CHAPTER XXX.

THE SUPPLY OF POWER—(*Continued*).

(2) Rolling Mill Engines.

Type of Engines Originally Employed.—Enormous improvements have been made in the engines used for driving rolling mills, since the introduction of the Bessemer process. The engines employed in the ironworks in 1856-7 were all beam engines, running at low speeds, with low-pressure steam, usually throttled by small pipes and faulty valves, so that in spite of their imposing dimensions they really gave very little power, one-half of which was often wasted by the heavy gearing employed.

The type of engine in common use was a condensing beam engine, with a cylinder 36 to 48 inches diameter, by 6 to 8 feet stroke, making from 14 to 20 revolutions per minute, which gave a mean piston speed usually of 200 to 225 feet per minute; the pressure of steam carried in the boilers was rarely more than 20 to 30 lbs., often much less; and owing to the low pressures employed a condenser was almost always a necessity. Such an engine gave out 150 to 250 indicated horse-power, that is the gross power exerted by the steam, from which had to be deducted 20 to 30 per cent. for the friction of the engine itself, and generally as much more for the friction of the gearing. One engine usually drove two or three mills.

To prevent the mill being pulled up or "stalled," as the men call it, when the piece was first put in the rolls, a heavy flywheel was necessary; so necessary with the low-powered engines then in use, that when Mr. Ramsbottom proposed to roll plates without using a flywheel, he was told by the mill managers that it was utterly impossible to do so. The power of a flywheel is proportional to the square of the velocity of the rim, which can be safely run up to 5,000 lineal feet per minute. As the crank-shafts of beam engines only made 14 or 15 revolutions per minute, the flywheel, if put on the crank-shaft, would have required to be over 100 feet in diameter to take full advantage of its weight; to avoid such impossible dimensions the wheel was made 20 or 25 feet diameter and secured on a second shaft which was geared to the crank-shaft, so as to run at 60 to 80 revolutions per minute. This shaft was again geared to the mills, which would run at 45 to 55 revolutions if a "forge train," 25 to 35 if a plate mill, and up to 100 or 200 if small merchant or guide mills were driven from the engine, all these various speeds being obtained by gearing up or down from the fly shaft, by means of heavy toothed wheels, which were continually breaking, and so causing serious loss in repairs and stoppage of production. To reduce the time lost while new ones were being cast, enormous stocks of spare wheels had to be kept on hand, while several days were lost in changing the broken wheels and hanging the new ones. These had to be laboriously set true on the rough square shafts by means of wooden blocking expanded by iron wedges, the boring of the wheels and the turning of the shafts being then almost unknown.

Objections to Beam Engines.—There was a belief common amongst the older workmen that a beam engine needed less repairs than was required by "direct acting engines," so-called because the connecting-rod was coupled direct to the piston-rod without the intervention of a beam. This impression was probably due to the fact that the newer types of engines were set to run faster and with higher steam pressures; but if we consider that a cylinder of the size which was allowed for the exertion of 300 H.P. in an old beam engine, often gives out 3,000 H.P. with the speeds and pressures common to-day, we shall find the cost of repairs per useful horse-power exerted is not in favour of the older type. So strong was the prejudice in favour of the beam engine that it was used exclusively to drive the mills in the Barrow Works, which were designed in 1866, and were intended to surpass every works which until then had been constructed.

But the beam engine, apart from its enormous first cost, is not a convenient engine to employ for driving a rolling mill, as the great weight of its reciprocating parts renders it impossible to drive it beyond a very moderate piston speed, 350 to 400 feet per minute being the extreme limit of safety; this necessitates a low speed of revolution of the crank-shaft, and requires gearing between engine and mill, and engine and flywheel.

