which must be done rapidly, or the steel will cool too much before the rolling is completed. As such bars are rolled in long lengths, the engines run a considerable number of revolutions before requiring to be reversed, and the percentage of steam wasted at each reversal is not so great a proportion of the supply as in the case of the cogging engines, which only make a few revolutions each way. Accordingly these engines are coupled direct to the mill, without any gearing, thus securing a high speed of rolling. The pair of engines figured weigh about 270 tons.

The wheel between the two engines is constructed with a weight on the side opposite the two cranks, in order to balance their weight, so that the engines will stand equally well in any position, and will not pause when the two cranks are on the lower side, which would make them more inconvenient to handle.

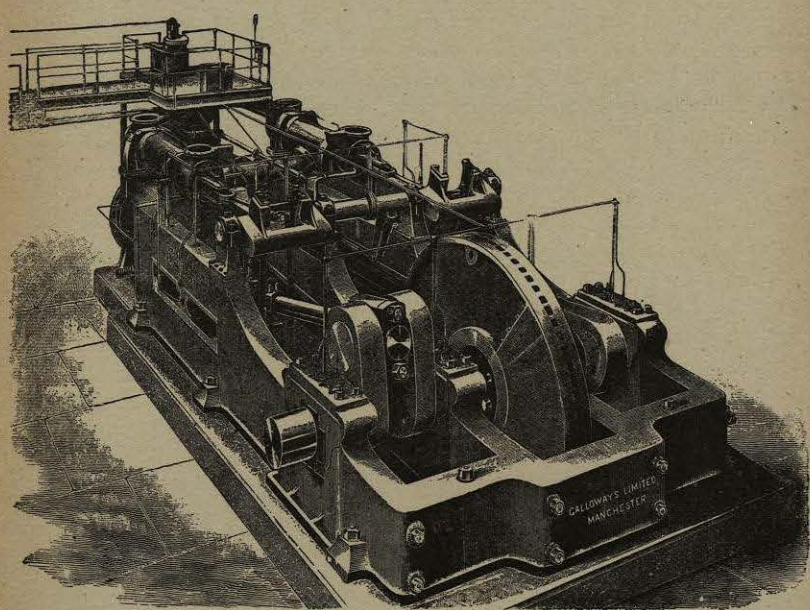
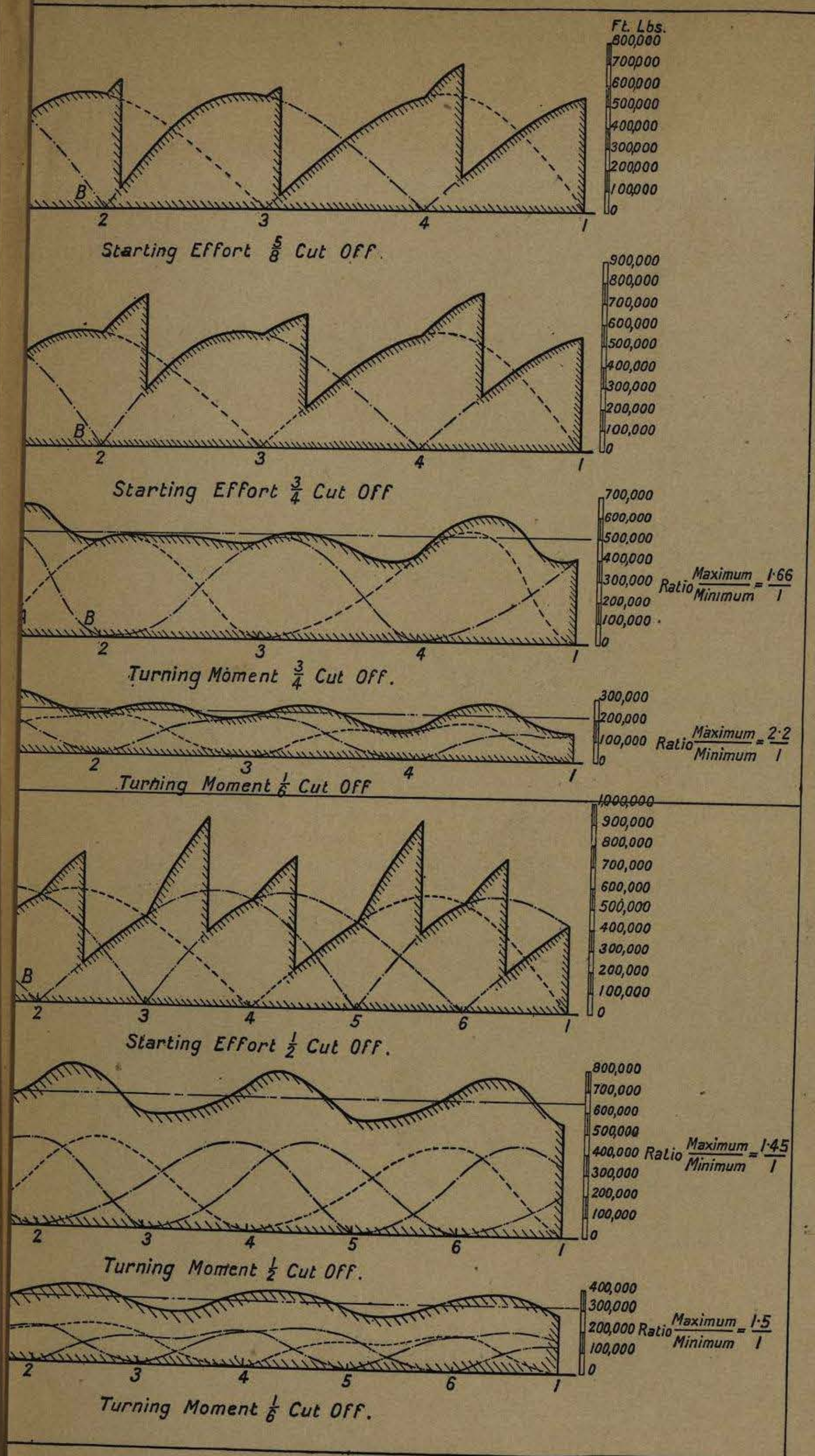


