No. 1,621.—"Description of Machinery for the Production and Transmission of Motion, in the large Factories of East Lancashire and West Yorkshire." By GEORGE WILLIAM SUTCLIFFE, Assoc. M. Inst. C.E.

THOUGH the subject treated in this Paper is, of necessity, more or less familiar to the members of the Institution, it is written in the hope that it may prove of interest to many who are not directly conversant with the district in question.

## PRODUCTION OF MOTION.

For the production of motion—commonly called power—steam is employed, to the almost total exclusion of any other agent. Water power is not so abundant in England as it is in some foreign countries; but still there are opportunities for its profitable use, if judiciously supplemented by steam power. So many turbines and water-wheels have failed to give successful results, that water motors have as a rule been condemned, though there are several kinds possessing superior merit. These failures have resulted, mainly, either from ignorance of the laws of hydraulics, or from defective proportions and finish of the machinery. Every square mile of freely drained area of watershed in this district will furnish 0·2 HP. per foot of fall (with very little storage), except during a few weeks in summer, when steam machinery is in its state of maximum efficiency, and steam is less required for heating purposes.

The universal type of steam generator is the Lancashire boiler, having a shell 28 to 30 feet long, by 6 feet 6 inches to 7 feet in diameter, the fore end being attached by outside angle irons, the rear end by inside angle irons, or flanging of the end plate; and two furnace tubes about 2 feet 9 inches in diameter, fixed longitudinally below the water level. In the original arrangement the furnace tubes were separate throughout their whole length. Messrs. W. and J. Galloway, of Knott Mill, Manchester, form the rear ends of furnace tubes into one combustion chamber, with a cross section as shown in Plate 3, Fig. 1. When first adopted the cross section was elliptical. All good boilers are provided, at the rear end of the furnace tubes, or in the combustion chambers, with water tubes, of which many forms are made, Galloway's being shown in Figs. 1 to 5. These tubes promote circulation of water, and utilise much heat that would otherwise be lost. The section of combustion chamber shown in Fig. 1 allows almost absolute interchangeability of tubes; it is also much more accessible on the lower side than the previously arranged chambers. Various systems of riveting are adopted for the transverse joints of furnace tubes; but the best, in the Author's opinion, is Mr. Adamson's flanged joint (Fig. 6), made by flanging the edges of the adjoining plates outwardly, then inserting an annular thin plate between the two, and passing the rivets through the three pieces, the whole being caulked. This joint allows the expansion and contraction of the tubes to take place with at least as much freedom as any other joint, and is eminently suited for use in the neighbourhood of the fire, while it presents no rivets or double thickness of plates to the direct action of the fire; it also strengthens the tube against collapse caused by external pressure. The longitudinal joints of the tubes are usually single riveted lap joints, breaking joint with each other, but all below the centre horizontal plane of the furnace tube; they are usually about 90° apart in circular tubes. No riveted joint ought to be acted upon by the flame, either by being situated over the fire or in the upper part of the combustion chamber near the furnace; the thickness of metal along a riveted joint being double the ordinary thickness of the tube plates, the difference between the temperatures on the opposite sides of plates along the joints is double the amount elsewhere, so that straining is caused by the unequal expansion. The longitudinal joints of the shell are double-riveted joints, either lapped or outside-strip butt joints. The transverse joints of the shell are single-riveted lap joints, or, in rare cases, they are butt joints.

The relative merits of drilled and punched holes remain an open question. The rivets have little hold in the drilled hole, and, unless the drills are well sharpened, they tend to cause a "burr," which, if not removed, almost prevents the possibility of making a tight joint. Rivet holes are often punched with little care as to accuracy, or with deficient clearance in the die, and because these are weaker, or show leakage, a case is made out against all punched holes. A boiler is supposed to stand much waste from corrosion before becoming unsafe, and as corrosion does not take place inside the joint, there is ultimately a greater thickness of material along the joint than in the body of the plate, even in those cases where actual grooving does not occur. Of course, if the corrosion is entirely on one side this fact possesses no value; but where corrosion takes place equally on the two sides, the extra remaining thickness along the joint does much to compensate for that removed in making the holes, and in many cases prevents the somewhat greater strength of drilled holes over that of punched holes from possessing any real value. Fig. 7 shows this action.

The end plates are stayed (in addition to the staying effect of the furnace tubes) by two bolts, passing longitudinally through the boiler from end to end, with nuts and cast-iron washers at each end; gusset stays (Figs. 1 to 4) are often added to, or are substituted for, the stay bolts. Four or five gusset stays are in this case provided at each end, some of which are attached to the first ring of the shell, and the others to the second ring. All stays (but more especially the gusset stays) must be arranged to allow of free expansion and contraction of the furnace tubes relatively to the shell.

In the majority of cases the material used is iron. Steel has been often and satisfactorily employed; and with due regard to experience, it may be used slightly thinner under equal conditions, thus offering less resistance to the passage of heat. Great care must, however, be exercised in selecting steel of the proper temper, and in annealing the plates, otherwise there is a risk of cracks forming in the boiler after use, more especially if suddenly or rapidly cooled down.

The furnace is provided with grate bars carried on cast-iron cross pieces. The grate bars are of cast iron, 2 feet to 3 feet long, taper in cross section, with distance pieces cast with each bar at the ends and at the centre of the length. Each bar is cast singly; the bars have been often made in nests of three or four, but without success, chiefly owing to the mass warping. When the grate bars are parallel to the axis of the boiler they can be more easily cleared than when in any other position, but they cannot be perpetually cleared, and the local heating occurs during the intervals. However, no mischief results with single bars during ordinary work, and the pattern appears likely to endure, at least for hand-fired boilers, though the material may be displaced by some other and more durable one. Many other patterns—transverse, zigzag, and with differently-formed apertures for the admission of air—have been introduced, and withdrawn after a brief experience of their disadvantages. These patterns have been devised with a view to increased air admission, but all become filled with cinder.

The production of smoke may be reduced to a small amount by careful stoking. Some firms give a quarterly premium to the stoker for low smoke production; and a mirror is often set outside the boiler-house, so that the stoker may see what quantity of smoke is issuing from the chimney top.

Several systems of mechanical stokers are now largely used, and so far appear tolerably successful when the steam supply is nearly uniform. Some stokers regulate the coal supply by means of indented plates reciprocating across the mouth of the coal hopper, and thus delivering the coal on the dead plate; in this system the bars must be arranged to reciprocate, so as to propel the coal towards the rear of the grate.

Another class of mechanical stoker is provided with a sliding measure, to place the coal in a suitable position for the action of a plunger, which, being propelled by a cam, suddenly throws the coal on the fire. This is not yet completely successful.

Probably the best and most popular system is that in which the coal is withdrawn from the mouth of the hopper by a pair of spiked rollers (occasionally only one), which break up the coal, and drop it on a roller set directly beneath, and provided with measuring recesses. These recesses transfer the crushed coal to a fan or beater, which throws it on the fire, deflecting plates being provided to spread it over the required surface. All mechanical stokers appear more or less incompetent to deal with wet coal, and they are often for this reason inapplicable in cases where they might otherwise be successful. Being driven by the main engine, they are inoperative before the engine is started; it is therefore necessary that means should be provided for allowing the fires to be stoked by hand at such times, also for "banking up," and in cases of casualty.

Each boiler requires provision for the admission of air over the fire, for the thorough combustion of the gases, and for the prevention of smoke when hand fired; this is not so necessary with mechanical stokers, owing to the greater regularity of the fuel supply. Air is usually admitted by means of a perforated door provided with a cover, which is set according to experience and observation, requiring care and discretion to yield good results. Many methods of variable admission—full when first stoked, and afterwards decreasing—have been applied to furnace doors, and to bridges, but with limited success. The usual direction of the flues outside the boiler is such as to allow the gases from the combustion chamber to divide without control, passing along each side of the boiler from rear to front, then to meet and enter one passage beneath the boiler, and pass from front to rear, and away to the chimney. A damper is provided in some part of the passage to the chimney; this damper usually works up and down, being connected by chains over pulleys to a balance weight in the stokehole. Dampers, however, are sometimes applied on a vertical axis, worked either by a shaft and mitre-wheels, or by a push-rod and lever from the stokehole.

A valve is required to regulate the feed supply; the best is the ordinary screw-down stop valve, with the necessary condition that the valve itself is not collared to the spindle, but is left free to drop, and act as a retaining valve at the same time; otherwise a separate retaining valve becomes necessary. A good position for the feed valve is a few inches below low-water level, on either side of the boiler. The feed pipe should be continued 3 or 4 feet inwards, parallel to the furnace tubes, and grooved along one side, care being taken to avoid placing the groove so as to allow the feed water to impinge upon the furnace tubes. As the feed water is now usually admitted at a high temperature, the last condition is not so imperative as when cold feed water was the rule. An apparatus is sometimes required for blowing off scum, which forms on some waters. Floating "scummers" are seldom used. With fixed "scummers" a definite unmistakable mark must be made on or near the gauge glass, at which the water surface must stand when scumming, or the operation is useless. Various waters differ exceedingly in their liability to form scum.

One or two gauge glasses must be provided, with cocks in good condition, to be closed every night. If only one gauge glass is used, a set of four plain gauge cocks will be required instead of the second one. A dial pressure-gauge should be placed on the front of the boiler, so as to be constantly in view of the stoker. It may be attached to any part of the steam space, with the intervention of an inverted siphon, to accumulate water and prevent heating of the instrument.

Two safety valves are necessary, one of which is either a deadweight valve, or a lever valve carefully constructed with knife-edge bearings and a narrow valve bearing, so as to furnish a ready and reliable test for accuracy of the steam-pressure gauge, and to be frequently used for that purpose. The steam-engine indicator is often applied for the same object, and if in good condition it is trustworthy, but on the whole the former test is preferable.

The blow-off valve is usually in the form of a cock with taper plug. A cast-iron casing is occasionally employed in conjunction with a brass valve-plug, and when the mud is blown off under pressure it is often difficult, or even impossible, to close the valve before the boiler is emptied, owing to the fact that brass expands with heat to a greater degree than iron, and thus the heat from the water passing the cock causes it to become fixed.

Until lately every boiler was provided with a dome, but scarcely any at the present time are so fitted, a few feet of cast-iron piping, grooved lengthways, being equally efficient (if not more so) in collecting dry steam.

Many kinds of tubular boilers have been tried, but, so far, without practical success, owing in a great measure to the impure feed water which is almost universal in the district under consideration. This is more especially the case in towns where (as in Oldham) every firm must provide a cooling pond, or work without condensation. A sewer pipe of 4 inches bore is often regarded as providing a valuable water supply. This class of water is not adapted for tubular boilers, even if the construction and arrangements are otherwise perfect, as the tubes become blocked-up by the impurities. Even the water tubes before mentioned, placed across the furnace tubes and combustion chambers, are often stopped up, unless the manager of the works exercises a strict personal supervision of the interior during cleaning operations.

The feed water is usually heated before admission into the boiler, either by the waste gases from the furnaces or by the exhaust steam. In the former case, the system almost invariably pursued is to provide a series of vertical cast-iron pipes in the flue; these pipes are collected at the top and bottom into nests, and the nests are then connected to one delivery and one supply pipe. The pipes are covered with a layer of soot on the outside, and of scale on the inside; the soot is removed by continuously-reciprocating scrapers, while the latter must be removed by a rude process of boring. The flues are arranged so that the gases may be passed at pleasure through this apparatus, which is instrumental in effecting a marked economy by raising the feed water to boiling point. As steam cannot be raised to an appreciable extent with it, the possible economy is strictly limited, and is usually somewhat less than is supposed. Messrs. Green, of Wakefield, make the best apparatus of the class, and it is applicable to condensing or non-condensing engines.

The exhaust steam is often used as a feed-heating agent in non-condensing engines. In some cases the feed water passes the boiler feed-pump before entering a worm, placed in an enlarged part of the exhaust pipe; in other cases the exhaust steam and feed water are intimately mixed in entering a vessel somewhat similar to an ordinary injection condenser. In the latter system, much suspended, and some dissolved, impurity is removed from the water by the use of the heater; but, on the other hand, trouble is sometimes caused by the admission of an excessive quantity of tallow into the boiler. The water must then be made to pass the pump after leaving the heater, which causes frequent trouble unless the pump is well designed.

Feed pumps often possess a defect which, annoying in any case, is especially so when they are required to pass hot water. The intermediate space between the suction and discharge valves ought to be at a lower level than the discharge valve, so that all air may tend towards that point, and pass away. Even the small space existing around the ram—where the barrel does not fit the ram to a level below the passage to the delivery valve—is often prejudicial when working at a high pressure, from the expansion and contraction of its contained air with the motion of the ram. On referring to the section of the feed pump in the lower part of Fig. 11, Plate 4, it will be seen that the cover to the suction valve is projected downwards to reduce the intermediate space and to avoid an accumulation at that point.