Limits of Speed of Engines.—The speed of any engine is limited by the weight of its reciprocating parts, which, if too heavy or if driven unduly fast, will break up the engine by the magnitude of the efforts demanded to start them and bring them to rest twice in each revolution of the shaft. If we take a direct-acting engine of the size usually employed to drive a large rolling mill, with a cylinder, say, 40 to 45 inches diameter, by 5 feet stroke, and drive it by some external force at a speed of 75 revolutions per minute, equivalent to 750 feet of piston speed per minute, without admitting any steam at all to the cylinder, we shall find that the resistance to motion offered by the inertia of the piston with its rod, crosshead, and connecting-rod, at the beginning of the stroke, and the momentum stored in the same parts resisting the stoppage of motion at the end of the stroke, with ordinary good design and material, is equivalent to about 30 lbs. pressure per square inch on the piston. Even by using light-coned steel pistons, and employing every possible care in designing, the weight can scarcely be kept below the equivalent of 20 lbs. per square inch of piston. The resistance increases in proportion to the square of the velocity, thus rising at 100 revolutions per minute to from 30 to 53 lbs., and at 125 revolutions to 83 lbs. per square inch.

The forces above mentioned, which are generated *by the mere act of revolving the crank-shaft*, with the engine empty (without steam in it) and doing no external work whatever, are transmitted to the crank-pin and main bearing, and any load caused by the steam pressure is so much *additional* shock to be resisted by the engine framing; from which it is evident that high piston speeds are only possible with high pressures of steam and low weights of reciprocating parts, and, to obtain this, high-class materials having great strength in proportion to their weight must be employed. As the beam with its attachments doubles or trebles the weight of the reciprocating parts, an engine so constructed as to need a beam is evidently unsuited for running at high speeds.

Influence of Inertia and Momentum.—The great effect which the inertia and momentum of the reciprocating parts mentioned above produce on the working of an engine will be most readily understood by a study of the following diagrams, which represent graphically the effects due to these forces when compared and conjoined with those exerted on the piston by the pressure of steam in the cylinder.

In the diagram, fig. 422, let the length of the line A B represent to any convenient scale the travel of the piston of an engine from A to B. At any convenient number of points in the length, A B, let there be set off, to any convenient vertical scale, heights representing the pressure of steam in the cylinder when the piston is at the points selected, as it travels from A to B, and let a line be drawn through the points thus found. Similarly, when the piston reaches the same points on its return journey, let there be set off to the same scale any pressure that may remain in the cylinder owing to the incomplete escape of the steam through the exhaust port, and let a line be drawn through these points also. If the pipes and valves admitting the steam to and releasing it from the cylinder could be both made of sufficient size and to open and close instantaneously, the full pressure of the steam in the boiler would come on the piston at the commencement of the stroke, as soon as the valve admitting the steam opened; and the pressure would remain constant until the end of the stroke, when the exhaust valve would open, and the charge of steam would be released instantly and completely, so that the piston on its return journey would have no "back pressure" behind it. In that case the lines joining the points marking the pressure at each portion of the stroke would form the rectangle A C D B, the line A B being the minimum of no pressure, and the line C D the maximum pressure possible — namely, the pressure in the boiler.