Fig. 425.—Reversing Engines without gearing, for directly driving a Finishing Mill.

Three-cylinder Engines.—The balance weights necessary to prevent the two-cylinder engine "hanging" when the two cranks are on the bottom, add to the revolving masses, and so make the engine less prompt in reversing. The necessity for a balance weight can be avoided by applying three cylinders working on cranks set at angles of 120° from each other, when the cranks and connecting-rods balance each other in any position.

But the use of three cranks has other important advantages, as will be seen by a study of Plate xxxii., fig. 426, which gives in graphic form certain properties possessed by reversing engines. The engines taken for purposes of illustration are of a size commonly employed in steelworks, having cylinders 48 inches diameter by 5 feet stroke, and working at the not unusual pressure of 120 lbs. per square inch at the engines, which are assumed to be non-condensing.

Near the centre of Plate xxxii, fig. 426, are shown, in each instance, an end view of the crank-shaft, from which the position of the cranks is seen,



s, having cylinders 48 inches diameter × 5 feet stroke, running at 120 revolutions per
Weight of one piston with its attached reciprocating parts, $4\frac{1}{2}$ tons.

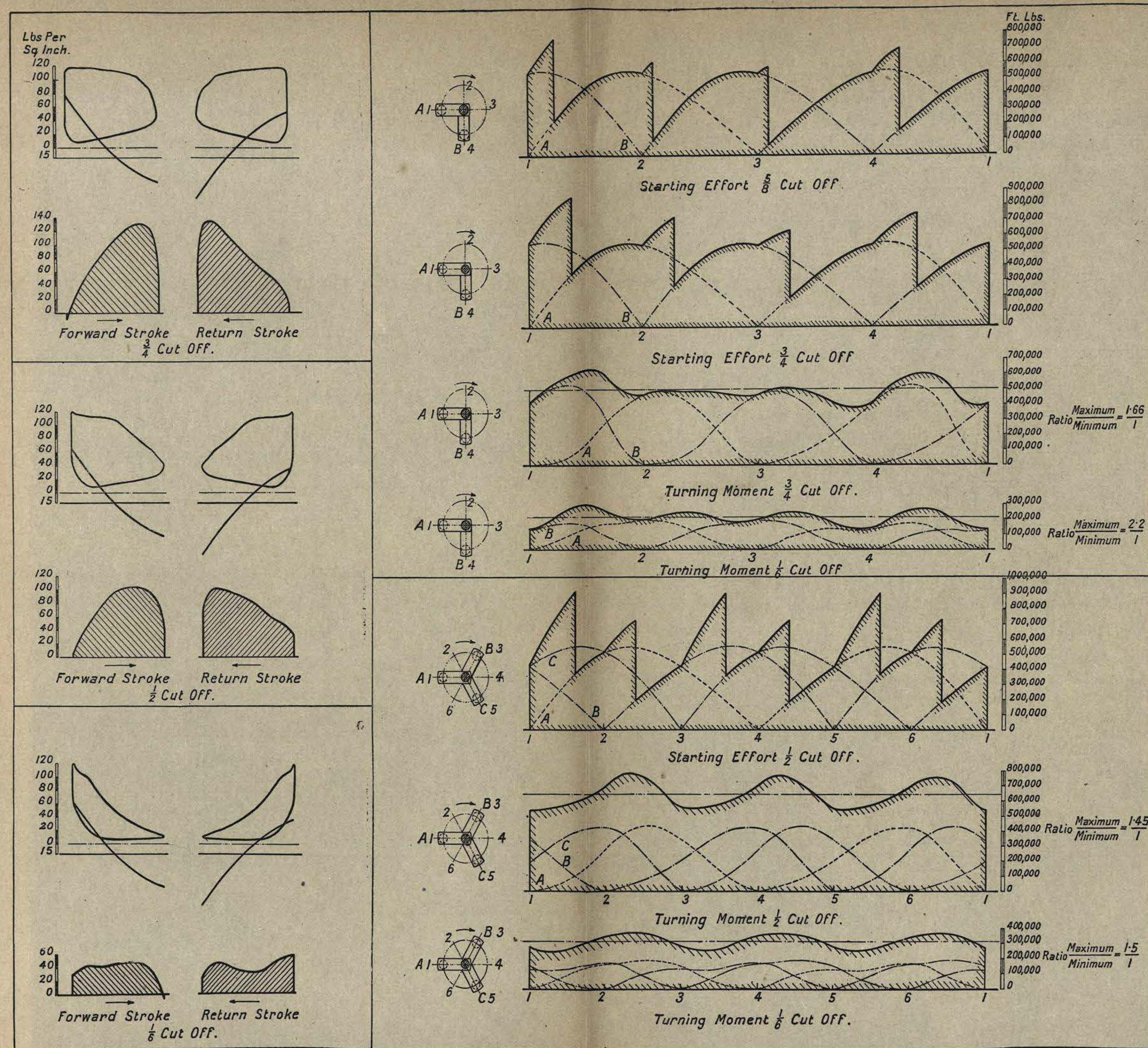


Fig. 426.—Starting Effort and Turning Moment for Two- and Three-Cylinder Engines, having cylinders 48 inches diameter × 5 feet stroke, running at 120 revolutions per minute, with steam pressure of 120 lbs. per square inch at steam chest. Weight of one piston with its attached reciprocating parts, $4\frac{1}{2}$ tons.

showing that they are all revolving in a clockwise direction. The cylinders are in all cases supposed to be to the left of the crank-shaft, so that when the crank-pin, A, travels in the direction of the arrow, over the centre of the shaft, from station 1 to station 3 (1 to 4 in the case of the three-cylinder engine), the piston attached to A is travelling outwards towards the crank-shaft; as the revolution continues, and the crank-pin passes from station 3 (or 4) back again to 1, the crank-pin is passing under the centre of the shaft, and the piston moving inwards, away from the crank-shaft.

To the right of the crank-shaft in each case is a base line representing the distance the crank-pin travels in its circular path from station 1 through 2, 3, &c., until it returns to 1 again. The length of this line then represents the full 360° through which the crank travels in one complete revolution, and any point on the line between the limits 1-1 represents the position of the crank, A, at any intermediate point between the commencement and completion of one revolution, during which period, while the crank has travelled continuously forward in the direction of the hands of a clock, the piston has travelled to and fro, and so also returned to its original position.