In many works a small engine combined with a pump, or one of the crankless steam pumps, is substituted for a feed pump connected to the engine. This arrangement at a low speed feeds the boilers, at a moderate speed washes out the boilers after cleaning, and at full speed acts as a steam fire-engine. These combined engines are of various capacities, capable of pumping from 150 to 800 gallons per minute at ordinary speed. Neighbouring works frequently agree to give mutual assistance in case of fire; and when the pipes are well arranged a large quantity of water can be quickly brought to bear on any given point.

Greater care is necessary in these steam fire-engines than in feed pumps to prevent air accumulations, as time is too valuable (when a fire breaks out) to allow for priming before water can be forced out. The steam pumps are usually in, or near, the stoke-hole, so as to be under the easy control of the stoker; but when no room can be spared for such a pump, a Giffard's injector is often employed, either alone, or as an auxiliary to the boiler pump in the engine room. The substitution of an injector for the feed pump admits of the omission of a feed-escape valve, which is otherwise necessary.

Such is a sketch of the boiler and other appliances ordinarily found in the boiler house. In conclusion it may be stated, that the number of boilers used for any given power varies widely according to their age, the pressure, and the condition of the feed water, &c. New boilers work safely at a higher pressure than old ones, and less frequently require to be laid off for repairs. Bad feed water also causes boilers to be more frequently laid off, for cleaning in case of muddy water, or for repairs when water is obtained from mines, or is contaminated by chemical residues.

In ordinary work the Lancashire or Galloway boilers of the present day appear to give the highest evaporation when burning 18 to 20 tons of good coal per week. With interest on outlay and all items of cost added, power appears to be more cheaply supplied when the boilers consume about 24 tons per week of fifty-six hours; the amount of water evaporated, and therefore effective HP., varying with the state of the boiler and other appointments.

In current practice the steam engine is usually of the horizontal class. The power required for any works not already in active operation is not easy to calculate, and is often absolutely impossible, owing to the machinery supplied by different makers for the same purpose differing widely in the amount of power required to propel it. A prevalent pattern is that in which four cylinders are used, one high-pressure cylinder 18 inches in diameter, and one low-pressure cylinder 36 inches in diameter, acting on each crank at the two ends of the crank shaft, the stroke usually being 6 feet; the two cylinders on each side of the centre line forming separate compound engines. The steam ports are ample, $1\frac{1}{2}$ inch by 12 inches being frequently adopted in the high-pressure cylinders, with an ordinary slide valve, $1\frac{1}{2}$ inch lap, 6 inches travel, cutting off at about $\frac{3}{4}$ stroke. The steam passes from this cylinder to one of the larger bore without any intermediate receiver, except that constituted by the intermediate pipe. This cylinder has ports 3 inches by 24 inches, lap 2 inches, travel 10 inches, cutting off at about $\frac{5}{6}$ stroke. Both cylinders are set on one common centre line, and often have one piston rod with the two pistons cottered on; sometimes two piston rods are used, and joined in one piston by means of a cotter through both rods; at other times two rods are used and jointed outside the cylinders; but as the joint cannot enter either cylinder, the total length of the whole engine is thereby much increased. Some of the best engines, of nearly the above dimensions, carry almost the whole boiler pressure of 80 lbs. on the piston, indicating about 600 HP., with a consumption of little more than 2 lbs. per HP. per hour, including steam for heating purposes in the works, of which no separate account is kept.

Cut-off motions are often applied to new engines, frequently to be removed after a few years' use. The Corliss valve has not been so largely employed in this country as it deserves, or might have been expected to be. Messrs. Galloway apply a cut-off motion on a similar principle to the Allen engine, and with success. The leading feature is that the eccentric centre is in the same plane through the crank-shaft centre as the crank-pin centre. The clip of this eccentric is held by a link working on a stud at a distance below the centre plane, and is provided with a slot, in which a block is carefully fitted to slide vertically, being actuated by the governor lever. This block carries the stud which fits the end of the link that works the valve spindle. The whole arrangement ensures a variable cut-off and a constant lead or steam admission, one valve only being used for steam supply.

A variable cut-off motion is not of equal value in all classes of work. A cotton or woollen mill usually works up to its maximum capacity, while a saw mill or machine shop is liable to sudden and great variations in the power required. In the former case, there is no necessity for any reserve force; but in the latter the absence of a reserve force would cause the engine to stop. As a waste of power is caused by working on the throttle valve, a good cut-off motion is a necessary part of an economical engine exposed to irregular work. In all cases, however, a simple variable cut-off motion is a valuable addition to an engine; while (where not absolutely necessary) a complicated one may introduce more evils than are compensated by its beneficial properties. Frequently a throttle valve may be found on engines which are supposed to be fitted with an efficient cut-off motion, also worked by the governor.

Recently it was reported that a cylinder 18 inches in diameter had only 3 square inches of steam opening in maximum condition, though the actual opening of the steam port was 11 square inches. This was caused by adding lap to the valve, so as to cut off at one-third to one-fourth of the stroke, with a view to an increased economy. The exhaust opening is, of course, not visible, and the inevitable early release is not noticed. It may easily be seen that this must result in extravagant working; yet this condition of affairs is of frequent occurrence in small works, where usually there is no indicator.

The proportionately great length of the cylinder causes the adoption of short ports and separate valves at each end of the cylinder. The exhaust steam is thus in contact with the cylinder over an excessive amount of surface; and the amount is often increased by continuing the exhaust passage as a belt to the bottom of the cylinder, so as to place the exhaust and intermediate pipes out of sight in the foundations. This is shown in the exhaust-passage arrangement of a low-pressure cylinder, Plate 4, Fig. 13. The valve spindle is generally carried through the valve, a tube for that purpose being cast in each valve. The engagement may be either by locked nuts, collars (both of which are shown in Fig. 15), or by a tube fitting the valve-spindle, and bearing against the inner surface of the nuts, preventing them approaching sufficiently near together to hold the valve. In all these arrangements the valve must freely lift to a slight extent to let the water from the cylinder escape. Yokes are now seldom used for the valves of engines of this class. The laps of valves are often cut off obliquely, or to a V shape, to allow of the more gradual admission of steam.

The usual valve motion is produced by eccentrics on the crank shaft, separate eccentrics being used for each cylinder, so as to give greater range of adjustment. The eccentric sheaves are split and held together by turned bolts, and the clips are of cast iron (Fig. 15a) often (though unnecessarily) bushed with brass. When the crank pedestals are of sufficient length, it usually occurs that the eccentric rod is farther from the centre line of the engine than the valve spindle; some method is then necessary for the connection of the two. The best connection is by means of a short, stout, cast-iron lever, carried by a rocking shaft, and with a stud through the top boss, one end of this stud engaging the eccentric rod, and the other end the link which is attached to the valve spindle. A more usual and elegant method is to provide two separate wrought-iron levers keyed on one shaft; but in this case the torsion of the shaft and the deflection of the levers cause the motion to be unsteady. Where the eccentric rod is of great length, it must be provided with a joint near the centre of the length, and a supporting lever. A good joint for valve spindles and eccentric rods is shown in Plate 3, Figs. 8, 9, and 10, care being taken in employing ample proportions. The valve spindle is usually supported by a bracket outside the stuffing-box, a good packing space being left. A solid cast bracket, bored to fit a bush of cast iron, which fits the valve spindle (and is easily renewed when worn), is a very successful form of valve-spindle guide.

It may here be noticed that in some cases (as in Figs. 11 to 14) the larger or low-pressure cylinder is nearer to the crank shaft than the smaller or high-pressure cylinder, while in others the reverse order is observed. The valve gear is more compact in the latter case, as the eccentric nearest to the engine centre works the valve spindle; while in the former the reverse order obtains, two levers on one vibrating shaft being sometimes quite necessary. The valve gear of the rear cylinder ought always to be arranged to allow the rear cover of the fore cylinder to be removed for inspection and cleaning, without interfering with the valve gear in any way, and without demanding an excessive intermediate space between the two cylinders. It is obvious that this condition can be much more easily fulfilled when the larger cylinder is in the rear position, because, first, the cover which requires to be left free is then the smaller one; and secondly, the valve gear is farther from the centre of the engine, and therefore not so obstructive.

Slide valves are seldom balanced to relieve the pressure on their faces, and consequently require powerful gear to drive them quite steadily. Many gears transmit an unsteady motion to the valves.

The system of framing, in which the working strains are directly opposed, has not yet met with extensive adoption in the district. The leading idea is to provide a bed plate of sufficient weight and strength in the estimation of the builder and the owner. This bed plate has an approximately flat upper surface, prepared to receive the loose crank-pedestals, the slide bars or parallel motion preparation, and the two cylinders, the common centre line of which is kept as low as possible in order to reduce the bending action. Though not strictly correct in principle, this class of bed plate is successful in practice, and has several good features. Two different arrangements of the end carrying the crank shaft are extensively adopted, both of which are shown in Fig. 12, one in full lines, and the other in dotted lines. When the part directly beneath the connecting rod is carried far towards the crank shaft, a recess becomes necessary, in which the connecting rod is to work; and this is essentially necessary when the centre of the cylinders approaches the top of the bed plate. This recess is frequently condemned as an unworkmanlike proceeding, but its use is often compulsory. When the bed plate exceeds 30 feet in length, it is more convenient, for casting and transport, to cast it in two or more lengths, which are then connected by dowelling. The bed plate must be held to the foundation by a sufficient number of holding-down bolts; where practicable these should be arranged at intervals not greater than 6 feet apart; but this is only imperative near the crank shaft, where, however, there is seldom any obstruction. Where the bolts cannot be placed at intervals of about 6 feet, the builder must use his discretion as to the proper arrangement to be adopted. Lewis bolts are sometimes used, but long bolts, reaching to the bottom of the stone foundations, are to be preferred, and are almost universally used; a hand hole is provided which allows access to the lower end of the bolt, and is occasionally convenient in permitting the bolt to drop a short distance. The preparations for all attachments are planed in good work.

The air pump is generally worked by a bell-crank lever, by means of a pair of short links connecting the top end of the lever to the crosshead; therefore, the air-pump centre is nearly under the centre of the crosshead. The bell-crank lever shaft is carried by a pair of pedestals, which are either secured in or to the stone foundation, or to a pair of cast-iron frames, fastened to the lower side of the bed plate. This arrangement is only adopted in engines possessing slide bars. The pump rod requires guides, either in the form of slide bars or a trunk; if slide bars are employed, they must be sufficiently far apart to permit the free withdrawal of the air pump, cover, and bucket. The feed pump is most advantageously worked by a lever on the end of the air-pump rock shaft, as shown by its situation in Fig. 11. When this shaft is abolished, as when horizontal air pumps are employed, the feed pump may be worked from the crosshead; but this proceeding never fails to cause the crosshead to follow a crooked course, unless special means be taken to prevent it.

If a horizontal engine be provided with a parallel motion, the air pump is often placed below the engine-room floor, to the rear of the cylinders. There it is actuated by a bell-crank lever, but the top of the lever is worked by a connecting rod from the centre bar of the parallel motion. The connecting rod passes beneath the cylinder or cylinders, and the whole arrangement is directly derived from that of the beam engine, with the addition of the bell-crank lever to change the direction from horizontal to vertical motion. This is more usually adopted where only one cylinder is placed on one frame, otherwise the connecting rod becomes excessively long.

Vertical air pumps are firmly attached to the stone foundation beneath the engine-room floor, and are provided with brass foot valves, indiarubber bucket valves, and indiarubber delivery valves, either whole or in the form of a number of smaller circular valves, attached to a cover plate provided with a stuffing box, through which the pump rod slides. The acting surfaces of guards for indiarubber valves are often curved in vertical section to a radius equal to their diameters, in which case very little lift is allowed at the centre of the valve. Many owners prefer flat guards, in which the lift is equal throughout the valve, and the friction on the stud greater; therefore the wear in the eye of the valve is greater, but the tendency of the valve towards breaking across is reduced. The preference is often ruled by the action of water on the rubber. The outer surface of the bucket is sometimes solid; or it is packed with wood, of which alder is the best; more frequently it is provided with metallic packing. The part of packing which comes into contact with the bore of the pump barrel must be of brass, which, though actually wearing faster than iron, is not damaged by corrosion, and does not damage the bore of the barrel to the same extent.