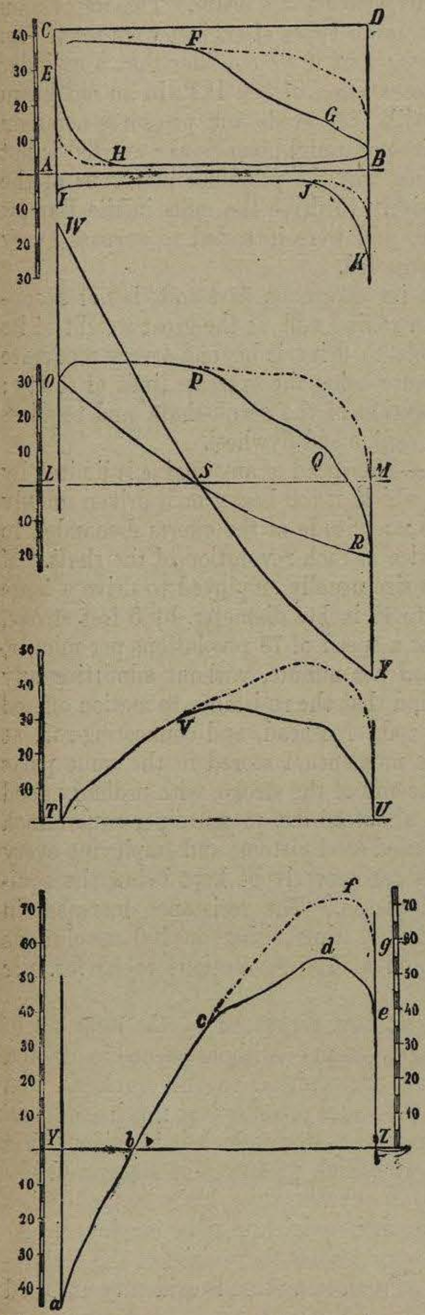

Fig. 422.—Diagrams to explain the Influence of Inertia and Momentum of Moving Parts in a Steam Engine.

Such conditions are unobtainable in practice, the steam being always to some extent "throttled" in its passage through the pipes and valves, and the figure described by the curved lines contained within the rectangle A C D B, shows the character of curve expressing the pressures,

which is obtainable in actual practice from an engine whose steam admission valve closes at about half stroke, and whose waste steam is discharged into the atmosphere: an instrument known as an indicator is employed to record the pressure automatically by describing the figure in question, the upper line indicating the pressure at each instant on the left-hand face of the piston as it travels from A to B, and the lower the pressure on the same face as it travels back from B to A. The steam is admitted at E just before the piston begins its stroke, but owing to the resistance to the flow of steam through the pipes and valves, the pressure is never equal to that in the boiler, and as the piston increases its speed, and the valve connecting the cylinder to the boiler closes, the resistance to the flow of the steam increases, and the pressure falls off until the point F is reached. The connection with the boiler is then closed entirely, and the steam imprisoned in the cylinder expands in volume and falls in pressure until the point G, where the exhaust valve begins to open. During the pause of the piston at the end of the stroke most of the steam escapes up the exhaust pipe, and when the piston returns there is little back pressure until the point H is reached, when the exhaust valve is closed, and what steam remains is compressed by the advancing piston until, under suitable conditions, it may nearly reach the boiler pressure.

So far we have considered only the pressure on the left-hand face of the piston, but on its forward journey from A to B it has experienced the resistance due to the back pressure on the right-hand face due to the waste steam contained in the right-hand end of the cylinder, which back pressure, as it resists the forward motion of the piston, must be deducted from the pressure driving it, to arrive at the real effective force which the steam is exerting. As this back pressure is a minus quantity, we will set it off on the under side of the line, A B, remembering that the sequence of pressures must be read from the curve in the opposite order, because the piston is travelling in the opposite direction at the time. The curve, I J K, set off below A B, now shows the amount of the negative pressure or resistance experienced by the piston at the same periods of its travel, as those at which the upper curve of the diagram gives the forward pressure.

Draw a fresh line, L M, at any convenient distance below and parallel to A B, as a new base line of zero pressure, and at every point on it corresponding to the points selected on the original line A B, set off the difference between the forward and back pressure shown by the lines E F G and I J K. If the quantity is positive measure upwards above L M, and if negative measure downwards below the line, thus forming the curve, O P Q R, which represents exactly the effective pressure at every portion of the stroke tending to hasten or retard the motion of the piston from L to M.

When the crank-shaft is rotating at a uniform speed, we can find the precise speed at which the piston must be travelling at every portion of its stroke. The inertia resisting or momentum assisting motion is equal to the weight in lbs. of the reciprocating parts multiplied by the square of the velocity in feet per second, and if we divide this by the area of the piston, we obtain the amount of these forces in terms of lbs. per square inch of piston area, the same terms in which the pressure of steam on the piston is measured, and can thus compare the amount of these disturbing forces directly with the force of steam on the piston.

We will assume that our engine is running at a not uncommon piston speed of 800 feet per minute, and that the reciprocating parts are made so light that the greatest inertia experienced at this speed is only equal to 30 lbs. per square inch on the piston, a result obtainable by careful designing.

