fig. 248. There MN is the path in which the piston-rod head, or cross-head, as it is often called, is to be guided. ABC is the middle line of half the beam, C being the fixed centre about which the beam oscillates. A link BD connects a point in the beam with
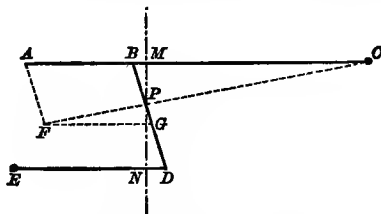
FIG. 248. Watt's Parallel Motion.

a radius link ED, which oscillates about a fixed centre at E. A point P in BD, taken so that $BP : DP :: EN : CM$, moves in a path which coincides very closely with the straight line MPN. Any other point F in the line CP or CP produced is made to copy this motion by means of the links AF and FG, parallel to BD and AC. In the ordinary application of the parallel motion a point such as F is the point of attachment of the piston-rod, and P is used to drive a pump-rod. Other points in the line CP produced are occasionally made use of, by adding other pairs of links parallel to AC and BD.

Watt's linkage gives no more than an approximation to straight-line motion, but in a well-designed example the amount of deviation need not exceed one four-thousandth of the length of the stroke. It was for long believed that the production of an exact straight-line motion by pure linkage was impossible, until the problem was solved by the invention of the Peaucellier cell. The Peaucellier linkage is not applied to the steam-engine except in isolated cases.

32—2

In by far the greater number of modern steam-engines the cross-head is guided by sliding on planed surfaces. In many beam engines, even, this plan of guiding the head of the piston-rod has taken the place of the parallel motion.

300. Direct-acting Horizontal and Vertical Engines. The use of guiding surfaces and sliding motion for the piston-rod head is practically universal in engines of the direct-acting class: the piston, the connecting-rod, the crank and the frame or bed-plate of the engine constitute a kinematic chain of four elements, of the "slider-crank" type.

No form of steam-engine is so common as the horizontal direct-acting. For small powers the engine is generally self-contained, in other words, a single frame or bed-plate carries all the parts, including the main bearings in which the crank-shaft with its fly-wheel turns. The cylinder either rests on the bed-plate, or overhangs at the back, being in the latter case bolted to a vertical part of the frame which forms a cover for the front end of the cylinder. The frame is often given what is called a girder shape, which brings a portion of it more directly into the line of thrust between the cylinder and the crank centre, and allows the upper as well as the lower of the two surfaces which serve as guides for the cross-head to be formed on the frame itself. This construction is usual in large as well as in small horizontal engines. The feed-pump plunger is usually driven from a separate excentric: in some cases it is directly attached to the cross-head, and in others to the valve-rod. When a condenser is used with a small horizontal engine it is usually placed behind the cylinder, and the air-pump, which is within the condenser, is a horizontal plunger or piston-pump worked by a "tail-rod"—that is, a continuation of the piston-rod past the piston and through the back cover of the cylinder. In large horizontal engines the condenser generally stands in a well between the cylinder and the crank-shaft, and the pump, which has a vertical stroke, is worked by means of a bell-crank lever attached by a link to the cross-head of the engine. A tail-rod, however, is often introduced in such engines to assist in supporting the weight of the piston, which would otherwise tend to cause undue wear on the lower side of the cylinder.

When uniformity of driving effort or the absence of dead points is specially important, two independent cylinders often

work on the same shaft by cranks at right angles to each other, an arrangement which allows the engine to be started readily from any position. Such engines are called *coupled*. The ordinary locomotive is an example of this form. Winding engines for mines and collieries, in which ease of starting, stopping, and reversing is essential, are very generally made by coupling a pair of horizontal cylinders, with cranks at right angles to each other, on opposite sides of the winding-drum, with a link-motion as the means of operating the valves.

Direct-acting engines of the larger class are generally compounded either (1) by having a high- and a low-pressure cylinder side by side, working on two cranks at exactly or nearly right angles to each other, or (2) by placing one cylinder behind the other, with the axes of both in the same straight line. The former is called the *cross-compound* and the latter is called the *tandem* arrangement. In tandem engines one piston-rod is generally common to both cylinders; occasionally, however, the piston-rods are distinct, and are connected to one another by a framing of parallel bars outside of the cylinders. Another construction, rarely followed, is to have parallel cylinders with both piston-rods acting on one crank by being joined to opposite ends of one long cross-head. In a few compound engines the large cylinder is horizontal, and the other lies above it in an inclined position, with its connecting-rod working on the same crank. In another form, already referred to in speaking of the balancing of engines (§ 263), the combination consists of one horizontal and one vertical cylinder, with connecting-rods working on the same crank.

In tandem engines, since the pistons move together, there is no need to provide a receiver between the cylinders. It is practicable to follow the " Woolf" plan (§ 180) of allowing the steam to expand directly from the small into the large cylinder; and in some instances this is done. In any case, however, the connecting-pipe and steam-chest form an intermediate receiver of considerable size, which will cause loss by having too much "drop" (§ 182) unless steam be cut off in the large cylinder before the end of the stroke. Hence it is more usual to work with a moderately early cut-off in the low-pressure cylinder than to admit steam to it throughout the whole stroke. Unless it is desired to make the cut-off occur before half-stroke, an ordinary slide-valve will serve

to distribute steam to the large cylinder. For an earlier cut-off than this a separate expansion-valve is required on the low-pressure cylinder, to supplement the slide-valve; and in any case, by providing a separate expansion-valve, the point of cut-off is made subject to easy control, and may be adjusted so as to reduce drop or to divide the work as may be desired between the two cylinders. For this reason it is not unusual to find an expansion-valve, as well as a common slide-valve, on the low-pressure cylinder even in tandem engines. In many cases, however, the common slide-valve only is used. In the high-pressure cylinder of compound engines, the cut-off is often effected either by an expansion slide-valve or by some form of Corliss or other trip-gear.

For mill engines the compound tandem and compound coupled types are the most usual, and the high-pressure cylinder is very generally fitted with trip-gear. In the compound coupled arrangement the cylinders are on separate bed-plates, and the fly-wheel is between the cranks. In triple expansion horizontal engines it is not unusual to use a tandem arrangement for two of the cylinders but to cross-couple as regards the third, and in some cases the third stage is divided between two cylinders, making four in all which are grouped as two cross-coupled pairs, one pair consisting of the high-pressure cylinder tandem* with one low-pressure cylinder, and the other pair consisting of the inter-mediate cylinder tandem with the other low-pressure cylinder. Where considerations of floor-space admit of it a horizontal arrange-ment is generally preferred with separate admission and exhaust valves for each end of each cylinder. The condenser is generally put in a pit at a somewhat lower level, and the air-pump is worked by a bell-crank lever.

The general arrangement of vertical engines differs little from that of horizontal engines. The cylinder is usually supported above the shaft by a cast-iron frame resembling an **A** or inverted **V**, whose sides are kept parallel for a part of their length to serve as guides for the cross-head. Sometimes one side of the frame only is used, and the engine is stiffened by one or more wrought-iron columns between the cylinder and the base on the other side. *Wall-engines* are a vertical form with a flat frame or bed-plate, which is fixed by being bolted against a wall; in these the shaft is generally at the top. Vertical engines are compounded, like

horizontal engines, either by coupling parallel cylinders to cranks at right angles (or at 120° if triple expansion is to be used with three cylinders) or, tandem fashion, by placing the high-pressure cylinder above the other. In vertical condensing engines the condenser is situated near the base under the back limbs of the frame, and the air-pump, which has a vertical stroke, is generally worked by a horizontal lever connected by a short link to the cross-head. In some cases the pump is horizontal, and is worked by a crank on the main shaft.

301. Condensers. In land engines using condensation the simple jet form of condenser is the most usual, but surface condensation is resorted to in special cases, when the available supply of water is unsuited for boiler feed, and it is therefore important to return the condensed water to the boiler. For the same reason surface condensation is universal in marine engines. The most usual form of surface condenser is a group of brass tubes extending horizontally between two tube plates, with an arrangement by which the cooling water circulates towards one side through the lower half of the group and back towards the other side through the upper half, while the steam to be condensed comes into contact with the outside of the tubes. The aggregate surface of the tubes is about $1\frac{1}{2}$ square feet per indicated horse-power. The quantity of cooling water supplied to a condenser is in general 20 to 30 times more than the quantity of condensed steam. When the supply is limited means may be taken to cool the condensing water, as by letting it trickle down over a high scaffolding constituting a *cooling tower,* before it is used over again. But an interesting alternative in such cases is to employ what is called an *evaporative condenser,* consisting of a stack of pipes into which the exhaust steam is admitted and over which a comparatively small amount of cold water is allowed to drip. Such a condenser is placed in the open air, generally on the roof, and the amount of water used by it need not exceed the amount that is condensed. It can therefore be applied to what would otherwise be non-condensing engines, giving the thermodynamic advantage of condensation without any additional expenditure of water, the feed-water being saved by condensation while the quantity of cooling water is no more, and may even be less, than the quantity would be required for feed.

Any of these forms of condenser requires the use of an *air-pump* to maintain the vacuum, the same pump serving, as a rule, to remove the condensed water and (in a jet condenser) the water admitted by the jet. But in some cases the main or wet air-pump is supplemented by what is called a *dry air-pump* which is so placed as to pump air and vapour only. This allows the main pump to be reduced in size, as its duty is only to remove the water, and the supplementary dry pump can be worked at a good speed. The combination allows a high vacuum to be attained.

With some land engines it is practicable to remove the water from the condenser by a *gravity drain*, namely a pipe with a vertical drop of about 35 feet, ending in a water seal to prevent the return of air, a dry air-pump doing the rest.

Reference has already been made in Chap. VIII. to the *vacuum augmenter* devised by Parsons to serve the object of securing a higher vacuum than is usually obtainable with an air-pump of the ordinary kind.

The *ejector condenser* is a device giving jet condensation without the use of any air-pump. It acts on the principle of the injector, delivering water against the pressure of the atmosphere and at the same time condensing the exhaust steam from the engine. It consists of a series of coaxial cones set in a vertical line with the wider ends above, with annular spaces between them through which the exhaust steam enters, and with a central channel down which the condensing water flows from an overhead tank. The group of cones may be said to form a combining-nozzle, in which the exhaust steam, entering by the circumferential channels, meets the stream of water in the centre and is condensed. Then the stream passes on to a divergent or delivery-nozzle, in which its velocity is gradually reduced and its pressure consequently increased, and this terminates in the hot-well. As the pressure there is that of the atmosphere it follows that in the combining-nozzle, where the velocity of the stream is greater, the pressure is much less than atmospheric, and in fact a vacuum equivalent to about 25 inches of mercury may readily be maintained by this device.

302. Single-acting High-Speed Engines. With the development of electric lighting, soon after 1880, a demand arose for a type of engine which would run fast enough to drive a

dynamo directly, without the use of belting or other intermediate gearing to multiply the speed, the required speed being often as much as 500 revolutions per minute or even more. To some extent this demand was met by adapting the direct double-acting engine for high-speed work, by enlarging the bearing surfaces and reducing, as far as possible, the masses of the reciprocating parts. But the single-acting type of engine came into favour as being specially suited for this class of work, and for a good many years had a great vogue. The most successful example of this type is the "central-valve" engine of P. W. Willans, which was used in the equipment of many of the early stations for electric supply in London and elsewhere. It owed much of its success to the excellence of its workmanship, and also to the scientific manner in which Willans dealt with questions of steam-engine efficiency, in the researches to which reference has been made in Chapter VII.

Apart from the advantage of simplifying the valves, single action allows alternation from push to pull at the joints to be entirely avoided, which was a matter of much importance in early high-speed engines, before the introduction of forced lubrication made it easy to use double action without causing serious "knock" and wear of the bearings at the crank-pin and crosshead-pin. If much play were left at the bearings the engine knocked badly: if the play were reduced to a minimum the bearings were liable to seize as an effect of unequal expansion when the engine warmed to its work. These troubles could be escaped in single-acting engines, which could be made to run smoothly and noiselessly at a high speed by avoiding reversals of stress.

In most single-acting engines the rods are kept in a state of thrust throughout the revolution. Steam is admitted to the back of the piston only. To secure, however, that the connecting-rod shall always push when the engine is running at a high speed there must be much cushioning during the back or exhaust stroke, for reasons which have been explained in § 253. From a point near the middle of the back stroke to the end the piston is being retarded; cushioning must begin at that point or earlier, and the work spent upon the cushion must at every stage be at least as great as the loss of energy on the part of the piston and rods. In some single-acting engines this cushioning is done by compressing a portion of the exhaust

steam ; in others the rod is kept in compression by help of a supplementary piston, on which steam from the boiler presses ; in the Willans engine the cushioning is done by compressing air.

In the Willans, as in most other high-speed engines, the arrangement is of the inverted vertical type, and the engine is completely enclosed within a casing to prevent the scattering of oil. In the compound and triple forms the successive cylinders are set tandem, in a vertical line, and the space below the upper piston serves as intermediate receiver. In some cases a single crank is used, but generally two or three sets of cylinders are grouped in parallel lines above a corresponding number of cranks. The piston-rod of each set of cylinders is hollow, and has a piston-valve in it, worked by an eccentric on the crank-pin, and arranged so that the relative movement of this valve with respect to the hollow rod determines the admission, transfer, and exhaust of the steam. The cross-head is itself a piston, working in a hollow cylindrical guide, which becomes closed during the up-stroke so that air may be compressed in it to serve as a cushion and prevent the stress at the crank-pin from ever changing from push to pull. The work stored in this air-cushion during the up-stroke is restored during the down-stroke, almost without loss. The eccentric-rod which works the valve is also as a rule kept in compression by the pressure of the live steam on the topmost piston-valve. The rod is split up into two parallel parts and the valve-eccentric is set between them, its sheave being forged on the crank-pin so that the relative motion of the valve to the piston-rod which encloses it may be the same as the motion which a valve in an ordinary engine performs relatively to its fixed seat when the valve is moved by an eccentric on its shaft. In this and several other forms of high-speed single-acting engine lubrication was effected by making the cranks splash about oil and water inside the enclosing case.

303. Double-acting High-speed Engines. When it was realized that the smooth running at high speed of a double-acting engine could be secured by the device of continuously pumping oil into the bearings under a moderately high pressure the special merit of the single-acting type disappeared. For the same power the double-acting type has of course the advantage of saving weight and space. Messrs Belliss and Morcom introduced in 1890

a double-acting high-speed engine with forced lubrication which has been highly successful. It is now made in sizes ranging up to 2500 horse-power, and for speeds ranging from 750 revolutions per minute in the small sizes to 200 in the largest. It is of the inverted vertical class and is enclosed. The bottom of the crank case serves as a reservoir for the lubricating oil which is pumped up by a pump worked from the valve-eccentric and delivered through a system of small pipes to all the bearings at a pressure of 10 to 20 lbs. per square inch. These engines are generally two-crank compound, three-crank triple, or three-crank two-stage compound with two low-pressure cylinders. In the two-crank compound an ingenious arrangement of piston-valves, placed between the cylinders, enables both valves to be worked by one eccentric-rod. The cranks are set at 180° apart, and the valves have therefore to be opposite in phase. Both valves are of the hollow piston type : they are set in tandem on the one rod, and the high-pressure valve admits steam by its inner edges (inverting the ordinary action) and exhausts at its outer edges. A shaft governor with external spring control regulates the speed, generally through a double-beat throttle valve, but in the larger engines automatic cut-off gear is fitted. The tension of the external controlling spring is adjustable by means of a hand wheel while the engine is running. From published tests it appears that the steam consumption of these engines, in sizes of about 1000 horse-power and at speeds of about 250 revolutions per minute, using triple expansion, may be as low as 11 lbs. per brake horse-power-hour, with steam at 180 lbs. per square inch and about 125° C. of superheat.

304. Pumping Engines. In engines for pumping water and other liquids, or for blowing air, it is not essential to drive a revolving shaft, and in many forms the reciprocating motion of the steam-piston is applied directly or through a beam to produce the reciprocating motion of the pump-piston or plunger without the intervention of any revolving part. On the other hand, pumping and blowing engines are frequently made rotative by adding a fly-wheel. When the level of the suction water is sufficiently high, horizontal engines, with the pump behind the cylinder and in line with it, are often preferred; in other cases a beam engine or vertical direct-acting engine is more common.

Horizontal engines are, however, employed to pump water from any depth by using triangular rocking frames, which serve as bell-crank levers between the horizontal piston and vertical pump-rods. Often two parallel lines of pump-rods are used driven from the two ends of a rocking double bell-crank or inverted T so that one rod is going down while the other is coming up and the effects of gravity are balanced. For deep-well or mine pumping the Cornish type (§ 215) still finds employment with its single cylinder, single action, and cataract control. The non-rotative engine frequently takes a double-acting and compound form, sometimes with the two cylinders in the inverted vertical position above the rods, and the rods are coupled by means of an inverted beam or a system of levers so that they differ in phase by a single stroke.

Non-rotative engines of this kind, like the old Cornish pump, are able to work expansively in consequence of the great inertia of the reciprocating pieces, the chief of which are the long and massive pump-rods. Notwithstanding the comparatively low frequency of the stroke, enough energy is stored in the movement of the rods to counterbalance the inequality with which the expanding steam works in different portions of the stroke, and the rate of acceleration of the system adjusts itself to give, at the plunger end, the nearly uniform effort which is required in the pump. In other words, the motion, instead of being almost simply harmonic as it is in a rotative engine, is such that the form of the inertia curve, when drawn as in fig. 201, is nearly the same as that of the steam curve, with the result that the distance between the two, which represents the effective effort on the pump-plunger, is nearly constant. The massive pump-rods may be said to form a reciprocating fly-wheel.

It is however only to deep-well pumping that this applies, and a very numerous class of direct-acting non-rotative steam-pumps have too little mass in their reciprocating parts to allow such an adjustment to take place at any ordinary speed. This is the case for example in the direct-acting feed-pumps already referred to in speaking of steam boilers (§ 288). In such engines an auxiliary rotative element is often introduced, partly to secure uniformity of motion and partly for convenience in working the valves; a con-necting-rod, for instance, is sometimes taken from a point in the piston-rod to a crank shaft which carries a fly-wheel, or a slotted cross-head is fixed to the rod and gives motion of rotation to a

crank-pin which gears in the slot, the line of the slot being perpendicular to that of the stroke. But many pumps of this class are purely non-rotative, and in such cases the steam is generally admitted throughout the whole or nearly the whole of the stroke, since the inertia of the parts is not sufficient to give the means of reconciling uniformity of pump-effort with expansive working.

The Worthington duplex engine, referred to as a boiler pump in §288, has been extensively applied, on a large scale, to deliver water for the supply of towns and to force oil through "pipe-lines" in the United States. In the larger sizes it is made compound or triple, each of the two parallel pump-rods having two, or three, steam-cylinders on it arranged in tandem. To allow of expansive working, an ingenious device is added which compensates for the inequality of effort on the pump-piston that would result from an early cut-off. A cross-head A (fig. 249) fixed to each of the piston-rods is connected to the plungers of a pair of cylinders B, B, which are free to oscillate on fixed trunnions. These cylinders contain water which can pass between them and a reservoir in which the pressure is maintained at 200 or 300 pounds per square inch, the water passing to and fro through
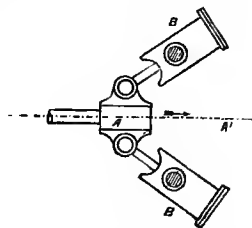
FIG. 249.

the trunnions, which are made hollow for the purpose. When the stroke (which takes place in the direction of the arrow) begins these plungers are at first forced in, and hence work is at first done by the main piston-rod, through the compensating cylinders B, B, by forcing water into the reservoir against this pressure. This continues until the cross-head has advanced so that the oscillating cylinders stand at right angles to the line of stroke. Then for the remainder of the stroke their plungers assist in driving the main piston, and the reservoir gives out the energy which it stored in the earlier portion. Any leakage from the compensating cylinders or their reservoir is made good by a small pump. One advantage which this method of equalizing the effort of a steam-engine piston has (as compared with making use of the inertia of the reciprocating masses) is that the effort, when adjusted to be uniform at one speed, remains nearly uniform although the speed be changed, provided the inertia of the reciprocating parts be small. In the Worthington "high-duty" engine this plan is in use to make it

possible to have an early cut-off in the steam cylinders. The high-
and low-pressure cylinders are each provided with a separate
expansion-valve of the rocking-cylinder type, as well as a slide-
valve; the cut-off is early, and the efficiency is as high as in other
pumping engines of the best class.

Another method of compensating for the inequality of the
piston thrust during expansion in non-rotative pumping engines
is to connect the pistons not directly but through a rocking piece
in such a way that the steam-piston gets a mechanical advantage
over the pump-piston as the stroke proceeds. This has been done
by Mr Davey in cases where the reciprocating pieces have not
enough inertia to make a compensating device unnecessary. A
rocking sector between the pistons causes their velocity ratio,
which is nearly one of equality in the early portion of the stroke,
to alter as the stroke goes on, with the result that in the later
stages as the steam-pressure falls off the pump-piston moves more
slowly than the steam-piston.

305.   The Pulsometer.   Mr Hall's " pulsometer " is a peculiar
pumping engine without cylinder or piston, which may be regarded
as the modern representative of the engine of Savery (§ 6). The
sectional view, fig. 250, shows its principal parts. There are two
chambers A, A', narrowing towards the top, where the steam-pipe
B enters. A ball-valve C allows steam to pass into one of the
chambers and closes the other. Steam entering (say) the right-
hand chamber forces water out of it past the
clack-valve V into a delivery passage D, which
is connected with an air-vessel. When the
water-level in A sinks so far that steam begins
to blow through the delivery-passage, the
water and steam are disturbed and so brought
into intimate contact, the steam in A is con-
densed and a partial vacuum is formed. This
causes the ball-valve C to rock over and close
the top of A, while water rises from the suction-
pipe E to fill that chamber. At the same time
steam begins to enter the other chamber A',
discharging water from it, and the same series
of actions is repeated in either chamber alter-
nately. While the water is being driven out
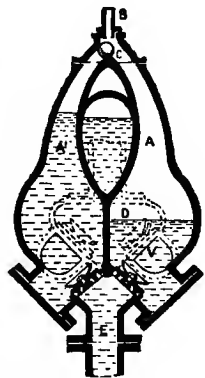
FIG. 250.   Pulsometer.

there is comparatively little condensation of steam, partly because the shape of the vessel does not promote the formation of eddies, and partly because there is a cushion of air between the steam and the water. Near the top of each chamber is a small air-valve opening inwards, which allows a little air to enter each time a vacuum is formed. When any steam is condensed, the air mixed with it remains on the cold surface and forms a non-conducting layer. Further, when the surface of the water has become hot the heat travels very slowly downwards so long as the surface remains undisturbed. The pulsometer of course cannot claim high efficiency as a thermodynamic engine, but its suitability for situations where other steam-pumps cannot be used, and the extreme simplicity of its working parts, make it valuable in certain cases. Trials of its performance have shown that under favourable conditions a pulsometer may use no more than 150 lbs. of steam per effective horse-power-hour. This consumption, large as it is when judged by the standard of an efficient large engine using steam expansively, does not compare very unfavourably with that of small non-rotative steam-pumps[1].

306. Rotary Engines. From the earliest days of the rotative engine attempts have been made to avoid the intermittent reciprocating motion which an ordinary piston-engine first produces and then converts into motion of rotation. The design of *rotary* engines, to use the name generally applied to non-reciprocating forms, has exercised the ingenuity of many inventors, with results which have little value or interest except to the student of applied kinematics. Murdoch, the contemporary of Watt, proposed an engine consisting of a pair of spur-wheels gearing with one another in a chamber through which steam passed by being carried round the outer sides of the wheels in the spaces between successive teeth[2].

In a more modern wheel-engine (Dudgeon's) the steam was admitted by ports in side-plates into the clearance space behind teeth in gear with one another, just after they had passed the line of centres. From that point to the end of the arc of contact the clearance space increased in volume ; and it was therefore possible,

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[1] *Proc. Inst. Mech. Eng.* 1893, p. 456. An automatic valve is described in the same place which enables the pulsometer to use steam expansively.

[2] See Farey's *Treatise on the Steam-Engine*, p. 676.

by stopping the admission of steam at an intermediate point, to work expansively. The difficulty of maintaining steam-tight connection between the teeth and the side-plates on which the faces of the wheels slide is obvious; and the same difficulty has prevented the success of other forms of rotary engine. These have been devised in immense variety, in many cases, it would seem, with the fallacious idea that a distinct mechanical advantage in respect of power was to be secured by avoiding the reciprocating motion of a piston[1]. In point of fact, however, very few forms entirely escape having pieces with reciprocating motion. In all rotary engines, with the exception of steam turbines,— where work is done by the kinetic impulse or reaction of steam,— there are steam chambers which alternately expand and contract in volume, and this action usually takes place through a more or less veiled reciprocation of working parts.

A comparatively modern example of the rotary type in which reciprocating motion occurred only to a trifling extent was the spherical engine of Mr Beauchamp Tower[2]. This engine was, like several of its predecessors[3], based on the kinematic relations of the moving pieces in a Hooke's joint. Imagine a Hooke's joint, connecting two shafts set obliquely to one another, to be made up of a central disk to which the two shafts are hinged by semicircular plates, each plate working in a hinge which forms a diameter of the central disk, the two hinges being on opposite sides of the disk and at right angles to one another. Further, let the disk and the hinged pieces be enclosed in a spherical chamber through whose walls the shafts project. As the shafts revolve each of the four spaces bounded by the disk, a hinged piece, and the chamber wall will suffer a periodic increase and diminution of volume, between limits which depend on the angle at which the shafts are set. In Mr Tower's engine this arrangement was modified by using spherical sectors, each nearly a quarter sphere, in place of semi-circular plates, for the hinged pieces in which the shafts terminate. The shafts were set at 135° to each other. Each of the four enclosed cavities then altered in volume from zero to a quarter

[1] A large number of proposed rotary engines are described, and their kinematic relations to one another are discussed, in Reuleaux's *Kinematics of Machinery*, translated by Sir A. B. W. Kennedy.

[2] *Proc. Inst. Mech. Eng.*, March 1885.

[3] One of these, the disk-engine of Bishop, was used for a time in the printing-office of *The Times*, but was discarded in 1857.

sphere, back to zero, again to a quarter sphere, and again back to zero, in a complete revolution of the shafts. One shaft was a dummy and ran free, the other was the driving-shaft. Steam was admitted and exhausted by ports in the spherical sectors, whose backs served as revolving slide-valves.

Another rotary engine of the Hooke's-joint family was Mr Fielding's, in which a gimbal-ring and four curved pistons took the place of the disk. Two curved pistons were fixed on each side of the gimbal-ring, and as the shafts revolved these worked in a corresponding pair of cavities fixed to each shaft.

No rotary engine has been permanently successful, and with the advent of the steam turbine, which does all that a rotary engine ever attempted to do, and far more, interest in this class of mechanism may be said to have ceased.

307. Marine Engines. The early steamers were fitted with paddle-wheels, and the engines used to drive them were for the most part modified beam engines. Bell's " Comet " (§ 21) was driven by a species of inverted beam engine, and another form of inverted beam, known as the side-lever engine, was for long a favourite with marine engineers. In the side-lever engine the cylinder was vertical, and the piston-rod projected through the top. From a cross-head on the rod a pair of links, one on each side of the cylinder, led down to the ends of a pair of horizontal beams or levers below, which oscillated about a fixed gudgeon at or near the middle of their length. The two levers were joined at their other ends by a cross-tail, from which a connecting-rod was taken to the crank above. The side-lever engine is now obsolete. In American practice engines of the beam type, with a braced-beam supported on **A** frames above the deck, are still found in river-steamers and coasters.