Horizontal air pumps have recently been applied to a large number of horizontal engines. Some are double acting, and others single acting, very similar to that adopted in the Allen engine. Unless great attention is paid to the piston in double-acting air pumps, the leakage may become serious; and the same may also be said of the pistons in steam cylinders. The single-acting air pump has a ram or plunger working through a stuffing box into the pump and into the atmosphere. The box should be packed by first placing in it a little flax or hemp, then a metal ring of H section, and afterwards more flax or hemp to fill the box. The ring must have one plane face against the lower and one against the upper fibrous packing, and also a few holes bored through the distance piece, or what may be called the web. A pipe of $\frac{1}{4}$-inch or $\frac{3}{8}$-inch bore enters the stuffing box opposite this ring, and conveys water into it from the hot well. The pump is thus found to make and maintain its vacuum better than without it, and the gland may be slacker in the stuffing box, thus diminishing the friction. Horizontal air pumps are chiefly employed where some obstacle prevents the use of a vertical air pump below the floor; they might be more frequently adopted with advantage. From their usual position behind the cylinders, and on the same bed plate, they possess one objection, that when a short stop is made during ordinary working hours the injection water ought to be stopped. This, however, is often omitted during the confusion attendant on such stoppages, when, first, the air pump cavities are more or less filled, and then the water overflows into the cylinder, ready to cause mischief if the start is suddenly made. This may, of course, be obviated by raising the exhaust pipe, and by opening the water-escape valves on the cylinder, so as to destroy the vacuum by the admission of air. The injection valve need not then be touched (unless the water is delivered under pressure), which is an advantage if the valve is closely adjusted to give proper opening.

In some cases, two cylinders only are employed, whose equivalent diameters are 25-inch and 48-inch, or 50-inch, with a stroke of 6 feet as before. Each piston propels a separate crank on the opposite ends of a crank shaft; the crank, which is driven by the high-pressure piston, being set 135° in advance of that driven by the piston of the low-pressure cylinder. This arrangement is somewhat cheaper in first cost and maintenance, while it is little, if at all, inferior in producing an equable motion. When the cranks approach positions directly opposite to each other, there is a small saving in coal, at the sacrifice of some degree of equable motion.

Steam cylinders are usually moulded in loam, the inside being formed by a separate core; both mould and core must be thoroughly dried, either in a stove or by a special fire built underneath. The cylinders are moulded and cast in a vertical position, whatever position they may be intended to assume when complete; the upper end in casting is left longer, forming a head, which is cut away when the casting is bored, and leaves the effective casting more dense in texture. The ports are made as direct as possible, to reduce the clearance (Fig. 15); the slide valve is then divided into two; also two separate exhaust ports are required. As before mentioned, this causes the exhaust passages to have a large area of surface in contact with the cylinder body, and leads to a larger area of valve and increased friction, unless (as is seldom the case) the valves are balanced. It may be noticed that a cylinder, 18 inches in diameter by 6 feet stroke, is a somewhat awkward casting; and the ports, if a single valve was employed, would require more delicate treatment than moulders usually bestow upon any work, so that practically the short ports are most successful. The covers project 2 to 3 inches into the cylinder, because the port must reach to the cover, and there must be sufficient material outside the port to hold the studs of the cover. The studs ought to avoid the horizontal and vertical planes through the centre of the cylinders; this may also be advantageously extended to the covers of the valve chests. The covers must be provided with ample stuffing-boxes, so as to avoid frequent replenishing, and also the friction consequent on screwing down the gland excessively, which is necessary to hold the steam if the boxes are shallow. The cylinders must be carefully bored in the first instance, and the bore tested throughout; they also require re-boring from time to time, and as the covers must still fit the bore, the cylinder is counterbored in the first instance, $\frac{3}{4}$ inch to 1 inch larger in diameter at each end than the working bore of the cylinder; this counterbore is not afterwards touched. The parallel working bore of the cylinder is of a length equal to the stroke + the width of one piston-casing ring, where two casing rings are employed without an intermediate flange. By this means each casing ring works beyond the parallel bore of the cylinder at each end of the stroke to the extent of half its width, thus preventing the formation of shoulders in the bore of the cylinder.

An efficient elastic piston of the spiral class is made by many makers, and is well received by owners. The steam piston is shown in section in Fig. 15; it is provided with a loose junk ring, to allow access to the packing. The junk ring is secured by set screws, with brass nuts and brass washers; there are two or three set screws, which may be removed, and the holes may receive forcing screws for easing off the junk ring. Beneath this junk ring a circumferential groove is turned to receive the packing. The packing consists of two parts: an inner spiral spring and an outer pair of casings. The L-shaped casings only are shown; they are turned of a diameter somewhat larger than the bore of the cylinder, and are cut obliquely across, one part to the right hand, and the other to the left hand; they are then forced apart, and also radially from the centre, by the inner spiral spring. Some makers employ a special kind of cast iron in the construction of these spiral springs, while others employ steel. The casings are made from good free cast iron when for steam cylinders, and from brass when for air-pump buckets. The casings and spiral spring are obstinate in "putting up," but after a little practice this is overcome, steel being the more refractory in this respect. When well made, and the bore of cylinder in fair condition, these pistons and buckets are invariably successful.

Each cylinder is secured against longitudinal movement at one end only; this is to allow the expansion and contraction caused by variation in temperature to take place freely, as the other end is only bolted down, and the bolt holes are larger than the bolts. Formerly, the practice was prevalent to hold the cylinder by separate feet at each end; but latterly, the foot is continuous throughout the length, or is placed in the centre and made of ample length. Longitudinal movement is prevented by a key, sunk one-half into the bed plate and the other half into the cylinder foot, either at the centre of the length or near the end towards the crank shaft, so that the cylinder and the piston rod expand simultaneously when heated. Should the arrangement or dimensions of the valve chest preclude the existence of a foot in the ordinary manner below the valve chest, the bolts holding the cylinder to the bed plate are sometimes passed through the valve chest, in which case great care must be exercised to avoid contracting the steam or exhaust passages. It is occasionally convenient to continue a bed plate holding-bolt through the foot of the cylinder, and thereby save one bolt, introducing a bed plate holding-bolt where it would be otherwise impossible to do so.

Each cylinder should be provided at both ends with a valve, opening outwards, and of area sufficient to allow the escape of any condensed water which may exist in the cylinder. These valves are often closed by weights and levers, but the inertia of the weights frequently prevents the prompt release of the water before the cylinder is cracked. Spring valves are much more prompt in action, though not so easy of adjustment. Probably the best arrangement would be a combination of the two, by inserting a spring between the lever and the weight.

The best piston rods are of steel, which is stronger than iron, and not liable to flute. They are usually fixed in the piston, as shown in Fig. 15, by a split cotter, the piston rod being bossed to a larger diameter at that point, and turned taper. The piston rod is turned taper to fit the bore of the crosshead without increase of diameter, which would necessitate the splitting of the gland (or its brass bush) to pass over the end of the piston rod. The crosshead cotter may be inserted horizontally or vertically as may be most accessible for insertion and removal.

The crosshead is usually of wrought iron, but cast iron may often be advantageously employed, allowing somewhat larger dimensions; the patterns vary very much.

Connecting rods are commonly arranged with the distance of the centres equal to the length of stroke multiplied by 3; sometimes both ends are single, but more frequently the crosshead end is forked. The forking of the connecting-rod end is, however, not so desirable as forking the solid crosshead where possible. By forking the connecting rod the forging is more elaborate, the brasses, strap, gibs, cotter, &c., are all duplicated, and a great risk of occasional side strains (caused by careless cottering) is introduced. The latter contingency prevents the adoption of the same refinement of proportion that could otherwise be attained, because each side must be strong enough to withstand the whole strain in itself; the possible existence of this side strain also demands a more secure arrangement of slide bars or parallel motion. If the crosshead is forked instead of the connecting rod, the number of parts is reduced to a minimum, and the greater complexity of the crosshead, whether of cast or of wrought iron, is little more than equivalent to that of the forging only of a forked connecting rod. In the case of a forked connecting rod, whether the crosshead end is single or double, it is almost invariably provided with brasses, straps, gibs, and cotters, as shown in Figs 17 to 19 (Plate 3), of various proportions suitable for either connecting rods, pump rods, or valve gear; the gib nearest to the end of the rod is usually omitted in light work. In Figs. 20 to 22 a class of rod end is shown which is equally suitable for either end of the connecting rod, and is occasionally employed in other situations. The end of the rod is first forged solid, and a rectangular eye is cut through it; a pair of brasses and a wrought-iron wedge of the form shown are then fitted, and the wedge is provided with an adjusting screw, which is fixed either by a pinching screw, or by manipulating the lock nuts.

The marine pattern of crank pin and connecting rod, Figs. 23 to 25, is now coming into extensive use; here the tension is borne by two stout bolts, and the brasses are more bulky than appears at first sight to be necessary. They give much less trouble by heating than any others; this is probably due to their bulky character, which, when assisted by the bolts, tends to prevent the distortion caused by heat. Whatever the arrangements of connecting-rod ends, the adjustment for wear at both ends ought to be towards the same cardinal point, with a view to preserving a constant distance of the centres. This is, however, not absolute, as the crank-pin brasses usually wear more quickly than those at the crosshead end.

Bearings of wrought iron working on gun metal ought not to sustain a greater pressure perpendicularly to the axis of the shaft than—the diameter in inches $\times$ the length in inches $\times$ 400 lbs.,—otherwise the risk of overheating becomes serious. In other words, the load may not exceed 400 lbs. per square inch in the longitudinal plane section of the bearing.

Phosphor bronze now enters largely into the construction of important bearings, allowing a greater pressure to be placed on it than the above. But nevertheless it is advisable to work as near the above figure as possible, to maintain the bearing surfaces in first-class condition. Crosshead and other slowly-moving oscillating bearings may be (and usually are) loaded to a much higher degree than would be possible under other circumstances.

R 2

The most common form of crank is that shown in Figs. 11 and 12 (Plate 4), being the old pattern wrought-iron one, fixed upon the shaft and the crank pin when hot. A key is driven between the crank and shaft, to prevent rotation on the shaft. The crank pin is fixed, either in a way exactly similar to the shaft (in which case the hole is bored parallel), or the hole is bored taper, and a cotter driven through the boss of the crank and pin, to prevent it from working out. Cast-iron disc cranks are occasionally employed, but are liable to split across in fixing upon the shaft. This is more frequently the case when the crank is arranged to balance the connecting rod, and the work is not skilfully performed.

The crank is the only admissible situation for a weight to balance the half connecting rod. In single-crank engines, however, the fly wheel is often weighted with that object in view, though situated several feet from the crank. This balance allows the engine to be more easily moved by hand, with the assistance of a bar; but when the engine is steaming, its action is decidedly prejudicial. The subject of balancing shafting is subsequently referred to.

The crank shaft of an engine of the class in question is 11 to 12½ inches in diameter in the journals; these dimensions being sufficient to cause the pressure per square inch not to exceed the figure (400 lbs.) before mentioned. In many cases, however, less than half that pressure is really applied, such proportions being justified by the results. The majority of crank shafts are made from wrought-iron forgings, but mild steel is frequently and successfully employed at the present time. Sir Joseph Whitworth's fluid-pressed steel makes the best shafts in the market, but comparatively few owners care to meet the cost of this material.

The crank shafts of horizontal engines are usually carried in angular pedestals with top and bottom brasses, the joint being inclined at an angle of 40° to 50° with the horizontal plane. Fig. 16 is an elevation of such a pedestal. Several of the old engines with flat pedestals are more reliable as to freedom from heating in the crank bearings. The fact that some makers turn the cap towards the cylinders, while the majority take an opposite course under similar conditions, tends to show that the flat, or level-jointed pedestal, would do as well as the angular form. The angular pedestals are often arranged so that the oil is scraped off the journal, at the lower side of the joint, before it reaches the point of greatest effect at the bottom of the bearing. In the class of engines in question, one-half of the weight of the fly wheel, crank shaft, &c., which rests on each bearing, is quite competent to prevent the shaft from climbing out of its proper position in the bottom brass. This view is confirmed by the fact, that brasses which are worn sufficiently to require renewal have suffered most in the part directly under the centre of the shaft; any deviation that occurs is towards the lower side of the joint of the brasses. In high speed single-crank engines, with a light fly wheel, or with a fly wheel placed much farther from the crank pedestal than from the other pedestal, the fly wheel is incompetent to perform the same office satisfactorily, and some other arrangement becomes necessary, whereby the thrust of the connecting rod may be efficiently and directly resisted. This is not so in the case under consideration. From an engineer's point of view the fly wheel appears unnecessarily heavy, but the owners find that their interests are best served by the employment of such wheels; when this is the case their effect upon the bearings ought to be carefully studied.