The height of the line above this base line represents the tangential turning moment in foot-lbs. exerted on the crank-pin to turn it round in any position, between the commencement and completion of one revolution. The height of the dotted line represents the effort exerted on the crank-pin, A, and that of the chain line that on B, while the full thick line, which is hatched with shading for greater clearness, gives the combined effort of the two cranks, when the effort of each is added to the other.

The height of this thick line above the base thus shows, in graphic form, the extent of the twist which the steam can exert to start the crank-shaft from a state of rest in any position. And if the height of this line above the base is measured by the scale placed to the right of it, the figure obtained will be in every instance the weight in pounds, which, if hung at the end of a cord wrapped round a pulley on the crank-shaft, having an effective radius of 1 foot to the centre of the cord, would exactly balance the effort which could be exerted by the steam, when the crank-shaft was in the position indicated.

The power available is not alike in each quadrant of the circle, because, in addition to the variation in the effective leverage of the crank, the irregularity introduced by the angle which the connecting-rod forms with it, has to be taken into account. The length of the connecting-rod assumed is the common proportion of five times that of the crank.

In the case of the three-cylinder engine, the effort exerted by each cylinder is plotted in the same way, and marked with the letter corresponding to the crank-pin in question, and then the forces of all three are similarly added up to obtain the total tangential twist exerted on the crank-shaft at any point of the revolution, taking the starting point of crank-pin, A, in each instance, as the point in the circle from which to measure the movement of the crank-shaft, and expressing the results in the same manner as in the case of the two cylinder engine.

Having obtained these starting effort diagrams, let us see what is to be learnt from them. In the first two, which relate to the two-cylinder engine, there are four positions of the crank where the slide valve of one cylinder has already closed the steam port against the admission of steam, so that none can enter that cylinder unless and until the other cylinder can turn round the crank-shaft through a sufficient distance to open the valve. It will be seen from the first diagram that a two-cylinder engine having valves which cut off the steam as early as $\frac{5}{8}$ of the stroke, has not sufficient power to do this, because the starting effort at three points is only 150,000 foot-lbs., and at one point only 60,000 foot-lbs. Such an engine, with this cut-off, and with only two cylinders, would, therefore, be unworkable, as it would fail to start in many positions.

The second diagram shows that with a cut-off as late as $\frac{3}{4}$ of the stroke, the minimum effort available for starting would be 190,000 foot-lbs.