An old form of direct-acting paddle-engine was the steeple-engine, in which the cylinder was set vertically below the crank. Two piston-rods projected through the top of the cylinder, one on each side of the shaft and of the crank. They were united by a cross-head sliding in vertical guides, and from this a return-connecting-rod led to the crank.

Paddle-wheel engines are usually either *oscillating cylinder engines* or *diagonal engines*. In the former the cylinders are set under the crank-shaft, and the piston-rods are directly connected to the cranks. The cylinders are supported on trunnions

E.   33

which give them the necessary freedom of oscillation to follow the movement of the crank. Steam is admitted through the trunnions to slide-valves on the sides of the cylinders. In some instances the mean position of the cylinders is inclined instead of vertical; and oscillating engines have been arranged with one cylinder before and another abaft the shaft, both pistons working on one crank. The oscillating cylinder type is best adapted for what would now be considered comparatively low pressures of steam. Diagonal engines are direct-acting engines of the ordinary connecting-rod type, with the cylinders fixed on an inclined bed and the guides sloping up towards the shaft.

When the screw-propeller began to take the place of paddle-wheels in ocean-steamers, the increased speed which it required was at first supplied by using spur-wheel gearing in conjunction with one of the forms of engines then usual in paddle-steamers. After a time types of engine better suited to the screw were introduced, and were driven fast enough to be connected directly to the screw-shaft. The smallness of the horizontal space on either side of the shaft formed an obstacle to the use of horizontal engines, but this difficulty was overcome in several ways. In Penn's trunk-engine, which was at one time a usual form in war vessels, the engine was shortened by attaching the connecting-rod directly to the piston, and using a hollow piston-rod, called a trunk, large enough to allow the connecting-rod to oscillate inside it. The trunk extended through both ends of the cylinder and formed a guide for the piston. The trunk-engine had the drawback of requiring very large stuffing-boxes, of wasting cylinder space, and of presenting a large surface of metal to alternate heating by steam and cooling by contact with the atmosphere.

The return-connecting-rod engine was another horizontal form also used in the navy. It was a steeple-engine placed horizontally, with two, and in some cases four, piston-rods in each cylinder. The piston-rods passed clear of the shaft and the crank, and were joined beyond it in a guided cross-head, from which a connecting-rod returned.

Ordinary horizontal direct-acting engines with a short stroke and a short connecting-rod have also been used in warships, where the horizontal was at one time generally preferred to the vertical type of engine for the sake of keeping the machinery below the water-line.

All these types are now abandoned in favour of the inverted vertical engine which is in universal use for those sea-going ships, whether naval or mercantile, which have not adopted the steam turbine. In many instances the engine takes the three-crank form, using triple expansion, but the tendency is to prefer four cranks even when triple expansion is retained, especially in ships designed for high speed. With a four-crank triple-engine the steam in the last stage in the expansion is divided between two cylinders usually of the same size. This is done partly to avoid the structural objections to a large cylinder, which would be serious in engines of very high power, and partly for the sake of the better balance which, as was pointed out in § 258, is attainable with four cranks.

When three cranks are used they are, as a rule, set at 120°. In some large engines using three cranks (such as those of the "Campania," "Lucania," and "Spree") there are five cylinders, two of these being high-pressure cylinders put tandem-fashion on top of two low-pressure cylinders, while the intermediate-pressure cylinder stands over the third crank. This arrangement, however, does not represent the best modern practice. The four-crank engine without tandem-cylinders is the normal type for large marine engines, whether triple or quadruple, and is rarely departed from unless the power to be developed is so great as to make more than four cylinders desirable. In the quadruple expansion engines of the "Deutschland," where each of the twin sets develops 17,500 horse-power, though there are four cranks there are six cylinders. The high-pressure stage is divided between two cylinders, and so is the fourth stage. The two H.P. cylinders are tandem with the two L.P. cylinders, the two intermediate stage cylinders standing over the other two cranks. In the "Kaiser Wilhelm II." there are six cranks and eight cylinders in each twin-set of engines, developing about 20,000 horse-power. Each of the stages of expansion is divided between two cylinders, and each of the two high-pressure cylinders is set tandem on one of the first intermediate-pressure cylinders. These engines make 80 revolutions per minute, with a stroke of 5 ft. 10 in.

Slide-valves, placed as a rule between the cylinders, with reversing gear worked from eccentrics on the shaft, are used in nearly all marine engines, the valves being generally of

the piston form for the high-pressure and intermediate-pressure stages and of the flat form for the low-pressure stage (usually double-ported). The slide-valve serves, when "notched up" by the link motion or other reversing gear, to give such variations of cut-off as are required, without the use of a separate expansion-valve. The cylinders of large marine engines, like those of large land engines, have an internal *liner* with a space between it and the outer wall, which is made steam-tight and may be used as a jacket.

The air-pump is often driven by a rocking-lever from one of the cross-heads of the main engine, but in modern practice especially with large engines it is frequently an independent auxiliary. This is also the case with the circulating, bilge, and feed pumps. In many cases the circulating pump is of the centrifugal type.

In the last large ships of the Royal Navy to be fitted with reciprocating engines, before the turbine came into general use, the speed had increased until the number of revolutions per minute ranged from 120 to 145, with a piston speed of 1000 feet per minute. In destroyers the engines were designed to run at 400 revolutions per minute with a piston speed of about 1200 feet per minute. In the engines both of the larger and smaller ships forced lubrication was introduced, with very good results, in the crank-pins, eccentrics, and main bearings.

308. Condensers for Marine Engines. Surface condensation was introduced in marine practice by S. Hall in 1831, but was not brought into general use until much later. Previous to this it had been necessary, in order to avoid the accumulation of too dense brine in the boiler, to blow off a portion of the brine at short intervals and replace it by sea-water, a process which of course involved much waste of heat. By the use of surface condensers it became possible to use the same feed-water over and over again. The very freedom of the condensed water from dissolved mineral substances was for a time an obstacle to the adoption of surface condensers, for it was found that the boiler, no longer protected by a deposit of scale, became rapidly corroded through the action of acids formed by the decomposition of the lubricating oil, which in those days was of a much less stable character than the heavy mineral oils now in use. This objection was overcome by introducing a sufficient amount of salt water to allow some scale to form, and the use of surface condensers

soon became universal in steamers plying in sea-water. The marine condenser is made up of thin brass tubes about $\frac{5}{8}$ or $\frac{3}{4}$ of an inch in external diameter, the surface of which has an aggregate area of from 1 to 1·5 square foot per indicated horse-power. At each end the tube passes through a tube-plate, and a ferrule is screwed in round it to form a gland which is packed with cotton. The tubes are generally horizontal, and the cooling water circulates through them with the steam outside.

309. Locomotives. The ordinary locomotive consists of a pair of direct-acting horizontal or nearly horizontal engines, fixed in a rigid frame under the front end of a boiler of the type described in § 276, and coupled to the same shaft by cranks at right angles, each with a single slide-valve worked by a link-motion, or by a form of radial gear. The engine is non-condensing, and the exhaust steam, delivered at the base of the funnel through a blast-pipe, serves to induce a draught of air through the furnace. The demand for power in a locomotive changes frequently and quickly with changes of gradient in the line; and this method of forcing the draught has the great practical convenience that it makes the production of steam respond automatically to variations in the rate at which steam is taken by the engine. In some instances a portion of the exhaust steam, amounting to about one-fifth of the whole, is diverted to heat the feed-water. In tank engines the feed-water is carried in tanks on the engine itself; in other engines it is carried behind in a tender.

On the shaft which the engine drives are a pair of driving-wheels, which furnish tractive force by their frictional adhesion to the rails. A single pair of driving-wheels being generally insufficient to furnish as much tractive force as is required, two, three, or more pairs are used, and these are connected together by *coupling-rods* outside the wheels, which cause the work to be divided between as many of the wheels as are coupled in this way, and enable a greater fraction of the whole weight of the locomotive to be utilized in furnishing the frictional adhesion. Thus we have what are called "four-coupled," "six-coupled," "eight-coupled," and even "ten-coupled" locomotives. For express traffic the four-coupled arrangement is the most usual, with driving-wheels some 6 ft. or 6 ft. 8 inches in diameter: for goods traffic with its heavier and slower trains the engines are more generally six-

coupled with smaller driving-wheels. In *inside-cylinder* engines the cylinders are placed side by side within the frame of the engine, and their connecting-rods work on cranks in the driving shaft. In *outside-cylinder* engines the cylinders are spread apart far enough to lie outside the frame of the engine, and to work on crank-pins on the outsides of the driving-wheels. This dispenses with the cranked axle, which is apt to be the weakest part of a locomotive engine. Owing to the frequent alternations of strain to which it is subject, a locomotive crank-axle is peculiarly liable to rupture, and has to be removed after a certain amount of use.

In a few locomotives the leading wheels act as drivers, being coupled to driving-wheels behind them, but it is now generally preferred to have under the front of the engine two or four smaller wheels which do not form part of the driving system. These are carried in a *bogie*, that is, a small truck upon which the front end of the frame rests by a swivel-pin or plate which allows the bogie to turn, so as to adapt itself to curves in the line, and thus obviate the grinding of tyres and danger of derailment which would be caused by using a long rigid wheel-base. The bogie appears to have been of English origin[1]; it was first brought into general use in America, but it is now a common feature of locomotives everywhere. Instead of a four-wheeled bogie, a single pair of leading wheels is also used, carried by a Bissel *pony* truck, which has a swing-bolster pivoted by a radius bar about a point some distance behind the axis of the wheels. This has the advantage of combining lateral with radial movement of the wheels, both being required if the wheel-base is to be properly accommodated to the curve. Another method of getting lateral and radial freedom is the plan used by F. W. Webb of carrying the leading axle in a box curved to the arc of a circle, and free to slide laterally for a short distance, under the control of springs, in curved guides[2].

Locomotives are often classified by stating successively the numbers of leading-wheels, driving-wheels, and trailing-wheels. Thus a locomotive of what is called the "Atlantic" type, in which there is a four-wheeled bogie in front, then four coupled wheels on two axles, and then an axle with a pair of trailing-wheels, is

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[1] *Min. Proc. Inst. C. E.* Vol. LIII. p. 50.

[2] *Proc. Inst. Mech. Eng.* 1883.

described as a "4-4-2." Similarly a "4-6-0" means an engine with a bogie, three coupled driving-axles, and no trailer; and a "4-6-2," which is called the "Pacific" type, differs from it in having a pair of trailing-wheels added. We shall have other examples of this nomenclature in speaking of compound locomotives.

In inside-cylinder engines the slide-valves used frequently to be placed back to back in a single valve-chest between the cylinders. The width of the engine within the frame leaves little room for them there, and they are reduced to the flattest possible form, in some cases with split ports, half above and half below a partition in a central horizontal plane. In a few engines the valves are below the cylinders, with faces sloping down towards the front, while the cylinders themselves slope slightly up. In many more the valves work on horizontal planes above the cylinders; this position is specially suitable when a form of radial gear is used instead of the link-motion. Radial valve-gears have the advantage, which is of considerable moment in inside-cylinder engines, that the part of the crank-shaft's length which would otherwise be needed for eccentrics is available to increase the width of main bearings and crank-pins, and to strengthen the crank-cheeks. Walshaert's gear is almost universally used in Continental locomotives, and Joy's has been applied to a considerable number of British engines.

The outside-cylinder type is universal in America[1]. There the two castings which form the cylinders are bolted together to make a saddle on which the bottom of the smoke-box sits. The slide-valves are on the tops of the cylinders, and are worked through rocking levers from an ordinary link-motion or a radial gear. Outside-cylinder engines are very general on the Continent, and increasingly common in England, owing to the difficulty of finding room within the frame for the larger cylinders which are now required in consequence of the need of augmented power. Engines are now set to draw much heavier loads than was formerly the case, and often at higher speeds. Their boilers have been enlarged and their weight greatly increased, but this growth necessarily takes place subject to the limitations imposed by a fixed gauge between the rails, which determines, within narrow limits, the width of the frame.

[1] For details of American locomotive practice see a paper by P. J. Cowan, *Min. Proc. Inst. Civ. Eng.* Vol. cliv. 1903.

310. Compound Locomotives. Locomotive engines have been compounded in several ways; but it is still very doubtful whether the application of compound working to locomotives offers any distinct advantage, when regard is had to convenience in driving and cost of repairs as well as to economy in fuel.

In 1876 Mr A. Mallet introduced, on the Bayonne and Biarritz Railway, a type of compound locomotive in which one small high-pressure cylinder and one large low-pressure cylinder were used in place of the two equal cylinders of a common locomotive. Outside cylinders were used, but Mallet's system has also been applied to inside-cylinder engines. The pipe from the high- to the low-pressure cylinder took a winding course through the smoke-box; this gave it a sufficient capacity to serve as intermediate receiver, and also dried the steam before it entered the large cylinder. A reducing valve was provided through which steam of a pressure lower than that of the boiler could be admitted direct to the low-pressure cylinder to facilitate starting, as the engine might have stopped with the high-pressure port covered. The reversing gear was arranged to act on both cylinders by one movement, and also to permit a separate adjustment of the cut-off in each. Engines on Mallet's system have been used with some success on various . Continental railways and in India. He has also introduced an engine with four cylinders,—one high-pressure cylinder tandem with one low-pressure cylinder on each side. In another type of four-cylinder compound the high- and low-pressure cylinders are placed side by side, the high on top of the low, one pair on each side of the engine. The "Santa Fe" is a heavy locomotive of this type, a 2-10-2, built at the Baldwin Locomotive works, weighing 128 tons, or nearly double the usual weight of an express loco-motive. Another symmetrical form has been used, in which a pair of outside high-pressure cylinders is compounded with a pair of inside low-pressure cylinders. This arrangement was adopted by F. W. Webb, after trials of another method (now obsolete) of compounding in which two high-pressure cylinders acting on one axle delivered low-pressure steam to a single low-pressure cylinder acting on another axle. The De Glehn compound locomotives which are successfully used on the Northern Railway of France are also of the type having two outside high-pressure cylinders compounded with two inside low-pressure cylinders. The latter drive the crank-shaft of a leading pair of driving

wheels, and this pair of driving wheels is coupled to another pair of wheels driven by the high-pressure outside cylinders. The engine can be worked as a four-cylinder simple engine by means of a valve which allows the high-pressure cylinders to exhaust directly into the atmosphere and admits boiler steam directly (through a reducing valve) into the low-pressure cylinders. An interesting example of a specially powerful compound locomotive is the Mallet "articulated" engine built by the American Locomotive Company for the Baltimore and Ohio Railroad. It may be described as a 0-6-6-0, for there are no idle leading or trailing wheels, but two sets of six coupled drivers, arranged as two separate groups, the foremost group being carried on a truck permitting a little angular movement with respect to the boiler, just as in a bogie. The high-pressure cylinders drive the rear group, which are rigidly connected to the frame, and the low-pressure cylinders drive the forward group. This exceptionally heavy locomotive weighs 150 tons. Its cylinders are 20 and 32 inches in diameter, by 32 inches stroke.

The advantage of compounding locomotives can only be judged under the conditions of service which hold on a railway with its variable gradients, when the locomotives are hauling trains with the usual variations of speeds and with the usual intervals of rest. No trial is so satisfactory as one in which a compound and a single engine are compared by repeated use on the same line under like conditions of traffic. Important trials of this kind have been carried out on the Great Northern Railway by Mr H. A. Ivatt[1], which go far to show that there is no material advantage in compounding, even as regards consumption of coal, and that when other charges, namely those for oil and repairs, are taken into account, the total cost per train-mile or per ton-mile is higher in the compound than in the simple engine. The experience obtained on other lines is somewhat more favourable to compounding, but in the face of such results it is not surprising that the use of compound locomotives makes but little headway, and that in some cases where they have been tried there has afterwards been a reversion to the simple type.

311. Use of Superheating in Locomotives. A much more promising departure from customary locomotive practice is to employ a superheater giving highly superheated steam. This

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[1] *Proc. Inst. Mech. Eng.* May 1907.

has passed so far beyond the experimental stage as to be in
regular use in some thousands of locomotives on the Prussian
State Railways, the Belgian State Railways, the Canadian Pacific
and many other lines. Its successful introduction is chiefly due
to Dr W. Schmidt, whose work in connection with the use of high
superheat has been referred to in § 126[1]. As was mentioned in
§ 297, the usual method of superheating steam in locomotives is
to have in the upper part of the water space of the boiler shell a
number of specially large tubes, about four or five inches in
diameter, extending like the other tubes from the fire-box to
the smoke-box. Inside each of these is placed a superheater coil,
consisting of a long small tube through which the steam passes
on its way to the cylinder. Each of these small inner superheater
coils is a tube bent on itself three times, entering at the smoke-
box end, extending back nearly to the fire-box, then forward again,
and back again, and finally making its exit at the smoke-box end.
These various superheater coils are grouped in parallel; their inlet
ends are connected to a box to which the boiler steam is admitted,
and their exit ends to another box from which the cylinder steam
is drawn; the two boxes form parts of one casting with a partition
between. Thus every portion of the steam on its way to the
cylinders has to traverse one or other of the coils, outside of which
the hot gases from the fire-box are passing on their way to the
smoke-box. The steam is consequently superheated to a tempera-
ture generally between 300° and 350° C. About 315° C. is
considered best. To regulate the degree of superheat dampers
are fitted at the smoke-box end by which the flow of hot gases
through the large tubes containing the superheater coils can be
more or less checked or stopped altogether. In this way the
temperature of the steam is prevented from exceeding 350°, and
when the engine is standing still the superheater is put out of
action.

Comparative trials in practice are said to show an economy in
these locomotives, due to superheat, over ordinary non-compound
locomotives, amounting to about 20 or 25 per cent. in coal con-
sumption and 25 to 30 per cent. in water consumption. Besides
this saving they have the important advantage of giving a
materially increased amount of power with an engine of a given
weight.

[1] See R. Garbe, *The Application of highly superheated Steam to Locomotives*, 1908.

To make locomotives run well with highly superheated steam Schmidt has introduced certain modifications in the usual design of pistons, cylinders and valves. The pistons, which are made as light as possible, carry three light packing rings, which bear none of the weight, for the piston-rod is carried through the front of the cylinder, forming there a tail-rod, which runs in a fixed guide outside, beyond the stuffing-box, and the weight of the piston and piston-rod is borne by this guide and by the cross-head guide. The piston-rings are pressed out against the cylinder surface by admitting steam behind them. The stuffing-boxes have metallically-packed glands and are designed to allow a limited amount of lateral freedom so that the outer guides will still act as bearers although there may be some lateral displacement of the stuffing-boxes, due to unequal heating. Piston-valves are used for the distribution of the steam, generally with wide rings, which have steam-tight spaces at the back. By allowing the steam to enter these spaces through small holes in the rings a balance of steam-pressure on the rings is obtained, and the pressure against the port surfaces consequently depends only on the elasticity of the rings. Heavy mineral oil is used for lubrication and is forced in by means of oil presses to ensure a proper supply to the rubbing surfaces of the cylinder pistons and valve pistons.

CHAPTER XVI.

312. Air-engines with external or internal combustion.
The term Air-engine may be used in a restricted sense to denote
an engine in which the working substance is atmospheric air, but
it may with advantage be extended to apply to any heat-engine
which employs a gaseous working substance, as distinguished from
a vapour which becomes condensed during some part of the cycle
of operations. In this more extended sense the term would in-
clude engines in which the working substance is the mixed gas
resulting from the combustion of fuel, whether gaseous, liquid or
solid, within the engine itself. In other words, it would include
gas-engines and oil-engines. These will be more particularly con-
sidered in the next chapter, but some of the remarks which follow,
relating to air-engines in general, apply to them as well as to
engines using atmospheric air.

When air alone forms the working substance, it receives heat
from an external furnace by conduction through the walls of a
containing vessel, just as the working substance in the steam-
engine takes in heat through the shell of the boiler. An engine
supplied with heat in this way may be called an *external-com-
bustion* engine, to distinguish it from the very important class of
engines in which the combustion which supplies heat occurs within
a closed chamber containing the working substance. These last
are *internal-combustion* engines.

By far the most common kind of internal-combustion engines
are those in which the heat constituting the supply is generated
within the cylinder itself by the combustion there of inflammable
gas or vapour, generally mixed with air so that the combustion
takes place in an explosive manner. These engines will be con-

sidered more particularly in the next chapter. But besides these there is a limited class of internal-combustion air-engines in which the combustible may be solid fuel burning in a vessel apart from the cylinder, in which the working substance takes up heat by coming into direct contact with the burning fuel. The air which maintains the combustion is itself the working substance. Mixed with products of combustion it passes into the cylinder to do work there; and so far as the action in the cylinder is concerned the engine may operate in the same manner as if the working gas were heated by conduction from an external furnace. Engines of this class will be briefly noticed in the present chapter, along with air-engines of the kind that are externally fired.

313. Advantage in respect of Temperature Range. Compared with engines using saturated steam, engines using air or other gases have the advantage that the temperature and the pressure of the working substance are independent of one another. In the steam-engine, and in any other heat-engine in which the working substance is a saturated vapour, the upper limit of temperature is comparatively low in consequence of the high pressure with which high temperature is, in such cases, necessarily associated. But in an air- or gas-engine it becomes possible to use an upper limit of temperature greatly higher than the limit practicable in a steam-engine, and if the lower limit is not correspondingly raised an increase of thermodynamic efficiency results. It is true that the upper limit of temperature may be raised in the case of steam, by superheating; but even when the amount of superheating is exceptionally great a steam-engine continues to take in the greater part of its supply of heat at the comparatively low temperature at which the feed-water is converted into steam, and the direct thermodynamic advantage of the superheat is consequently small.

So long as external combustion is used, there must still be some considerable drop in temperature, of an irreversible and therefore wasteful kind, between the temperature which is produced in the furnace by combustion gases and the temperature at which the working air receives its heat, since without this no sufficiently rapid conduction of heat through the walls of the heater could occur. Internal-combustion engines have the advantage that the temperature which is produced in the

combustion is itself the upper limit in the thermodynamic cycle. This gives them, from the thermodynamic point of view, a great superiority, for the temperature reached in combustion is immensely higher than any temperature that it is practicable to give to a working substance so long as the supply of heat has to pass through the wall of a containing vessel capable of resisting pressure. When the heat has to pass through iron a temperature exceeding that of a low red heat, say 600° C. or 700° C. is out of the question, and in any actual air-engines of this class the maximum has been much lower. But in the cylinder of a gas- or oil-engine the temperature reached in the explosion is very usually 1600° C. and may approach 2000° C. Such temperatures are practicable within the cylinder because the heat that produces them does not have to pass through the metal, and the metal itself may be kept cool by means of a water-jacket so that it forms a safe container of the excessively hot gas.

314. Air-engine using Carnot's cycle. A theoretically simple and thermodynamically perfect form of external-combustion air-engine would be one following Carnot's cycle, in which heat is received while the air is at the highest temperature τ_1, the air meanwhile expanding isothermally. After this the supply of heat is stopped, and the air is allowed to expand adiabatically until its temperature falls to the lower extreme τ_2. At this it is compressed isothermally, giving out heat, and finally the cycle is completed by adiabatic compression, which restores the initial high temperature τ_1. The indicator diagram for this cycle has been sketched in fig. 12, § 43. Practically this action would be attended by the serious drawback that the volume to be swept through by the piston would be very great in relation to the work done. The inclination of adiabatic to isothermal curves for a gas is slight, and hence the area of the diagram, or the effective work done per revolution, is small in comparison with the two quantities of which it is the difference, namely, the work done by the substance during the forward stroke and the work spent upon it during the backward stroke. The mean effective pressure would be very small in comparison with the greatest pressure for which provision would have to be made in designing the parts for strength. An air-engine using Carnot's cycle would consequently be excessively bulky and heavy, and also mechanically inefficient.

315. External-Combustion Air-engine with Regenerator: Stirling's Air-engine.

This objection is lessened when the use of a regenerator (§ 53) is substituted for the adiabatic steps of the Carnot cycle. In Stirling's engine, where the regenerator was first used, the working substance was cooled from the upper limit τ_1 to the lower limit τ_2 by passing in one direction through a regenerator, which stored the heat it extracted from the gas in such a way that when the gas was passed through the regenerator in the opposite direction the heat was again taken up and the temperature consequently rose from τ_2 to τ_1. The cycle of operations has been described

FIG. 251. Ideal Indicator diagram of Air-Engine with Regenerator (Stirling).

in § 54, and an ideal indicator diagram sketched there is reproduced in fig. 251.

Several forms of engines were designed by Stirling in which the action approximated to the cycle of fig. 251. The characteristic parts of one of them are shown in section in fig. 252. A is the heater, a closed iron vessel containing air, externally heated by a furnace beneath it. A pipe from the top of A leads to the working cylinder B. At the top of A is a cooler C, consisting of pipes through which cold water circulates. In A there is a displacer plunger D, which is driven by the engine; when this is raised the air in A is in the lower part of the vessel and is consequently taking in heat from the furnace, whereas when D is lowered the air in A is transferred from the lower to the upper part and is thereby brought into contact with the cooler. On its way from the bottom to the top of A, or from the top to the bottom, the air must pass through an annular lining of wire-gauze E. This is the regenerator, and the air in passing up through it becomes cooled, and in passing down again through it becomes heated. At the beginning of the cycle D is at or near its highest position. The air is then receiving heat at temperature τ_1, and is expanding isothermally; this is the first stage in the diagram. Then the plunger D descends. The air is driven through the regenerator, where it deposits heat, and its temperature on emerging at the top is τ_2. Next, the working piston

makes its down-stroke; this compresses the air isothermally, the heat produced by compression being taken up by the cooler C. Finally the plunger is raised, and the working air again passes down through the regenerator, taking up the heat it left there, and rising in temperature to τ_1. In the actual engine the working cylinder was double-acting, another heating vessel, precisely like A, being connected with the cylinder B above the piston.
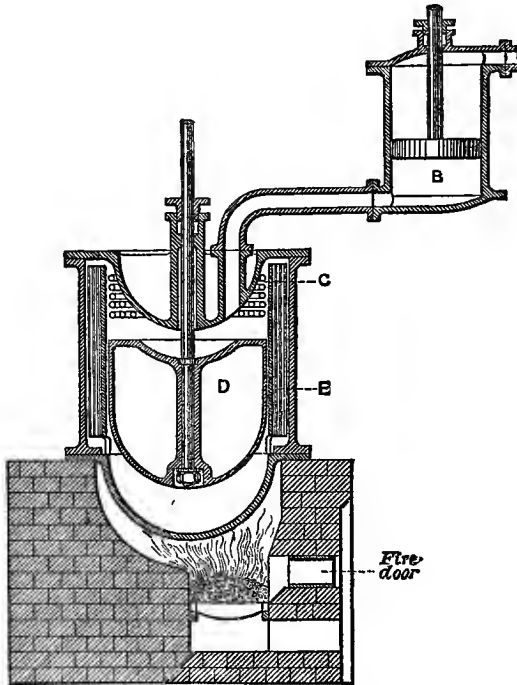
Fig. 252. Stirling's Air-engine

The actual forms in which Stirling's engine was used are described in two patents by R. & J. Stirling (1827 and 1840[1]). An important feature in them was that the air was compressed by means of a pump which formed an additional organ of the engine, so that its average pressure was kept much above that of the atmosphere. The pressure was commonly as much as 150 lbs. per

[1] The 1827 patent is reproduced in Fleeming Jenkin's Lecture on Gas and Caloric Engines, *Inst. Civ. Eng.*, Heat Lectures, 1883–84. See also *Min. Proc. Inst. C. E.* 1845 and 1854.

square inch with the air cold. Stirling's cycle is theoretically perfect whatever be the density of the working air, and the use of compression affects the theoretical thermodynamic efficiency only if the ratio of adiabatic expansion and compression be altered. But it gives a higher mechanical efficiency, and also, what is of special importance, it increases the amount of power developed by an engine of given size. To see this it is sufficient to consider that with compressed air a greater amount of heat is dealt with in each stroke of the engine, and therefore a greater amount of work is done. Practically the use of compressed air also increases the thermodynamic efficiency by reducing the ratio of the heat wasted by external conduction and radiation to the whole heat.