We shall find that the curve expressing the forces of retardation and acceleration will be the line O S R, which is concave upwards as the piston runs out on its forward stroke towards the crank-shaft, and convex upwards as the piston returns home on its back stroke, the difference in the curves being due to the irregularity of motion introduced by the angle formed by the connecting-rod. The distance between the curve, O S R, and the line, L M, will show what is the amount of the inertia and momentum due to the speed and weight of the reciprocating parts at every point of the piston's travel from A to B, expressed in lbs. per square inch of the area of the piston; the portion of the line from O to S set off above the base line, L M, expresses the resistance opposing motion at the beginning of the stroke, and must, therefore, be deducted from the pressure of steam on the piston; and the portion from S to R below L M, giving the momentum stored in the piston hurrying it forward during the latter portion of its stroke, must therefore be added to the effective steam pressure, in order to arrive at the total available force which is actually exerted by the piston and its connections, in the direction of the crank-shaft, at each instant of its motion. To obtain a graphic picture of these forces, set off the vertical distances between the lines O P Q R and O S R, on the third base line, T U, which will give the curve, T V U.

Utilising Momentum.—The first person to clearly recognise the importance of the foregoing facts was the American, Mr. Charles T. Porter, who showed how this property, apparently a serious defect, could under suitable conditions be converted into a valuable aid in equalising the effort exerted on the crank-pin, and astonished the engineering world by exhibiting at the Paris Exhibition of 1867 an engine with a cylinder 12 inches diameter by 2 feet stroke constructed on these principles, which ran perfectly smoothly at the then unheard-of speed of 200 revolutions, or 800 feet of piston speed per minute.

The foregoing diagrams show that by selecting a piston speed and weight of reciprocating parts suitable for the pressure of steam, and a convenient point at which the steam is cut off from the boiler, and by closing the valve to the exhaust at a suitable period, the pressure on the crank-pin may be practically nil at the commencement of the stroke, increasing steadily to its maximum, which may be made considerably less than the initial pressure of steam on the piston, and may steadily decrease to nothing at the end of the stroke, imitating the action of the leg of a skilled bicyclist. Consequently shocks are avoided, the engine turns the centre silently, and the strains to which it is subjected are much reduced, with a corresponding saving in wear and tear.

Direct Coupling.—As we have seen, the smooth running of any engine is dependent on the speed at which the piston travels, and not on the number of revolutions which the crank-shaft makes in a given time, which depends only on the length of stroke selected; for instance, with a speed of piston of 800 feet per minute our engine might be set to run at 300 revolutions per minute with a stroke of 16 inches down to 50 revolutions with a stroke of 8 feet. When Mr. Porter showed how safely to employ piston speeds which were three or four times as high as had been common previously, he enabled the engine builder, by the selection of a convenient stroke and other suitable conditions, to offer to the mill manager engines capable of running at almost any number of revolutions per minute that might be desired. High speeds of revolution are usually required for small mills, and low speeds of revolution for large mills, and so it became possible to couple any rolls direct on to the end of the crank-shaft of the engine, and dispense with gearing entirely. The Americans were quick to perceive the advantages of such a method of driving, and the Porter-Allen engine, a view of which is shown in fig. 423, is still one of the best known and most commonly used engines in the States for moderate powers, while some are in use which have cylinders as large as 45 inches diameter by 5 feet stroke. Many mills, running at speeds up to 200 or 220 revolutions per minute, are still driven direct by this engine, or by one of a similar type.

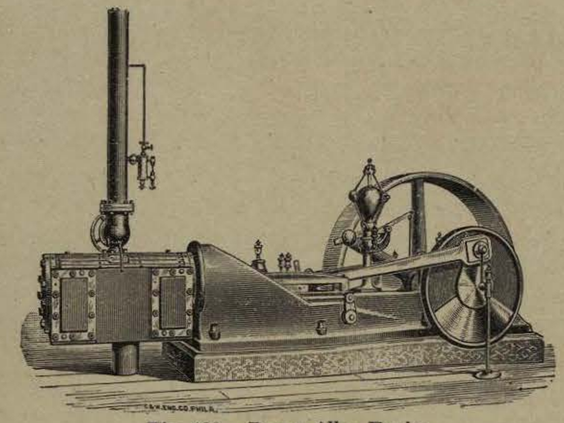

Fig. 423.—Porter-Allen Engine.