Looking at the starting effort diagram of the three-cylinder engine, we find that, although this engine cuts off the steam as early as half-stroke, yet it can exert at least 190,000 foot-lbs. of twist at any point.

Now, the approximate consumption of steam per horse-power exerted by non-condensing rolling mill engines, when cutting off steam of 120 lbs. pressure at $\frac{3}{4}$ of the stroke is 37 lbs.; but, if the cut-off can be made as early as half-stroke, the consumption falls to about 26 lbs. per horse-power, so that the three-cylinder engine would prove far more economical in steam consumption.

To find the twisting moment on the crank-shaft, measured in foot-lbs., at any instant while the engine is in motion, we require to know (1) what is the momentary pressure on the pistons, (2) how this is modified by the momentum and the inertia of the reciprocating parts.

The pressure on the piston at any point is found by means of the indicator, and on the left of Plate xxxii. are given the indicator diagrams which would be obtained from the engines were the steam cut off at $\frac{3}{4}$, at $\frac{1}{2}$, and at $\frac{1}{6}$ stroke respectively.

The speed of the engines is taken to be the not uncommon one of 120 revolutions, equal to a mean piston speed of 1,200 feet per minute.

The weight of each piston, with its piston-rod, crosshead, and guide shoes, and connecting-rod, is assumed to be $4\frac{1}{4}$ tons, a practicable weight if care is exercised in the design of the engines, and equivalent to nearly 5 lbs. per square inch of piston area.

Below each diagram is drawn a line representing the effect of inertia and momentum, similarly to the line, O S R, in fig. 422, from which are obtained the secondary diagrams, shown shaded, giving the equivalent effective pressure at every instant, as in the third diagram of fig. 422, when the inertia is deducted and the momentum is added. Multiplying these equivalent pressures by the effective leverage of crank and connecting-rod at each point, we obtain the turning moment of each cylinder, and adding these as before, we obtain the total twisting moments on the crank-shaft at each instant, when the engines are running, expressed in the same terms as the starting effort. A line drawn through the crests and hollows of the waves which the line forms gives the mean turning moment.

It will be seen that the fluctuation in the twisting moment on the crank-shaft, in the case of the two-cylinder engine, when running at its full power, is as 1·66 is to 1, and in the case of the three-cylinder engine, although it cuts off the steam earlier in the stroke, is only as 1·45 is to 1. Also, that when both engines are linked up so as to cut off the steam at one-sixth of the stroke, the fluctuation is as 2·2 is to 1 with two cylinders, and as 1·5 is to 1 with three cylinders. It is easy to see from this why the three-cylinder engine in practice runs so much more smoothly than the two-cylinder engine.

It is obvious that the nearer the maximum twist is to the mean, the less is the maximum stress induced in the machinery, while any fluctuations of effort are liable to set up tortional vibrations in shafts, which may chance to become cumulative. The more regular is the turning effort therefore the better.