A double-acting Stirling engine of 50 I.H.P., used in 1843 at the Dundee foundry, appears to have realized an efficiency of 0·3, and, notwithstanding very inadequate means of heating the air it consumed only 1·7 lbs. of coal per I.H.P.-hour[1]. This engine remained at work for three years, but was finally abandoned on account of the failure of the heating vessels, though the highest temperature reached by the air was only about 343° C. In one form of the engine as described in Stirling's patent the regenerator was a separate vessel; in another the plunger D was itself constructed to serve as regenerator by filling it with wire-gauze and leaving holes at top and bottom for the passage of the air through it.

316. Ericsson's Air-engine. Another mode of using the regenerator was introduced in America by Ericsson, in an engine which also failed, partly because the heating surfaces became burnt, and partly because their area was insufficient. In Ericsson's engine, which was tried on a considerable scale on the steam-ship "Caloric," the temperature of the working substance was changed by passing through the regenerator while the pressure remained constant. Cold air was compressed by a pump into a receiver, from which it passed through a regenerator into the working cylinder. In so passing it absorbed heat from the regenerator and expanded. The air in the cylinder was then allowed to expand further by taking in heat from a furnace under the cylinder until its pressure fell to near that of the atmosphere.

[1] See Rankine's *Steam-Engine*, p. 367. The consumption per brake H.P. was much greater.

The cycle was completed by the discharge of the air through the regenerator. The indicator diagram approximates to a form bounded by two isothermals and two lines of constant pressure[1].

Compared with Stirling's engine Ericsson's had the practical disadvantage that the mean pressure of the working substance was much lower, the lower limit of the pressure range being the pressure of the atmosphere. This required the engine to have an excessive bulk relatively to its power. With four working cylinders, each 14 feet in diameter, and a stroke of six feet, it developed only about 300 indicated horse-power, the mean effective pressure being only a little over 2 lbs. per square inch. The consumption of coal was reported to be 1·87 lbs. per I.H.P.-hour, but a large part of the indicated power must have been expended in overcoming the friction of the engine itself.

317. Modern Air-engines of the Stirling type. Externally-heated air-engines are now employed only for very small powers—from a fraction of 1 H.P. up to about 3 H.P. Powerful engines of this type are scarcely practicable, partly on account of the relatively enormous bulk they would have and partly on account of the difficulty which would be experienced in the heating of large quantities of air. By keeping the working substance highly compressed, giving it a mean density much in excess of that of atmospheric air, the bulk of the working cylinder and displacer might be reduced, but the difficulty would remain of getting enough heating surface and of preserving the heater from being burnt through its exposure to oxygen at a high temperature. The small externally-fired air-engines that are now manufactured resemble the original Stirling engine very closely in the main features of their action, and comprise essentially the same organs.

One of these modern Stirling engines is the small domestic motor designed by Mr H. Robinson (fig. 253). In this case there is no compressing pump and the mean pressure of the working air is equal to the pressure of the atmosphere. The range of pressure is slight—so slight indeed that no packing is needed in the piston or other working parts—and the engine develops only a fraction of one horse-power. It is used for such purposes as pumping water for the supply of country houses.

[1] For a diagram of Ericsson's engine see Rankine's *Steam-Engine*, or *Proc. Inst. Mech. Eng.* 1873.

A is the heater and displacer cylinder; B is the working
cylinder, which communicates with A by a passage D. A is
heated externally by a small coke fire at C or by a gas flame
from a Bunsen burner.   The displacer E takes its motion from a
rocking lever F connected by a short link to the crank-pin, and is
about 90° in phase ahead of the working piston.   In the figure the
displacer is at the bottom of its stroke and the piston has still
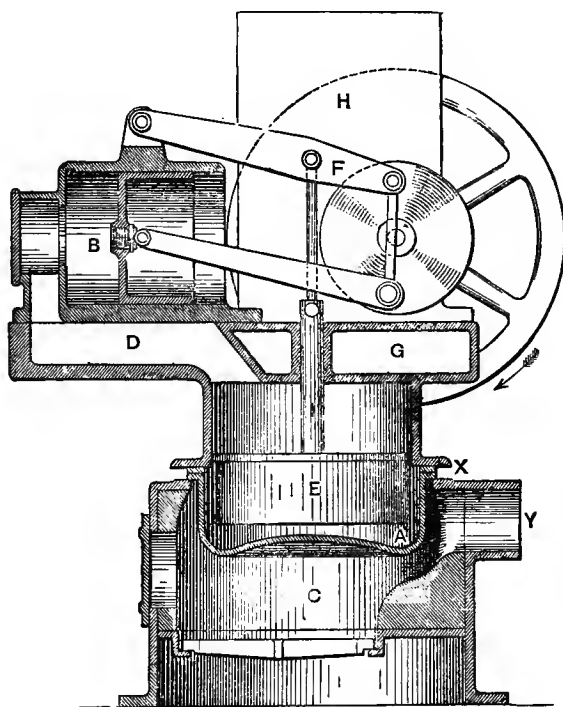
Fig. 253.   Robinson's form of Stirling Engine.

half the back stroke to perform.   The displacer E is itself the
regenerator, its construction being such that the air passes up and
down through it as in one of the original Stirling forms.   On the
top of the displacer cylinder is a water vessel G, which is the
cooler, and this is kept in communication with the circulating
water tank H.   The account which has already been given of the
Stirling cycle will serve as a description of the action in this
engine.   A conspicuous feature is that there are neither valves,

packing, nor glands; but the absence of compression, which makes
this possible, limits the effici   y of the engine as well as its power.

A larger engine of the Stirling type, working up to some
3 horse-power, is made by Messrs Bailey, of Salford. Another, the
Rider engine, made by Messrs Hayward and Tyler, follows sub-
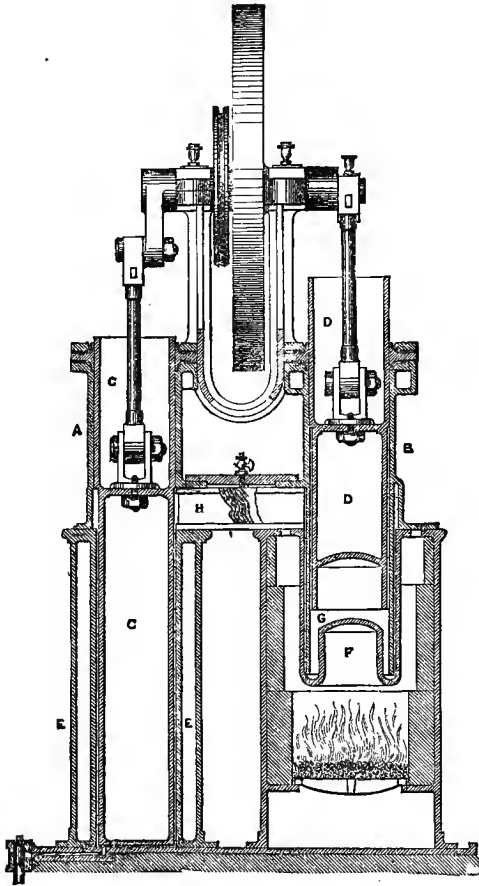
FIG. 254. Rider's Air-engine.

stantially but not exactly the Stirling cycle. A sectional view of
this engine is given in fig. 254. *A* and *B* are two cylinders,
open at the top, with plunger pistons *C* and *D*, which are connected
to cranks nearly at right angles. Between the two is the re-
generator *H*. Round the lower part of *C* is the cooler *E*, a

jacket through which cold water is kept circulating. Under the lower part of B is the furnace F which heats the air contained in the space G below the plunger D. In the position shown in the figure, D is rising, and C is just beginning to rise. Nearly all the working air has been compressed into G and is expanding as it takes in heat, doing work against the plunger D and also against C. By the time D reaches the top of its stroke C is about half way up: air is passing rapidly through the regenerator from G to the space under C, and is cooled first by the regenerator and then by the water-jacket E. As D comes down this transfer of the air continues and the pressure falls. Then C follows, compressing the air beneath it while the cooler E absorbs the heat, and finally forcing all the working air back through the regenerator into the heater, when the cycle begins again. The maximum pressure reached during the cycle is about 20 lbs. per sq. inch.

The action is of course continuous, but we may broadly distinguish the following four stages :

(1) The air, previously compressed to a small volume (in G), takes in heat at its highest temperature and expands, doing work on D and subsequently to some extent on C.

(2) After this expansion it is transferred through the regenerator to the cold cylinder A, storing heat in the regenerator and losing pressure. During this process little work is done on or by the air, since the actions on the plungers nearly balance. In other words, the volume does not materially change.

(3) The air which is now in A, expanded to large bulk and at a low temperature, is compressed by the descent of C and gives out heat to the cooler E. During this process work is done upon the air by the fly-wheel.

(4) The compressed air is transferred through the regenerator to G, rising in temperature and pressure. In this process, again, little work is done by or upon the air.

This engine differs from the pure Stirling type chiefly in having a displacer which is also a working piston. The Rider engine is mainly used to pump water for domestic supply, and the cooling jacket E is kept cold by making the water which the engine pumps circulate through it.

318. Internal-Combustion Air-engines. The difficulty already referred to of getting heat into and out of a gaseous

working substance is a fundamental objection which has prevented the external-combustion air-engine from coming into use on any large scale. The activity of a heating surface is vastly greater when the substance that is being heated is changing its state from liquid to vapour than when the substance is already a gas. And similarly a gas that is being cooled by conduction through a surface will part with its heat far less readily than a vapour which is being condensed in the process.

This objection applies when the air-engine is of the external-combustion type, but so far as the heating process is concerned it is removed by causing the combustion to occur within the engine itself. So far as cooling is concerned the difficulty also disappears when the substance which is to reject heat is expelled to the atmosphere instead of being used over again after cooling. Hence it has been possible for the internal-combustion engine to attain a much greater efficiency.

The earliest practical example of the internal-combustion engine (if we leave guns out of account) appears to have been the hot-air engine of Sir George Cayley[1], of which Wenham's[2], Buckett's[3] and Bénier's engines are more recent forms. In these engines coal or coke is burnt under pressure in a closed chamber, to which the fuel is fed through a species of air-lock. Air for combustion is supplied by a compressing pump, and the engine is governed by means of a distributing valve which supplies a greater or less proportion of the air below the fire as the engine runs slow or fast. The products of combustion, whose volume is increased by their rise in temperature, pass into a working cylinder, raising the piston. When a certain fraction of the stroke is over the supply of hot gas is stopped, and the gases in the cylinder expand, doing more work and becoming reduced in temperature. During the return stroke they are discharged into the atmosphere, and the pump takes in a fresh supply of air. Fig. 255 is a diagram section of the Buckett engine. A is the working piston, the form of which is such as to protect the tight sliding surface (at the top) from contact with the hot gases; B is the compressing pump, and C is the valve by which the governor regulates the rate at which fuel is consumed by admitting more or less of the air under the grate through the channel F. D is the

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[1] *Nicholson's Art Journal*, 1807.   See also *Min. Proc. Inst. C. E.* Vol. ix.
[2] *Proc. Inst. Mech. Eng.* 1873.   [3] Fleeming Jenkin, *loc. cit.*

air-lock and hopper through which fuel is supplied, and E is the exhaust valve through which the products of combustion are finally expelled. In a reported trial of an engine indicating 20 horse-power the number of lbs. of fuel burnt per hour was 1·8 per I.H.P. and 2·5 per B.H.P.

In an engine such as this it was practicable to let the working substance reach a higher temperature than could be reached with
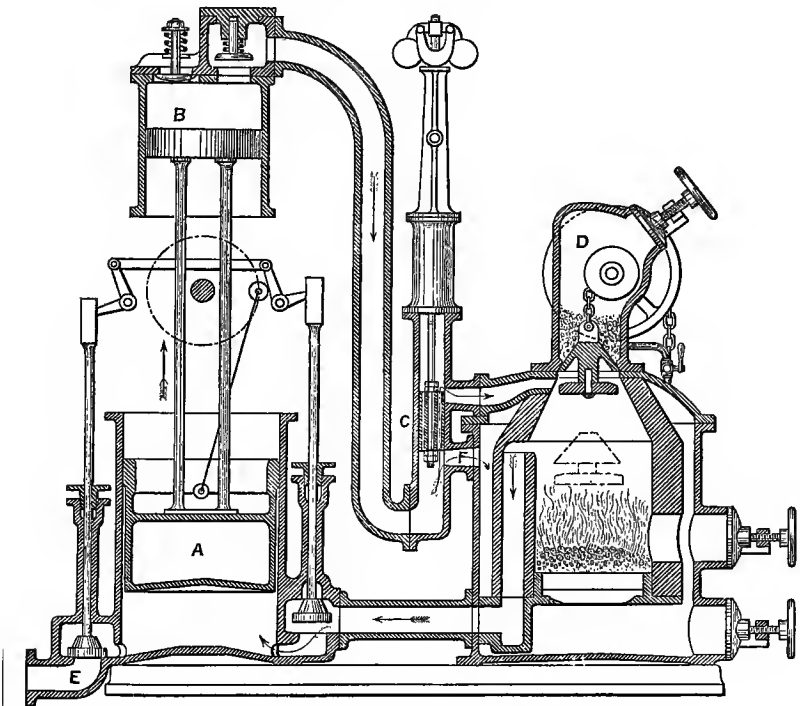
FIG. 255. Buckett's Internal-Combustion Air-engine.

external firing, and in fact in Wenham's engine it was about 600° C. But a limit was still imposed by the fact that the hot gas had not only to be contained in iron vessels, under pressure, but had to go through passages and valves between the heating chamber and the cylinder. Hence in practice the highest temperature in the working cycle was necessarily much lower than it is in a gas-engine, where no transfer of the gas takes place after it is heated by explosion, until it has cooled itself by doing work.

In engines of this class the degree to which the action is

thermodynamically efficient depends very largely on the amount of cooling the gases undergo by adiabatic or nearly adiabatic expansion under the working piston. Without a large ratio of expansion the thermodynamic advantage of a high initial temperature is lost. In any kind of internal-combustion engine the gases have to be discharged at atmospheric pressure, and consequently a large ratio of expansion is possible only when there is much initial compression. Compression is therefore an essential condition, without which a heat-engine of this type cannot be made efficient. It is also, as has already been pointed out, an essential feature in any air-engine which is to develop a fair amount of power without excessive bulk. When we come to discuss in the next chapter the theoretical action of gas-engines we shall see how dependent the thermodynamic efficiency is on the amount of compression which the working substance undergoes before ignition takes place.

Internal-combustion engines using solid fuel have been but little used, and that only for small powers. But the use of liquid and especially of gaseous fuel has given the internal-combustion engine a position of great and constantly increasing importance. Gas-engines, that is to say, engines acting by the combustion or explosion of a mixture of air and a combustible gas or vapour, have in recent years entered into serious competition with the steam-engine. In some of these the fuel is the ordinary coal-gas supplied for illuminating purposes, but where any considerable amount of power is to be developed it is much more economical to use a cheap combustible gas, which may be made for the purpose from coke or coal, and plants using such "producer gas" are now common. In many instances gas-engines of large size utilize the combustible gas discharged from coke-ovens or from blast furnaces. In another group of internal-combustion engines the fuel is liquid paraffin (kerosene), vaporised before admission to the cylinder; and in another the fuel is petrol, a lighter and more readily vaporised product of the distillation of petroleum. Engines of this last class have become widely known through their use in motor cars: it is their successful development that has made mechanically propelled road vehicles common and has enabled a solution to be arrived at of the problem of artificial flight. These types of internal-combustion engines will be considered in the next chapter.

CHAPTER XVII.

GAS-ENGINES AND OIL-ENGINES.

319. Early Gas-engines. Lenoir. The first gas-engine to be brought into practical use was that of Lenoir (1860) though before that several inventors had described engines none of which got beyond the experimental stage. In 1820 the Rev. W. Cecil read a paper before the Cambridge Philosophical Society giving details of experiments made with an engine of his own design driven by the explosion of a mixture of hydrogen and air. It is probable that this was the earliest gas-engine to be actually worked. A patent by W. Barnett (1838) makes the important proposal to use compression of the explosive mixture: it also describes a method of ignition which was afterwards applied for a long time in other engines with success. Another patent by Barsanti and Matteucci, in 1857, describes a free-piston engine of a type which afterwards took practical form in the hands of Otto and Langen. But none of these, or other, early proposals bore immediate fruit, and it was not until Lenoir brought out a design in which various difficulties were met that the gas-engine entered on its practical career[1].

Lenoir's engine was of great simplicity. There was no compression, and in this respect the action (from the thermodynamic point of view) was seriously defective and was a retrograde step after Barnett's proposal. During the early part of the stroke air and gas, in proportions suitable for combustion, were drawn into the cylinder. At about half-stroke the inlet valve closed, and the mixture was immediately exploded by an electric spark. The heated products of combustion then did work on the piston by

[1] For accounts of the early history of the gas-engine see D. Clerk's *The Gas, Petrol, and Oil Engine*, Vol. I., 1909, also B. Donkin's *Text-Book of Gas, Oil, and Air-engines*, Fourth Ed., 1905.

expanding during the remainder of the forward stroke, and were expelled during the back stroke. The engine was double-acting, and the cylinder was prevented from becoming excessively heated by a casing through which water was kept circulating. This water-jacket is a feature that has been retained in nearly all later gas-engines. In Lenoir's engine every stroke was active, two explosions taking place per revolution, one on each side of the piston.

An indicator diagram from a Lenoir engine is shown in fig. 256. After the explosion the line falls, partly from expansion, and partly from the cooling action of the cylinder walls. In consequence of there being no compression of the explosive mixture before the combustion takes place the amount of subsequent expansion is very small:

FIG. 256. Lenoir Engine Diagram.

the final volume is less than double the volume after explosion. Hence the amount of cooling through expansion is comparatively small, and the efficiency is low; the waste heat discharged in the products of combustion as they leave the cylinder, or taken up by the water-jacket by conduction through the walls, forms a very large part of the whole heat generated. Only some 4 per cent. of the energy of the gas was converted into work, whereas in a good modern gas-engine the indicated thermal efficiency exceeds 30 per cent. Notwithstanding what we should now consider its excessive consumption of gas the Lenoir engine found considerable application between 1860 and 1870, and engines of the class then built continued to work well for many years. Hugon's engine, introduced five years later than Lenoir's, was a very similar non-compressive engine. A novel feature in it was the injection of a jet of cold water to keep the cylinder from becoming too hot. These non-compressive engines are now obsolete; the last to survive was a small engine designed by Bischoff, the mechanical simplicity of which gave it an advantage in cases where but little power was required.

320. **Otto and Langen's Atmospheric Gas-engine**. In 1866 Otto and Langen gave working shape to the suggestions of Barsanti and Matteucci by introducing a practical free-piston engine which, as to economy of gas, was distinctly superior to

engines of the Lenoir type[1]. Like them it did not use compression. The explosion occurred early in the stroke, in a vertical cylinder, under a piston which was free to rise without doing work on the engine shaft. The piston rose with great velocity, so that the expansion was much more nearly adiabatic than in earlier engines, and the ratio of expansion was greater, for the momentum of the piston carried it so far that the pressure of the gas fell below that of the atmosphere. After the piston had reached the top of its range the gases became further cooled by giving up heat to the walls of the cylinder, and, their pressure being below that of the atmosphere, the piston came down, this time in gear with the shaft, and doing work upon it. The burnt gases were discharged during the last part of the down-stroke. A friction-coupling allowed the piston to be automatically thrown out of gear with the shaft when rising, and into gear when descending. This "atmospheric" gas-engine came into somewhat extensive use in spite of its noisy and spasmodic action. It achieved an indicated thermal efficiency of about 16 per cent. in favourable cases, and about 11 per cent. on the brake. After a few years it was displaced by a greatly improved type, in which the direct action of Lenoir's engine was restored, but the gases were compressed before ignition.

321. The four-stroke cycle of Beau de Rochas and Otto. The advantage of compressing the explosive mixture before igniting it, in order to make the subsequent expansion large, was recognized by several inventors when Lenoir's engine came into use. It was discussed in a remarkable pamphlet published by Beau de Rochas in 1862. He pointed out that compression might be carried to any extent short of that which would cause the mixed gas to explode in consequence of its elevation of temperature. He further suggested a means of compressing the explosive mixture without using a separate compressing pump. His plan was to have the following four operations take place, on one side of the working piston, during four successive strokes or two revolutions of the crank-shaft.

(1)   Drawing in the charge of gas and air during one whole
        stroke of the piston.

[1] *Proc. Inst. Mech. Eng.* 1875.

(2) Compression during the return stroke (into a comparatively large clearance space below the piston).

(3) Ignition at the dead point, followed by expansion during the third stroke.

(4) Discharge of the burnt gases from the cylinder during the fourth and last stroke.

This was the earliest account of the "four-stroke" cycle of operations which is now used in most gas-engines. Beau de Rochas further pointed out that, besides compression, high speed and small cylinder surface were conditions to be aimed at as favourable to economy. Extremely valuable as were the suggestions contained in his pamphlet they were for a long time unproductive. It was not till 1876 that Otto, who with Langen had been engaged in the manufacture of the "atmospheric" engine, introduced the highly successful gas-engine in which this action is carried out. The new engine was called the Otto "silent" engine to distinguish it from its noisy predecessor, the engine of Otto and Langen. Not only was it the first gas-engine to come widely into use, but it has formed the model to which, since the expiry of Otto's master patent, other gas-engines are for the most part indebted for the chief features of their action. The manufacture of the Otto engine in England by Messrs Crossley led to its rapid introduction in thousands of cases. At first the only fuel available was illuminating coal-gas, and the engine was used for rather small powers only, in which the cost of gas fuel though greater than that of the fuel for a steam-engine of corresponding power was more than balanced by the greater convenience and saving in attendants' wages. For a good many years nearly all gas-engines were of less than 20 horse-power. In 1881 Mr J. Emerson Dowson applied a cheap form of gaseous fuel, obtained by the partial combustion of anthracite in a "producer" of his own design, to the driving of a gas-engine. Since then the application of producer gas has been effected in many ways, and in 1895 it was shown by Mr B. H. Thwaite that gas-engines might be driven by the so-called waste gases of the blast-furnace. The use of cheap gas has given an immense impetus to the design of large gas-engines, and they are now built in sizes generating hundreds and even thousands of horse-power. In some instances the dimensions are such that over 1000 horse-power are developed in a single cylinder, and engines working up to 3000 or 4000 horse-power are

designed by combining four cylinders, usually by placing two cylinders in tandem and coupling two such tandem pairs.

Nearly all gas-engines of small or moderate power, and many of the largest, use the four-stroke cycle of Beau de Rochas. For brevity they are spoken of as *four-cycle* engines. The chief alternative is the *two-cycle* action introduced by D. Clerk which will be referred to later (§ 325).

322. The Four-cycle Engine. In the original Otto engine, and in most other small examples of the four-cycle type, the cylinder is single acting, with a trunk piston, and the explosive mixture is compressed before ignition into a large clearance space at the back end of the cylinder. The volume of the clearance depends on the amount of compression which is desired. In the early forms of Otto engine it was generally more than half the volume through which the piston moved, but a much larger amount of compression is now common. To complete the action requires two revolutions of the crank-shaft.

During the first forward stroke of the cycle gas and air are drawn in by the piston. During the first back-stroke the mixture is compressed into the clearance space. The mixture is then ignited as the crank reaches the dead point, and the second forward stroke, which is the only working stroke in the cycle, is performed under the pressure of the heated products of combustion. During the second back-stroke the products are discharged into the atmosphere through an exhaust valve, with the exception of so much as remains in the clearance space, which (except where special means are taken to remove it) is allowed to dilute the explosive mixture in the next cycle. The cylinder is kept cool enough to admit of lubrication, by means of a jacket through which a continuous circulation of water is kept up. The admission and exhaust valves are worked by cams on a side-shaft which is geared to run at half the speed of the crank-shaft, so that the several events take place once in two revolutions of the engine.

323. Ignition in Gas-engines. In early forms of the Otto engine the ignition of the compressed gases was effected by carrying a flame, through a narrow port in a slide-valve, from a gas jet that was kept burning outside to the mixture within. The slide-valve was worked from the half-speed side-shaft. To

prevent the gases from blowing back through the valve when the explosion took place the slide was arranged so that the port in it which served as a vehicle for the flame had passed under a cover which shut it off from the atmosphere, before it reached the fixed port in the cylinder cover through which the flame passed in to ignite the contents of the cylinder. This mode of igniting the gases is now obsolete. The device which took its place was an ignition tube, namely, a small closed tube of metal or porcelain maintained at a bright red heat by a flame playing on its outside surface. A portion of the explosive mixture was allowed to enter the tube from the cylinder at the time when it should be fired. In most cases the ignition tube was used in conjunction with a "timing valve" which determined the instant at which the explosion occurred by being opened to allow a portion of the compressed explosive mixture to enter the ignition tube, but in some gas-engines the "timing valve" was dispensed with and the ignition tube was in free communication with the cylinder throughout, the instant of firing being determined only by the compression of some of the explosive mixture into the tube. The ignition tube is now used only in some small engines : in larger engines the use of electrical ignition is practically universal. For this purpose many engines employ a "sparking plug" with an insulated terminal from which a high-tension spark or rather stream of sparks is discharged at the desired instant by the use of an induction coil or high-tension magneto machine. In other cases a low-tension spark is produced by the mechanical breaking of a contact within the cylinder, in a circuit having a considerable amount of self-induction.

324. Governing of Gas-engines. The speed of a gas-engine is usually regulated by a centrifugal governor, and in small engines this often acts by cutting off the supply of gas when the speed exceeds a certain limit, making the engine miss one or more explosions. The governor determines whether the gas valve shall or shall not be opened, by means of a "hit and miss" arrangement of the kind briefly referred to in § 235. A cam fixed on the side-shaft, so that it makes one revolution for every two revolutions of the engine, opens the gas-admission valve by acting on a lever through an intermediate roller. This intermediate roller is carried by an arm which is caused to move sideways by the governor, in

such a manner that when the speed exceeds a certain value the roller is removed and consequently the cam fails to act on the lever, and the admission valve remains closed. In some instances a stepped cam is used, giving admission to various amounts of gas corresponding with various positions of the centrifugal governor, with the effect that the explosive mixture is weakened when the speed rises. The tendency, however, to miss fire with weak mixtures is an obstacle to this method of regulating, and more generally the gas is freely admitted when the speed is below the limit, and completely cut off when the limit is passed. In some small gas-engines the inertia of a reciprocating piece is used instead of the inertia of revolving pieces to determine the admission or non-admission of gas. When the speed exceeds a limit the acceleration of the oscillating piece becomes sufficient to displace it in such a way that the gas-admission valve misses the stroke.

For large gases the usual method of governing is to control the quantity of explosive mixture taken in per cycle, without altering the proportion in it of gas to air. For this purpose the mixed gas enters through an admission valve, which, like the exhaust valve, is of the mushroom or lift type. The governor acts by determining the time during which the admission valve is lifted, and consequently the volume drawn in to form the charge.