The pedestal is bored out, and the exterior of the brasses is turned to fit the bore of the pedestal. Snugs are provided on brasses to prevent their rotation, and consequently the finishing of the brasses outside, over a strip of surface equal to the width of the snugs, cannot be accomplished; this strip is cast to stand below the turned surface, and is therefore unsupported. The best holding snugs are situated close to the joint of the brasses, and only on the effective side of the bottom brass, so that when the weight of the shaft is relieved, the brass may be easily slipped around to the upper side of the shaft, and removed. These snugs are generally cast on both sides for use in turning and boring the brasses, and afterwards one snug is cut away. A hole in the end of each pedestal, drilled and tapped to receive a set screw of sufficient strength to lift the shaft, wheel, &c., is convenient for adjustment, inspection, and repairs.

The fly wheel is often made with one hundred and fifty-two teeth, 5 inches pitch by 15 inches wide, if eight arms are employed; or one hundred and fifty similar teeth, with ten arms: in both cases the diameter is nearly 20 feet, and the weight 25 to 35 tons. The arms of the wheel are driven into sockets, which are cast, and afterwards bored out in a central boss; the boss is secured to the shaft by four keys, which fit flats on the shaft and keyways in the boss; the boss is not bored, but has the eye cast of a diameter larger than that of the shaft. Two keys have heads and two have points towards each side. This allows some adjustment of the wheel, which would be impossible if the boss was bored; but its chief value lies in the fact that the facility of handling both the wheel and the shaft, either during erection or subsequently, is thereby increased. Each arm is secured in the boss by cotters, and is prepared at its outer end (by a flange or plate) for the reception of the adjacent ends of two fly segments, whose office is to give sufficient weight to the wheel, and which are bolted to arm ends. The fly segments may be furnished with snugs, to admit the use of keys or floats against the ends of the arms, and thereby increase the stability of the wheel. Two arms are shown (Fig. 11) prepared for keys, and two without such preparation. When snugs for keys are adopted they prevent the possibility of turning at the inside of segments to receive the ends of the arms, unless the snug is only carried part of the way across the segment, and the bedding performed on the remaining part. In such cases the end of the arm is often surfaced in the lathe with the same centres as are employed in the turning of the spigot to fit the boss; this surface is then plane and perpendicular to the centre of the arm, while the surfaces (chipped strips) of segments, which fit to the ends of the arms, must be finished by hand. The outer surfaces of the fly segments are prepared for the reception of an equal number of tooth segments, the joints of which are arranged midway between those of the fly segments. These tooth segments are cast with the utmost regard to the attainment of a suitable section, with a view to prevent sacrifice of strength in any part, which may be caused by the presence of large or unequal masses of metal. Formerly the custom was to arrange one set of segments only to serve both purposes, though sometimes cast with recesses, which were afterwards filled with a bulky casting in each; these constantly gave trouble from the teeth breaking close to the rim, and the system is now abandoned in this district. Figs. 11 and 12 show a good ordinary wheel of the class described.

The pendulum governor is the one most largely used, being the Watt governor with a slide on the spindle, at about the same height as the centre of the balls, instead of the parallelogram at the top of the spindle. The controlling motion is taken off, either by a hollow spindle containing a rod passing upwards, or by a groove in the bottom slide which contains a collar (or its equivalent) with a pair of opposite studs working a forked lever, if the motion is required downwards. The arms are usually suspended at points on each side of the spindle, about 2 inches from the centre of the spindle, or 4 inches from each other. The more correct plan, of fixing the point of suspension on the centre of the spindle, is seldom employed. The diameter of the balls is 10 to 11 inches, and the length of the arm is such as to give about the same working number of revolutions (thirty-three to thirty-six) to the governor spindle as to the crank shaft. Neglecting the influence of the arms and links, the number of revolutions per minute $= \frac{187 \cdot 5}{\sqrt{h}}$, where $h$ = height in inches of the intersection of the centre of the arms with the spindle centre, above the straight line joining the centres of the two balls. In ordinary cases the number of revolutions, obtained from the above well-known rule, may be increased by the addition of three to the required number of revolutions, to compensate for the influence of the arms, links, and parts sliding on the spindle, thereby dispensing with complicated formulæ and calculations, while still arriving at a good practical result. If the arms, links, and sliding parts are of extra weight, a greater number must be added.

Governors have been successfully employed with the point of suspension of each arm, and the weight attached to each arm, on opposite sides of the spindle; therefore, the actual arms (instead of the arms produced) cross in the centre. In many cases governors on the high speed principle, with light balls driven at a great speed, and a heavy balance weight sliding on the spindle, to counteract the centrifugal force at a certain speed, have been employed; many forms have been adopted, of which Porter's and Galloway's may be taken as types. The two latter systems of governor are more sensitive than the first or ordinary one (in fact, Porter's is sometimes too sensitive), but they have not met with the extensive adoption which they deserve. As the necessity for equable motion in most of the works of this district is of the highest importance, this is surprising.

As before mentioned, the usual means for regulating the supply of steam is by a throttle valve worked by the governor. The old disc valve swivelling across the steam-supply pipe is fast disappearing in favour of the equilibrium valve, having an upper and lower seating, each with corresponding conical valve, the steam being admitted either between the two or outside. The two discs are sometimes absolutely of equal diameters, in which case no ready means of taking the apparatus apart exist; more frequently the lower valve is made of a smaller diameter, and with specially narrow bearing, so as to have the difference as small as possible; the valve may then be thoroughly taken apart in the minimum time. No difference in working effect can be observed until the valve is practically closed, and then it is very small.

Two classes of stop valves are shown in Figs. 26 and 27; one is usually called a stop valve, and the other a junction valve, but at some shops the names are reversed. In Fig. 26 the entry is at the bottom, and the exit at the side. The valve shown in Fig. 27 is merely inserted in a straight, continuous pipe. The valve disc itself in Fig. 26 is winged; in Fig. 27 it is a spindle or mushroom valve. In Fig. 26 the spindle is provided with a nut below the bonnet; in Fig. 27 the nut is above the bonnet; either kind of bonnet, nut, valve, and seating may be employed in either kind of casing. The casing shown in Fig. 27 provides a fluid way throughout equal to the clear area of the bore of the pipe; the valves mostly employed, however, fall far below fulfilling this necessary condition. The dotted lines in Fig. 27 represent a more common form of casing, and it is not rare to find the area reduced to less than one-half that of the bore of the pipe. The flanges ought to be well secured to the body of the casing by ribs or brackets, but these are often omitted because they interfere with the bolts.

The steam-supply, intermediate, exhaust, and water-injection pipes, are usually all below the floor of the engine room. The steam-supply pipe is provided with a stop valve. When two high-pressure cylinders are employed, a main stop-valve is inserted in the steam pipe before branching, and a smaller auxiliary valve in the branch leading to each cylinder. Ordinarily, the smaller valves are kept wide open, and only the main valve is worked, except when one cylinder is stopped. A similar but smaller valve is also employed to communicate between the steam and intermediate pipes, for convenience of warming by the admission of steam before starting. These valves are sometimes replaced or supplemented by brass by-pass cocks, which communicate between the steam chest and the steam port (independently of the position of the slide valve), and which, besides convenience in warming, are of great assistance in starting the engine from a fixed position; or, by introducing a retarding steam pressure, they assist in stopping it in a required position. The valve which regulates the supply of injection water is usually a sluice valve worked by a screw; some firms employ cocks for this purpose. The water reservoir is seldom above the injection inlet, owing to the fact that the water is to be repeatedly used, and if the reservoir was elevated, the back water would require pumping accordingly. When the injection water is not delivered under pressure, and the work is regular, the valve may be untouched for weeks together. All the valves, when beneath the engine-room floor, are worked by continuations of their spindles to a distance of about 2 feet 6 inches above the engine room floor, and supported by pillars. In Fig. 11 the intermediate pipe is shown to take an overhead course, because more direct, and avoiding the awkward joint necessary on the high-pressure cylinder beneath the valve chest; but usually this inconvenience is disregarded for the sake of appearance. It will be noticed that the joint of the exhaust pipe with the low-pressure cylinder is inaccessible; it is made before lowering the cylinder to the bed plate. The exhaust pipe is not well arranged in Fig. 11 to expand and contract freely; but as the range of temperature is small, there is no trouble on that account. Very few expansion joints are now fitted in steam pipes, as an elbow with the two arms at right angles, and of considerable length, is found to act satisfactorily. In the class of engine under description, the diameter of the steam pipe is equal to about one-third that of the high-pressure cylinder, the intermediate pipe being equal to one-half that, or one-quarter of the low-pressure cylinder, and of equal diameter throughout its length. The diameter of the final exhaust pipe is equal to that of the low-pressure cylinder multiplied by 0·32 to 0·35 at its upper end, being usually somewhat larger at its lower end or junction with the condenser. The description just given applies with little modification to a large portion of the engines now being erected, or recently set to work, in the district.

Twenty years ago the beam engine was largely adopted, and many examples are now at work yielding creditable results. The strains to be resisted in such engines act at many points, in a great measure independently of each other, which causes total cost of the engines and of the foundations to be heavy. The beam is frequently a weak point, giving much trouble and expense by breaking, and causing the fracture of other details. On the whole, therefore, the beam engine is an expensive form of steam machinery. On the other hand, when the details are all well proportioned as regards strength, the system gives a uniform motion, offering great facility for working the pumps, and comparing favourably for general accessibility with many others.

Beam engines are occasionally applied at the present day, but their use is limited, and is decreasing. The great majority of beam engines now working, whether new or old, are compounded on McNaught's system; the cylinder, which in the original form was used alone (engaging the end of beam), being employed as the low-pressure cylinder, while a high-pressure cylinder is erected below a point in the beam about half-way between the main centre and the connecting-rod stud. The capacity of this high-pressure cylinder is from one-third to one-fourth of that of the original, or low-pressure cylinder. When well-proportioned to the required work and to the boiler pressure, the addition of such a high-pressure cylinder to an existing beam engine conduces to economical working. At the same time another important advantage is obtained, and this is often the primary reason for such alteration, viz., the two cylinders being on opposite sides of the main centre of the beam, assuming the work performed in the two separate cylinders to be equal in amount, there is no more pressure on the beam centres when the engine is working than when it is standing. The degree of closeness with which the above assumption agrees with the facts rules the action on the beam centres; but in every case the tendency is towards the same result, viz., to relieve the beam.

In Woolf's system of compounding, the two cylinders are arranged beneath the same end of the beam, and the connecting rod is applied to the other end; this is with a view to reduce the cubical capacity of the passages, and to promote economy. Obviously this system does not relieve the beam, and is only to be attempted when the appointments are good.

Another principle (which in approximate chronological order may be placed between the beam engine and the horizontal form) is that of the vertical engine, with the cylinder on the foundation and the crank shaft overhead. In this case the air pump is below the level of the cylinder, is worked by means of a lever from the crosshead, and is often only accessible with difficulty. The cost of foundations for this type of engine may be small (as also the total cost) by judicious arrangement, but this is never a conveniently-worked engine, and is now seldom employed in new work. As in all engines, the crank shaft is the part most difficult to control; here it is situated very inconveniently for being well secured.

Several firms are erecting vertical engines with overhead cylinders with fair success, as might be expected from the results of their use as marine engines. They occupy less floor space than others, except the last described. The motion is at least as steady as that produced by any other reciprocating engine. The question of the side adjustment of the crank-shaft brasses is solved in a decided manner, unless materially influenced by the direction in which the work is taken off, and even then an advantage over the horizontal engine still remains. The whole weight of the engine rests on the foundation plate near the crank, and where most desirable for steadiness. No side stays are required, as all motion above the crosshead takes place in an absolutely vertical direction. The air pump may be worked by a lever from the crosshead, and placed on the engine-room floor, allowing free access to all its parts. The piston is also readily accessible, as the cylinder cover may be removed. The wear of the piston and bore of the cylinder are equal throughout the circumference; while assurances are freely offered to the effect that the wear on the lower side of pistons and cylinders is very small, or even absent, yet the contrary is constantly shown to be the case, many pistons being in a wretched state from this cause. Some engines of this class scatter the drip from the piston rods and valve spindles, causing a slovenly appearance; but this is not an insurmountable obstacle, and even if it were so, it would not prove a serious objection.

A most important point to be regarded in the design of this type of engine is, that the frame must be scrupulously symmetrical with respect to the vertical plane traversing the axis of the crank shaft, and also with respect to the vertical plane which is perpendicular to this one, and traversing the axis of cylinder. If any departures be necessary for the attachment of the valve or pump gear, they must be reduced to the lowest limits. If the crank bearing be firmly supported (or, better still, be duplicated and well supported on the opposite side of the above second plane), the whole engine may be confidently expected to perform its work well and without vibration.