Limitations to the Useful Employment of Momentum.—Up to the present we have been considering the running of the engine under one set of conditions only, selected because they are specially suitable to it. Though it is rarely possible to do more than approximate such conditions in daily practice, it is nevertheless most important to know the nature and extent of the disturbance which will be introduced by an alteration of any one of the factors in question, for inattention to these considerations may lead to serious trouble. We will, therefore, consider the effect of altering the point at which the steam is cut-off, the weight of the moving parts, the speed at which they travel, and the steam pressure.

(1) If, instead of cutting off the supply of steam when the piston has travelled half the length of its stroke, the supply is continued for three-quarters of the stroke, we shall find that the pressures during the latter half of the stroke are as shown by the dotted lines (fig. 422); and if the valve controlling the exhaust is not worked independently of that controlling the admission of steam, but, as is frequently the case, is the same valve, or is coupled rigidly to it, the later closing of the steam port involves a later closing of the exhaust port also. This imprisons less of the exhaust steam in the cylinder, thus reducing the pressure and time of action of this steam which is used as a cushion to bring up the piston at the end of the stroke. The effect of this, also shown in dotted lines, is to increase the impulse driving the piston towards the end of the stroke, the consequence of which is a blow and a knock as the engine turns the centre.

(2) An increase in the weight of the moving parts will increase the inertia and momentum in direct proportion to the weight, increasing the height of the point O above L, and increasing the depth of the point R below M, thus decreasing the force exerted at the beginning of the stroke and increasing it towards the end. There are circumstances, such as a very early cut-off,

when an increase in the weight is purposely made, with the express object of equalising the power exerted; but in by far the larger number of cases the great difficulty is to keep down the weight.

(3) The momentum stored in the moving parts varies as the square of the speed, so that any alteration in the speed is of vastly greater importance than an alteration in any of the other conditions. Fifteen or twenty years ago, when steam pressures of 50 to 70 lbs. per square inch were common, piston speeds of 300 to 400 feet per minute were usual, and to-day most engine-builders, where possible, prefer to limit the speed of their engines to 600 or 700 feet per minute, at which speeds the effect of momentum bears a moderate proportion to the power exerted by the steam, and consequently does not too seriously alter the results if the other conditions are departed from.

If, in the case of the engine we nave been considering, in fig. 422, we increase the piston speed 50 per cent.—viz., from 800 to 1,200 feet per minute —we shall increase the effect of inertia and momentum 225 per cent., and the curve, W S X, will represent the extent of the forces thus set up, while the curves, *a b c d e* and *a b c f g*, described on the base line, Y Z, will give the total resultant force at each point of the piston's travel when running with the same steam pressure and cut-off, and with the same working parts, the speed alone having been increased. We observe that during the first quarter or so of the stroke the pressure of the steam is insufficient to drive the piston at the speed desired; therefore, to attain so high a speed it must be dragged forward by the crank-pin which is being itself hurried on by the flywheel (or by a second piston coupled to a crank set at another angle, should there be no flywheel), until the point *b* is reached; then only the piston just begins to perform its proper function of driving the crank-pin, and the direction of pressure on the bearings being suddenly changed, a heavy knock is the result. Towards the end of the stroke the piston rushes on with an impetus that suggests the action of a steam hammer rather than that of an engine intended to turn a crank, and the stopping of the piston at the end of the stroke has to be done by the engine frame, which, if not strong enough, will be broken by the succession of blows caused by these alternate rushes and snatches. The moving parts must themselves be of high-class material to stand the sudden snatches of the crank-pin to which they are exposed, for any increase in their dimensions, seeing it increases their weight, merely adds to the evil, thus practically limiting the speed of any piston to about 1,200 feet per minute, and the advisable speed to from 600 to 800 feet.