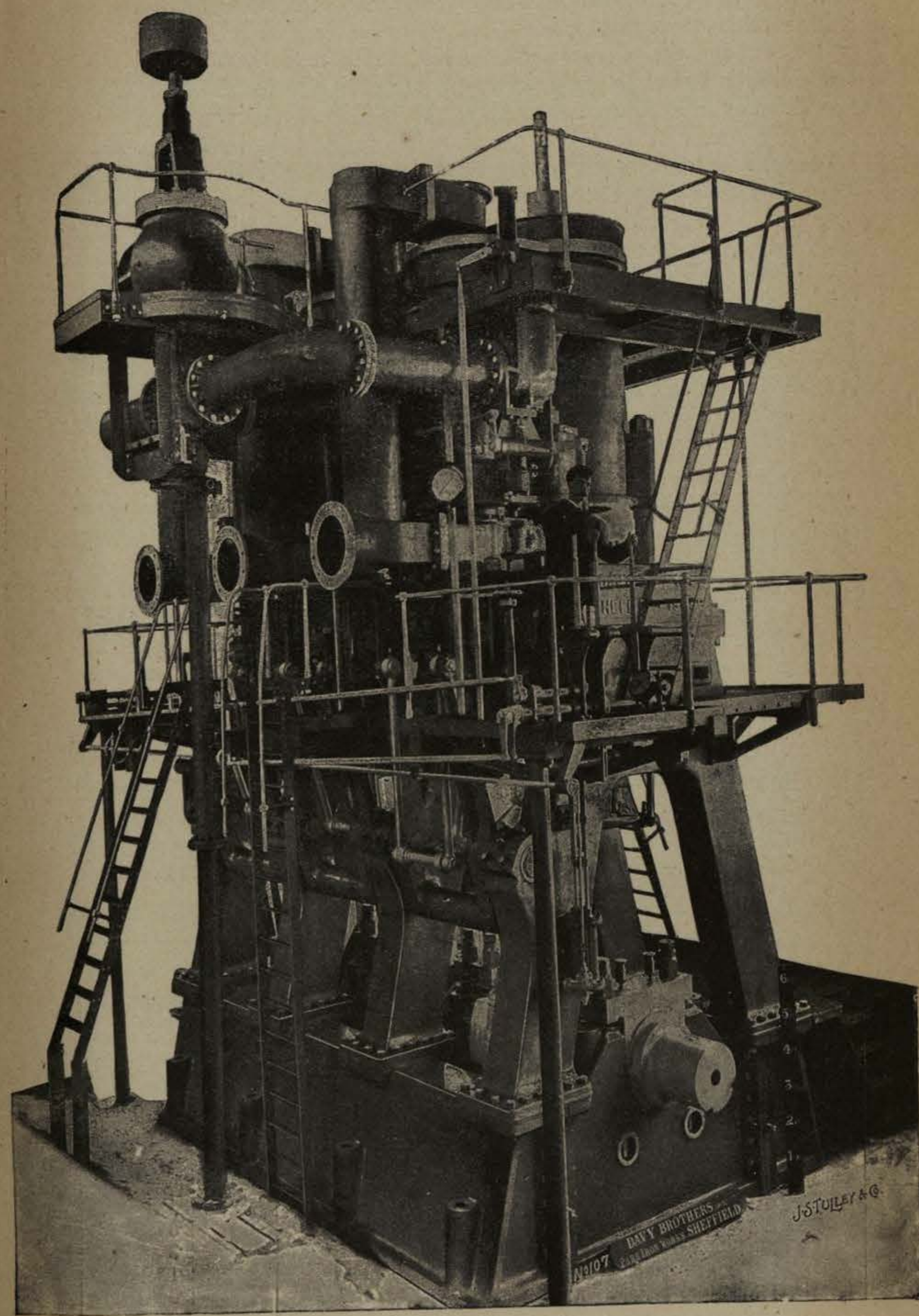


Fig. 427.—Three-cylinder Vertical Reversing Engine.

Bringing the rolls and other revolving masses to rest, and starting them running again in the opposite direction, absorbs a considerable amount of power. As the three-cylinder engine, in spite of its earlier cut-off, can exert at full power a twist nearly one-third greater than the two-cylinder engine, it is, therefore, more prompt in reversal.

To sum up, the three-cylinder engine can be made handier and more economical in steam than the older form, and, being better balanced, causes less vibration, and throws less strain on the machinery it drives.

Fig. 427 shows a three-cylinder reversing engine built by Messrs. Davy Brothers, Ltd., of Sheffield, having vertical cylinders, 48 inches diameter by 52 inches stroke, proportioned for a steam pressure of 140 lbs., and intended to run at 120 revolutions per minute, with forced lubrication to the main bearings. The crank-shaft, which is 21 inches diameter, and in one piece, weighs 21 tons. If the engine were run up to its full speed and power, it would be capable of exerting 16,000 horse-power.

The vertical three-cylinder engine has been recently adopted in several English works. It occupies little floor space, and avoids the wear caused in the cylinders of horizontal engines by the weight of the pistons. Against this has to be set the fact that there often is not sufficient head room in old works to accommodate them, and that they are somewhat less accessible for repairs than the horizontal type.

Economy due to Expanding Steam.—Full economy in the consumption of steam when used at high pressures is only obtainable by cutting off the supply from the boiler after a certain portion of the cylinder has been filled, and permitting the charge of steam imprisoned to expand and drive the piston forward for the remainder of the stroke. Taking steam at the pressures previously given, and cutting it off after the piston has travelled for that percentage of the stroke which, from a commercial point of view, is found to be the most economical, the consumption of feed will be that shown below in column 4. By comparing those figures with the others in column 2, which give the consumption when the steam is cut off at 75 per cent. of the stroke, the saving obtainable by an earlier cut-off is readily appreciated.

TABLE CXIV.—CONSUMPTION OF FEED-WATER BY NON-CONDENSING ENGINES.

1 Initial Pressure of Steam above Atmosphere in Lbs. per Square Inch.	2 Gallons of Feed-Water which must be evaporated per Hour per Indicated Horse-Power Developed when Steam is cut off at 75 per cent. of the Stroke.	3 Percentage of the Stroke at which it is generally found to be (Commercially) most Economical to cut off the Steam.	4 Gallons of Feed-Water which must be evaporated per Hour per Indicated Horse-Power Developed when Steam is cut off at the points given in column 3.
30	5·0	75	5·0
60	4·2	30	3·0
90	3·9	25	2·6
120	3·7	22	2·3
150	3·6	20	2·1
175	3·5	19	2·0
200	3·4	18	1·9
250	3·4	17	1·8

Economy due to Condensers.—In the previous case we have supposed the engine to discharge its waste steam into the atmosphere, which is present everywhere at a pressure averaging 14·7 lbs. per square inch. If, however, the steam is discharged into a closed vessel known as a condenser, where it can be cooled to about 100° F. by contact with a jet of cold water, or with the surface of tubes cooled by a flow of water through or around them, a partial vacuum is formed within the condenser, thus removing the pressure of the air which opposes the motion of the piston, so in effect increasing the mean effective pressure of steam on the piston by about 13 lbs. per square inch. Although the vacuum adds 15 to 20 per cent. to the power developed on the piston of an engine using steam at high pressure, and may add twice this amount if the boiler pressure is low, this is not all net gain. A deduction, rarely under 3 per cent. of the total gross power developed, must be made for the power absorbed in driving the air pump, needed to remove the air, which finds its way into the condenser with the steam and water, or leaks in through defective joints, and which, if not removed, would soon equalise the pressure. Moreover, a larger proportion of the steam entering the cylinder is liquified, and wasted, by the colder cylinder.