Steam jackets have been repeatedly adopted and discarded, chiefly because attention was not paid to the conditions necessary to their economical working. The steam should be supplied to the jacket independent of the supply to the cylinder, or at least should be taken away from that pipe before passing the stop valve, with a view to obtaining as nearly as possible the full boiler pressure (and therefore the full boiler temperature) in the steam jacket. The drainage also requires careful attention, and must be arranged so that every drop of water is removed at once on its formation. If the cylinder is sufficiently elevated above the level of the boiler, the drainage pipe may be connected to the water space of the boiler, but such connection is not to be recommended. Another reason why less benefit is derived from the use of jackets is, that the measure of expansion obtaining in the high-pressure cylinder is not great, therefore the range of temperature, with or without jacket, is not so great as when the whole expansion occurs in one cylinder. The steam jacket ought to be applied to both cylinders, but this is seldom the case in compound engines.

Chiefly for these reasons steam jackets have failed to give appreciably beneficial results where they have been adopted. As a small failure goes a long way against any principle, it may be expected that steam jackets will form no exception, and that they will not be largely adopted for a long time to come.

The usual rate of piston speed is about 400 feet per minute; some firms adopt a rate of 500 feet, and others 600 feet or more, with success, but they are exceptions. Occasionally an engine is driven at a higher speed, to relieve it a little when overloaded. If these alterations are made without judicious corresponding ones in the valve gear, the coal consumption rises; more especially when a cumbrous engine constitutes a considerable part of the load. From repeated observations the conclusion is arrived at that high speed in itself necessitates a greater coal consumption. A high piston speed, when judiciously adopted in connection with an efficient expansion arrangement, forms a valuable feature in economical working arrangements, but otherwise it is a snare. In many cases the driving wheels have been removed in favour of a pair of different ratio, allowing the engine to run at a higher speed, without any alteration having been made in the valves and gear to provide for greater expansion; the only possible logical result being arrived at, that the coal consumption is much greater; more especially when the exhaust ports and passages are restricted in area.

The design of steam engines tends to become plainer with each change. Moulds, fluted pillars, carved bases, and other distortions, are replaced by simple curves, either single or double, in which approximate ellipses and parabolas predominate. This is caused partly by the exigencies of prime cost, but also largely by the tardy recognition of the fact that a style, which is beautiful when skilfully wrought in marble, and seen in the open air of a sunny climate, appears under adverse circumstances in the confined space of an engine room, when reproduced in cast iron, with the defects of moulding added. The current practice is to avoid scooping out a space for a nut and washer, or a bolt-head, when these occur obliquely on a surface. A boss is added, instead of removing material, in the large majority of cases; in fact, the principle is often pursued to excess, as, for instance, where an internal angle is carefully filleted, and then the hollow is practically obliterated by the addition of a bolt-hole boss, or facing. This occurs when the length of the internal angle is only a little greater than the width of the boss or facing.

The practice of providing, during the manufacture and erection of all engines, for the quick and easy removal, for inspection, cleaning, and repairs, is extending. This is chiefly effected by liberally drilling and tapping holes for the insertion of forcing screws, obviating by this means the unmechanical forcing apart of details by chisels, &c. Holes are also drilled and tapped, to receive eye bolts (which also make good forcing screws), to which the lifting tackle is readily applied. The cost of these small provisions is amply covered by their convenience during the erection of the engine.

Another practice, recently adopted, is to build up the walls of the engine room at once, before erecting the foundation. At a suitable height, provision is made, either by a cornice or corbels, for the reception of a range of rolled joists along each of the longer sides of the engine room, usually leaving about 3 feet above the top of the rolled joists to the ceiling of the room. Rails are then placed on the rolled joists, and the rails carry a travelling crane, movable in two directions. The crane is arranged to work from below by endless ropes or chains. The hoisting winch consists of a chain barrel, worm wheel, and worm, mounted on two or more pieces of the frame by two shafts. The use of the worm and wheel gives the simplest motion, being reliable, and dispensing with the use of a brake or ratchet wheel, with its attendant cord. The crane used in erecting the foundation and engine afterwards remains as a permanent appointment in the engine room. The success of this practice is modified by the fact that, for the attainment of an economical result, the stonemason requires that his hoisting be performed at a greater speed than the engine builder can admit, therefore the same arrangement is not suitable for both parties. To some extent this objection may be surmounted by changing the worm and worm wheel; the chain barrel, or the whole winch, still retaining the longitudinal and transverse joints.

A few years ago the practice in building engine foundations was to commence by laying a sheeting course, 4 to 6 inches thick, over the whole area, projecting a few inches on each side. The centres of the holding bolts in horizontal projections were next marked on the sheeting course, and a course of brickwork built up about 12 inches high, leaving 7½ inches on each side of each bolt centre, forming an open hand hole for access to the lower end of the bolt. On the top of this the solid stonework was carried up to the required height, in courses of 12 inches to 18 inches, leaving the holes for the holding bolts during building, except the length through the top course, which was marked from the bed plate and drilled afterwards. Recently, however, owing to the increased cost of stone, the intermediate courses have been dispensed with, only the base course (next above the brick hand hole course) and the top course being retained; while brickwork is substituted for all the intermediate courses. Rough wood boxes are built in the brickwork at each bolt hole, to preserve the vertical opening. The brickwork is set in cement, and appears fully as substantial as stonework. Sometimes also the course of stone next over the hand hole course is omitted, cast plates being laid in its place, and over these the brickwork. Occasionally the sheeting course is stepped out to different levels, as there is no necessity for the whole area to be carried as low as the part below the air pump, if a solid bed is reached at a higher level.

Before leaving the steam engine, it may be mentioned that there are two methods in use for heating the works. In the older system, cast-iron pipes, about 4 inches in diameter, were employed as radiators, the steam being supplied by wrought-iron pipes of smaller diameter, and the condensed water conducted from the cast-iron pipes by traps to wrought-iron pipes. The traps for allowing the escape of the condensed water are of various patterns, the older ones working chiefly on the principle of the ball-lap and cistern, more or less modified.

The more recent forms of traps depend upon the expansion and contraction caused by changes of temperature for their action. One arrangement is similar to the ordinary screw-down stop valve, the valve disc of which is connected to an iron pipe, while the seat is connected to an inner copper pipe. In a decreasing temperature the upper pipe contracts more, opening the passage for the exit of the contents; the reverse effect is caused by an increasing temperature, and the action is found to be very regular. Another trap acts similarly by the expansion and contraction of ether contained in a sealed vase. All traps are in a greater or less degree troublesome, and their use is avoided in a system now largely adopted. In place of cast-iron pipes of 4 inches in diameter, wrought pipes of about 2 inches in diameter are employed. One end of each range of piping is connected to the steam space of the boiler, and is nearer (following the circuit of the pipe) to the highest point of the range, while the other end of the range is connected to the water space of the boiler. The pipes, where not required as radiators, are smaller than elsewhere. This arrangement gives much less trouble in obtaining uniform results than the former one, but is less durable.

All outlet pipes in the works ought to be conducted into one tank in some conspicuous part of the yard. This applies to outlets from heating apparatus, cylinder cocks, drains from steam supply, mud cocks of boilers, safety-valve drips, water-gauge fittings, &c. Defective fittings are thus promptly shown, and the flooring and walls kept more sightly than with the usual promiscuous arrangements.

## Transmission of Motion.

The fly wheel gives motion to a pinion about 6 feet in diameter, making one hundred and five to one hundred and twenty revolutions per minute, and staked on the second-motion shaft in a way similar to that in which the fly wheel is fixed. This shaft is usually carried in angular pedestals, and is placed in the same horizontal plane as the chief line of shafts, with a view to direct driving by simple bevil wheels. The engine is often erected with its crank-shaft centre on the same level, but frequently 6 feet or even 10 feet lower. As the shafts for the upper rooms are in the same vertical plane, one upright shaft, placed directly opposite the series, suffices to drive the whole, and only one pair of bevil wheels is required for each story.

The form of teeth in the wheels is in the last degree arbitrary. Each shop possesses a different rule, and to some extent individual workmen exercise influence over the shape of the tooth, often according to the fancy of the hour. The general law, that the normal to the common surface of the teeth described through the point of contact of the teeth, in any position, in any pair of wheels, should pass through the pitch point, is quite unknown to the mass of wheel makers. Doubtless the wisest course is to use the closest approximation to the cycloidal form. That is possible by means of arcs of circles; but is not easy of performance without the use of two arcs of circles for the face of the tooth, though one is sufficient for the root. Workmen are not to be relied upon for following instructions absolutely in such a case.

Professor Willis's form gives good working results, but involves the use of angles, which diminishes its value. Workmen are accustomed to employ some given radius for both roots and faces, and to fix the centre from which the root is described on the pitch circle, while the centre from which the face is described is fixed at a definite distance below the pitch circle.

A system has been in use for some years with successful results, which corresponds more nearly with the ordinary and simpler method just mentioned than does Professor Willis's. It was devised by Professor Smith. The teeth described by it are slightly superior to those of Willis's on the face, which is the most important, while the accuracy of the roots (compared with a true hypercycloidal curve) is about the same. The system has been applied to wheels of various pitches, diameters, and ratios, spur and bevil. Wheels of nearly equal diameters are more usually found to work together, without actual inconvenience, than pairs which differ largely, or have a high ratio. The evils of defective form culminate in the case of a rack and pinion, as may be seen on a planing machine, in the reproduction on the work executed of the pitch of the rack. In this system the radius of curvature employed in describing both the face and the root = pitch $\left(1 - \frac{4}{u} + \frac{5}{u^2}\right)$, the root being described from a centre on the pitch circle, and the face from a point below the pitch circle, at a distance equal $\frac{\text{pitch} \times \left(5 - \frac{90}{u} + \frac{500}{u^2}\right)}{32}$. Wheels thus developed are interchangeable. This is chiefly advantageous in spur wheels, but also occasionally in bevil wheels, when shafts form acute or obtuse angles with each other. In bevil wheels, the virtual numbers must be taken. This may be done either by the development of conical surfaces normal to the pitch surfaces, and completing the circles, or by the formula $\left(\frac{u \times \sqrt{u^2 + u^2_1}}{u_1}\right)$, in which $u$ = number of teeth in the wheel under consideration, and $u_1$ = number of teeth in its fellow. This applies only to wheels revolving on shafts at right angles to each other. In other cases the best way is to obtain the virtual radius at once by development, and thence the virtual number of teeth. The above expressions for the radius for rounding, and for the distance below the pitch circle, may be reduced to a table or to an ordinary diagram scale or direct measurement.[1]

Wheels which are well made, but with defective form of teeth, when set to work produce an equal, continuous, rumbling sound, varying in intensity and pitch according to the degree of divergence from a correct form, and to the number of teeth in contact per minute, or per second. Wheels are made of cast iron, moulded by machine or from a pattern. From a new pattern, divided on a good wheel-cutting machine and securely constructed, the best castings are produced. Machine-moulded wheels are much better

[1] This method has since been improved by the originator, but the details are not to hand.—G. W. S.

than those produced from a pattern which has been stored sufficiently long to become distorted, and with all wood patterns this is only a question of time. Wheel-moulding machines are exposed to great wear and tear; therefore they should be frequently subjected to the most stringent tests for verification of accuracy. Notwithstanding the utmost care and skill that may be exercised in the manufacture of wheels, there is frequently found to be one tooth, or more, of greater thickness than the others; this is usually trimmed off by pitching and chipping. If allowed to pass without adjustment, a blow is caused at each revolution. If its fellow possesses a similar defect, the severity of the blow is increased when the two come in contact. This is a frequent defect, for which the only remedy appears to be, to finish the wheels by machine. It is often most prominently displayed on wheels which are otherwise good.

The pitch and diameter are usually decided by rule of thumb, combined with experience. The annexed table (page 258) has been found serviceable in determining the diameter of the shaft, the pitch of the wheel, the indicated HP., and the number of teeth, after some of the details have been settled. In this table, the first horizontal line gives the diameter of the shaft; the second the HP. values for separate diameters. The proper shafting value, multiplied by the number of revolutions per minute, gives the actual HP. capable of transmission by the shaft, if made from reasonably good iron and in ordinary steady work. The first vertical line gives the pitch of the wheels; the second the pressure in lbs. (safe working load); the third the relative HP. values, the width of the teeth being assumed to be equal to the pitch $\times\ 2\frac{1}{2}$. To obtain the indicated HP. capable of transmission by any wheel, the corresponding number is selected from the table, then multiplied by the number of teeth in the wheel, and the product multiplied by the number of revolutions per minute.