325. The Clerk or two-cycle Engine. Mr Dugald Clerk, who has done much for the development of the gas-engine both by his inventions and by his investigations in points of theory, introduced in 1881 an engine in which the explosion occurs once in each forward stroke of the piston instead of once in each alternate forward stroke as in the Otto or Beau de Rochas cycle. The Clerk cycle has therefore the advantage of reducing the proportion of idle strokes by two-thirds. His two-cycle engine requires a pump or displacer, and is in this respect less simple than the Otto four-cycle engine, in which the working cylinder itself acts as pump. The Clerk engine, though it met with some favour as a competitor of the Otto during the life-time of Otto's patent, fell after that out of use and remained in disuse so long as gas-engines of small power only were built; but in the modern development of large gas-engines the Clerk cycle has been revived, and some of the most successful large engines now employ it.

In the Clerk engine of 1881 (fig. 257) the gas and air were

inhaled by an auxiliary piston in a separate cylinder A forming the pump, from which they were delivered at a low pressure (about 4 lbs. per square inch) to the main cylinder just after the main piston had completed its working stroke. They entered at B passing through a trumpet mouth or large cone C forming the back end of the cylinder, which had the effect of removing the kinetic energy of the stream, and hence of allowing the fresh charge to
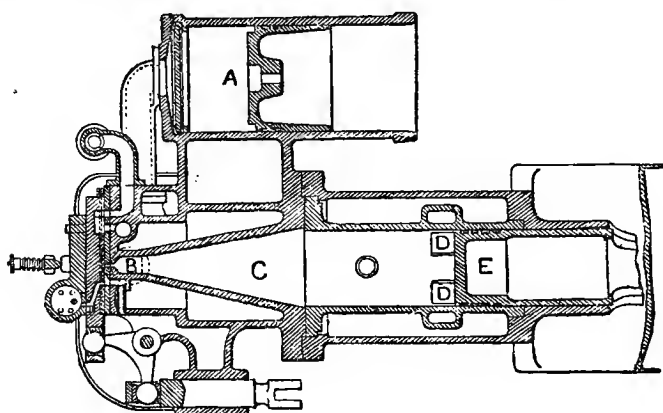
FIG. 257. Clerk engine of 1881. Horizontal section through cylinder and pump.

enter without intermingling much with the products of combustion already in the cylinder. The fresh charge drove the products of combustion in front of it, causing these to be expelled at exhaust ports D in the side of the cylinder close to the front end of the stroke. The piston E, returning, closed these exhaust ports and compressed the fresh mixture, which was ignited as usual when its compression was completed by the piston passing its dead-point at the back end. The indicator diagram was almost identical with that given by the Otto engine.

326. Atkinson's Engines. It is a defect of the ordinary Otto cycle that the ratio in which the gases are expanded after ignition is no greater than the ratio in which the explosive mixture is compressed. A larger ratio of expansion is desirable, for the temperature and the pressure are still high when release occurs. The ingenious "differential" engine of Mr Atkinson (1885) was an attempt to avoid this drawback. Its working chamber consisted

of the space between two pistons working in one cylinder. During exhaust the pistons came close together; they receded from each other to take in a fresh charge; they approached for compression; and finally during the working stroke they receded again very rapidly and farther than before, after ignition of the mixture, thus giving a comparatively large ratio of expansion with the further advantage that the working stroke took place so quickly as to give the burning gases comparatively little time to give up heat by conduction to the metal. At the same time, by moving bodily along through the cylinder, the pistons uncovered admission and exhaust ports in the sides, as well as an ignition tube which was kept permanently incandescent. The pistons were connected to a single crank-pin through a pair of beams or levers with connecting links at each end; this had the important disadvantage of introducing a large number of working joints.

A year or so later Mr Atkinson abandoned the use of two pistons, but succeeded in giving a single piston long and short strokes alternately, by connecting it to the crank-pin through a species of toggle joint which made two complete oscillations for each revolution of the shaft. The first oscillation served to inhale and compress the gases; the second, the amplitude of which was about twice as great, served for the working and exhausting stroke. Competitive tests made under the auspices of the Society of Arts showed that the "cycle" engine, as this form was called, had an exceptionally high thermal efficiency for that date, but the mechanical complication of the toggle with its still numerous joints stood in the way of its success. Both of these engines are obsolete, but they are interesting as attempts to secure a ratio of expansion which should not be limited by the ratio of compression.

327. Scavenging. In the ordinary Otto cycle the clearance space is left at the end of the exhaust stroke full of products of combustion, which mix with the incoming charge. To avoid this some of the early gas-engines working on the Otto cycle had a pair of idle "scavenging" strokes added to the four strokes necessary for the cycle, with the object of clearing the burnt gas out more completely before the next admission of explosive mixture. After the usual exhaust stroke the two scavenging strokes took place: in the first of them air was drawn in atmospheric pressure and in the return stroke it was expelled, with the result that the clearance

E. 35

space was left full of nearly pure air instead of products of combustion. There was, however, no sufficient advantage in this to compensate for the drawback of having additional idle strokes, and the use of such strokes was soon given up. The presence of some burnt gas in the mixture does not prevent ignition nor interfere with the completeness of combustion. A considerable amount of scavenging can be secured by making use of the momentum of the stream of escaping exhaust gases to draw in fresh air, by opening the air admission valve before the exhaust valve is closed, a device first used for the purpose by Mr Atkinson. The clearance spaces are now so small, in consequence of the high compression used in modern engines, that the quantity of burnt gas that remains has become relatively unimportant. The principal advantage in sweeping out the clearance space with fresh air before the charge enters is to reduce the risk of premature ignition.

328. Ideal Action in the Four-Stroke Cycle. Consider an ideal engine in which there is no transfer of heat between the working substance and the metal, and in which the combustion is completed while the piston is at the dead-point, after compression
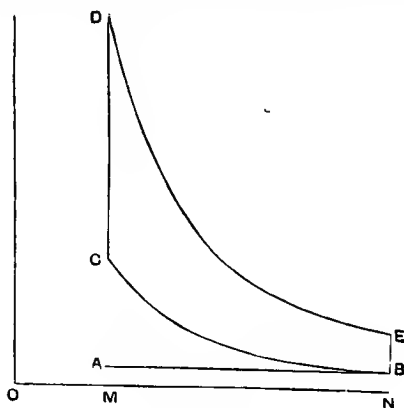
Fig. 258.

is over and before expansion begins. The indicator diagram of such an ideal engine would take the form shown in fig. 258. OM is the volume of the clearance, which is the volume occupied by the mixture during combustion, and MN is the volume swept through by the piston. AB is the admission at atmospheric pressure; BC is the compression, which by assumption is adiabatic; CD is the rise of pressure caused by the explosion; DE is

adiabatic expansion constituting the working stroke; and *EBA* is the exhaust.

From *C* to *D* the whole energy due to the chemical reaction goes to heat the mixture, for by hypothesis none is lost by conduction to the walls. The rise of pressure from *C* to *D* can be readily calculated when the rise of temperature resulting from this accession of heat is known, provided we also know what is the change in specific volume due to the change in chemical constitution brought about by the explosion. With the mixture used in gas-engines there is very little change in specific volume: that is to say the burnt products, if brought to the same pressure and temperature would fill very nearly the same volume as they filled before chemical union took place. With mixtures of coal-gas and air the specific volume is reduced after explosion by about two per cent. in ordinary cases. With some explosive vapours the specific volume is slightly increased. The changes being in any case small, it is convenient in considering an ideal engine to ignore them, and to treat the working substance as if it were air to which between *C* and *D* there is imparted a definite quantity of heat, which may be calculated when we know the composition of the explosive mixture and the heats of combustion of its various constituents.

Hence, provided we knew the temperature at *C*, we could calculate the rise of temperature and consequently the rise of pressure at *D*, if the average specific heat (at constant volume) between *C* and *D* were known.

When this calculation is made on the basis of assuming that the same value may be assigned to the specific heat as we know it to have at low temperatures, the calculated rise of temperature turns out to be very much larger than is ever found in a gas-engine, and the difference is much too great to be explained by the loss of heat which, in a real engine, takes place by conduction or radiation to the cylinder walls during the time combustion is going on.

At one time it was commonly supposed that this difference was to be accounted for by incompleteness in the combustion itself. But recent researches have shown that the specific heat of a gas increases very considerably at high temperatures, and it is mainly for this reason that the temperature and consequently the pressure reached after explosion falls much short of the value which might be expected if this increase of specific heat did not take place.

329. Ideal Cycle with Constant Specific Heat. It is nevertheless instructive to study an ideal cycle in which the working substance is supposed to have a constant specific heat, and in which the action takes place in the manner assumed in fig. 258. This was done by a Committee of the Institution of Civil Engineers, who on this basis determined what is generally called the "air standard" for comparison with the performance of actual internal-combustion engines[1]. The assumptions made are that there is no transfer of heat between gas and metal, that there is complete combustion at the dead-point, and that the specific heat is constant. Let τ_0 be the (atmospheric) absolute temperature at which the working mixture is taken in, τ_1 the temperature to which it is compressed, τ_2 the temperature after explosion, and τ_3
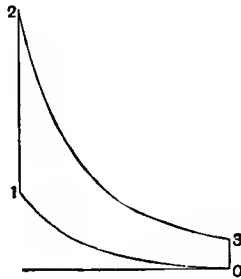
FIG. 259.

the temperature after expansion. Fig. 259 shows the cycle, with the stages numbered to correspond with these suffixes. Let K_v be the (constant) specific heat at constant volume. Then the heat taken in, namely the heat generated in the explosion, is $K_v(\tau_2 - \tau_1)$. The heat rejected is $K_v(\tau_3 - \tau_0)$, for it makes no difference whether the products of combustion are cooled on release to the atmosphere or kept in the cylinder after expansion and cooled there to atmospheric temperature, at constant volume, before release. Hence the efficiency is

$$\frac{K_v(\tau_2 - \tau_1) - K_v(\tau_3 - \tau_0)}{K_v(\tau_2 - \tau_1)},$$

or

$$1 - \frac{\tau_3 - \tau_0}{\tau_2 - \tau_1}.$$

[1] Report of the Committee on the Efficiency of Internal Combustion Engines, *Min. Proc. Inst. C. E.* Vols. CLXII. and CLXIII.

Writing r for the ratio of compression, which is also the ratio of expansion, we have by § 41

$$\frac{\tau_0}{\tau_1} = \left(\frac{1}{r}\right)^{\gamma-1} \text{ and } \frac{\tau_3}{\tau_2} = \left(\frac{1}{r}\right)^{\gamma-1}.$$

Hence also

$$\frac{\tau_3 - \tau_0}{\tau_2 - \tau_1} = \frac{\tau_0}{\tau_1} = \left(\frac{1}{r}\right)^{\gamma-1}.$$

The efficiency of this ideal cycle may accordingly be written

$$1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

This conclusion is practically important as showing the beneficial influence of compression in augmenting the efficiency. With increased amounts of compression the efficiency of the "air standard" increases as follows, taking γ to be 1·4:

Ratio of Compression	Efficiency
2	0·242
3	0·356
4	0·426
5	0·475
7	0·541
10	0·602
15	0·661
20	0·698
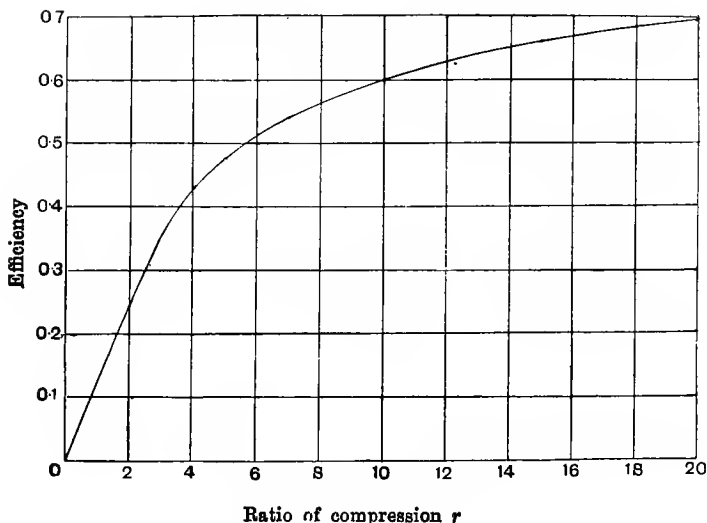
Fig. 260. Efficiency of "air standard."

It will be seen from these figures and from the curve (fig. 260) that there is at first a very rapid gain of efficiency with compression but that the advantage of increased compression becomes less as the compression increases. The efficiencies actually realized fall much short of these values, not only because the actions in a real engine are far from adiabatic but also because the real working substance does not have a constant specific heat. In favourable cases something like 60 per cent. of the air-standard efficiency is attained, and in one test of a large engine cited by Clerk[1] the measured efficiency was as high as 69 per cent. of the air standard.

330. Variation of Specific Heat. Attempts have been made by several independent methods to determine experimentally the variation of specific heat with temperature, or rather the relation between the internal energy of the gas and its temperature, the internal energy being identical with the heat taken in when the gas is heated at constant volume. For a summary of these experiments and their results reference should be made to the Reports of the British Association Committee on Gaseous Explosions[2]. Though the results are not very accordant they leave no room to doubt that the internal energy increases not uniformly with the temperature but in a more rapid ratio as the temperature rises. Thus at 1000° C. the internal energy of a gas-engine mixture appears to be about 22 per cent. higher than it would be on the hypothesis of constant specific heat, at 1500° C. about 30 per cent. higher, and at 2000° C. about 41 per cent. higher.

In gas-engine calculations it is convenient to reckon the specific heat and internal energy per unit of volume, the unit volume being measured at standard pressure and temperature (760 mm. and 0° C.). This has the great advantage that the reckoning is practically independent of the character and strength of the mixture, since, as was pointed out in § 38, equal volumes of the various gases require very nearly equal quantities of heat to raise their temperatures by the same amount. Taking K_v for dry air to be 0·173, which seems the most probable value (§ 38), and its volume at standard pressure

[1] *Min. Proc. Inst. C. E.* Vol. CLXIX. 1907.

[2] Brit. Assoc. *Reports* from 1908. See also D. Clerk's book on *The Gas, Petrol, and Oil Engine*, Vol. I. 1909.

and temperature to be 12·39 cub. ft. per lb. we should have for the specific heat at constant volume, reckoned in foot-pounds per cubic foot at standard conditions,

$$\frac{0 \cdot 173 \times 1400}{12 \cdot 39} = 19 \cdot 6 \text{ foot-pounds.}$$

This figure may be taken to apply to any gas-engine mixture at low temperatures. The increase at higher temperatures raises this figure, so far as can be told from the available experiments, to about 36 foot-pounds at 2000° C., the mean value between 100° C. and 2000° C. being about 27·5. These figures must be taken as provisional and may require correction in the light of later measurements.

331. Comparison of Efficiencies. When allowance is made for the variation of specific heat which occurs in a real working substance it appears, from estimates by Callendar[1], that the ideal efficiency of an engine working on the cycle of fig. 258 is roughly about 75 per cent. of the efficiency of the air standard. Thus in the example already referred to where the test of an engine gave an efficiency of 69 per cent. of the air standard, the real margin for improvement is not 31 per cent. but only the difference between 69 and 75. This is the loss due to the action not being adiabatic. It is a comparatively small quantity in large engines, where the volume is large in comparison with the surface; but in small engines it is much more considerable.

Comparing three engines differing much in size Callendar finds that a good approximation to the loss of indicated power which arises from heat passing through the walls is obtained by deducting the fraction $1/D$, where D is the diameter of the cylinder in inches. In other words, the indicated power to be expected is roughly $\left(1 - \dfrac{1}{D}\right)$ of the power that would be realized if the engine worked without losses, in which case it would approach, as has just been said, to 75 per cent. of the air standard. The three engines compared had cylinder diameters of $2\frac{1}{3}$, $5\frac{1}{2}$, and 14 inches respectively. Compared with the air standard the ratios of their actual to ideal performance were respectively

<div align="center">0·42,   0·61,   0·69.</div>

[1] *Min. Proc. Inst. C. E.* Vol. CLXIX. 1907, pp. 163—166.

Compared with the modified ideal standard which is 75 per cent. of the air standard these figures become

$$0.56, \qquad 0.81, \qquad 0.92,$$

which agree well with the values of $1 - \dfrac{1}{D}$, namely,

$$0.57, \qquad 0.82, \qquad 0.93.$$

332. The Process of Explosion. Much light has been thrown on the action which takes place when a gas-engine mixture explodes, by the researches of D. Clerk, B. Hopkinson and others[1]. The following may be taken as a brief summary of the chief facts established by experiments. In these researches the gaseous mixtures were ignited in a closed vessel, in some cases at atmospheric pressure initially, in other cases at higher pressures. The changes of pressure in relation to the time were observed, and also in some cases the changes of temperature as they occurred at various points within the vessel.

When the mixture is ignited at any point, a flame spreads in all directions from the point in question, taking a finite time to travel, so that each portion of the mixture ignites in turn, the most distant portion last. The rate at which the flame spreads depends on the pressure. When the initial pressure is atmospheric, it may travel only about five feet a second even in a rich mixture such as 1 of gas to 9 of air. The rate depends on the richness of the mixture as well as on the pressure: in a weak mixture it takes much longer for the combustion to extend to the more distant parts.

As the combustion spreads and more and more heat is given out the average temperature rises and the pressure rises with it. At the same time losses of heat occur to the walls, first by radiation and later by conduction as the combustion spreads to the gas in contact with the walls. If there were no losses the pressure would continue to increase until combustion was complete, but the losses of heat to the walls tend to reduce the pressure, and make it pass its maximum before all the gas is burnt. It is probable that the layer of gas adjacent to the walls burns with difficulty and that its combustion requires a comparatively long time to be quite complete.

The view was formerly held that even after any portion of the

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[1] See Mr Dugald Clerk's book cited above.

gas had become ignited a considerable time elapsed before the chemical reaction in that portion was complete, and the name "after-burning" was used to describe the supposed slow continuance of the process. According to this theory, it was believed that after the flame had spread throughout the whole mixture a large fraction—perhaps 40 per cent.—of the whole heat due to the chemical action had still to be evolved and that its evolution went on slowly, after the temperature and pressure had begun to fall through the cooling action of the walls, or, in a gas-engine, through both that and the expansion of the gas. The phenomena which seemed to support this view are now explained as being due to variation of specific heat; and it is generally accepted that little or no after-burning in this sense occurs, any continued combustion of the mixture as a whole being mainly or entirely due to delay in the ignition of all portions of the gas, especially those portions which lie in close contact with metal. In mixtures of ordinary strength, nearly though not quite all of the heat is evolved by the time the maximum pressure is reached. "It seems probable that very shortly after the attainment of maximum pressure, that is, within a time small compared with that required to reach maximum pressure, the transformation of the chemical energy into thermal form is everywhere complete except in a thin surface layer where this transformation is retarded by the cooling action of the walls[1]."

The distribution of temperature throughout the gas after explosion is far from uniform though the mixture may have been perfectly homogeneous before ignition. The part which is first ignited, in the neighbourhood of the ignition plug, first receives heat at a nearly constant pressure, through its own combustion, and is afterwards compressed in a nearly adiabatic manner because of the subsequent combustion of other portions. Its temperature consequently rises to a value much higher than that reached by outlying portions, which are first compressed and then heated by combustion without further rise of pressure. In a closed-vessel experiment by Hopkinson[2], where ignition took place at the centre of the vessel full of gas at atmospheric pressure, the temperature at the centre rose quickly on ignition to about 1200° C. and subsequently, through compression, to about 1900° C. This was when the pressure reached its maximum of 6·5 atmospheres. The mean temperature throughout the vessel was then

[1] *Brit. Assoc. Rep.* 1908.   [2] *Proc. Roy. Soc.* Feb. 1906.

1600° C. and the temperature 1 cm. for the walls was about 1200° C. These differences soon disappeared through convection of the heat, and the burnt mixture became a hot core of nearly uniform temperature surrounded by a layer chilled by conduction to the enclosing wall. Assuming that the loss by radiation from the flame was about 15 per cent. Hopkinson calculates that the mean specific heat of the burnt products between 1200° C. and 1900° C. was 1·3 times the value at low temperatures, and that the mean value of γ, the ratio of specific heats, within the same range was 1·25.

To illustrate the difference which richness of the mixture makes in the time-rate at which an explosion occurs, by affecting the
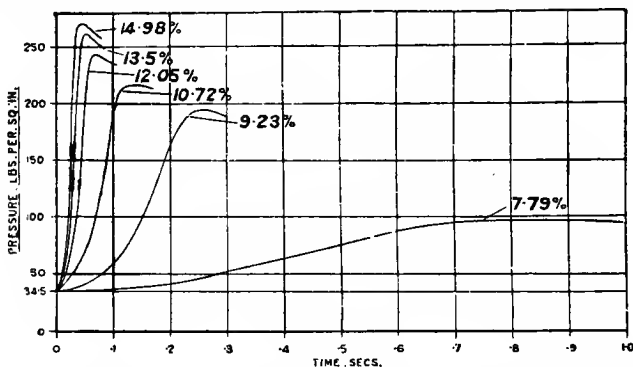
Fig. 261. Explosion curves of various mixtures of gas and air at initial pressure of 34·5 lbs. per sq. in. absolute (Bairstow and Alexander).

speed of travel of the flame, fig. 261 is taken from a research by Messrs Bairstow and Alexander. It shows on a time base the rise of pressure to a maximum in the explosions of a series of mixtures of coal gas and air in which the proportion of coal gas (by volume) had the values indicated. In the strongest mixture 14·98 per cent. of the volume was coal gas, in the weakest 7·79 per cent. The initial pressure was 34·5 lbs. per square inch (absolute) throughout the series. The explosions took place in a vessel 18 inches long and 10 inches in diameter. With the strongest mixture the time taken to reach maximum pressure was 0·04 second, with the weakest it was 0·8 second and the maximum pressure was of course much lower. Other parts of the same research illustrate the point that given a mixture of one strength the time of explosion is shortened by initial compression. The

rise of pressure on explosion goes on much faster in a highly compressed mixture[1].

Any *dissociation* to which the products of combustion are liable at high temperatures may be regarded as contributing to the increase of specific heat, for during the heating of the gas it implies that part of the heat is being expended in maintaining the dissociated molecules in a state of separation, which is restored by their entering into chemical union as the gas cools. The effect is indistinguishable from an augmentation of specific heat.

333. Indicator Diagrams of the actual engine. An indicator diagram from an early example of the Otto engine working in its normal manner is given in fig. 262, where AB is the
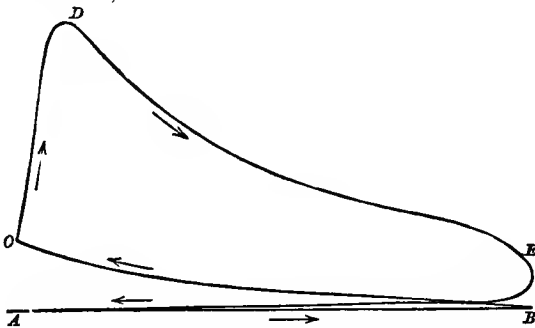
FIG. 262.   Indicator diagram from an Otto Gas-engine.

admission stroke, BC the compression, CDE the working stroke, and EA the exhaust. There is a rapid rise of pressure on explosion, so rapid that the volume has not very materially altered when the maximum of pressure is reached; and the specific heat at constant volume may therefore be used without serious error in calculating the amount of heat which this rise accounts for. By the time the maximum pressure is reached, at D, combustion is nearly complete and some heat has been lost to the walls by radiation and conduction. In the expansion curve, from D to E there is a continued loss of heat to the walls, along with (in some cases) a slight further generation of heat within the gas. The high specific heat during expansion, resulting from the fact that the temperature is then ranging from say 2000° C. to 1000° C. makes the index γ for

[1] *Proc. Roy. Soc.* 1905.   See also Mr D. Clerk's book.

adiabatic expansion comparatively low, and so tends to keep the curve up notwithstanding the losses of heat.

With a very weak mixture (fig. 263) the spread of the ignition is so slow that the mixture is burning during nearly the whole of the working stroke, with the effect (in this example) that the pressure is kept nearly constant up to release.   In such abnormal cases the gases discharged to the exhaust may contain some unburnt fuel.

In the engine which gave the indicator diagram of fig. 262 the compression was much smaller than is now usual: the volume of the clearance was about 40 per cent. of the volume swept through
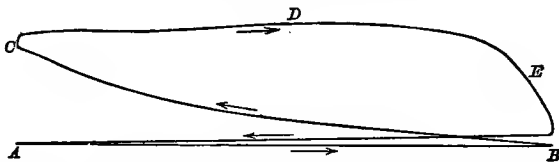
FIG. 263.   Otto Engine diagram with weak explosive mixture.

by the piston, and hence the compression ratio was 3·5.   Compression ratios of 6 or more are common in modern engines, with the result that the mixture is compressed to a pressure between 150 and 200 lbs. per square inch before explosion.   This makes the combustion very rapid and raises the maximum pressure to values approaching 500 lbs. per square inch.   Two typical indicator diagrams from one of Hopkinson's papers [1] are given in fig. 264 for a 40 horse-power Crossley engine, using coal gas in charges of two grades of richness.   In one case the mean effective pressure is 102 lbs., in the other 88½ lbs.   The thermal efficiency in this engine is about 0·35.

334.   Examples of engines.   Fig. 265, which is taken from one of Mr Clerk's papers, shows in section a typical gas-engine of moderate power and of the usual single-acting pattern—a 68 B.H.P. engine of the National Gas-engine Company.   It illustrates the common features: the trunk piston; the water-jacket surrounding the whole of the cylinder barrel as well as the combustion chamber, through which a constant circulation of cooling water is maintained by the pipes which appear above and below the

[1] *Proc. Inst. Mech. Eng.* April 1908.
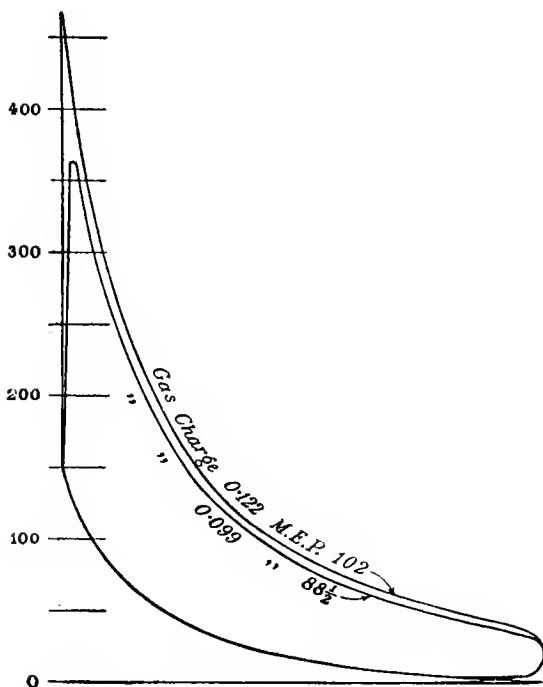
Fig. 264.   Indicator Diagrams from a Crossley Otto Engine, 1908 (Hopkinson).
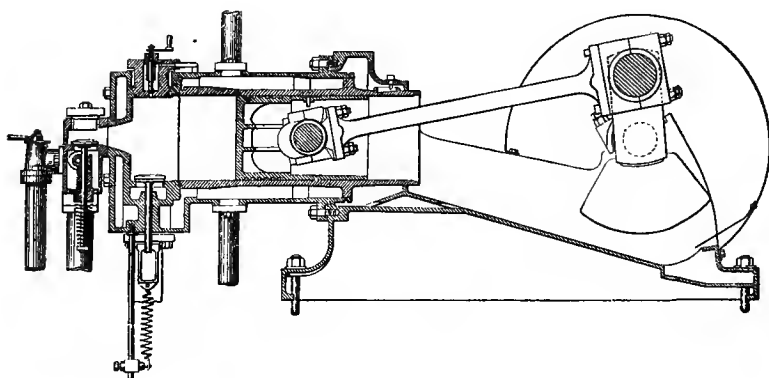
Fig. 265.   "National" Gas-engine.

cylinder in the figure; the mushroom valves opening inwards for admission and exhaust, which are worked by levers from cams on the half-speed side-shaft.