The table below shows the consumption of condensing engines, under conditions comparable with the previous table, which gave the consumption of non-condensing, commonly, but improperly, called "high pressure" engines.

TABLE CXV.—CONSUMPTION OF FEED-WATER BY CONDENSING ENGINES.

1 Initial Pressure of Steam above Atmosphere in Lbs. per Square Inch.	2 Gallons of Feed-Water which must be evaporated per Hour per Indicated Horse-Power Developed when Steam is cut off at 75 per cent. of the Stroke.	3 Percentage of the Stroke at which it is generally found to be (Commercially) most Economical to cut off the Steam.	4 Gallons of Feed-Water which must be evaporated per Hour per Indicated Horse-Power Developed when Steam is cut off at the point given in column 3.
30	3·5	25	2·2
60	3·4	19	1·9
90	3·35	14	1·7
120	3·3	11	1·55
150	3·25	8	1·45
175	3·25	7	1·4
200	3·25	6	1·35
250	3·2	5	1·3

Columns 3 and 4 in the above table must be taken as applicable only to compound engines, to be described later, because the points of cut-off in column 3 are considerably earlier than, in practice, are found advantageous, when only a single cylinder is employed.

The advantages of condensing appliances are being now more fully appreciated in steel works, and many works which have previously allowed their engines to exhaust into the atmosphere are putting down central condensing plants, and leading the steam from all their engines into them, thus obtaining considerable reduction in the consumption of fuel under the boilers, or are sending the exhaust steam to turbines to be used over again before passing it finally to the condenser.

However large an exhaust valve may be, it must offer some resistance to the escape of the spent steam. That portion which fails to escape from the cylinder of an engine discharging its waste steam into the air, will have a pressure generally of about 2·3 lbs. per square inch *above the atmosphere*, or 17 lbs. absolute; but if the engine exhausts into a condenser, the steam left in the cylinder will have a pressure of 2·3 lbs. *above the vacuum in the condenser*, or about 4 to 6 lbs. absolute. When compressed, the pressure of steam (absolute) varies inversely as the volume into which it is compressed. Therefore, if the exhaust valve be closed at points such that the returning piston must compress the steam still remaining in the cylinder two, three, and fourfold respectively, the cushion of steam so provided to bring the piston to rest will, in a non-condensing engine, have a final pressure opposing the piston's motion of 34, 51, and 68 lbs. per square inch respectively, while the pressures in a condensing engine with a good vacuum in the condenser can only be 8, 12, and 16 lbs. respectively, under like conditions.

Engines which exhaust directly into a condenser without the intervention of an exhaust turbine must, therefore, be stronger than if exhausting into the atmosphere, while reversing engines, whose pistons run at very high speeds, are particularly affected by the change, and require to be much stronger in proportion than the mere addition of 10 or 15 lbs. to the boiler pressure would render necessary, and, in any case, they must run less smoothly.

Central Condensing Plants.—When engines run continuously in one direction, fluctuations in the demand for power are largely met by drawing upon that reservoir of power, the flywheel, and each engine may then have its own condenser, and drive its own air-pump. But, with a reversing engine, the speed and direction of revolution is continually varying, even if there are not frequent periods of entire rest, and driving the air pump from the engine is then quite impracticable. If a separate condenser, fitted with an air-pump driven continuously by some independent means is employed for each engine, apart from the additional consumption of power this involves, and from questions of cost, maintenance, and superintendence, the supply of steam from reversing engines is so irregular that the condensing plant must either be excessively large, or the vacuum desired will fall off seriously, just at the very instant when there is the greatest demand for it. To avoid this difficulty, the exhaust from all the engines in the works is frequently led to one central condensing plant, which can then be proportioned to deal with the average quantity of steam flowing from all, without fear of the fluctuations in the supply from any individual engine greatly affecting the vacuum obtained.

Any leak in a steam pipe makes itself visible at once, because the steam escapes into the air, but a leak from the air into an exhaust pipe, in which there is a more or less complete vacuum, is quite invisible, and most difficult to discover. An exhaust pipe to convey the steam from several large engines scattered about a big works must be long, and unless the resistance to the flow of the steam is to be excessive must have a diameter of 5 or 6 feet. It is necessarily made from plates rivetted together, and every plate joint and rivet hole is a potential leak. What appear very trifling leaks admit sufficient air considerably to affect the vacuum obtainable, and, therefore, central condensing plants require constant attention to maintain their efficiency.