If the necessary diameter of a shaft and the pitch of wheels to drive it be decided upon (assuming the work to be undivided), the necessary number of teeth may be found in the table at the intersection of the two columns. If the pitch be two-thirds the diameter of the shaft, the number of teeth is found to be fifty-seven, which is generally a good proportion to employ. Many special cases arise in which it is admissible or necessary to depart from the proportion shown in the table, but even in such cases the table is useful for reference.

In order to reduce the pressure per square inch on the larger bearings to a sufficiently low point for safe working, it is often

| Diameter of shafts . . | | | Inches. 2 | Inches. 2¼ | Inches. 2½ | Inches. 2¾ | Inches. 3 | Inches. 3¼ | Inches. 3½ | Inches. 3¾ | Inches. 4 | Inches. 4¼ | Inches. 4½ | Inches. 4¾ | Inches. 5 | Inches. 5½ | Inches. 6 | Inches. 7 | Inches. 8 | Inches. 9 | Inches. 10 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Wheels. | | I. HP. Values. Shafts. | 0·0987 | 0·1406 | 0·1927 | 0·259 | 0·33 | 0.4238 | 0·5293 | 0·651 | 0·79 | 0·9477 | 1·125 | 1·3231 | 1·5432 | 2·054 | 2·6 | 4·2345 | 6·3299 | 9·0 | 12·3456 |
| Pitch. | Pressure. | Wheels. | | | | | | | | | | | | | | | | | | | |
| Ins. | lbs. | | Numbers of teeth in wheels. | | | | | | | | | | | | | | | | | | |
| 1 | 291 | ·000736 | 134 | 191 | 262 | | | | | | | | | | | | | | | | |
| 1¼ | 454 | ·001436 | 69 | 98 | 134 | 180 | 232 | | | | | | | | | | | | | | |
| 1½ | 655 | ·002481 | 40 | 57 | 78 | 104 | 134 | 171 | 213 | | | | | | | | | | | | |
| 1¾ | 891 | ·00394 | 25 | 36 | 49 | 66 | 85 | 108 | 134 | 165 | 200 | 240 | 285 | | | | | | | | |
| 2 | 1,164 | ·005882 | 17 | 24 | 33 | 44 | 57 | 72 | 90 | 111 | 134 | 161 | 191 | 225 | 262 | | | | | | |
| 2¼ | 1,473 | ·008378 | 12 | 17 | 23 | 31 | 40 | 51 | 63 | 78 | 94 | 113 | 134 | 158 | 184 | 245 | 317 | | | | |
| 2½ | 1,819 | ·011488 | | 12 | 17 | 23 | 29 | 37 | 46 | 57 | 69 | 83 | 98 | 115 | 134 | 179 | 232 | | | | |
| 2¾ | 2,201 | ·01529 | | | 12 | 17 | 22 | 28 | 35 | 42 | 52 | 62 | 74 | 87 | 101 | 134 | 174 | 277 | | | |
| 3 | 2,620 | ·019854 | | | | 13 | 17 | 21 | 27 | 33 | 40 | 48 | 57 | 67 | 78 | 103 | 134 | 218 | | | |
| 3¼ | 3,074 | ·02524 | | | | | 13 | 17 | 21 | 26 | 31 | 37 | 44 | 52 | 61 | 81 | 105 | 167 | 251 | | |
| 3½ | 3,566 | ·031525 | | | | | | 13 | 17 | 21 | 25 | 30 | 36 | 42 | 49 | 65 | 85 | 134 | 201 | 285 | |
| 3¾ | 4,093 | ·03877 | | | | | | | 14 | 17 | 20 | 24 | 29 | 34 | 40 | 53 | 69 | 109 | 163 | 232 | |
| 4 | 4,658 | ·047056 | | | | | | | | 14 | 17 | 20 | 24 | 28 | 32 | 44 | 57 | 90 | 135 | 191 | 263 |
| 4½ | 5,895 | ·067023 | | | | | | | | | 12 | 14 | 47 | 20 | 23 | 31 | 40 | 63 | 94 | 134 | 184 |
| 5 | 7,277 | ·09191 | | | | | | | | | | 10 | 12 | 14 | 17 | 22 | 29 | 46 | 69 | 98 | 134 |
| 6 | 10,480 | ·15882 | | | | | | | | | | | | | | 14 | 17 | 27 | 40 | 57 | 78 |

necessary to enlarge the bearings to a greater diameter than that absolutely essential for the transmission of power. This must be decided mainly from experience, but the requirement is only local, the diameter of the shaft being usually reduced by tapering before reaching the next bearing.

It may be observed that the pressures given as permissible in the higher pitches are greater than 400 lbs. per inch in width, as decided by the late Sir W. Fairbairn.[1] The accuracy of the table is, however, often and independently corroborated, and it agrees closely with ordinary practice. Much greater durability is attained in such wheels by increasing the width of the tooth when the foundations are thoroughly secure, and the bearings are prevented from wearing much or unequally. Frequently such wheels are, through carelessness, allowed to bear on one side only, in which case the increased durability conferred by increased width exists only in imagination. For this reason chiefly, the present tendency is to reduce, rather than to increase, the width of the tooth. This tendency on the part of workmen to allow all wheels to drift into an unsatisfactory state of adjustment is often observed. As a rule, it is more mischievous in spur wheels than in bevil wheels, owing to the fact that the latter are more easily adjusted, and are therefore often attended to, when the somewhat more difficult operation of adjusting spur wheels would be avoided. For the reason just given, it is usually unnecessary to draw any distinction between spur and bevil wheels in using the table. If wheels and shafts are exposed to frequent or severe shocks, or are worked unsteadily, it is wise to increase the strength of each, still maintaining the same relation between them.

Where an approximate ratio in a pair of wheels is sufficient, it is wise to add a "hunting tooth," unless the numbers of wheels are already prime, as causing more steady work. Too much reliance is laid on the value of wheel-moulding machines, and the hunting tooth is considered to be unnecessary.

The ends of the teeth of bevil wheels were formerly of spherical form, described from a centre at the intersection of the centre lines of shafts, but at present they are conical, and normal to the rolling surfaces or pitch cones. This is more convenient in construction, whether for pattern or machine-moulded wheels, and produces the same result. Wheels were devised thirty years ago with the roots strengthened on the non-working side, to prevent breakage by shocks. These have been recently revived and patented, great

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[1] *Vide* Minutes of Proceedings Inst. C.E., vol. lv., p. 405

advantages being claimed over the ordinary form in respect to the working load. Such claims may be justified in wheels subjected to severe and repeated shocks; but with steady work, teeth whose working and idle faces are well-shaped and similar, are found to wear away on the working faces before breaking. In ordinary work the improved form has no effect whatever till the wheels become partially worn, and may be said to add a fictitious life to them.

In small wheels, it is advisable to dispense with arms, and to substitute a continuous plate. This causes the resulting wheel to be more circular in form and rather more sound. If machine-moulded wheels are thus arranged, their cost is somewhat reduced; both kinds are shown in the figures.

Heavy wheels have their bosses split in casting, to prevent the breakage of the arms by contraction. The boss is then surrounded by a strong wrought-iron hoop on each side, to prevent bursting or springing in driving the keys. Similar hoops are often added when the boss is not split, as additional safeguards, in the case of wheels and pulleys. The pulley shown in section in Fig. 44, Plate 3, is prepared to receive such hoops.

All shafts which carry toothed wheels ought to have bearings on both sides of each wheel where possible. Any arrangement by which the wheel is between the bearing and the end of the shaft should be avoided, unless the shaft is specially strong. Toothed wheels, unless in unfrequented situations, are covered by cases of sheet iron, formed to fit closely, and so to prevent the scattering of grease and other matter from them.

Toothed wheels are occasionally used to connect two parts of a shaft which form a small angle, each with the produced centre line of the other one, and which are usually regarded as one continuous bent shaft. Another means to the same end consists in employing a pair of claw couplings, whose driving surfaces are portions of plane surfaces, traversing the axis of the shaft. These are useful for rough slow work, but otherwise must be condemned, as the communicated motion is very variable. Hooke's joint, when well proportioned and constructed, ought to be as nearly perfect for such a purpose as any other method, but it is seldom used except in small work.

Shafts 4-inches in diameter and upwards are turned from forgings; those of smaller diameters are turned from rolled bars, both being polished throughout, except where covered by wheels, couplings, &c. Large wheels are usually fixed by four keys, as before described, the shaft being bossed to a larger diameter for convenience of manipulation. This arrangement is shown in elevation in Fig. 28, Plate 5, and in the sections of many other wheels. Smaller wheels are usually bored to fit bosses, provided on the shaft and secured by one key, which fits at the top and bottom on the flat of the shaft and the keyway of the wheel. In either case, preparation must be made to slip the driven wheel out of gear so as to stop the shaft.

When many wheels are required on the same line of shaft, they are sometimes fixed on a plain parallel shaft by conical cast-iron keys (Figs. 35 and 36, Plate 5). This arrangement is subsequently described in connection with pulleys.

In ordinary practice, shafts are secured with a view to prevent longitudinal movement at the end to which power is imparted. This is done by means of one or two journals, each provided with collars on each side. If the shaft is of great length, and carries heavy bevil wheels at points more than 40 feet from these bearings, the expansion and contraction caused by atmospheric changes become very troublesome. In such cases, other bearings are provided near such wheels, and an expansion coupling arranged in some convenient intermediate position. Ordinary claw couplings are sometimes used as expansion couplings, but the more usual form is a solid cylindrical coupling, into the bore of which the end of each shaft projects to the centre of the length. One end is then keyed securely, and the other is provided with two keys, secured in grooves on opposite sides of the shaft. The shaft and keys are made to fit accurately but easily, allowing for expansion and contraction.

The diameters of the shafts may be obtained from the table, when the speed and HP. are known. They are usually somewhat greater than are strictly required, having reference to ultimate strength alone. By this means excessive torsion is avoided, which otherwise forms a serious evil in the employment of long shafts of small diameters. The shafts are reduced by small steps; often the consecutive lengths differ in diameter by only $\frac{1}{8}$ inch, and usually by $\frac{1}{4}$ inch. As the disposition of the machinery is permanent, the number of different sizes may be much larger without inconvenience than where the arrangements are liable to frequent alteration.

The separate lengths of shafting are generally from 20 to 30 feet long; they are secured together by solid cylindrical couplings, which are turned on the outside and bored to fit the ends of the shafts. The end of each shaft reaches to the centre of the coupling, and the two are secured by a key, either continuous or divided in the centre of length. The ends of the shafts are turned off square, the half-lap coupling being now discarded. This coupling is the simplest that is made, and if the ends of the key are properly finished off, it presents no projection when running.

Several systems of adjustable couplings are in use, and appear to answer well, but they are more complicated, and none have so neat an appearance as the solid one; neither can they claim any important advantage in their adjustability, except with shafts that ought to be rejected on account of careless turning. Sellar's coupling, and one or two others, however, by changing the bushes, can be adapted to the ordinary small changes of diameter of shafts, without reducing the larger to the diameter of the smaller one. Their chief advantage, however, lies in the fact that they dispense with the use of the key sunk into the shaft (or, at least, they reduce its importance greatly), also with the bosses thereby rendered necessary, on the adjacent ends of shafts. Again, by dispensing with bosses the majority of shafts may be run through a stretching machine, thus dispensing with hand stretching.

The couplings are placed as close to the bearings as possible, and on that side of the bearing which is farthest from the driving point, from whatever source the power may be derived. The chief reason for this is, that if any part of a shaft requires to be stopped, the proper coupling may be taken off, and the shaft will still be supported throughout the running part. Also because the shaft is generally secured against longitudinal movement at the driving point; then, if it becomes necessary to disconnect any single length of the shaft, the facility of longitudinal movement is a great convenience. The addition of an extra pulley in a bay which contains a coupling may by this means be performed without any hoisting of shafts, which would be impossible with a reserved arrangement.

All shafts are now erected overhead; in some old works, a single shaft was arranged to serve two floors. The fire risk absolutely prohibits such an arrangement in modern factories. Very few machines in such works require the use of a countershaft, being driven directly from the line shaft, and hence the question of length of belts does not arise to cause the underground system to be desirable.

The driving pulleys are fixed to the shaft almost invariably in the same manner. A conical hole is bored in the boss of each pulley, whose smallest diameter is sufficiently large to allow the pulley to pass the bosses on the shaft. This hole is fitted with a bush which is bored to gauge fitting the shaft, and split longitudinally in three equidistant places; the three pieces so produced are driven into the pulley boss, being held by friction alone; they are fixed and removed with great facility. If necessary, the pulley may be taken to another shaft of nearly equal diameter, by merely providing a fresh set of these conical keys of different bore to fit the second shaft. Readjustments of the machines or the renewal of the belts often necessitate slight removal of the driving pulleys, and this is easily done when fixed as described. Fig. 36, Plate 5, shows the application of this fastening to wheels, and Figs. 45, 46, and 47, Plate 3, to pulleys.