Fig. 266 illustrates a different type, showing the cylinder of a "Premier" gas-engine of 250 horse-power which differs from the ordinary construction of a four-cycle engine in having a scavenging device. This is obtained by giving the piston an enlarged front end A and making it act as a pump. During the exhaust stroke air is compressed, in consequence of the difference in area between the piston proper B and the pump extension A, into an air space C (surrounding the water-jacket) which it fills at a pressure
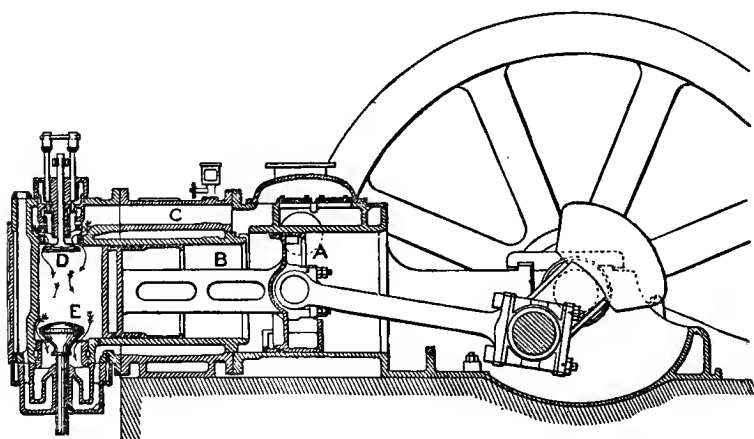
Fig. 266.   "Premier" Gas-engine with scavenging device.

slightly above that of the atmosphere. Consequently, when the admission valve D is opened, connecting this space with the combustion chamber, the residual products are swept out through the exhaust valve E by an inrush of fresh air before the working charge is drawn in, the exhaust valve being kept open for a short interval after the admission valve opens.

In some of the large gas-engines of the National Company the front end of the cylinder is enclosed to form an air-pump which is used to add air after the charge has been drawn in, with the effect of increasing the pressure to some 7 lbs. on the square inch at the beginning of the compression stroke. During the working stroke the same pump serves to compress air for scavenging

purposes. This device, which is due to Clerk, of adding air under pressure to the charge inhaled by the forward movement of the piston is sometimes called "super-compression." It increases the weight of the working charge by some 40 per cent. in a cylinder of given size, and augments the mean effective pressure without producing an inconveniently high maximum temperature.

In gas-engines of moderate size (up to 100 horse-power or so), the water-jacket on the barrel and end of the cylinder suffices to keep the working surface cool enough for lubrication, and the whole interior cool enough to prevent the risk of premature ignition. The piston, which receives much heat from the working substance through its face, is cooled by radial conduction in its own substance to the side walls, and the valves by radial conduction to their seats. But in large engines this cooling is insufficient, and it becomes necessary to maintain a circulation of cooling water in the interior of the piston itself and also in the interior of the exhaust valve. The Premier engine of fig. 266, for example, has its piston and exhaust valve water-cooled.

In some engines water injection into the working space is used as a supplementary means of keeping the surface sufficiently cool. This allows the compression to be carried further than would otherwise be practicable, without risk of premature ignition.

335. Large Gas-engines of the Otto Cycle type. In large gas-engines the cylinders are generally made double-acting so that with the Otto cycle a working stroke is obtained in each cylinder once in each revolution. Very commonly two cylinders are set in tandem on the same piston-rod, giving one impulse, in one or other of the cylinders, in each stroke; and in the largest engines two such pairs of tandem cylinders are often coupled, making four cylinders in all working on two cranks.

Engines of this kind are used in many iron and steel works to operate blowing engines and to develop power on a large scale for other purposes, the gaseous fuel being taken either from blast furnaces or coke ovens or from a producing plant erected for the purpose, in which gas is cheaply obtained from the distillation and partial combustion of coal, with recovery of the valuable ammoniacal products. With a separate plant for the production of cheap gas, large gas-engines form a means of obtaining power applicable to any sort of power station, and the cost of the power

so obtained compares favourably with the cost that is incurred in generating power by means of steam.

Gas-engines are built in some instances to develop as much as 1000 horse-power per cylinder with double-action, using the Otto cycle, and in exceptional cases this figure is exceeded. Thus in one of Messrs Thyssen's engines a pair of tandem cylinders four feet in diameter, with a stroke of $55\frac{1}{2}$ inches, running at 94 revolutions per minute, develops 2600 horse-power. By coupling twin pairs of tandem cylinders it is not unusual to obtain about 4000 horse-power from a single unit. In many cases, especially on the continent and in America, a number of such units are combined in one installation. Thus in the power-house of the Indiana Steel Company, at Gary, Ind. there are 17 Allis-Chalmers four-cycle gas-engines each generating 2500 kilowatts, or 42,500 kilowatts in all, equivalent to nearly 60,000 horse-power[1].

These engines and many other large examples of the four-cycle double-acting type, follow the general features of what is called the Nürnberg design, originally developed by the Augsburg and Nürnberg Company. Each crank is operated by a pair of tandem horizontal cylinders: the cylinders are alike and on each the admission valves are on the top of the barrel and the exhaust valves on the bottom of the barrel, leaving the cylinder ends free to take the form of simple hollow castings which can be readily removed to allow the pistons to be drawn for examination or repair. In general external appearance the valves somewhat resemble those of a Sulzer engine. They are worked by eccentrics on the half-speed side-shaft. There are two admission valves for each end of each cylinder, the first admitting gas to a space above the second valve, where it mixes with air, and the second valve admitting the mixture to the combustion space. The valves are of the mushroom type, opening inwards and closed by springs. The exhaust valves are cooled by internal water circulation. Governing is effected by tripping the gas inlet valve, thereby altering the quality of the mixture, or by tripping the second admission valve, thereby altering the quantity of mixture admitted per stroke, or in some cases by a combination of both methods. The governor may also be arranged to control the timing of the ignition so as to suit the varying strength of mixture.

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[1] See papers on "Large Gas Engines," by P. R. Allen, *Cassier's Magazine*, July, August and September, 1909.

The Westinghouse Company has introduced a vertical type of four-cycle gas-engine, with trunk piston and single-acting cylinders. Two of them are set tandem over each crank, and in some cases four cranks making right angles with one another are set side by side, the engine developing about 1000 horse-power. There is one admission and one exhaust valve for each cylinder, worked from a half-speed shaft which is enclosed in the case forming the engine frame. Governing is effected by throttling both the air and the gas.

336. Large Gas-engines of the Clerk Cycle type. Among large gas-engines mention must be made of two interesting designs which work on the Clerk or two-stroke cycle, and have achieved a large measure of success in practice. One of these is the Körting engine, the manufacture of which has been taken up in England by Messrs Mather and Platt, and in America by the De La Vergne Company. It has a long double-acting cylinder (fig. 267), with more or less conical ends as in the original Clerk
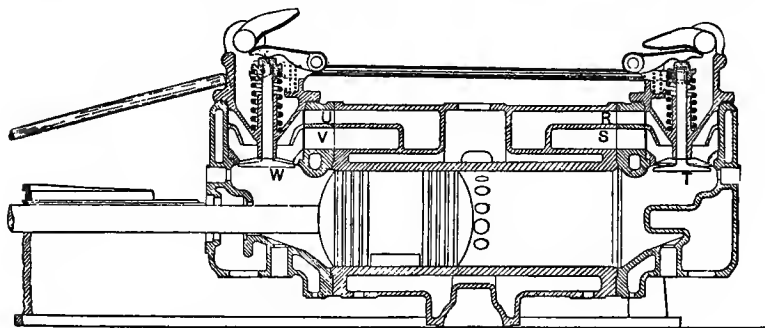
Fig. 267. Körting Engine (Mather and Platt).

design. The piston also is long, and at its extreme position to either side it just uncovers a ring of centrally situated exhaust ports, through which the burnt products escape after expansion. Alongside of the working cylinder are separate pumps for gas and air (fig. 268), the pistons of which are worked from a crank about 90° in advance of the main crank. There are three pumps in line, with a single piston-rod common to all three: the central one A is double-acting and supplies gas; the outer ones B and C are single-acting and supply air to the respective ends of the

cylinder. Between the gas and air-pumps are valve-chambers from which ducts lead to the spaces U, V, R and S above the admission valves of the working cylinder, gas coming through U and R and air through V and S. The supply from the pumps is so arranged that when either admission valve W or T opens the first thing to enter is air, which blows out the burnt products
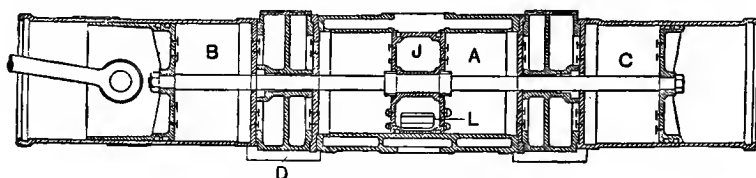
Fɪo. 268.   Gas and Air-pumps of Körting Engine.

through the open exhaust ports : gas follows only after scavenging has taken place, and by that time the exhaust ports are closing as the piston begins its compression stroke. The admission valves are operated from an eccentric on the main shaft by means of rocking cams in a manner which will be clear from the sketch, where the valve T has just begun to open.

The piston J of the gas-pump is hollow and the gas supply enters through a suction port L in the wall of the gas-pump cylinder. During the first part of the stroke, until the port is covered by the piston, no compression of the gas takes place, and consequently there is no delivery of gas to the main cylinder. During this time air only is entering the main cylinder, to effect the scavenging. In the later part of the gas-pump stroke gas is delivered, along with air from the air-pump, to constitute the charge. During the suction stroke of the gas-pump, gas can enter the suction side through the hollow piston and through automatic valves upon its face. The pressure at which gas and air are delivered is about 3 lbs. per square inch. The gas-pump delivers no gas for about 40 per cent. of its stroke while scavenging takes place. An indicator diagram from one end of the main cylinder is shown in fig. 269. It will be seen that release occurs before the end of the working stroke, through the uncovering of the central exhaust ports. The power required to operate the pump is not large, and it is stated that the mechanical efficiency of the engine reaches 84 per cent. even in moderately small sizes, and that the expenditure of coal per brake horse-power is about

0·75 lb. per hour, using bituminous slack in the producer. Engines of this type are extensively used especially as blowing engines in continental works, and they have been made in sizes
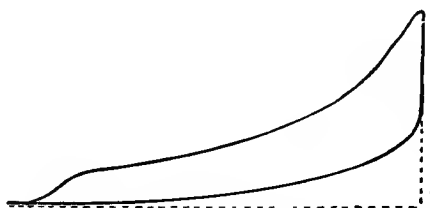
FIG. 269.   Indicator diagram of Körting Engine (Spring $\frac{1}{200}$).

capable of developing as much as 2000 horse-power in a single cylinder.   The twin-tandem arrangement may be applied, as in the four-cycle type, to multiply the power given by a single cylinder when larger units are wanted.

Another important modern example of the two-cycle action is found in the Oechelhäuser engine which is built by several continental makers, and in England by Messrs Beardmore.   In this case also the exhaust takes place through the uncovering by the piston of ports in the cylinder.   The cylinder is a long tube, open at both ends, in which there are two working trunk pistons which move in opposite directions, one being connected directly to a central crank on the shaft, and the other connected to two side cranks set at 180° from the first, through two side rods and a cross-head.   The pistons alternately approach one another, compressing the charge between them, and recede towards the ends during the working stroke, one of them finally uncovering a ring of ports, through which the burnt products escape.   At the same time the other piston in approaching the end of its stroke uncovers two rings of ports, through one of which air enters, first to perform the scavenging operation and then to supply a fresh charge along with gas which enters through the other ring of ports.

The air and gas come at a low pressure from receivers which are supplied by means of a pump worked by a tail rod from the back cross-head or by a rocking lever.   The air inlet ports are first uncovered, and the gas inlet ports after.   A conspicuous and valuable feature in the design is that valves as well as piston-rod glands are dispensed with.

36—2

337. Gaseous Fuels. A brief account may be useful of the kinds of gaseous fuel which are available for use in gas-engines.

Ordinary *coal gas* as manufactured for public supply, chiefly with a view to its use as an illuminant, is used in many small gas engines where the convenience of the supply compensates for its comparative high cost as an engine fuel. It is obtained by distilling the volatile constituents of coal in retorts and purifying the gases that are given off by extracting tar, ammonia, etc. A typical analysis by Bunsen and Roscoe, quoted by Clerk, is as follows:

Constituent	Parts by volume	Volumes of oxygen required for combustion	Volumes of resulting products in gaseous state
Hydrogen, H.....................	45·58	22·79	45·58
Methane, CH_4	34·9	69·8	104·7
Carbon monoxide, CO	6·64	3·32	6·64
Ethylene, C_2H_4...................	4·08	12·24	16·32
Tetrylene, C_4H_8.................	2·38	14·28	19·04
Sulphuretted Hydrogen, H_2S	0·29	0·43	0·58
Nitrogen, N	2·46	—	2·46
Carbon dioxide, CO_2	3·67	—	3·67
	100·00	122·86	198·99

The calorific value of average coal gas per cubic foot is about 333 pound-degree-centigrade units. Each cubic foot of gas requires rather more than 6 cubic feet of air for complete combustion, and when used in a gas-engine the gas is mixed with from 8 to 11 times its volume of air. Under favourable conditions a gas-engine will use 15 to 16 cubic feet of coal gas per hour per brake horse-power.

Natural gas issues from coal and oil-bearing strata in Pennsylvania, Baku, and other places. It consists largely of methane, sometimes to the extent of 93 per cent. by volume, along with a little hydrogen, ethane, and inert constituents.

Coke-oven gas, which is distilled from coal in the process of manufacturing coke, forms an engine fuel of comparatively high power, though short of that of the coal gas distilled in retorts. It contains about 63 per cent. of hydrogen, 23 of methane, and 5 of carbon monoxide, and has a calorific value of about 230 units[1].

[1] The unit used in these statements is the pound-degree-centigrade, which is 1·8 times the "British Thermal Unit" or pound-degree-Fahrenheit.

When this gas is used in an engine the compression pressure is not as a rule allowed greatly to exceed 100 lbs. per square inch, as the presence of dust along with a high percentage of hydrogen and hydrocarbons induces a greater danger of premature ignition than is found in using ordinary coal gas.

Water gas is made by passing steam through incandescent carbon. To maintain the carbon in a state of incandescence air is blown through it alternately with the steam, the alternation taking place at intervals of a few minutes. In many cases water gas is enriched for illuminating purposes by adding the gaseous products of decomposition of a certain amount of oil. Without enrichment the main constituents in water gas are 48 to 50 per cent. of hydrogen, and 36 to 40 per cent. of carbon monoxide, the remainder being inert. Its calorific value is about 160 or 170 units per cubic foot.

Producer gas is obtained, in its simplest form, by blowing air through coke or other carbon fuel, part of which is in a state of incandescence. The gas producer is essentially a device in which fuel is burnt with an insufficient supply of air. The combustion is therefore incomplete; an excess of carbon is taken up by the oxygen of the air, forming carbon monoxide, which is the effective constituent of the resulting fuel. Rather less than 30 per cent. of the whole heat of combustion of the carbon is generated in the producer, leaving fully 70 per cent. to be generated in the subsequent combustion of the gaseous fuel. In many producers coal is used instead of coke; consequently the gaseous fuel, while still consisting largely of carbon monoxide contains also some hydrogen and methane which add considerably to the calorific value. Producer gas made from coke has a calorific value of about 40 units per cubic foot, and that made from coal about 55 units.

An improved kind of producer gas was applied as a gas-engine fuel by Mr J. Emerson Dowson who designed a producer in which a certain proportion of steam was mixed with the air which passed through the incandescent coke or coal: a gas was thereby obtained which was intermediate in composition between water gas and the gas given by a simple air-fed producer. Many forms of producer are now used embodying this principle and yielding a mixed type of cheap gaseous fuel, very suitable for use in gas-engines, with carbon monoxide and hydrogen as its chief active constituents. The Mond producer is a large scale type, in which the process is

carried out with great economy owing partly to the scrubbing arrangements by which the ammonia in the gas is recovered, to form ammonium sulphate for fertilizing purposes, and to the very complete regenerative devices which are employed to save what would otherwise be waste heat. In generating Mond gas cheap bituminous slack serves for fuel and the proportion of steam is large, about $2\frac{1}{2}$ lbs. per lb. of fuel. Producers which work on a small scale generally use coke or anthracite, to escape the difficulty which arises when tar is one of the products.

Mond gas contains some 24 per cent. of hydrogen, 2 of methane, and 16 of carbon monoxide. Its calorific value is about 80 to 85 units per cubic foot. In other examples of mixed producer gas the proportion of hydrogen is as a rule lower, averaging something like 15 to 18 per cent., and the carbon monoxide is higher, about 20 to 25 per cent. The calorific value generally ranges from 80 to 90 units.

Blast furnace gas. A blast furnace acts like a gas producer of the air-fed type, yielding a gas which may contain some 28 per cent. of carbon monoxide, with more or less hydrogen and methane, according as the furnace burns coal or coke. The calorific value is in some cases only about 40 units, but may be higher. Notwithstanding its low calorific value blast furnace gas works well in large gas-engines. When properly freed from dust it may be subjected to high compression pressures without risk of premature ignition. In some large power installations the gas furnished by blast furnaces is more or less enriched by an auxiliary supply coming from coke ovens or special producers.

338. Suction Gas Plants. In most large gas producers the air is forced in under a low pressure. But for small or moderate powers it is very common to have a self-contained plant consisting of engine and producer combined, in which the engine sucks in air through the producer, thereby generating gas as it is wanted Plants of this kind are made in large numbers by various firms, in sizes ranging from about 5 to 200 horse-power. They furnish an exceedingly economical and compact means of obtaining power, requiring but little attention, and using only about 1 lb. of anthracite or $1\frac{1}{4}$ lbs. of coke per brake horse-power hour[1].

[1] See Report on Trials of Suction Gas Plants, *Journal of the Royal Agricultural Society*, Derby Meeting, 1906.

Fig. 270 illustrates a suction generator of this kind, by the National Gas Engine Company, along with the engine which it supplies. The gas generator consists of two main parts, the furnace or generator proper on the left, and the scrubber for cleansing the gas on the right. The scrubber is a cylindrical vessel forming a short tower filled with pieces of coke on top of which a stream of water is sprinkled. The gas percolates up through it, leaving by a pipe which leads to an expansion box from which the engine draws gas in each charging stroke. The effects of this suction are felt throughout the apparatus, and cause air to be drawn in between the fire-bars at the bottom of the furnace. On its way to the furnace the incoming air takes heat from the outgoing stream of gas between the furnace and the scrubber, and the heated air takes up a large quantity of moisture, evaporated in the annular space marked "vapouriser," through which the incoming air passes. The water supply which furnishes the moisture is itself also warmed by taking heat from the gases in the exit pipe. Combustion occurs in the lower part of the furnace only: fresh fuel (coke or anthracite) is added from time to time through a hopper at the top, arranged to form an air-lock. The foul water from the scrubber drains off through a water seal. When the apparatus is being started into action a fan is used to blow air through the furnace for a short time until combustible gas begins to be given off.

Besides being very extensively used on land gas-engines with suction plants have found a limited amount of employment at sea as a means of propelling trawlers and other small power craft.

339. Engines using liquid fuel. Little need be said here separately of internal-combustion engines using liquid fuel, for in nearly all cases the liquid is vaporized before admission to the combustion chamber, and the action proceeds precisely as in an engine using gaseous fuel. As regards the vaporization of the liquid a broad distinction may be drawn between cases where the liquid is so volatile that no special means have to be taken to heat it in order to cause evaporation, and other cases where a comparatively high temperature is required. Thus petrol (gasoline), which is a light spirit obtained from crude petroleum in the early stages of distillation, evaporates so readily that air which has passed through it, or into which a little of the liquid petrol has been
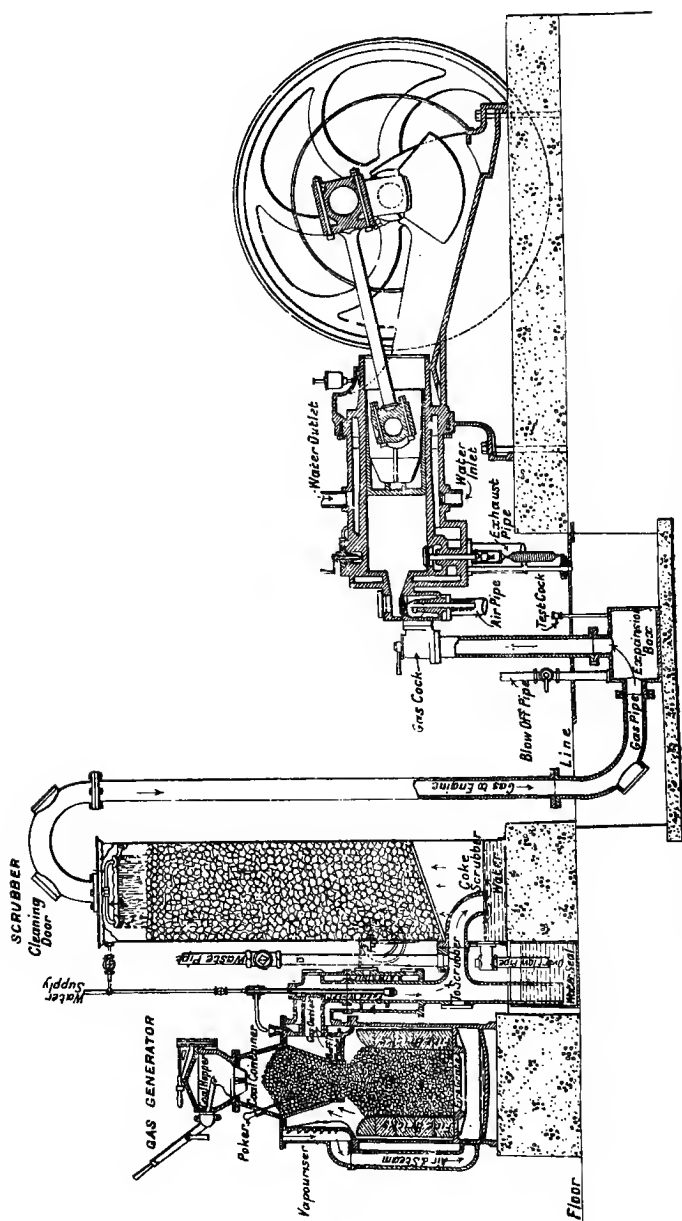
FIG. 270. Sectional Elevation of National Gas-Engine Suction Plant.

sprayed, will form an explosive mixture, and a simple spraying arrangement called a "carburettor" suffices as a means of introducing the fuel. On the other hand, paraffin oil (kerosene) with its much higher flash point requires a "vaporizer" which is kept at a fairly high temperature.

Petrol is familiar as the ordinary fuel of motor carriages and motor boats; and petrol driven engines may be said to have made mechanical flight possible. It consists largely of pentane, C_5H_{12}, and hexane, C_6H_{14}, the first and second liquid members of the paraffin group of hydrocarbons. It is so volatile even at low atmospheric temperatures as to require great care in use and storage. To form an explosive mixture with air at atmospheric pressure requires only about 2 per cent. of petrol vapour, and a smaller proportion still is used to constitute the charge which explodes under pressure in an engine. The calorific value of petrol per lb. is about 11000 lb.-degree (centigrade) units.

340. Petrol Engines. As a rule engines using petrol work on the Otto cycle and are single acting. The common arrangement in cars, motor boats, etc. is to have a group of vertical cylinders, often four and sometimes six, standing side by side above the crank shaft, each with a trunk piston. The most primitive method of producing the explosive mixture is to cause the air supply, on its way to the engine, to bubble through liquid petrol in a suitable vessel: but this is open to the objection that the charge undergoes a progressive change in quality, for the more volatile constituents of the composite liquid are the first to pass off. To secure a more uniform quality of charge many forms of carburettor have been devised. A usual type is one in which the liquid is aspirated from the open end of a pipe standing in the air inlet pipe; this is connected with a vessel which maintains the liquid in the aspirating pipe at a constant level by means of a float which operates an admission valve between this vessel and the supply tank. Thus each gust of air that is being sucked into the cylinder draws with it a definite quantity of petrol spray which is immediately dissolved as vapour in the air. The proportion by volume of petrol vapour in the charge is about 1 in 70. The temperature produced in compression tends to produce chemical changes in the mixture; these may lead to premature ignition unless the compression pressure is kept low, and may also

hinder complete combustion, with the result that the efficiency is reduced.   As an alternative fuel alcohol has the advantage not only of greater safety but of greater stability in the state of vapour, so that though its thermal value is much less (about 6000 units) a better proportion of that is realized in the output of the engine.

341. Oil-Engines. In engines which use paraffin oil (kerosene) the vaporizer is generally a tube or vessel kept hot by an external flame into which oil is injected or aspirated as it is wanted to form the charge.   An important exception is the Hornsby-Akroyd oil engine, in which the vaporizer is a vessel forming an extension of the clearance space at the end of the cylinder.   This is warmed to begin with by an external flame, which is extinguished when the engine is running.   The interior of the vaporizer holds so much heat, after each explosion, that it not only vaporizes the next charge of oil but ignites it when in the backward movement of the piston the mixed gases are fully compressed, and hence no electric or other igniting appliance is required.   It is found that by properly adjusting the degree of compression the charge will ignite at the right time.   During the forward (charging) stroke of the piston oil is delivered by a feed-pump to the hot chamber and its vapour mixes with the inert gases present there as residue from the previous explosion, while air is drawn in behind the piston.   During the compression stroke some of this air is compressed into the hot chamber : at the same time the temperature is raised by compression, and as the piston passes the dead-point a part of the charge containing vapour and air in proportions which make explosion possible is raised to ignition point.

The regulation of speed is effected by making the governor open a by-pass on the oil feed-pump, through which part of the oil returns to the oil-tank when the speed becomes too high.

342. Diesel Oil-Engine. An interesting type of internal-combustion oil-engine was introduced by R. Diesel about 1895 and has found considerable application.   In the Diesel engine a cycle resembling that of Otto is followed, but with this important difference that air only is compressed during the compression stroke, and then the oil is pumped into the cylinder in the early

stages of the working stroke. The compression is carried so far that the air is heated by it to a temperature such that the oil burns as it enters. The result is that during the first part of the working stroke, while the oil is entering the cylinder, the pressure is kept nearly constant by the heat which is then being generated. Hence the action approximates to one in which the heat is taken in at constant pressure instead of at constant volume as in the ordinary oil- or gas-engine. The compression ratio is very high, which makes for efficiency; and all risk of premature ignition is of course avoided, since the fuel only begins to enter the cylinder when it is desired that combustion should occur.

The air in the cylinder is compressed to about 500 lbs. per square inch, and a small quantity of air is separately compressed to 650 lbs. or so in order that it may serve as a means of injecting the oil. By the help of this auxiliary high-pressure air the oil required for the charge is sprayed in as the piston passes the dead-point. The oil used is generally paraffin. As the temperature of the air in the cylinder has been raised by compression to 600° C. or over, the oil immediately ignites, and the combustion goes on for a short portion of the working stroke, so long as oil continues to enter. During this time the pressure remains nearly constant, the increase in volume due to the movement of the piston being compensated by an increase in temperature due to the combustion. During the remainder of the stroke there is simple expansion with falling pressure, followed at the end by release, with discharge of the burnt product during the return stroke just as in the Otto cycle. In the next forward stroke air for a fresh charge is drawn in, and in the next return stroke it is compressed.