Exhaust Steam Turbines.—The perfecting of a turbine able to extract power from the steam rejected, after use, by non-condensing engines, has

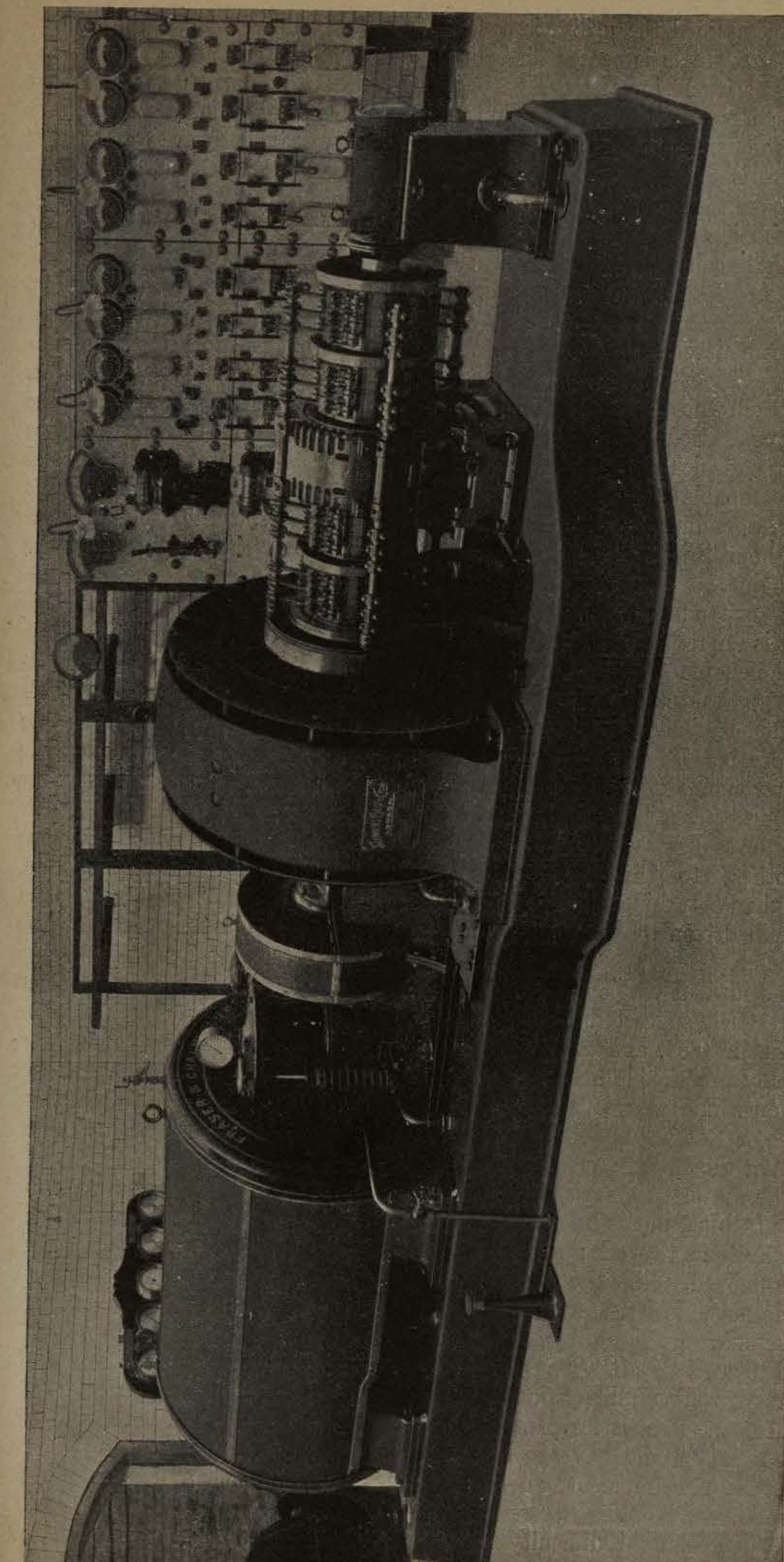


Fig. 428.—Exhaust Turbine driving Electric Generator direct.

greatly simplified the utilisation of exhaust steam in a steel works. When the exhaust steam is passed directly to a condenser a small leakage of air inwards seriously impairs the vacuum, but when the exhaust steam is led to a turbine the pressure in the long lines of exhaust pipes is only 2 to 4 lbs. above the atmosphere, and the quantity lost by leakage outwards, at this pressure, is trifling. Moreover, with a pressure in the exhaust pipes, the leaks make themselves visible, and are much more easily stopped.

In considering the action of an exhaust turbine it is necessary to remember that what really gives the steam the power to perform work is the heat it contains. Take steam at a pressure of 165 lbs. absolute, practically 150 lbs. by the gauge on the boiler, and it will have a temperature of 358° F.; expand this steam sixfold, which is as far as it pays to carry the work in a non-condensing engine, and it will then have a pressure of approximately 13 lbs. above the atmosphere. Its temperature will be 245° F., so that the engine will have utilised a fall in pressure of 113° F. If this steam, less the 15 per cent. or so which will have been reduced to water, were passed into a second engine, capable of expanding the steam until its temperature fell to 110° F., corresponding to a pressure of 1¼ lbs. above a vacuum, this second engine would be utilising a fall in temperature of 135° F., a greater range than the first engine.