It is therefore important to provide that a breakage in the mill, a matter of almost daily occurrence in many works, shall not, by suddenly removing the larger part of the load on the engine, give it a chance of "racing" at a speed much above that for which it has been designed. The inertia of the flywheel prevents a sudden increase of speed, and gives the governors time to shut off sufficient steam to keep the speed of the engine within safe bounds; but should the governors for any reason fail to act, the flywheel itself becomes a terrible source of danger, the stresses in the rim increasing as the square of the speed, so that an increase of speed of 100 per cent. will increase the stresses in the rim fourfold. As flywheels are made of such a diameter as to run normally at speeds as high as are safe, with the object of getting the maximum steadying effect possible from them, it is not surprising that when the engine runs away, the tensional stresses set up in the rim of the flywheel cause it to burst with such violence as to destroy everything in the neighbourhood.

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PLATE XXXI.—Reversing Engines, with Gearing for Driving a Cogging Mill.

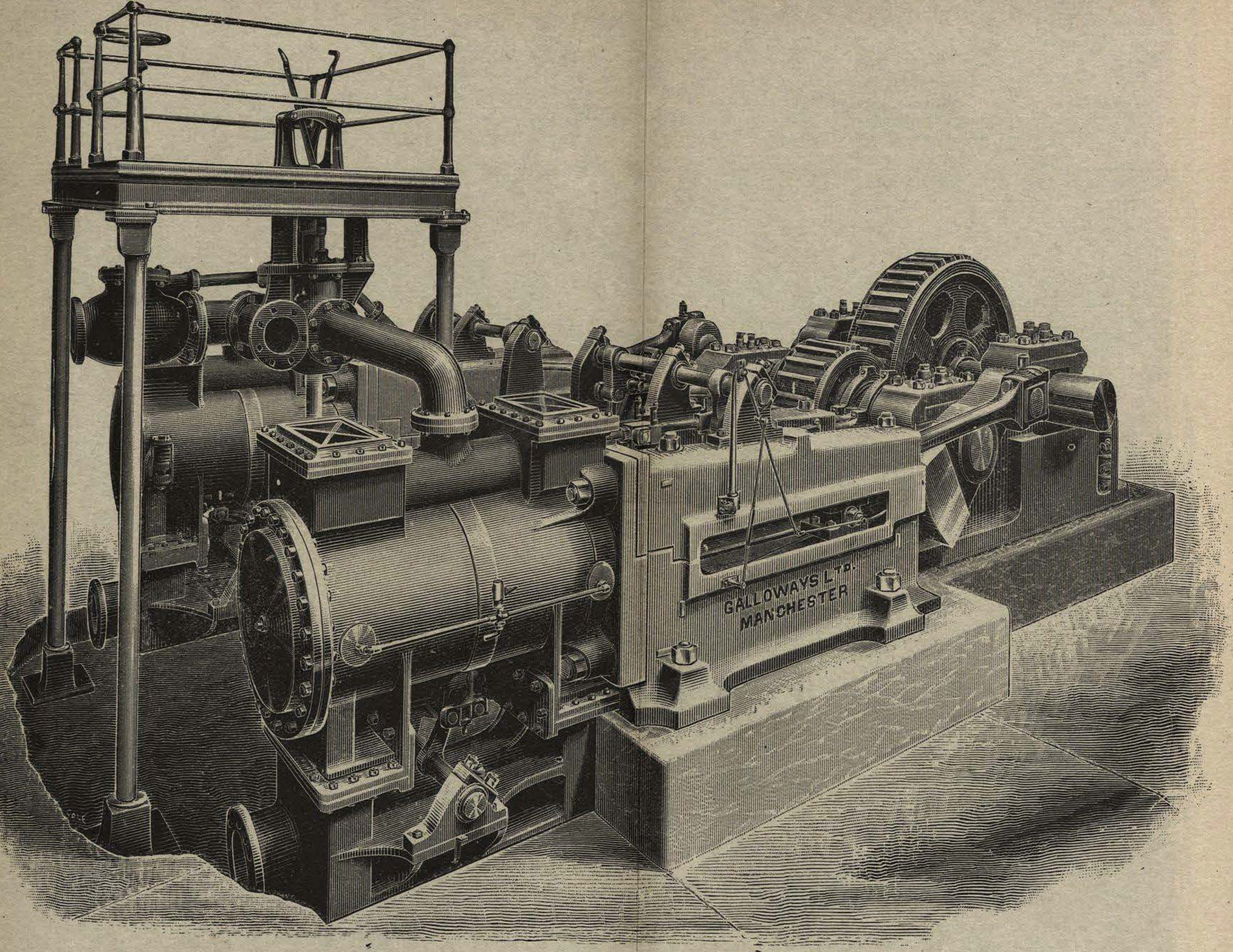

Fig. 424.

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PLATE XXXI.—Reversing Engines, with