If it were practicable to continue the expansion in the working stroke until atmospheric pressure were reached, by having the expansion stroke much longer than the compression stroke, we should have a constant pressure action as regards the discharge of heat as well as the reception of heat. In that case the ideal indicator diagram for the action would be that shown by the full lines of fig. 271, namely $ABCDEA$. With equal expansion and compression strokes the ideal diagram is $ABCDQBA$. The area QEB is lost through not rejecting the heat at constant pressure. To save it we should have to complicate the mechanism by arranging for strokes AE and AB of very different lengths, and the mean effective pressure in the working stroke would be much reduced,

which would involve a large addition to the weight of the engine for a given output of power.

Taking however the complete constant pressure cycle *ABCDEA*—that is to say the cycle in which heat is rejected as well as received at constant pressure—it is interesting to
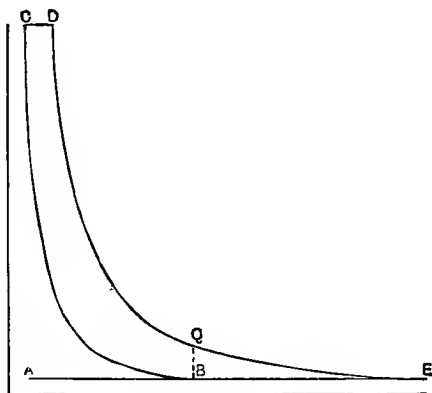
Fig. 271.

express the ideal efficiency in that cycle of an "air-standard," as was done in § 329 for a constant volume cycle: the assumption, as before, being that there is no transfer of heat between gas and metal, and that a constant value may be assigned for the specific heat. Calling the atmospheric temperature τ_0, the temperature after compression τ_c, after combustion τ_d, and after expansion τ_e, the heat taken in is $K_p(\tau_d - \tau_c)$ and the heat rejected is $K_p(\tau_c - \tau_0)$. The ratio of expansion $\dfrac{V_e}{V_d}$ is the same as the ratio of compression $\dfrac{V_b}{V_c}$, since $\left(\dfrac{V_b}{V_c}\right)^\gamma = \dfrac{P_c}{P_b}$ and $\left(\dfrac{V_e}{V_d}\right)^\gamma = \dfrac{P_d}{P_e}$, which is the same as $\dfrac{P_c}{P_b}$. Hence we find $\dfrac{\tau_e - \tau_0}{\tau_d - \tau_c} = \left(\dfrac{1}{r}\right)^{\gamma-1}$, where r is the ratio of compression, and the expression for the efficiency in this ideal cycle takes the same form as in the ideal constant volume cycle, namely

$$1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

In the ideal Diesel engine with equal expansion and compression strokes the air-standard efficiency is less, being

$$1 - \frac{K_v (\tau_q - \tau_o)}{K_p (\tau_d - \tau_c)}.$$

But it will be found, on working out numerical values, that with high ratios of compression such as are found in Diesel engines this is greater than the air-standard efficiency in the constant volume cycles of gas-engines with such ratios of compression as are permissible in them.

Tests of Diesel engines show that, in fact, a very high efficiency is reached. In trials by Mr M. Longridge of a 500 horse-power Diesel engine, quoted by Mr Clerk[1], the indicated horse-power was 595 after allowing for the negative work done in the pump by which the oil fuel was forced in. The consumption of oil was 207·2 lbs. per hour, or at the rate of barely 0·35 lb. per I.H.P.-hour. Taking the oil as having a calorific value (on the lower scale[2]) of 10000 centigrade units, these figures correspond to a thermal efficiency of 0·41, as between the energy of the fuel and the indicated power—a fraction which is about twice as high as is found in any form of steam-engine, and distinctly higher than the highest efficiency in ordinary gas-engines or other internal-combustion engines, where the compression is less.

343. Efficiencies of Internal-Combustion Engines. In judging of the performance of internal-combustion engines the brake power is for most purposes a more valuable criterion than the indicated power. This is not only because the idle strokes of the Otto cycle make a considerable deduction from the gross effect, but because there is, as a rule, much less certainty about measurements of the indicated power. In early gas-engine trials the indicators used were ill-adapted to give accurate readings in dealing with the sudden change of pressure due to explosion. Now, through the development of the optical indicator this difficulty is overcome, and an accurate record is obtainable for the action in any individual stroke. But a fundamental source of uncertainty remains, that the action in the cylinder is generally

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[1] D. Clerk, *The Gas, Petrol, and Oil Engine*, 1909, p. 320.

[2] That is, not including the latent heat of the water vapour which is formed in the combustion.

liable to wide variations from stroke to stroke, and it is difficult to ensure that the measurements of indicated power really represent a fair average. In some cases, as Professor Hopkinson has shown[1], the regularity is remarkable, but whenever governing is effected by altering the quality of the charge wide differences are bound to occur. The figures quoted here relate for the most part to trials where direct measurements were made of the brake horse-power.

Early Otto engines, using gas for fuel, gave on the brake a return of about 14 per cent. of the energy supplied in the fuel. With increased ratios of compression the efficiency has increased until the return is now generally 30 per cent. on the brake, and in exceptional cases as much as 32 per cent.   In a series of tests carried out in 1905 by a committee of the Institution of Civil Engineers a "National" engine developing about 50 horse-power was found to have a brake thermal efficiency of 0·30 and an indicated thermal efficiency of 0·35.   In tests of a Crossley engine of 40 horse-power with higher compression, by Professor Hopkinson in 1908, the brake thermal efficiency was 0·32 and the indicated thermal efficiency was nearly 0·37.   These figures correspond to mechanical efficiencies of about 0·86 for the full load.   At lower loads the mechanical efficiency is reduced, for the difference between the indicated and brake power remains nearly constant. It is in fact most conveniently determined by measuring the indicated power when the engine is running without load.

Comparing these figures with the "air-standard" it will be found that the indicated power in the best cases represents about 0·7 of that ideal.   Thus in Hopkinson's tests, when the ratio of compression was 6·37 the air-standard efficiency would be 0·52 against the 0·37 obtained with the indicator.   The difference, as has been pointed out in § 329, arises mainly from two causes— that the specific heat is not constant, and that the actions do not proceed in an adiabatic manner.

An analysis of the Institution of Civil Engineers' tests shows that, in round numbers, 40 per cent. of the energy of the fuel went off in heat in the exhaust, 25 per cent. in the jacket water and by radiation from the engine, and the remaining 35 per cent. was converted into (indicated) work.

In the trials of a Diesel oil-engine already referred to, it is

[1] B. Hopkinson, *Inst. Mech. Eng.* 1908.

calculated by Mr Clerk[1] that if the pumps supplying oil and air at high pressure had been driven by the engine itself, the mechanical efficiency should be reckoned as 0·77. Hence the thermal efficiency, which was 0·41 when estimated on the indicated power, becomes only 0·32 when estimated on the brake power. In other Diesel engine trials the indicated thermal efficiency was 0·40 and the brake thermal efficiency 0·31. These results show that while the Diesel engine gives a higher return in indicated thermal efficiency than any other existing form of heat-engine, the comparatively large mechanical losses involved in such high compression bring the net return, in brake horse-power, down to a value that is no higher than is given by a gas-engine working on the Otto cycle.

344. The Humphrey Internal-Combustion Pump. A new departure of great interest is the invention, by Mr H. A. Humphrey[2], of an internal-combustion pump of high efficiency in which nearly all the usual working parts of a pumping engine are dispensed with. Explosion takes place in a chamber which is closed by a column of the water or other fluid to be pumped: the effect is to set the column oscillating with the result that a part of it is discharged at a high level, or against a pressure, and the successive movements of the column complete the cycle of operations by first discharging the burnt gases, then inhaling a fresh charge, and then compressing it in preparation for the next explosion. What is substantially the Otto cycle (or, in a modified form of the apparatus, the Clerk cycle) is accomplished in an extraordinarily simple manner without the use of a piston, crank, or fly-wheel, and the effective return of work done in relation to the quantity of fuel burnt compares favourably with the best results obtainable in other ways. This is not only because of the absence of frictional mechanical losses, but because expansion in the working "stroke" of the column is carried so far as to bring the pressure down to a value approximately that of the atmosphere.

Fig. 272 shows one form of the apparatus designed to pump water from a low level source E to an outlet F some 30 feet higher. The two are connected by a long discharge pipe D, the water in which forms the oscillating column. This pipe terminates, on the

[1] D. Clerk, *The Gas, Petrol, and Oil Engine*, 1909, p. 320.

[2] *Proc. Inst. Mech. Eng.* December 1909.

left, in the working chamber A, which has an inlet valve B for the admission of the explosive charge, and an exhaust valve C, both opening inwards. The exhaust valve C is at the foot of a short pipe which projects some way into the chamber. It falls open by its own weight when a pawl H is released, which takes place when the valve G opens to admit water from the supply tank into the oscillating column. Imagine a charge in A to be already compressed and to be fired by the sparking plug K. Water is driven
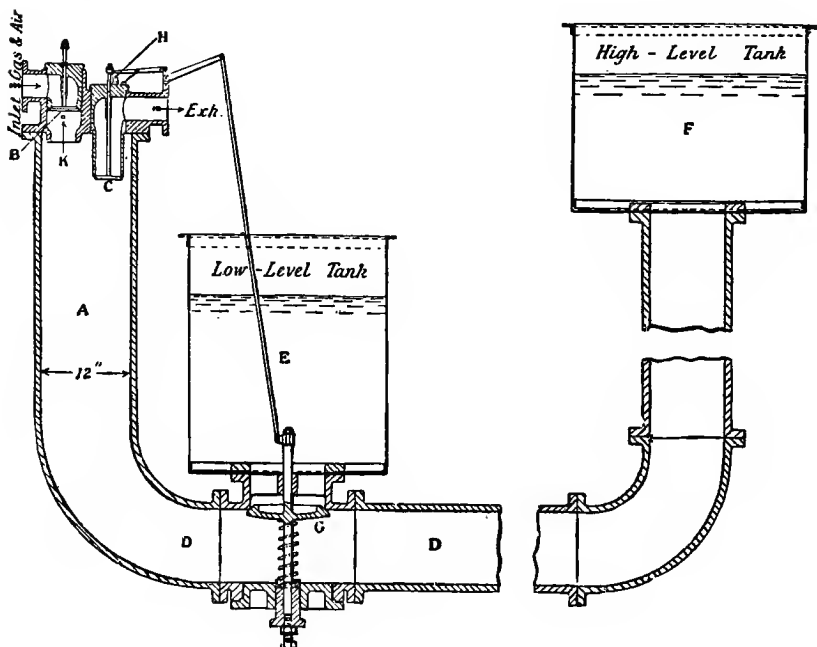
Fig. 272.

down in A and some is discharged at F. The momentum of the moving column carries the level in A down so far that the pressure of the expanding gases falls to about that of the atmosphere; the valve G opens and water is taken in from E. This also opens the exhaust valve C, which remains open until the column in the return movement hits the valve C and closes it, at the same time locking it in place by means of the pawl. The residue of burnt products left at the top of the combustion chamber becomes compressed until the backward motion of the column stops and it again begins to move down in A. During this downward movement the

inlet valve opens and a fresh charge is drawn in. In the next backward movement this is compressed and it is fired, just as the pressure passes its maximum, by means of a firing device operated by changes of pressure. This completes the cycle, which is the cycle of Otto with this advantageous modification, that the expansion stroke is longer than the inhaling and compression strokes. The column of water is maintained in a state of forced oscillation in which the alternate movements, to the same side, differ in range. A scavenging effect is obtained by letting the exhaust valve open while the pressure within the chamber is still falling, so that air is drawn in. Thus when the column returns, discharging the burnt gases, it leaves a mixture of air and burnt gases in the clearance space. By having a valve separate from the exhaust valve and at a higher level to admit air for scavenging, the residue in the clearance space is made to consist almost wholly of air.

To start the pump from rest a charge of compressed air is introduced until the water is lowered somewhat beyond the usual charge level. The exhaust valve is then forced open : the water consequently rises with momentum enough to close the exhaust valve, compress the residue in the clearance space, fall again drawing in the charge, and then rise compressing it, after which ignition takes place.

Reference should be made to Mr Humphrey's paper for an account of other forms of the apparatus, and of its possible uses, and also for particulars of the manner in which the three valves, for admission, exhaust and scavenging, are inter-connected so that their action takes place automatically without even the need of a connection to the water-admission valve such as is shown in fig. 272. A form of the Humphrey pump operating on what is in effect the Clerk cycle is shown in fig. 273. There the combustion chamber A ends in a pipe with two branches D and E, one or other of which is closed by a rocking valve F. The branch D leads to an auxiliary supply tank J, and the branch E to the rising main into which water can be drawn from the low-level supply tank H for discharge at the high level K. Starting from the condition sketched, with a compressed charge at the top of A and the valve F in the position which closes D, the charge is ignited, giving an impulse to the main water column AEK. When the expansion has caused the pressure to become atmospheric the exhaust valve opens and the higher pressure in D throws the

E.   37
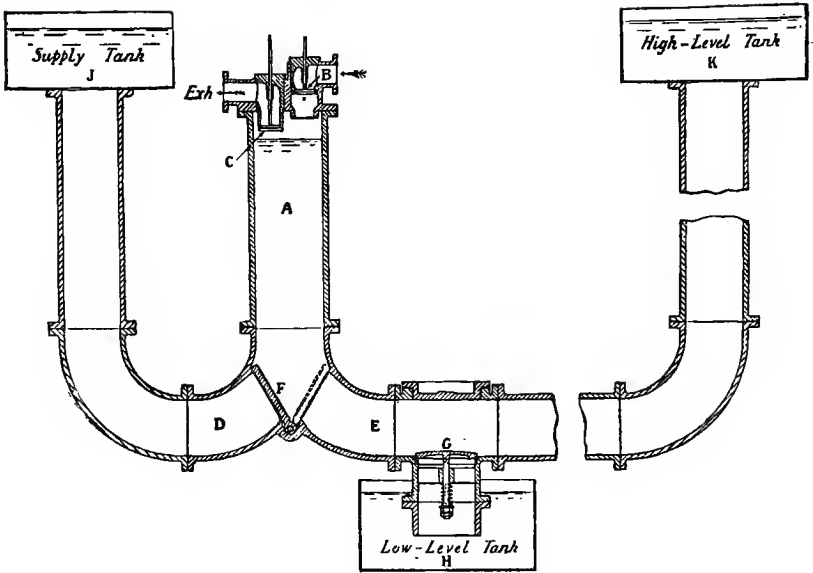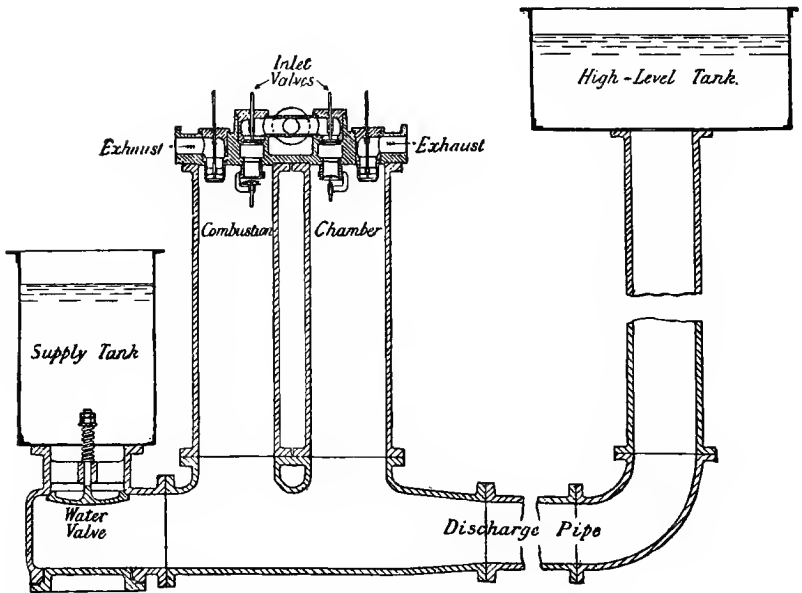
Fig. 273.

Fig. 274.

valve F over to the position shown by dotted lines. Water then comes down from J to discharge the burnt gases, and at the same time the main water column in E completes its movement towards the high level discharge, taking in more water from the low-level supply tank H through the self-acting valve G. The auxiliary water column JDA closes the exhaust valve C and compresses the residual gas into the cushion space. This stops and reverses its movement, and as it oscillates towards J a fresh combustible charge is drawn in. By the time this has happened, and the auxiliary column is beginning to return compressing the charge, the main water column in E returns, throwing F over to the initial position and completing the compression of the charge, after which the cycle repeats itself. A dashpot connected to the valve F makes it move over somewhat gradually, and by regulating this control it may be arranged that the backward movement of the main column keeps up the water supply of the auxiliary tank J, so that the net result of the whole action is simply a transfer of water from H to K. In this form of pump there is an explosion for every complete oscillation of the main column, instead of one for every two complete oscillations as in the four-cycle pump of fig. 272. The auxiliary column in D performs what is substantially the function of the displacing pump in the Clerk cycle. Mr Humphrey has also designed a form of pump which gives an effective impulse for each oscillation of the column, by having a pair of similar combustion chambers which act alternately on the same discharge pipe (fig. 274). In trials by Professor Unwin a pump of this kind delivered water through a lift of about 33 feet at the rate of 1621 gallons per minute, with an expenditure of gas equivalent to the consumption in a producer of 1·06 lbs. of anthracite coal per pump-horse-power-hour. The rate of working was 16·15 pump-horse-power, the compression pressure of the explosive charge was 44·7 lbs. per square inch, and 13·8 cycles were made per minute. Further trials showed that the pump adapted itself to work through lower lifts without any alteration and with no considerable reduction in efficiency.

APPENDIX.

PROPERTIES OF STEAM.

THE modern steam tables, which are adopted here, are founded on the work of Callendar[1], whose first paper on the subject was published in 1900. He assumed a characteristic equation connecting pressure, volume and temperature, applicable to water vapour generally whether saturated or superheated. This equation involves certain constants, and he adjusted these to accord with well-established results of experiment. He proceeded to show that it was practicable to deduce from the characteristic equation, in the form assumed by him, by aid of relations depending only on the general principles of thermodynamics, expressions for all the important properties of steam, from which numerical values could be deduced within the range to which the characteristic equation might be held applicable, namely for pressures such as correspond to saturation temperatures extending from 0° to 200° C. or so. Within this range the values so deduced are found to agree remarkably well with the results of such direct measurements as have been made, especially with those in which there are good grounds for believing the experiments to be accurate. At the time when Callendar devised his equation there was comparatively little material for such a comparison; but now, thanks especially to measurements of the volume and specific heat of steam carried out for the most part in the laboratory of technical physics at Munich, there is considerably more. The new data go to confirm the general correctness of Callendar's method, and to establish the conclusion that the values derived from his equation may be accepted with

[1] H. L. Callendar, "On the Thermodynamical Properties of Gases and Vapours as deduced from a modified form of the Joule-Thomson Equation, with special reference to the Properties of Steam," *Proc. Roy. Soc.* Vol. VII. June 1900. Also articles "Vaporization" and "Thermodynamics," *Encyclopaedia Britannica*, 1902; and a paper "On the Thermodynamical Correction of the Gas Thermometer," *Phil. Mag.* Jan. 1903.

confidence for the purposes of engineering calculation, within the stated range, which, moreover, is the range usual in engineering practice. These values have the great advantage of being thermodynamically consistent with one another, which cannot be said of the numbers in the older steam tables. For pressures such as correspond to saturation temperatures much beyond 200° C. experiments are lacking, and it may be expected that the Callendar equation will not apply to them with so high a degree of approximation as it does for pressures within the usual range.

The method of Callendar was taken up by Mollier[1] who, after making a slight readjustment of the constants, applied it to calculate numerical values of all the important properties of steam, which he has tabulated and has also represented graphically in the diagrams to which reference has been made in the text (§ 100). Mollier has also made a useful comparison of the values calculated by Callendar's method with those found in the Munich and other experiments. The calculations made by Mollier are followed in the tables which are given in this book.

The characteristic equation with which Callendar starts is in the form

$$V - b = \frac{R\tau}{P} - C\left(\frac{273}{\tau}\right)^n \quad \dots\dots\dots\dots\dots\dots(1),$$

where V is the volume of the unit mass of steam (either saturated or superheated);

P is the pressure;

τ is the absolute temperature;

and b, R, C and n are constants, the values of which have to be determined.

It is convenient to write Callendar's equation in the form

$$V - b = \frac{R\tau}{P} - c \dots\dots\dots\dots\dots\dots\dots\dots(1a),$$

where $c = C\left(\frac{273}{\tau}\right)^n$. Callendar calls c the "coaggregation volume," regarding it as an expression for the loss of volume which the vapour suffers (compared with the volume it would have if it were a perfect gas) in consequence of linking or grouping of the molecules. It is a quantity of the dimensions of a volume, and is a function of the temperature only.

[1] R. Mollier, *Neue Tabellen und Diagramme für Wasserdampf*. Berlin, Julius Springer, 1906.

Of the constants, b is taken as equal to the volume of water at a low temperature, when the vapour pressure is small, namely 0·001 cubic metre, when the unit of mass is the kilogramme. R, which corresponds with the constant to which the same letter is applied in the usual formula for a gas, is determined by reference to the molecular weight of water. It is taken as 47·00 kilogrammetres, or 0·110 heat unit.

Callendar deduces the constants C and n from the observed cooling effect when steam is throttled by expansion through a porous plug or constricted orifice. Calling K_p the specific heat at constant pressure, the cooling effect is measured by the product of K_p into the observed drop in temperature per unit drop in pressure in the process, or $\dfrac{dt}{dP}$ subject to the condition that the total heat I is constant. Callendar shows that the cooling effect

$$K_p \left(\frac{dt}{dP}\right)_{I \text{ const.}} = (n+1)\, c - b \ldots\ldots\ldots\ldots\ldots(2).$$

This product is the amount of heat which would have to be given to the throttled vapour to restore it, at constant pressure, to the temperature it had before throttling. From throttling experiments he finds numerical values of n and C. A probable value of n, separately, may be inferred by applying the kinetic theory of gases to steam as a tri-atomic gas, but the values both of C and of n which are used in the calculation of the tables are adjusted to bring the results into as close as possible an agreement with those of experiment. In Mollier's reckoning n is taken as $\frac{10}{3}$, the various quantities are calculated per kilogramme, and C is 0·075 cubic metre.

Callendar proceeds to show that

$$\left(\frac{dK_p}{dP}\right)_{\tau \text{ const.}} = \frac{n\,(n+1)\,c}{\tau},$$

and hence by integration,

$$K_p = K_p{}^0 + \frac{n\,(n+1)\,cP}{\tau} \ldots\ldots\ldots\ldots\ldots(3),$$

where $K_p{}^0$ is the limiting value of the specific heat at constant pressure, when the pressure is zero. It is assumed that at very low pressures the vapour approximates so closely to the state of a perfect gas that $K_p{}^0$ may be taken as constant, in the sense that it does not vary with the temperature. This assumption is well

supported by experiments on the specific heat of steam at very low pressures.

Similarly K_v^0, the specific heat at constant volume for steam of very low pressure, is treated as constant, the relation of K_v^0 and K_p^0 being (as in gases)

$$K_p^0 = K_v^0 + R.$$

According to Callendar K_p^0 for water vapour is equal to nR. Consequently K_p^0 is $(n+1)R$. Taking n as $\frac{10}{3}$ and R as $0\cdot110$ thermal unit the numerical values of the specific heats of very low-pressure steam accordingly are

$$K_p^0 = 0\cdot477 \text{ and } K_v^0 = 0\cdot367.$$

For the Entropy ϕ Callendar obtains the expression

$$\phi = K_p^0 \log_e \tau - R \log_e P - \frac{ncP}{\tau} + \text{const.} \dots\dots\dots\dots(4),$$

and for the Energy E,

$$E = K_v^0 t - ncP + \text{const.} \dots\dots\dots\dots\dots(5),$$

where the temperature t is reckoned from the usual zero and the constants are constants of integration. K_v^0 is the limiting value, at zero pressure, of the specific heat at constant volume.

Since the Total Heat $I = E + PV$ we have

$$I = K_v^0 t - ncP + PV + \text{const.} \dots\dots\dots\dots(6),$$

which may be written

$$I = K_p^0 t - [(n+1)c - b]P + \text{const.} \dots\dots\dots(6a),$$

since

$$K_p^0 - K_v^0 = R,$$

and by the characteristic equation $(1a)$

$$R\tau = P(V + c - b).$$

The constants of integration are found by applying these equations to particular cases where the values of I, ϕ, or E are already known from experimental results.

Taking the particular case of saturated steam at $100°$ C., for which the pressure is $1\cdot033 \times 10000$ kilogrammes per square metre, it appears from an examination of the best experimental determinations of latent heat that the value $639\cdot7$ may be assigned to I. Substituting this in equation $(6a)$ we have

$$639\cdot7 = 0\cdot477 \times 100 - \left[\frac{13}{3} \cdot 0\cdot075 \left(\frac{273}{373}\right)^{\frac{10}{3}} - 0\cdot001\right] \frac{1\cdot033 \times 10^4}{427} + \text{const.}$$

The term in square brackets is divided by J (namely 427) to reduce it to thermal units.

Hence the constant is

$$639 \cdot 7 - 47 \cdot 7 + 2 \cdot 66 \times 1 \cdot 033 = 594 \cdot 7.$$

In a similar manner the constant in equation (4) is found to be $-1 \cdot 0544$, and that in equation (5) to be $564 \cdot 7$.

To give the expressions a more convenient form for numerical evaluation we may, following Mollier, write

$$\frac{10000}{427}\left(\frac{13}{3}c - 0 \cdot 001\right) = Y,$$

and

$$\frac{10000}{427} \cdot \frac{10}{3} \cdot \frac{Y}{\tau} = Z.$$

Values of the quantities Y and Z, along with the values of c, have been calculated by Mollier for a series of temperatures going up by intervals of $5°$. They are given in the auxiliary table (Table D).

The equations for the properties of steam, whether saturated or superheated, then take the following comparatively convenient form :

The volume,

$$V = 47 \frac{\tau}{P} - Y + 0 \cdot 001 \dots\dots\dots\dots\dots\dots(7).$$

The total heat,

$$I = 594 \cdot 7 + 0 \cdot 477t - YP \dots\dots\dots\dots\dots(8).$$

The entropy,

$$\phi = 0 \cdot 477 \log_e \tau - 0 \cdot 11 \log_e P - ZP - 1 \cdot 0544 \dots\dots(9).$$

or, $$\phi = 1 \cdot 0984 \log_{10} \tau - 0 \cdot 2533 \log_{10} P - ZP - 1 \cdot 0544 \dots\dots(9a).$$

The internal energy,

$$E = 564 \cdot 7 + 0 \cdot 367t - \frac{10}{3} \cdot \frac{cP}{427} \dots\dots\dots\dots(10).$$

The specific heat at constant pressure,

$$K_p = 0 \cdot 477 + \frac{13}{3} ZP \dots\dots\dots\dots(11).$$

In these formulas,

V is the volume in cubic metres per kilogramme ;

P is the pressure in kilogrammes per square centimetre ;

t is the temperature on the centigrade scale ;

τ is the absolute temperature in centigrade degrees, or $t + 273$.

From these the tabulated values of the properties of saturated steam are found. But the formulas have the further important

use that they apply to superheated steam as well, and enable the volume, total heat, entropy, energy, and specific heat to be calculated for any assigned conditions as to temperature and pressure. They are true of water vapour generally, the saturated condition being only a special case. For any given pressure they hold good at all temperatures not lower than the corresponding saturation temperature; and for any given temperature they hold good at all pressures not higher than the corresponding saturation pressure. With the help of the auxiliary table containing values of c, Y and Z it is accordingly easy to obtain numerical values corresponding to any condition of superheat.