But expanding the steam in an engine of ordinary construction to this extent is impracticable. The average pressure on the piston would be so low that the engine would be of impossible size and cost, and the friction of the moving parts would absorb nearly all the power exerted by the steam. This difficulty is met by the exhaust turbine, in which, because there are no reciprocating parts, the moving blades which answer to the piston of the ordinary engine may move at half the velocity of the steam, which is about 30,000 feet per minute.

Without disrespect to the genius, patience, and practical skill of Parsons and others, to whose exertions its present marvellous perfection is due, the exhaust turbine (fig. 428) may not inaptly be described as a glorified windmill driven by the rush of the exhaust steam from the engine to the condenser.

The idea of employing some such appliance for the purpose had often been vaguely suggested, but it was not until Parsons produced his first steam turbine in 1884, and patented his exhaust steam turbine ten years later, that any practicable machine of the kind was available. The first exhaust steam turbine was constructed by the Hon. C. A. Parsons in 1902, and Professor Rateau applied one of his own invention to work with exhaust steam at the Bruay mines in France in 1901.

The steam turbine is designed on very similar lines to the much older water turbine, but is modified so as to suit the enormous head under which the wheel has to work; and, steam being an elastic fluid, a series of wheels are employed. Like the water turbine it is constructed in both the reaction and impulse varieties, and in some modern turbines both systems are used in one machine.

Rateau's Heat Accumulator.—The supply of exhaust steam from reversing rolling mill engines is very irregular and intermittent. At one moment the rush is far greater than the turbine could deal with, and the greater part would have to be thrown to waste through an escape valve, while the next moment, when the engines stopped, the supply would cease entirely.

To overcome this difficulty Professor Rateau devised his heat accumulator,

which can store potential energy in the form of heat, as a flywheel stores actual energy in the form of motion. The accumulator (fig. 429) consists

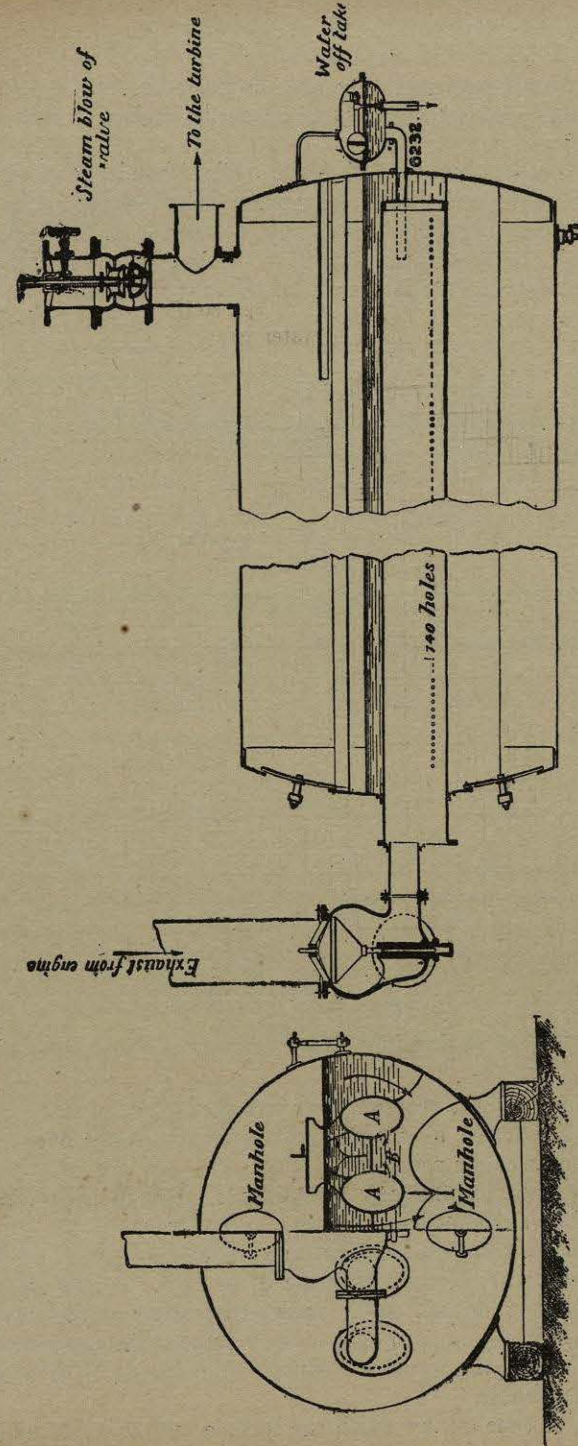


Fig. 429.—Professor Rateau's Heat Accumulator.