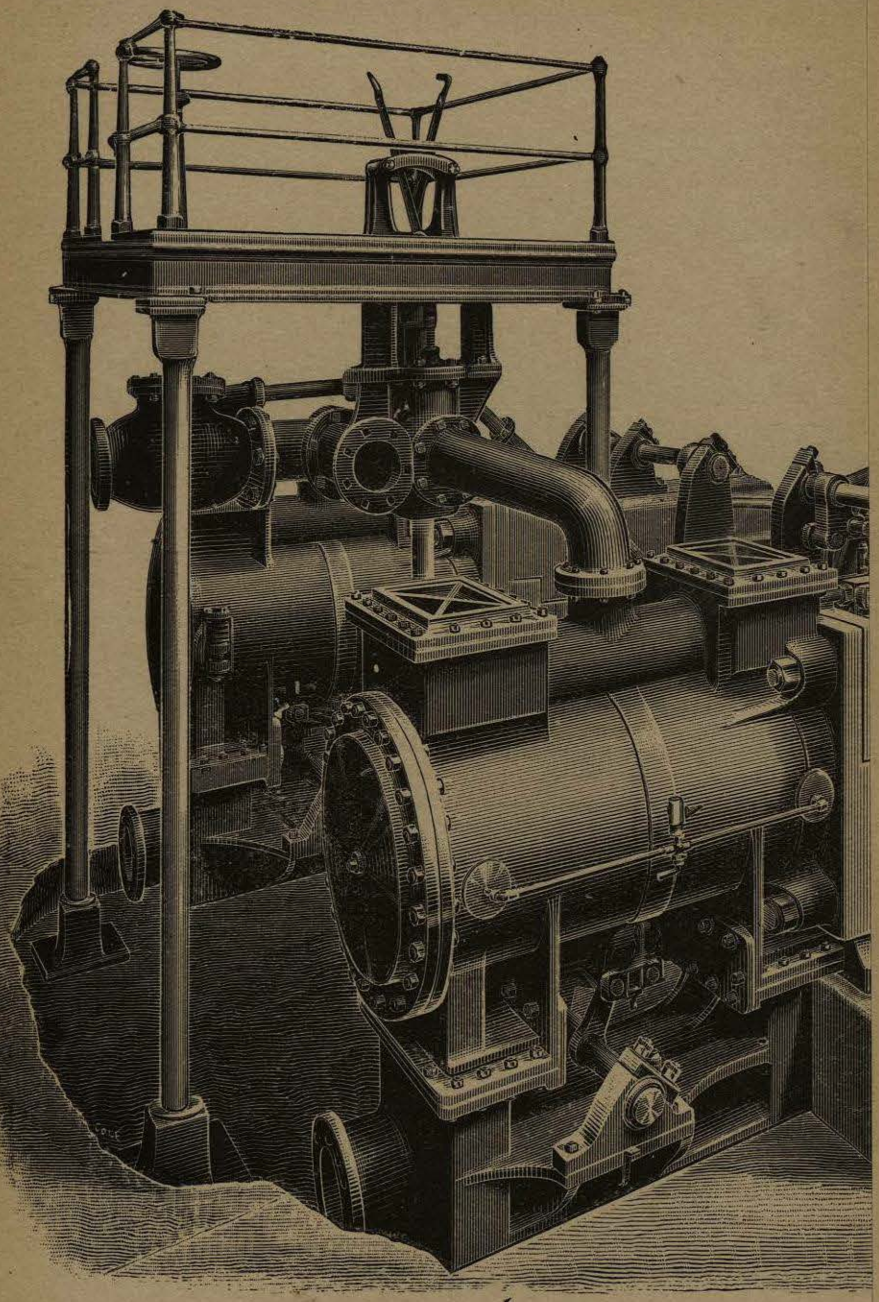

Fig. 424.

(4) In the case of our engine running at 1,200 feet of piston speed per minute, there is evidently not sufficient pressure of steam, by some 45 lbs., to overcome the inertia of the moving parts at the beginning of the stroke. Consequently, all engines to run smoothly at high piston speeds, must be worked with steam at high pressure, and the exhaust valve must be closed early in the return stroke to afford a sufficient cushion of steam to bring the piston to rest, while the engine must be carefully designed and of first-class material throughout.

Additional Stresses due to Reversing Engines.—In engines which run in one direction only, the speed of the piston can be controlled and kept within predetermined limits, by means of the flywheel and governor, but reversing engines, which must run intermittently in either direction, cannot be provided with a flywheel, or prompt reversal would be impossible; directly the resistance is removed by the piece leaving the rolls, the pistons dash away, and if the driver has not been very watchful and cut off the steam the instant before, may at once attain an enormous speed, certainly over 1,200 feet per minute; consequently no engine is subjected to such severe unknown strains as a reversing mill engine, whose speed can be controlled by nothing except the hand of the driver. As the blow struck by any part moving at a speed of 1,200 feet per minute, is nearly equivalent to allowing it to fall freely from a height of 6 feet, to which may be added the full boiler pressure, if the throttle valve is opened again suddenly, and as the whole force of this blow must be resisted by the main bearing, it is not surprising that most of the early horizontal reversing engines broke through their bed-plates between the cylinder and crank-shaft, provision for this insufficiently appreciated force not having been made by the designers of the engines.