Every piece mounted on a shaft ought to be balanced separately, more especially if the speed exceed one hundred and thirty to one hundred and fifty revolutions per minute, or the revolving pieces are comparatively of large diameter. When balancing is attempted, it is often by a weight sufficient for a standing balance, added anywhere when the pulleys or other details are in place. Neglecting the possibility that the position of details may be greatly altered, the whole arrangement is ineffective, because the balance must be restored in the same plane of revolution that contains the disturbing cause. Every running part ought therefore to be balanced in itself as far as practicable.

The bearings of shafts are at distances of 10 to 11 feet from centre to centre. They are usually fixed, that is, without facility of self-adjustment. Their length is equal to twice the diameter. They are provided with top and bottom brasses, or bottom brass and cast-iron cap to fit the shaft, and bored along with the bottom brass. The iron cap to fit the shaft may be replaced by a shell cap, containing a recess to hold tallow for use in case of accidental heating of the bearing. An ordinary flat pedestal is shown in Figs. 47, 48, and 49 (the apparatus on the side is subsequently referred to). It may be noticed that one end is cut away to shorten the base, in which case the cap bolt is provided with a collar, and nuts at each end, to act as cap and holding bolt; obviously, this can only be done when the lower end of the bolt is accessible for tightening the nut or nuts. The holding bolts are often arranged with the heads in T grooves, as shown in Figs. 28, 29, and 30, Plate 5. This is when the lower side of the plate is not disposed so that a bolt can be passed through from the reverse side; or, in other cases, to prevent the drip of oil which takes place with thoroughfare holes in a horizontal plate. Strips are often cast on the sides of brackets and other pedestal preparations, to prevent the flow of the oil as much as possible after it has left the bearing. These are useless when the bolt holes pass through the plate, and in any case a better arrangement is to cast an elongated dish on each side of the pedestal.

Loose oil dishes, with flat pedestals, are seldom employed, and never succeed well.

In the best practice, pedestals, and the surfaces on which they lie, are planed, but many are still bedded with a piece of soft wood between the rough cast surfaces of the pedestal and its plate. At each end of the pedestal, a snug is cast on the plate, preventing the motion of the pedestal; a small space is left between the pedestal and the snug, allowing a small adjustment (the bolt holes being arranged with the same view), and the space is afterwards fitted with a float of beech or iron. This space and snug are shown in Fig. 49, Plate 3. When the pressure is always tending to cause the pedestal to move towards one snug, the other is often omitted, more especially if the space is limited.

Wall boxes are often required for the support of the bearings of shafts, and to form and preserve openings in walls where required for the accommodation of wheels, &c. Examples are shown in Figs. 31 to 42, Plate 5. The best practice consists in arranging and providing the wall boxes as the work proceeds, then laying the stones beneath the position of the boxes as shown, and disposing the boxes so that the wall may be built up to every side; the covering stones are then placed on the top of the boxes as shown. Wherever practicable, wide outside flanges must be left on each side of the walls as extra preventives against movement when working. This cannot be so conveniently done if the boxes are inserted after the building of the walls. In any case, it is impossible to bestow too much care upon the secure execution of the work.

Small wall boxes are employed as shown in Figs. 34, 35, and 36, for the attachment of simple brackets, allowing an easy adjustment of the height of the centre of the shaft. They are best arranged to pass through the wall with a flange on each side, but are often stopped short, as shown in the Figs. 34 and 36, for the sake of appearance. When boxes are not provided with outside flanges on both sides, the difficulty of secure fixing increases with the decrease of the depth projected into the wall. Small boxes are shown in Figs. 38 and 39, whose office is to preserve the form of the opening made in the wall for the reception of a shaft or other object. All boxes which give a clear opening through a wall are provided with internal flanges, to which wrought-iron plates are bolted, forming a fire-proof barrier. Effective means ought always to be taken to prevent the smearing of the walls by the oil which runs off the bearings.

A small bracket is shown in Figs. 47, 48, and 49, Plate 3, for the reception of a single flat pedestal. It is bolted to the face of the wall, and the snug on the back is inserted in the wall to assist the bolts in carrying the weight. It is almost impossible in practice to ensure that the pedestal bed of a bracket of this class shall prove to be of the correct height. This class of bracket is therefore unsuited to the attainment of first-class work.

Side pedestals are bolted to pillars specially prepared, or to some other vertical plane surface. A pillar with side pedestal, and carrying an upper floor, is shown in Figs. 50 and 51. Side pedestals are fitted with brasses, or bottom brass, and solid or shell caps, in the same manner as flat pedestals.

The foot of an upright shaft is steeled, or has a short piece of steel welded to it. It is surrounded by a brass bush, which is sometimes whole, and in other cases split. A plate or disc of tough brass is placed beneath the shaft to bear the weight; this plate is often provided with a wrought-iron hoop, shrunk on, to prevent it from splitting under pressure. A large hole should be drilled in the centre of this disc, and preparation made for the free entry of the oil, the utmost care being taken to ensure this. A cap is provided, which forms an oil-tight joint with the bracket, preventing the escape of the oil, and yet allowing free access for inspection of the shaft and brasses. The brasses are released for withdrawal by slightly elevating the shaft; lifting screws for effecting this are provided in the first instance. Two wall boxes, with the wheels, shafting, &c., pertaining thereto, are shown in Figs. 37 to 42, Plate 5, one pair of the wheels driving the upright, and the other pair driving an upper line shaft from the upright shaft. The practice is becoming general of providing a separate walled tower around the upright shaft, with stages or floors at each pair of wheels. In this system the footstep is carried on a massive plate, which rests on the stone and brick foundation, carried up from the place at which a sufficiently solid footing is obtained. The superiority of this system is manifest. The foundation is infinitely superior to any arrangement of brackets, &c.; the risk from fire is greatly reduced, all openings being closed by iron doors; the facility of working at adjustments is increased, as the scaffolds are already fixed, and more than ordinarily secure; the bearings are kept more free from dust and other material; in case of wheels breaking, the damage is confined to the tower, the floors above and below the broken pair of wheels of course being sufficiently stout to resist the flying fragments or the mischief will be extended; the grease is also confined inside the tower, and prevented from smearing other objects. The same principle is sometimes extended to the whole second shaft, which is then enclosed in an elongated room.

Bearings which possess a facility of self-adjustment, with respect to the direction of the axis of the revolving shaft, have not met with extensive adoption in this district. They are largely adopted in the United States; one of the chief reasons being, that the buildings being lighter, and containing more wood in their structure, are not so stable as those in England. The same evil is encountered in England, though in a smaller degree, and the more frequent adoption of these bearings would produce good results. Another reason for their more extensive adoption in the States is, that the shafts are lighter, and more easily deflected by the tension put on them by the belts; this tension (being a variable, or even reversible quantity) soon destroys the bearings when not self-adjusting. The general adoption of self-adjusting bearings in the States allows the use of lighter shafting without encountering some of its chief drawbacks. But, even then, decreased first cost, and an appearance of elegance, scarcely balance greater wear and tear and a shorter life of the shafting and appointments.

The usual speed of line shafting is one hundred and twenty to one hundred and sixty revolutions per minute. This is exceeded in some rooms, where the machinery runs at a very high speed; and also in others where the machines are driven by quarter-twisted belts, to reduce the diameter of the pulleys on the main shaft, and therefore, in an equal degree, the lateral obliquity of the belts. Higher speeds than the above have been occasionally adopted, but the ultimate economy is found to be less. The outlay in the first instance is certainly less, but if the shafts are in each case rigidly proportioned to the work which is to be performed, the friction is not less, as has been assumed, because the speed must be increased in a high degree. When the diameter of the shaft is decreased, the power per revolution is decreased in a greater ratio, because the power varies as the cube of the diameter. Therefore, to transmit an equal power the reduced shaft must be driven at an increased speed, denoted by the original speed $\times$ $\left(\frac{\text{original diameter in inches}}{\text{reduced diameter in inches}}\right)^3$. But as the radius of friction on the surface is reduced at the same time as the revolutions are increased, the moment of friction, or the velocity of the surface, is only increased in the direct ratio of $\left(\frac{\text{original diameter}}{\text{reduced diameter}}\right)^2$. The weight of the shaft being reduced in the same ratio $\left(\frac{\text{reduced diameter}}{\text{original diameter}}\right)^2$, causes the actual amount of friction to be the same as before. But the difficulty arises, that the bolts are not adjusted to the utmost possible degree of accuracy, and the smaller shaft is in some cases required to resist the same side strain from tight belts to which its more sturdy predecessor was subjected; it therefore deflects, and if the bearings are solid, they wear out much faster than those of the larger shaft. Again, the angle of torsion varies inversely as the fourth power of the diameter ($\text{diameter}^4$), and this reason alone requires considerable allowances to be made on the diameters of small shafts of great length, when indeed it does not totally prohibit their use.

The line shafts are often driven by belts passing from a wide pulley on the crank shaft, and forming the fly wheel; or on the second-motion shaft, which is then driven by spur gearing, as before described. The drum or pulley in question is usually arranged to accommodate the same number of separate belts as there are of line shafts to drive, and a belt passes directly to each; or some of the more distant line shafts may be driven by the intervention of others more favourably situated. This system of belt gear, as it is commonly designated, has been the favourite one in some districts for several years, and it has one great advantage over ordinary toothed gearing in its almost absolute silence. Of course its transmitted motion is not positive, and in calculation it is wise to provide for a small amount of creeping, by giving a slightly excessive calculated speed to the line shafts; when this is done, and proper proportions are assigned, no difficulty is encountered. Cases often arise where a belt may be substituted for other and more complicated arrangements, with great success; more particularly with short-stroke high-speed engines. Notwithstanding this, its indiscriminate adoption cannot be too strongly deprecated. No general law can be given to decide its suitability or otherwise, as every case must stand on its own special merits, and be fully investigated, before any reliable decision can be given. It is impossible to give too much emphasis to the statement, that a high surface speed is necessary to the successful adoption of this system. This is not because high speed is a desirable factor when viewed alone, but because the speed increases directly with the number of revolutions, and with the radius of the pulleys, both of which are valuable factors. The work transmitted in HP. $= \left(\frac{\text{lbs. belt tension} \times \text{speed in feet per minute}}{33{,}000}\right)$, neglecting the unavoidable tension on the idle side of the belt. Evan Leigh's rule, published some years ago, is extremely trustworthy, yielding a good working and economical result, the width of the double belt in inches $= \left(\dfrac{66{,}000 \times \text{indicated HP.}}{\text{length of arc of contact on the lesser pulley} \times \text{the speed}}\right)$, speed in feet per minute. The correctness of this rule has been repeatedly confirmed, both by success and by failure. An impression is current to the effect that long belts possess advantages over short belts in their capacity for work. The rule makes no distinction as to length, as no difference can exist in the case of vertical belts, and the difference, if any, must vanish as the vertical is approached. It may be noted that the angle made by the plane containing the centre lines of the connected shafts, with the horizontal plane, would claim equal attention with the distance of the centres. When the centres are near together, and the connected pulleys are of different diameters, the length of the arc of contact on the smaller pulley is decreased; this is accounted for in the rule. Short belts also require more close attention to maintain an equal and constant tension, but this only acts to a small degree. A usual speed of belting at Lowell, Massachusetts, is 1 mile per minute (say 5,200 feet) with pulleys 10 to 12 feet in diameter; with these conditions the belts are loaded to transmit 10 indicated HP. per inch in width. This is less than they would safely carry according to the above rule; but a belt as made at the present day is stronger than those made at the date of the Lowell mills. At the present time some of the Lowell belts run very crookedly, thereby proving that they do not possess the same strength throughout. The width of many belts (more particularly when replacing arrangements of other kinds) has been fixed by rule of thumb, mostly with success because decided of excessive width. In replacing toothed gearing by belting, there is occasionally a difficulty in providing sufficient room for belts of ample proportion to cope with the required work. Under these circumstances, they are often applied of meagre proportions, causing failure, more or less complete. In the case of small pulleys, it is unsafe to take any given load as a basis, because the requisite tension on the idle side cannot be readily determined. With pulleys of 3 feet 6 inches to 4 feet 6 inches this back tension becomes excessive; the power required to bend the belt around the pulley becomes also an important element in the conditions to be regarded. With pulleys of less diameter than 3 feet 6 inches, double belts should not be used, as their economy then becomes doubtful.