The total heat of water is tabulated on the assumption that Regnault's formula for h may be applied without serious error, namely

$$h = t + 0.00002t^2 + 0.0000003t^3 \dots\dots\dots\dots(12),$$

and to this the quantity APw has to be added to find the total heat for water, or I_w in the notation of § 61.

By differentiating equation (12) we obtain an expression for the specific heat of water in terms of the temperature, which may be substituted for σ in the expression for the entropy of water, namely $\int_{\tau_0}^{\tau} \frac{\sigma d\tau}{\tau}$. On integration this gives for the entropy of water the formula

$$\phi_w = 2.4319 \log_{10} \frac{\tau}{273} - 0.000206t + 0.00000045t^2 \dots\dots(13).$$

Having obtained values of the entropy of saturated steam and of water at the same temperature, ϕ_s and ϕ_w, the latent heat may be found from the relation

$$L = \tau (\phi_s - \phi_w) \dots\dots\dots\dots\dots\dots\dots(14).$$

But the latent heat is also given directly by the relation

$$L = I_s - I_w \dots\dots\dots\dots\dots\dots(15),$$

where I_s is the total heat of saturated steam and I_w that of water at the same temperature.

Hence, equating these two expressions for L,

$$I_s - I_w = \tau (\phi_s - \phi_w) \dots\dots\dots\dots\dots(16),$$

and by substituting the expressions already found for these quantities in terms of P and t an equation may be obtained connecting P and t for saturated steam.

The saturated pressures found in this way for various temperatures throughout the range to which the equations apply are in very close agreement with those measured by Regnault throughout the same range, and the agreement between the calculated and measured pressures affords a confirmation of the soundness of Callendar's method. It is only at the highest pressures to which the table extends that there is any appreciable difference between the calculated pressures and those observed by Regnault, as the following comparison will show:

Temperature ° C.	Pressure of Saturated Steam in mm. of Mercury	
	By Mollier's calculations from Callendar's equation	By Regnault's measurements
0	4·60	4·60
25	23·6	23·55
40	54·9	54·91
50	92·0	91·98
75	288·5	288·5
100	760·0	760·0
130	2030	2030·0
160	4651	4651·6
190	9442	9426

Further confirmation of Callendar's method is obtained by comparing the volumes calculated from his formula with those obtained in the Munich observations[1]:

Temperature ° C.	Volume of Saturated Steam in cubic metres per kilogramme	
	By Mollier's calculations from Callendar's equation	By Munich experiments
100	1·670	1·674
110	1·207	1·211
120	0·889	0·892
130	0·664	0·669
140	0·507	0·509
150	0·3917	0·3921
160	0·3065	0·3073
170	0·2429	0·2430
180	0·1945	0·1943

[1] Knoblauch, Linde and Klebe, *Mitteilungen über Forschungsarbeiten herausgegeben vom Verein deutscher Ingenieure*, Heft 21, Berlin, 1905.

Equally striking agreement is obtained in the figures for super-heated steam:

Temperature °C.	Volume of Superheated Steam in cubic metres per kilogramme, at various pressures in kilogrammes per square centimetre			
	$P=2$	$P=4$	$P=6$	$P=8$
120	Calculated 0·902 Observed 0·903			
130	Calculated 0·928 Observed 0·929			
140	Calculated 0·953 Observed 0·954			
150	Calculated 0·978 Observed 0·979	0·481 0·481		
160	Calculated 1·002 Observed 1·004	0·494 0·494	0·324 0·324	
170	Calculated 1·027 Observed 1·029	0·507 0·507	0·333 0·333	0·2464 0·2454
180	Calculated 1·052 Observed 1·054	0·519 0·520	0·342 0·342	0·2532 0·2527

In dealing with superheated steam we may use equation (8) to calculate directly the total heat in the superheated condition, or we may take the value of the total heat of saturated steam from the tables and add the quantity of extra heat required for superheating, namely

$$\kappa \, (t' - t),$$

where t is the temperature of saturation and t' the temperature after superheating, and κ is the mean specific heat for the range from t to t'.

Values of κ have been given in § 65. They are found by calculating I' for superheated steam and I_s for saturated steam at the same pressure, using equation (8), and then dividing $I' - I_s$ by the difference of temperature $t' - t$. Direct measurements of the specific heat by Knoblauch and Jakob in the Munich Laboratory by the method of electrical heating give values which are on the whole slightly lower than these[1].

The tables which follow give the properties of saturated steam as calculated by Mollier. The centigrade scale of temperature is used throughout, and the total heat, latent heat, and energy are

[1] Knoblauch and Jakob, *Mitteilungen über Forschungsarbeiten herausgegeben vom Verein deutscher Ingenieure*, Heft 36, Berlin, 1906.

expressed in terms of the quantity of heat required to raise the temperature of unit mass of water through 1° C. at 15° C. In Table A the pressures and volumes are stated in British as well as in metric units. In that table the various quantities are given for a series of temperatures rising by equal steps of 5° C. In Table B the quantities are given for a series of pressures which are stated in kilogrammes per square centimetre.

A supplementary table (C) shows the temperature of saturated steam on the centigrade scale and its pressure in kilogrammes per sq. cm. for various pressures expressed in lbs. per sq. inch. This is to facilitate the application of the principal tables and diagrams to cases where, as is usual in English and American practice, the primary datum is the pressure in lbs. per square inch.

Table D is the auxiliary table calculated by Mollier, giving the values of c, Y, and Z for use in the various equations. This is specially serviceable in calculations relating to superheated steam. Particulars for saturated steam are in general more readily found by referring to the tables.

To a great extent the diagrams enable numerical tabulation and calculation to be dispensed with. They may be said to form graphical tables of a very comprehensive kind, exhibiting the properties of superheated as well as saturated steam, from which the required values may be obtained by direct measurement in a very simple and expeditious manner, and with as high a degree of accuracy as the experimental data warrant.

Of the various ways in which the properties of steam may be graphically represented it may be said that while the entropy-temperature diagram is invaluable in discussing the action of a heat-engine, Mollier's diagram of Entropy and Total Heat is in general more directly useful both as a means of graphic tabulation and as an aid in solving problems.

In reproducing this diagram here it has been slightly extended, as compared with Mollier's original, by adding additional constant pressure lines at the lowest pressures, to make it available in cases where (as with Parsons' turbines) a very high vacuum is obtained in the condenser. The other Mollier diagram, of Pressure and Total Heat, forms a useful supplement, especially in problems which are concerned with the volume of the steam. Both of these diagrams are given as folding plates at the end of the book.

In the diagram of Pressure and Total Heat the lines of constant volume in the region of superheat are straight, as was mentioned in § 98. This, as Callendar points out, follows from the relation $K_v^0 = nR$, which he takes as applying to water vapour within the limits of experimental error. Equation (6) may be written in the form

$$I = K_v^0 \tau - ncP + PV + \text{const.}$$
$$= nR\tau - ncP + PV + \text{const.}$$

Hence, substituting $P(V - b + c)$ for $R\tau$, from equation (1a), we obtain

$$I = (n + 1)PV - nbP + \text{const.},$$

showing that I is a linear function of V when P is constant. In the wet region the constant pressure lines remain very nearly straight, for the above relation still holds in that part of the steam which remains uncondensed, and its volume constitutes very nearly the whole volume of the wet mixture.

The constant temperature lines in this diagram are also straight, for if we differentiate equation (8) we have

$$\left(\frac{dI}{dP}\right)_{t\,\text{const.}} = -Y.$$

The slope of each line is therefore constant, but changes from line to line with the change of Y, which is a function of the temperature (see Table D).

| Tempera-ture ° C. | Pressure | | Volume | | Latent Heat |
	kilo-grammes per sq. cm.	pounds per sq. inch	cub. cm. per kg.	cub. ft. per lb.	
0	0·0063	0·089	204·97	3283	594·7
5	0·0089	0·127	146·93	2353·6	592·1
10	0·0125	0·178	106·62	1707·9	589·4
15	0·0173	0·246	78·23	1253·2	586·8
20	0·0236	0·336	58·15	931·48	584·1
25	0·0320	0·455	43·667	699·48	581·5
30	0·0429	0·610	33·132	530·72	578·8
35	0·0569	0·809	25·393	406·76	576·1
40	0·0747	1·06	19·650	314·77	573·4
45	0·0971	1·38	15·346	245·82	570·7
50	0·125	1·78	12·091	193·68	567·9
55	0·160	2·28	9·607	153·89	565·2
60	0·202	2·88	7·695	123·26	562·4
65	0·254	3·61	6·211	99·49	559·6
70	0·317	4·51	5·050	80·89	556·8
75	0·392	5·58	4·1353	66·24	553·9
80	0·482	6·86	3·4085	54·60	551·0
85	0·589	8·38	2·8272	45·29	548·1
90	0·714	10·16	2·3592	37·79	545·2
95	0·862	12·26	1·9797	31·712	542·2
100	1·033	14·70	1·6702	26·754	539·1
105	1·232	17·52	1·4166	22·692	536·1
110	1·462	20·79	1·2073	19·339	532·9
115	1·726	24·55	1·0338	16·560	529·8
120	2·027	28·83	0·8894	14·247	526·6
125	2·371	33·72	0·7681	12·304	523·3
130	2·760	39·26	0·6664	10·675	520·0
135	3·200	45·51	0·5800	9·291	516·6
140	3·695	52·56	0·5071	8·123	513·2
145	4·248	60·42	0·4450	7·128	509·7
150	4·868	69·24	0·3917	6·274	506·2
155	5·557	79·04	0·3460	5·542	502·6
160	6·323	89·93	0·3065	4·910	498·9
165	7·170	101·98	0·2724	4·363	495·2
170	8·104	115·27	0·2429	3·891	491·4
175	9·131	129·87	0·2171	3·478	487·6
180	10·258	145·90	0·1945	3·116	483·7
185	11·491	163·44	0·1748	2·800	479·8
190	12·835	182·56	0·1575	2·523	475·7
195	14·300	203·4	0·1423	2·279	471·7
200	15·89	226·0	0·1288	2·063	467·5
205	17·61	250·5	0·1170	1·874	463·3
210	19·49	277·2	0·1063	1·703	459·1
215	21·57	306·8	0·0965	1·546	454·7

Tempera-ture °C.	Total Heat		Entropy		Energy
	of Water	of Steam	of Water	of Steam	
0	0	594·7	0·0000	2·1783	564·7
5	5·0	597·1	0·0182	2·1479	566·5
10	10·0	599·4	0·0360	2·1188	568·3
15	15·0	601·8	0·0535	2·0909	570·1
20	20·0	604·1	0·0707	2·0643	571·9
25	25·0	606·5	0·0877	2·0389	573·7
30	30·0	608·8	0·1044	2·0146	575 5
35	35·0	611·1	0·1208	1·9912	577·3
40	40·1	613·5	0·1369	1·9688	579·1
45	45·1	615·8	0·1528	1·9474	580·9
50	50·1	618·0	0·1685	1·9268	582·6
55	55·1	620·3	0·1839	1·9070	584·4
60	60·1	622·6	0·1991	1·8880	586·1
65	65·2	624·8	0·2141	1·8697	587·8
70	70·2	627·0	0·2289	1·8522	589·5
75	75·3	629·2	0·2435	1·8352	591·2
80	80·3	631·5	0·2579	1·8189	592·8
85	85·3	633·5	0·2721	1·8031	594·5
90	90·4	635·6	0·2861	1·7879	596·1
95	95·5	637·6	0·2999	1·7731	597·7
100	100·5	639·7	0·3136	1·7589	599·2
105	105·6	641·7	0·3271	1·7452	600·8
110	110·7	643·6	0·3404	1·7319	602·3
115	115·8	645·5	0·3536	1·7190	603·8
120	120·9	647·4	0·3666	1·7064	605·2
125	126·0	649·2	0·3795	1·6943	606·6
130	131·1	651·0	0·3922	1·6824	608·0
135	136·2	652·8	0·4048	1·6710	609·3
140	141·3	654·5	0·4173	1·6599	610·6
145	146·4	656·1	0·4296	1·6490	611·9
150	151·6	657·8	0·4418	1·6384	613·1
155	156·7	659·3	0·4538	1·6280	614·3
160	161·9	660·8	0·4658	1·6181	615·4
165	167·1	662·3	0·4776	1·6082	616·5
170	172·2	663·7	0·4893	1·5986	617·6
175	177·4	665·0	0·5009	1·5893	618·6
180	182·6	666·3	0·5124	1·5802	619·6
185	187·9	667·6	0·5238	1·5713	620·6
190	193·1	668·8	0·5351	1·5626	621·5
195	198·3	670·0	0·5463	1·5541	622·3
200	203·6	671·1	0·5574	1·5458	623·2
205	208·9	672·2	0·5684	1·5377	624·0
210	214·1	673·2	0·5793	1·5298	624·7
215	219·4	674·1	0·5901	1·5220	625·4

Pressure		Temperature		Volume	Latent Heat
kilogrammes per sq. cm.	pounds per sq. inch	Cent.	absolute	cubic metres per kilogramme	
0·02	0·284	17·3	290·3	68·126	585·5
0·04	0·569	28·8	301·8	35·387	579·4
0·06	0·853	36·0	309·0	24·140	575·6
0·08	1·138	41·3	314·3	18·408	572·7
0·10	1·422	45·6	318·6	14·920	570·4
0·12	1·707	49·2	322·2	12·568	568·4
0·15	2·133	53·7	326·7	10·190	565·9
0·20	2·846	59·8	332·8	7·777	562·6
0·25	3·556	64·6	337·6	6·307	559·8
0·30	4·267	68·7	341·7	5·316	557·5
0·35	4·978	72·3	345·3	4·600	555·5
0·40	5·689	75·5	348·5	4·060	553·7
0·50	7·111	80·9	353·9	3·2940	550·5
0·60	8·534	85·5	358·5	2·7770	547·8
0·70	9·956	89·5	362·5	2·4040	545·5
0·80	11·38	93·0	366·0	2·1216	543·3
0·90	12·80	96·2	369·2	1·9003	541·4
1·0	14·22	99·1	372·1	1·7220	539·7
1·1	15·65	101·8	374·8	1·5751	538·1
1·2	17·07	104·2	377·2	1·4521	536·5
1·4	19·91	108·7	381·7	1·2571	533·7
1·6	22·76	112·7	385·7	1·1096	531·2
1·8	25·60	116·3	389·3	0·9939	528·9
2·0	28·45	119·9	392·6	0·9006	526·8
2·5	35·56	126·7	399·7	0·7310	522·2
3·0	42·67	132·8	405·8	0·6163	518·1
3·5	49·79	138·1	411·1	0·5335	514·5
4·0	56·89	142·8	415·8	0·4708	511·2
4·5	63·00	147·1	420·1	0·4217	508·2
5·0	71·11	151·0	424·0	0·3820	505·5
5·5	78·23	154·6	427·6	0·3494	502·9
6·0	85·34	157·9	430·9	0·3220	500·4
6·5	92·45	161·1	434·1	0·2987	498·1
7·0	99·56	164·0	437·0	0·2786	495·9
7·5	106·67	166·8	439·8	0·2611	493·9
8·0	113·8	169·5	442·5	0·2458	491·8
8·5	120·9	172·0	445·0	0·2322	489·9
9·0	128·0	174·4	447·4	0·2200	488·1
9·5	135·1	176·7	449·7	0·2091	486·3
10·0	142·2	178·9	451·9	0·1993	484·6
11·0	156·5	183·1	456·1	0·1822	481·3
12·0	170·7	186·9	459·9	0·1678	478·2
13·0	184·9	190·6	463·6	0·1556	475·3
14·0	199·1	194·0	467·0	0·1451	472·5
15·0	213·3	197·2	470·2	0·1360	469·8
16·0	227·6	200·3	473·3	0·1280	467·3
18·0	256·0	206·1	479·1	0·1145	462·4
20·0	284·5	211·3	484·3	0·1036	457·0

| Pressure | Total Heat | | Entropy | | |
kilo- grammes per sq. cm.	of Water	of Steam	of Water	of Steam	Energy
0·02	17·3	602·9	0·0616	2·0783	571·0
0·04	28·8	608·3	0·1004	2·0202	575·1
0·06	36·0	611·6	0·1240	1·9868	577·7
0·08	41·4	614·1	0·1411	1·9631	579·6
0·10	45·7	616·0	0·1546	1·9449	581·8
0·12	49·3	617·7	0·1659	1·9300	582·3
0·15	53·8	619·7	0·1799	1·9121	583·9
0·20	59·9	622·4	0·1984	1·8890	586·0
0·25	64·8	624·6	0·2129	1·8711	587·7
0·30	68·9	626·4	0·2252	1·8566	589·1
0·35	72·5	628·0	0·2356	1·8444	590·3
0·40	75·7	629·4	0·2448	1·8336	591·3
0·50	81·2	631·7	0·2604	1·8159	593·1
0·60	85·8	633·7	0·2734	1·8015	594·6
0·70	89·9	635·3	0·2846	1·7895	595·9
0·80	93·5	636·8	0·2944	1·7789	597·0
0·90	96·7	638·1	0·3032	1·7698	598·0
1·0	99·6	639·3	0·3111	1·7615	599·0
1·1	102·3	640·7	0·3183	1·7541	599·8
1·2	104·8	641·3	0·3250	1·7473	600·5
1·4	109·4	643·1	0·3370	1·7352	601·9
1·6	113·4	644·7	0·3475	1·7248	603·0
1·8	117·1	646·0	0·3569	1·7156	604·1
2·0	120·4	647·2	0·3655	1·7077	605·1
2·5	127·7	649·0	0·3839	1·6903	607·1
3·0	133·9	652·0	0·3993	1·6766	608·7
3·5	139·4	653·0	0·4125	1·6640	610·1
4·0	144·2	655·4	0·4242	1·6537	611·3
4·5	148·6	656·8	0·4347	1·6445	612·4
5·0	152·6	658·1	0·4442	1·6363	613·3
5·5	156·3	659·2	0·4529	1·6290	614·2
6·0	159·8	660·2	0·4609	1·6221	615·0
6·5	163·0	661·1	0·4683	1·6158	615·7
7·0	166·1	662·0	0·4753	1·6101	616·3
7·5	168·9	662·8	0·4819	1·6048	616·9
8·0	171·7	663·5	0·4881	1·5997	617·5
8·5	174·3	664·2	0·4939	1·5949	618·0
9·0	176·8	664·9	0·4995	1·5905	618·5
9·5	179·2	665·5	0·5048	1·5863	619·0
10·0	181·5	666·1	0·5099	1·5822	619·4
11·0	185·8	667·1	0·5194	1·5748	620·2
12·0	189·9	668·1	0·5282	1·5678	620·9
13·0	193·7	668·9	0·5364	1·5616	621·6
14·0	197·3	669·7	0·5440	1·5557	622·2
15·0	200·7	670·5	0·5513	1·5504	622·7
16·0	203·9	671·2	0·5581	1·5452	623·2
18·0	210·0	672·4	0·5707	1·5359	624·1
20·0	215·5	673·4	0·5821	1·5274	624·9

Pressure		Temp.	Pressure		Temp.
pounds per sq. inch	kilo-grammes per sq. cm.	Cent.	pounds per sq. inch	kilo-grammes per sq. cm.	Cent.
0·5	0·0352	26·6	62	4·359	145·9
1·0	0·0703	38·9	64	4·500	147·1
1·5	0·1055	46·6	66	4·640	148·3
2·0	0·1406	52·3	68	4·781	149·4
2·5	0·1758	57·0	70	4·922	150·4
3·0	0·2109	60·9	72	5·062	151·5
3·5	0·2461	64·3	74	5·203	152·5
4·0	0·2812	67·2	76	5·343	153·5
4·5	0·3164	70·0	78	5·484	154·5
5	0·3515	72·4	80	5·625	155·4
6	0·4218	76·7	82	5·765	156·4
7	0·4921	80·5	84	5·906	157·3
8	0·5625	83·9	86	6·046	158·2
9	0·6328	86·9	88	6·187	159·1
10	0·7031	89·6	90	6·328	160·0
11	0·7734	92·1	92	6·468	160·9
12	0·8437	94·4	94	6·609	161·7
13	0·9140	96·6	96	6·750	162·5
14	0·9843	98·7	98	6·890	163·3
15	1·0556	100·6	100	7·031	164·1
16	1·125	102·4	105	7·382	166·1
17	1·195	104·1	110	7·734	168·0
18	1·266	105·8	115	8·085	169·9
19	1·336	107·4	120	8·437	171·7
20	1·406	108·9	125	8·789	173·4
22	1·547	111·7	130	9·140	175·0
24	1·687	114·4	135	9·492	176·6
26	1·828	116·8	140	9·843	178·2
28	1·969	119·1	145	10·194	179·7
30	2·109	121·2	150	10·546	181·2
32	2·250	123·3	160	11·25	184·1
34	2·490	125·3	170	11·95	186·8
36	2·531	127·1	180	12·66	189·3
38	2·672	128·9	190	13·36	191·8
40	2·812	130·6	200	14·06	194·2
42	2·953	132·2	210	14·76	196·5
44	3·094	133·8	220	15·47	198·7
46	3·234	135·3	230	16·17	200·8
48	3·374	136·8	240	16·87	202·9
50	3·515	138·2	250	17·58	204·9
52	3·656	139·6	260	18·28	206·8
54	3·797	141·0	270	18·98	208·7
56	3·937	142·3	280	19·69	210·5
58	4·078	143·5	290	20·39	212·2
60	4·218	144·8	300	21·09	213·9

Temp. Cent.	c	Y	z	Temp. Cent.	c	Y	z
0	0·075	8	0·021	225	0·0101	1·00	0·00159
5	0·071	7	0·020	230	0·0098	0·97	0·00152
10	0·067	7	0·018	235	0·0095	0·94	0·00146
15	0·063	6	0·017	240	0·0092	0·91	0·00139
20	0·059	6	0·016	245	0·0089	0·88	0·00134
25	0·056	6	0·015	250	0·0086	0·85	0·00128
30	0·053	5	0·014	255	0·0083	0·82	0·00123
35	0·050	5	0·013	260	0·0081	0·79	0·00118
40	0·048	5	0·012	265	0·0078	0·77	0·00113
45	0·045	5	0·011	270	0·0076	0·75	0·00109
50	0·043	4·3	0·0103	275	0·0074	0·72	0·00105
55	0·041	4·1	0·0097	280	0·0071	0·70	0·00101
60	0·039	3·9	0·0091	285	0·0069	0·68	0·00097
65	0·037	3·7	0·0085	290	0·0067	0·66	0·00093
70	0·035	3·5	0·0080	295	0·0065	0·64	0·00090
75	0·033	3·4	0·0075	300	0·0063	0·62	0·00086
80	0·032	3·2	0·0070	305	0·0062	0·60	0·00083
85	0·030	3·1	0·0066	310	0·0060	0·58	0·00080
90	0·029	2·9	0·0062	315	0·0058	0·57	0·00077
95	0·028	2·8	0·0059	320	0·0057	0·55	0·00074
100	0·0265	2·66	0·00544	325	0·0055	0·53	0·00072
105	0·0255	2·55	0·00524	330	0·0053	0·52	0·00069
110	0·0243	2·44	0·00494	335	0·0052	0·50	0·00067
115	0·0232	2·35	0·00468	340	0·0051	0·49	0·00064
120	0·0223	2·24	0·00442	345	0·0049	0·48	0·00062
125	0·0214	2·14	0·00418	350	0·0048	0·46	0·00060
130	0·0205	2·06	0·00397	355	0·0047	0·45	0·00058
135	0·0197	1·97	0·00376	360	0·0046	0·44	0·00056
140	0·0189	1·89	0·00357	365	0·0044	0·43	0·00054
145	0·0181	1·82	0·00339	370	0·0043	0·41	0·00052
150	0·0174	1·75	0·00321	375	0·0042	0·40	0·00051
155	0·0168	1·68	0·00306	380	0·0041	0·39	0·00049
160	0·0161	1·61	0·00291	385	0·0040	0·38	0·00047
165	0·0155	1·55	0·00277	390	0·0039	0·37	0·00046
170	0·0149	1·49	0·00263	395	0·0038	0·36	0·00044
175	0·0144	1·44	0·00251	400	0·0037	0·35	0·00043
180	0·0139	1·38	0·00239	405	0·0036	0·34	0·00042
185	0·0134	1·33	0·00228	410	0·0035	0·33	0·00040
190	0·0129	1·29	0·00217	415	0·0034	0·33	0·00039
195	0·0124	1·24	0·00208	420	0·0034	0·32	0·00038
200	0·0120	1·20	0·00198	425	0·0033	0·31	0·00037
205	0·0116	1·15	0·00189	430	0·0032	0·30	0·00036
210	0·0112	1·11	0·00181	435	0·0031	0·29	0·00035
215	0·0108	1·08	0·00173	440	0·0031	0·29	0·00034
220	0·0105	1·04	0·00166	445	0·0030	0·28	0·00033

INDEX

850

800

M...
of Entr...
Sca...
Sca...

750

TOTAL HEAT I.

700

Pr...
Kilo...