To provide for this additional stress it is usual now to avoid placing the cylinders on the top of the beds, where the line of stress between the piston and crank-shaft being above the beds, sets up a cross-bending moment in the beds; the cylinders of reversing engines are now bolted to the ends of the beds, which are immensely increased in depth, so that the line of stress passes approximately through the centre of gravity of the castings, or at least falls within them, and the cross sectional area of the cast iron which has to resist these horizontal stresses has been increased until it is $\frac{1}{4}$ and even occasionally $\frac{1}{2}$ that of the piston area, instead of only $\frac{1}{8}$ to $\frac{1}{12}$ the area, as it was in some of the earlier engines.

Fig. 424 (Plate xxxi.), and fig. 425, which show two reversing engines, both made by Messrs. Galloways of Manchester, will make the construction of such engines clear. The first has cylinders 42 inches diameter, by 5 feet stroke, and is intended for driving a cogging mill, in which great power is essential. It is designed to deal with the heavy section of the ingot, which, being short, soon passes through the rolls, and the speed of revolution is, therefore, not a matter of importance. Accordingly the engine is geared to the shaft driving the mill in the proportion of $2\frac{1}{2} : 1$, thus gaining in power at the sacrifice of speed. The size and cost of the engine required to do the work are thus greatly reduced, and the consumption of steam is less, because, in any reversing engine, a cylinder full of steam must be wasted every time the engine is reversed. The teeth of the wheels are 8 inches pitch by 20 inches wide on the working face, and the weight of the engines complete is about 195 tons.

The other pair of engines (fig. 425) have cylinders 56 inches in diameter with 6 feet stroke, and are intended to drive mills for finishing sections,