Lines of cor...
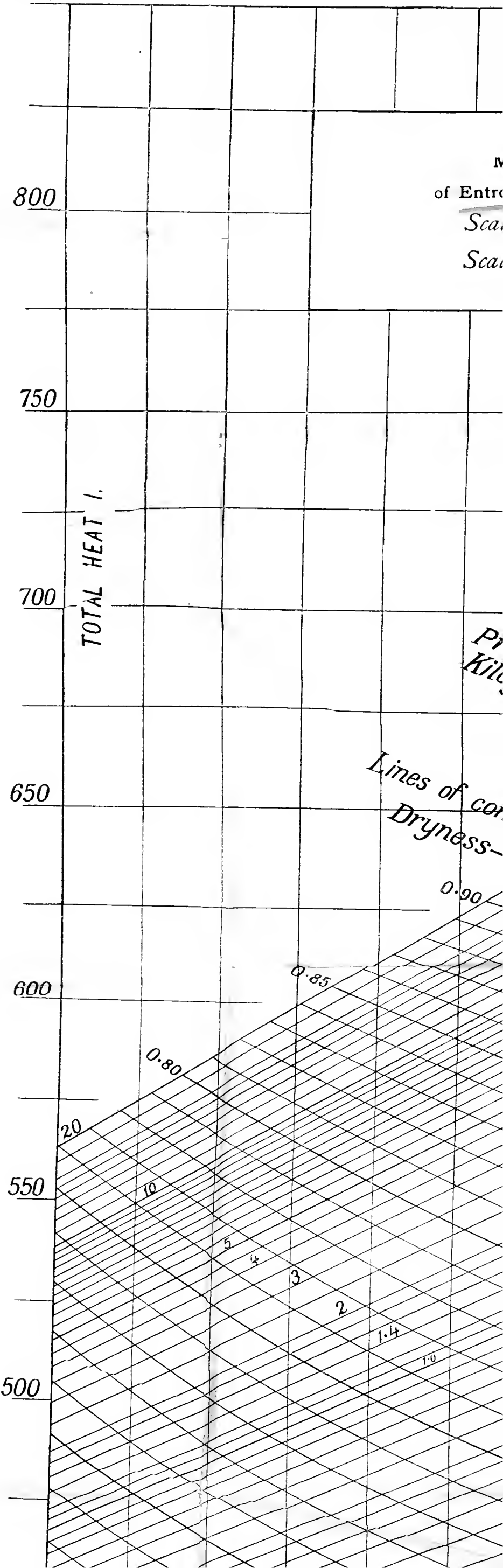Dryness—

650

0.90

0.85

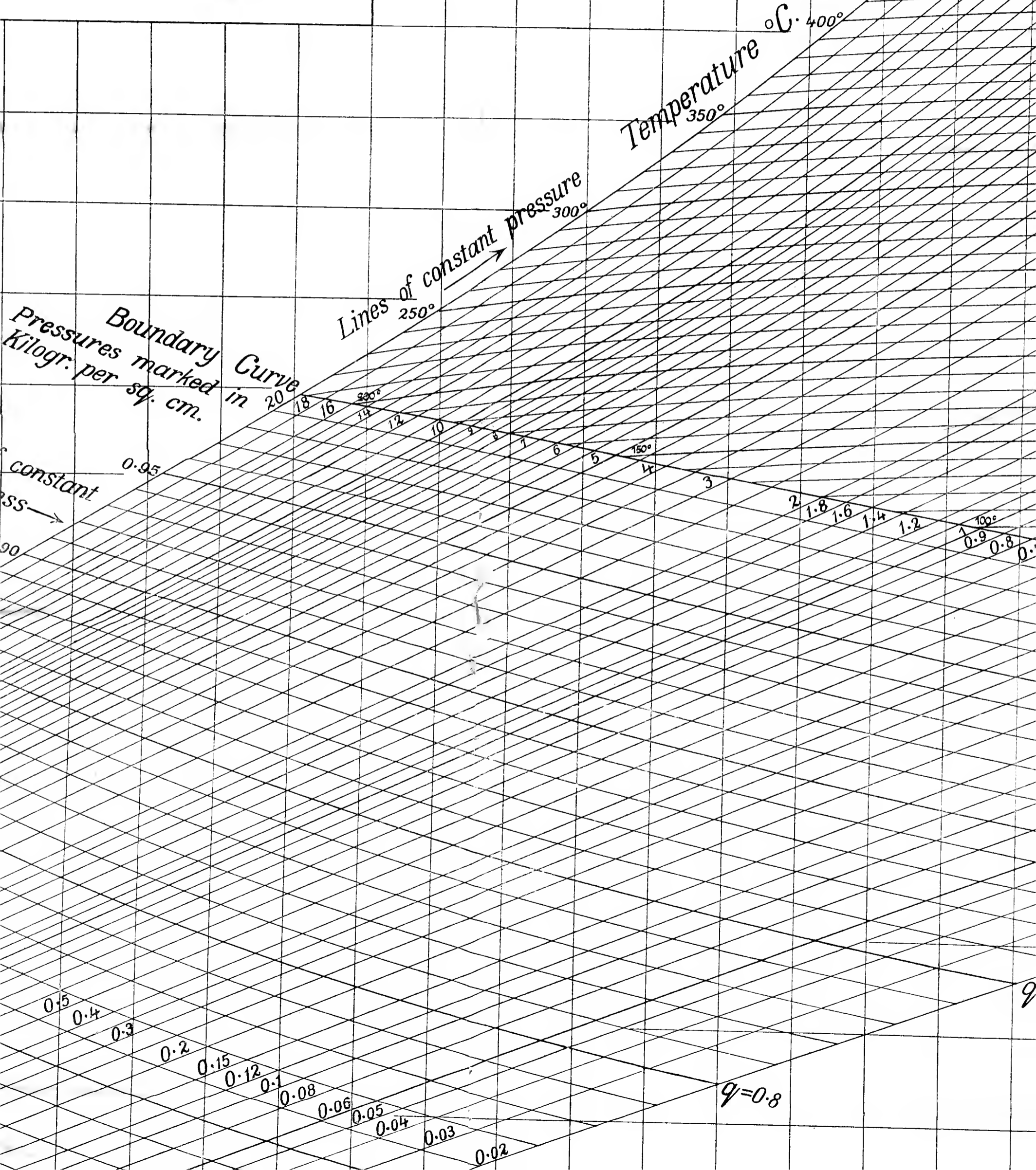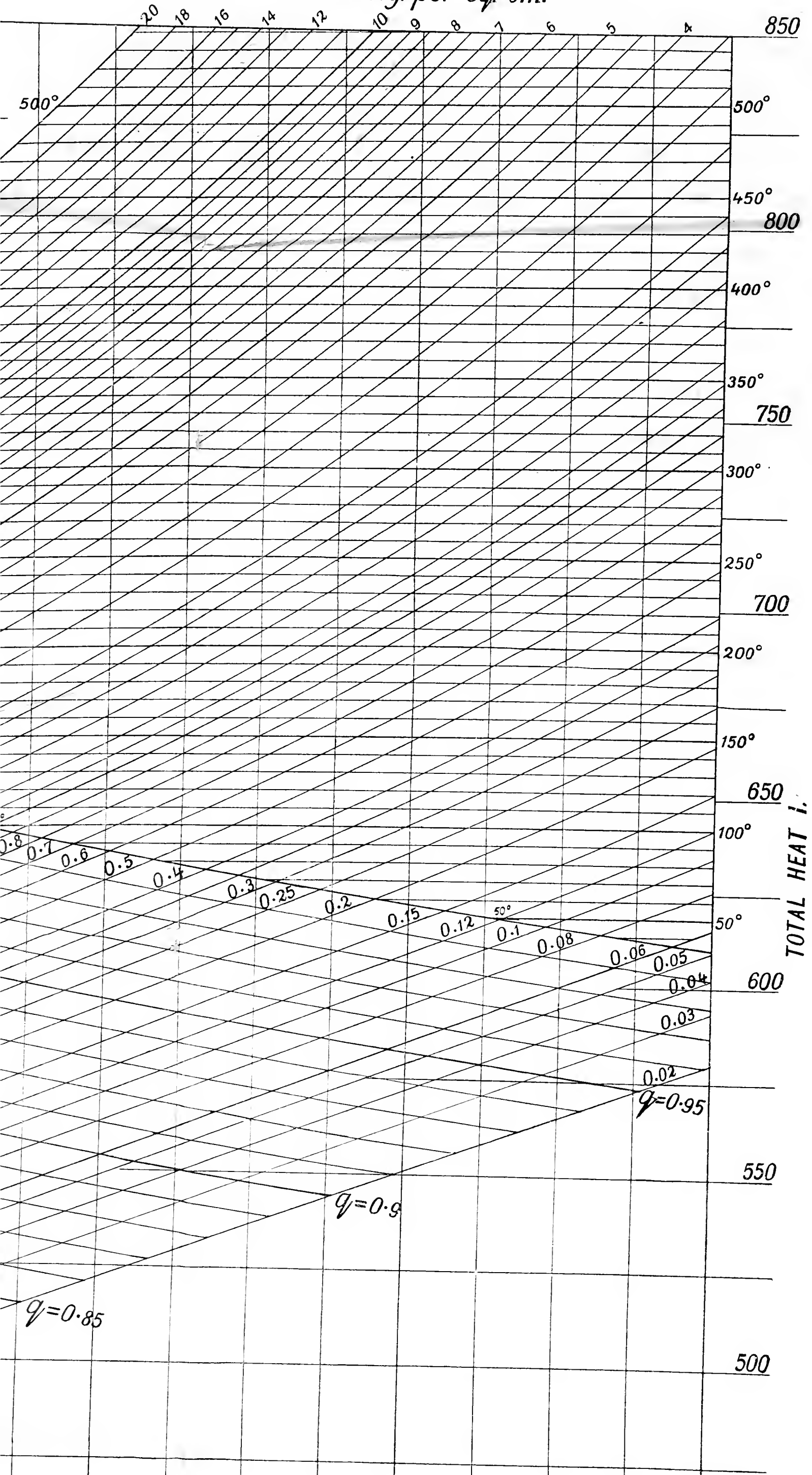600

0.80

20

550

10

5

4

3

2

1.4

1.0

500

MOLLIER'S DIAGRAM

Entropy and Total Heat for Steam
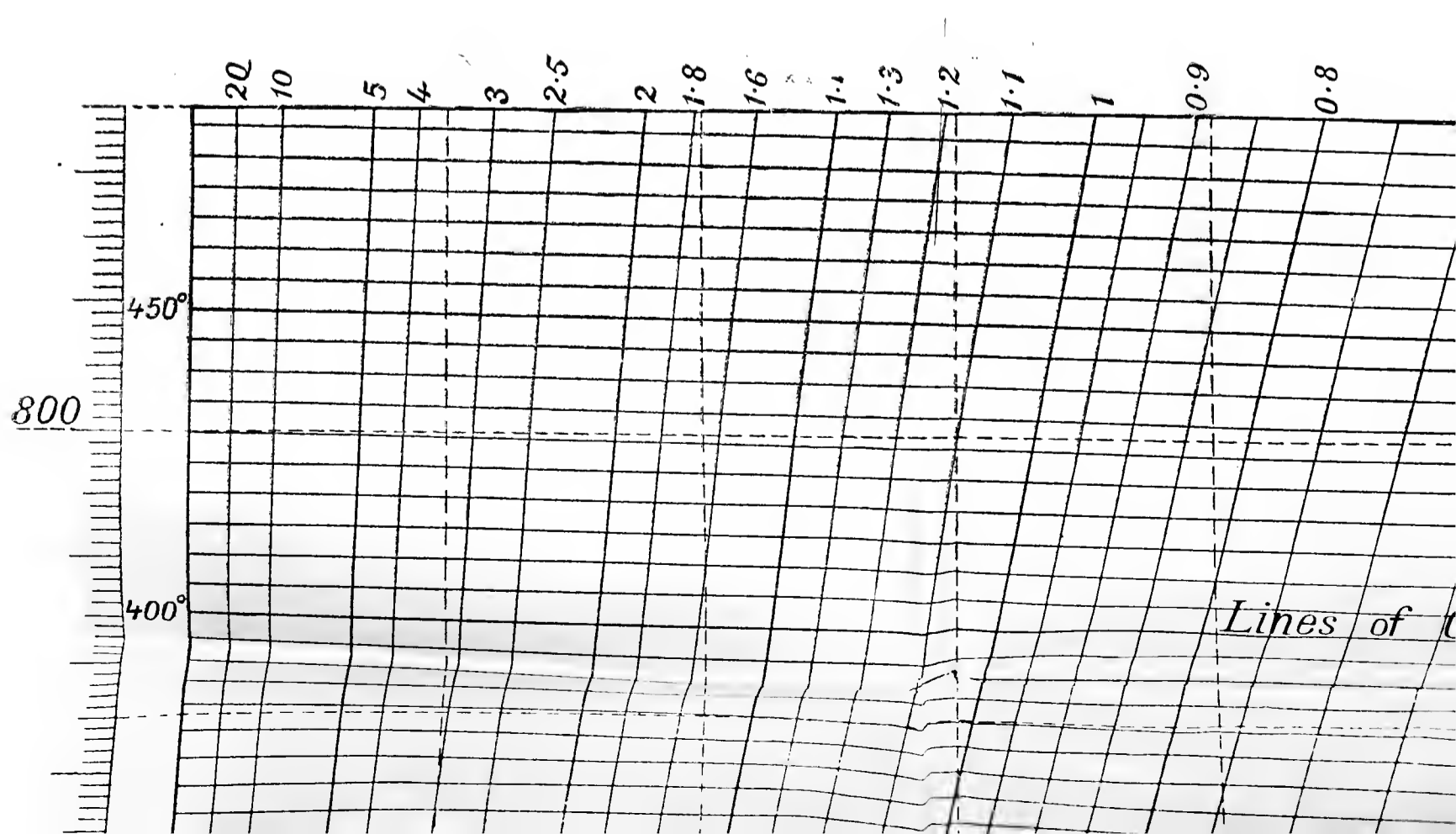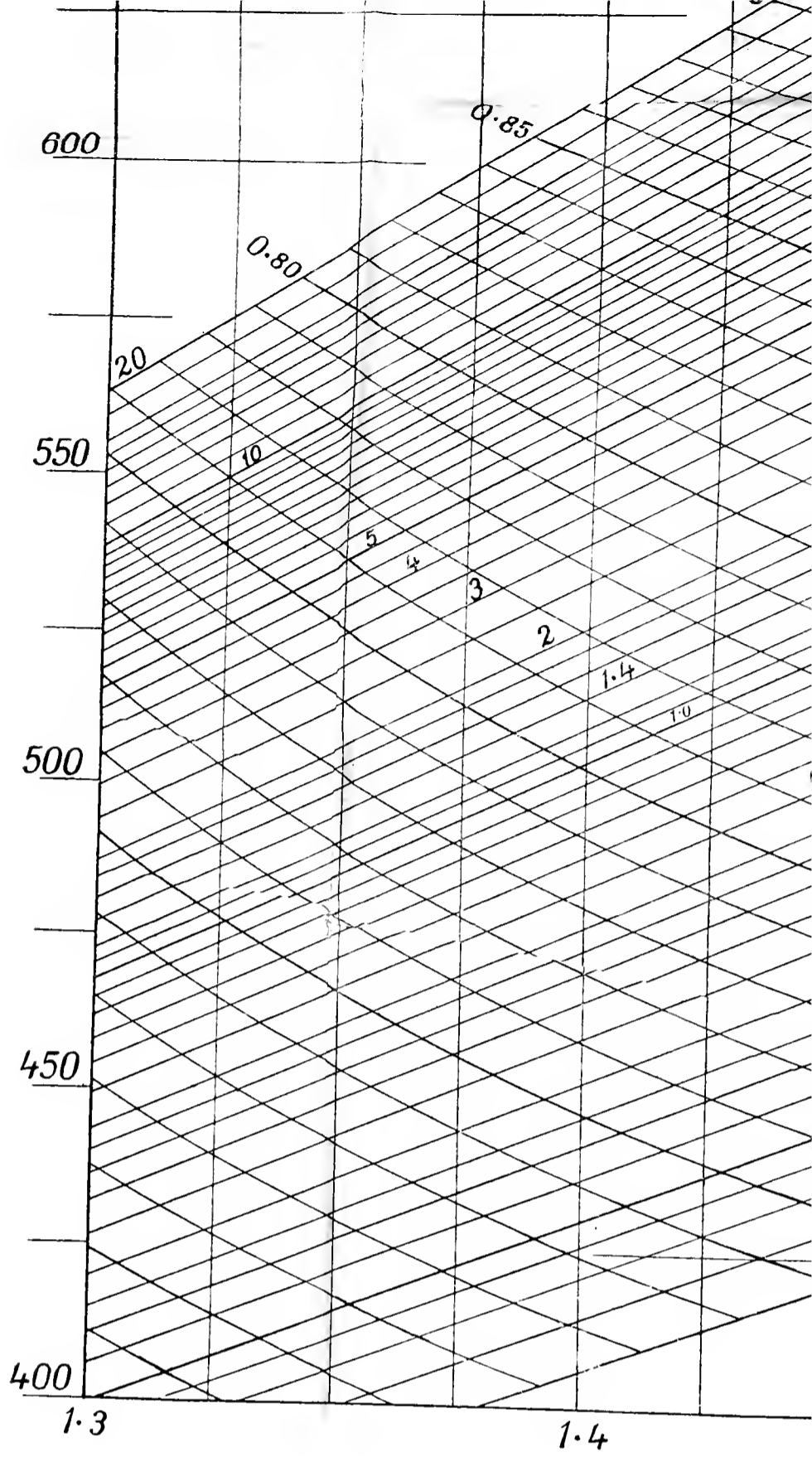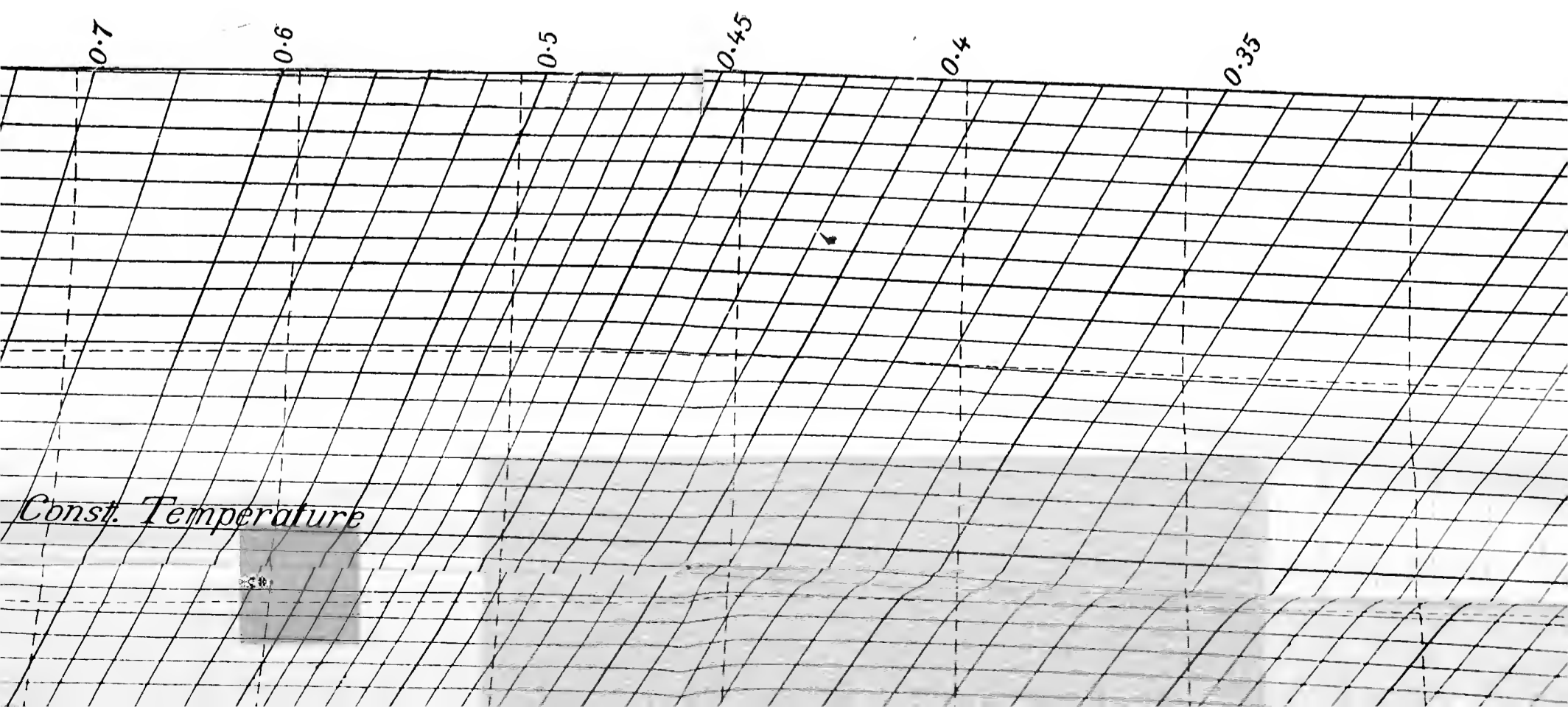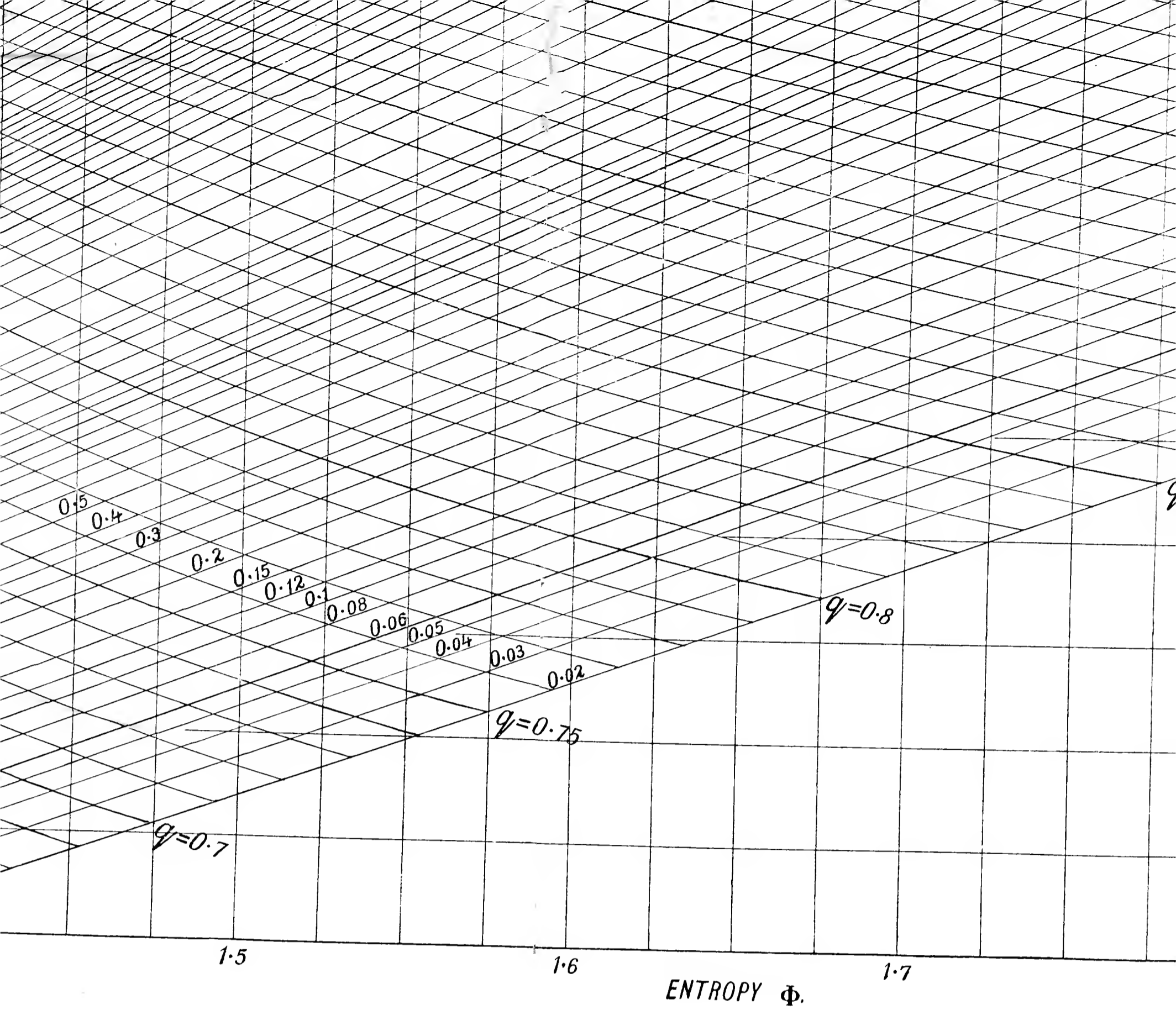
Scale of I, 1 *mm* = 1 *unit.*

Scale of **Φ**, 1 *mm* = 0.00125.

Pressure in Kg. per sq. cm.

600

0.85

0.80

20

550

10

5

4

3

2

1.4

1.0

500

450

400

1.3

1.4

20   10   5   4   3   2.5   2   1.8   1.6   1.4   1.3   1.2   1.1   1   0.9   0.85   0.8

450°

800

400°

Lines of

0.5 0.4 0.3 0.2 0.15 0.12 0.1 0.08 0.06 0.05 0.04 0.03 0.02

$q=0.8$

$q=0.75$

$q=0.7$

1·5   1·6   1·7

ENTROPY Φ.

0.7   0.6   0.5   0.45   0.4   0.35

Const. Temperature

0.3

0.275

0.25

0.225

0.2

400°

450°

775

800

825

$q=0.85$

$q=0.9$

$q=0.95$

1.8

1.9

2.0

0.3

0.25

0.2

0.15

0.12

0.1

0.08

0.06

0.05

0.04

0.03

0.02

50°

50°

400

450

500

550

600

TOTAL

Lines of Con...

Lines of Const. Volume

TOTAL HEAT I.

350°

750

300°

250°

700

200°

150°

650

100°

Volumes in cub. metres per Kg

600

0   1   2   3   4   5

PR

Temperature

0.35 0.3 0.275 0.25 0.225 0.2 *Boundary* 0.175 *Curve*

Lines o

0.9

6 7 8 9 10 11 12

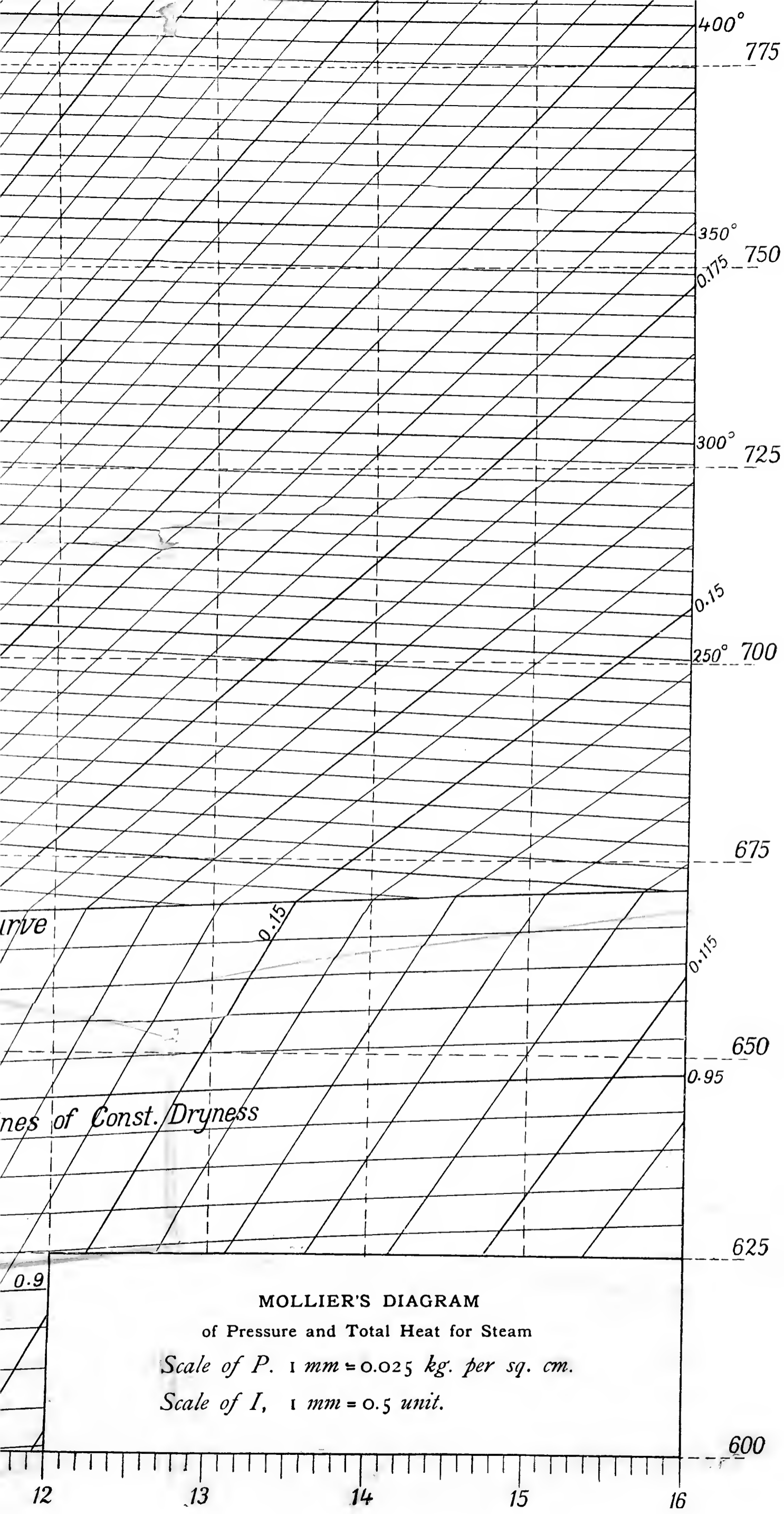*PRESSURE in kilogrammes per sq. centimetre.*

MOLLIER'S DIAGRAM

of Pressure and Total Heat for Steam

Scale of P. 1 mm = 0.025 kg. per sq. cm.

Scale of I, 1 mm = 0.5 unit.