The most usual material for belting is leather. In the best belts it is employed in the form of strips 2 inches to 3 inches wide.

These strips are carefully jointed together to form continuous strips, the whole length of the belt; they are then stitched, pegged, or riveted, in such manner as to break joint longitudinally, the transverse joints of the strips being also separated as much as possible. In completing the endless belt in position, it would be inconvenient to make the long splicings necessary to continue the same system throughout, even if it was possible to stitch, peg, or rivet it by hand firmly enough to prevent the rising of loose ends. The ends are therefore left square, and are connected by outside metal plates; these plates are in some cases attached by taper recess-head screws, very similar to those used for wood; in other cases bolts are used, with a thin feather-edged flush head against the pulley, and the nut on the outside of piecing plate, the head being provided with spikes to sink into the belt, and prevent its rotation in tightening the nuts, or in subsequent working. The plates fixed by bolts provide a secure and convenient method of joining the two ends together; they, however, cause a slight shock on coming into contact with the pulleys; they also prevent the use of a guide or tightening pulley pressing against the outside of the belt, which would be often convenient.

In the best double belts care is exercised to ensure that the points of all splicings are directed towards the same end on both faces throughout; then the end towards which the joints are directed is arranged to follow the other when the belt is ultimately set to work. Obviously, this cannot be done on both sides of a belt of single thickness, and the piecings are set to run the proper way on the inner side, as the most important one. If a guide pulley is applied (as is often the case) to the outer side of such a belt, the corners of the splicings on that side usually begin to rise, sooner or later, and can scarcely be made secure afterwards. With narrow belts the best way is to twist the belt through one half revolution, opposite the guide pulley if necessary, causing all the pulleys to work on the same side of the belt. The longitudinal direction of splicings is shown in Fig. 52, and the cross section of belt in Fig. 53, each referring to belts of double thickness.

Belts of American manufacture are now in use in England. They are of double thickness. The longitudinal joints are neatly spliced and arranged as above described; either the two layers are in single widths or the strips are much fewer in number than in the English belts. The final joint is made by means of stout leather thongs, or laces, so that no projection exists to interfere with the use of guide or tightening pulleys, on either side of the belt as may be most convenient. A novel form of belting is manufactured by Messrs. Aitken, of Helmshore, from cotton yarn: special looms are employed, which will make the belts of any length, width, or thickness. Hitherto they appear to answer well as main belts, but their applicability as machine belts is doubtful, unless with special prongs or strap-shifting forks. Steel belts provided with projections engaging recesses in the pulleys, are also employed; they give a positive transmitted motion, but their success remains to be proved.

All arrangements of belt driving are wasteful in power, as evidenced by the fact that longer journals are necessary than with toothed-gearing connections; this is especially the case when the only admissible pulleys are of small diameter. The supporting bearings ought to be as near to the pulleys as possible, to prevent deflection of shafts. Where the pulleys cannot be carried by bearings close to them, an arrangement is sometimes adopted which is a close approach to the self-adjusting bearings already mentioned. When the belt works in a vertical direction, the pedestals are held to a horizontal fixed plate by one pair of holding bolts to each pedestal; passing from one of these bolt holes to the other, a ridge is provided on the fixed plate or on the pedestal base, which ridge projects from the general plane of the face of the casting; or a loose strip of some material is inserted in that position (across the line or shaft in plan). The nuts of the holding bolts are then screwed down gently and locked. The pedestal will then follow the shaft when the latter is tilted up at either end a small amount from the level position. Then, notwithstanding a slight deflection, the whole length of the bearing carries the weight equally, and overheating is prevented, or stopped if not too far advanced. If, however, the heating has been at all serious, any labour expended with a view to its mitigation is usually wasted, unless directed towards re-bedding the shaft in its bearings. The reason of this is, that when a bearing brass is heated to any serious extent by friction, the bore tends permanently to assume a more concave form, as though its cross section formed a portion of a circle of smaller diameter than before; therefore, unless bored too large in the first instance, it is now too small, and nips the shaft, causing an increase of friction, which already is excessive. Many times trouble has also been caused by making the brasses of the full length (or nearly so) of the journal in the shaft. The expansion of the brass when heated is greater than that of the shaft. This is partly because the co-efficient of expansion by heat is greater in brass than in iron, and partly because the brass is of less bulk and greater conductivity with respect to heat. Both of the latter causes tend towards the same result as the first one; that is, the brass expands by heat much more quickly than the shaft. The more rapid increase of length of the brass causes it to press severely on the ends of the journal if not provided for by leaving clearance in the first instance; this action again heats the bearing still more, and the reciprocatory action continues till the brass is sufficiently shortened. Often a short time suffices for the total ruin of brasses and journals from this cause. When that occurs, the only safe proceeding is to take out the shaft, re-turn the journals, and fit new brasses.

Each pair of brasses ought, in the first instance, to be made shorter than the journal in which they are employed. This may be condemned as an unworkmanlike proceeding; but when the bearings in the same works under the charge of one man (and his time not exclusively devoted to them) may be counted by hundreds, it is easy to see that any one is liable to be overlooked at a critical moment in some period of its life. In old works, some important bearings are quite inaccessible when the shafting is in motion, and therefore in an unknown condition, but this is seldom found in new works. It then becomes necessary to adopt all simple and practicable measures whereby over-heating may be prevented. The amount of longitudinal clearance to be allowed may vary from zero to 10 per cent. In determining the amount due regard must be paid to the maximum period that the part may be expected to work at any time without attention, also to the possible effect on the connections of the shaft produced by its maximum longitudinal movement. If a shaft has no connection otherwise than by means of belts, a considerable longitudinal movement introduces no evil to compare with the good effect produced in smoothing the bearings. The fact that the first prolonged heating of the shaft and brasses causes incalculable mischief cannot be too strongly insisted upon.

Belts connecting two parallel shafts by means of pulleys whose surfaces are not absolutely cylindrical, tend to run towards the part of each pulley that is of the largest diameter. This is caused by the lateral stiffness of the belt, which leads it towards the higher side. If two shafts whose axes are in the same plane are not parallel to each other, the belt connecting them by pulleys with cylindrical surfaces will tend to run towards that part of the shaft in which they are nearest to each other. The opposite condition is usually deduced from the acknowledged fact with respect to non-cylindrical pulleys, but if this was the fact it would follow that the action of a strap-shifting fork would be a reversed one. In a belt connection by pulleys with cylindrical surfaces, the belt must leave each pulley in the plane traversing the centre and perpendicular to the axis of the pulley with which the belt comes next in contact, whether the system comprise two, three, or more pulleys. Paying due regard to the above facts, a belting arrangement may be sometimes employed for the connection of two shafts not parallel to each other, and at different heights, with or without the assistance of one or more guide pulleys.

The pulleys employed with belts are usually of cast iron, carefully turned, polished on the surface, and balanced. Occasionally, in wet situations, they are applied without turning on the surface; and when carefully made and balanced there is no reasonable objection to such pulleys. Some difficulty is encountered in casting them sufficiently true all round without showing the mould parting, and with constant amount of cross or lateral curvature. Disc pulleys are more reliable as to form than the usual arm pulleys, but are inadmissible for large diameters.

Wrought-iron pulleys have been largely made, most frequently in halves, with the rim jointed by lapping and passing screws through. These screws are sometimes troublesome in failing to hold the two parts securely together, and would probably be replaced by rivets with advantage.

Smaller pulleys for driving the separate machines are made of cast iron, with cylindrical faces when required to accommodate a shifting belt, and convex for a belt which is not shifting. A pulley for driving a light line of shafting is shown in Figs. 43 and 44, Plate 3.

Another system of connection is by ropes, usually of hemp, but occasionally of cotton. The conditions of application are nearly the same as for belting. The two pulleys are provided with rims of much greater thickness than for belting; V-shaped grooves are in the first instance cast, and afterwards turned with a greater or less degree of accuracy. The rope presses against each side of the groove with a kind of wedging action, taking firm hold of the same; the bottom of the groove is turned out to a circle of a radius smaller than that of the rope, with a view on the one hand to avoid the possibility of the rope touching the bottom of the groove, and on the other hand to reduce the depth of the rim. This system is not making so much progress as it would do if reasonable care were taken to ensure the same working radius for each rope in the separate grooves. Pulleys are occasionally to be found the turning of which is of the lowest class, while the utmost care and skill are necessary to success. A convenient test for the accuracy of the grooves is to fit a card or thin piece of wood to one groove quite closely, then to lay a straight-edge across parallel to the shaft, and mark off. The gauge may be applied to another groove and again marked. If in the second and subsequent grooves the sides are a close fit and the marks coincide, the pulley may be said to be satisfactory. Many pulleys are far from answering to the above tests, as the card does not fit the grooves in a similar manner, and indicates a difference in the working depth. Because the rope fits to the sides instead of the bottom of the groove, it sinks more or less into the groove, just as the cross mark on the card does. Now the lineal amount of rope paid off varies as the radius $\times$ 6·2832, and varies in the different ropes by an amount equal to $m \times 6 \cdot 2832$, where $m$ = maximum difference in the depth to which the gauge mark (and therefore the rope also) descends; in an exactly equal degree does the amount picked up on the other side of the pulley differ in the separate ropes. Ultimately, however, the amount picked up in each groove and paid off on opposite sides must be equal, unless the groove in the fellow pulley which engages the same rope is similarly proportioned, or unless (as is rarely the case) the errors are tolerably compensated by each other. The lineal amount can only be equalised by one rope being more stretched than the other in transit between the pulleys, and being simultaneously slackened on the opposite side. This evil becomes so large at times that the removal of some ropes actually relieves the remaining ones.

It is also advisable to obtain the full number of ropes for the connection of any pair of rope pulleys at once, thereby ensuring that they will be all of the same diameter. Grooves are usually provided in the first instance for a greater number of ropes than are at once applied, the deficiency being made up as the work transmitted increases. If four ropes are provided in the first instance, while grooves are made for six ropes, the four ropes inevitably stretch during the first period of work, correspondingly decreasing in diameter; if from one spinning, and with well-proportioned pulleys, they will gradually get into good working order. It may ultimately be found that another rope is required. This may (possibly) be obtained of an initial diameter equal to the ultimate diameter of the former ropes; after a short period of work it will be found to be stretched to a diameter smaller than the others, with the attendant evils as described. If, instead of this, the rope be obtained of a diameter equal to the diameter of the former ones before application, and of the same texture, &c., the difference will arise during the first part of the working life of the supplementary rope, while it may possibly become practically similar to the others, suffering more or less severely in the process.

It is obvious that if the grooves are correctly formed, and the pulleys of sensibly equal diameter, another rope might be added at any time, of any diameter that the grooves would accommodate; as an equal difference in the radius would result in each pulley, no inconvenience would result. But usually there is a great difference in the diameters, for often one pulley is about thrice the diameter of the other. The grooves are arranged so that, in section, the sides make an angle of 45° with each other, more or less, according to the practice of different engineers. It may prove desirable to make this angle more acute in pulleys of small diameter, and less acute in pulleys of large diameter. By that means the rope would be more securely held on smaller pulleys, where the tendency to slip is greatest. The power consumed in forcing the rope into the groove would also be reduced in the case of a large pulley, with the wear and tear attendant on the process, the acute angle being in that case not necessary in the same degree to prevent slipping. By this means, however, the inconvenience encountered in adding ropes would be aggravated.

The splices of ropes, however skilfully made, prove either thicker or weaker than the body. In the former case they cause an unsteady transmission, while in the latter they break before the rope is worn out. Rope-driving arrangements appear to be most satisfactory when a number of ropes amply sufficient for all probable requirements are provided in the first instance, carefully watched, and defects promptly attended to. In other cases it is difficult to ensure an equal strain upon each separate rope, even with good pulleys.

The rope-driving principle, has advantages over belting in first cost, and in greater range of applicability to non-parallel shafts without the use of guide pulleys; otherwise their merits are nearly equal when well proportioned and constructed.

Almost all machines are provided with fast and loose pulleys for convenience in stoppage and starting. Long machines which require frequent stoppage are provided with a rod throughout the whole length, so as to be promptly stopped or started from any part. Long machines with few stoppages have a lever at the driving end. In these machines the loose and fast pulleys are contained in the machine, and provision is sometimes made for automatic stoppage when certain contingencies arise. The loose pulley does not bear directly on the shaft, but on a cast-iron bush (which is held by a set screw) bored to fit the shaft, turned on the outside to fit the eye of the pulley, and furnished with a collar to retain the loose pulley close to the fast pulley. This cast bush fulfils several offices; two surfaces of cast iron wear better together than cast and wrought iron; the bush is easily replaced by one of larger diameter, when it or the pulley becomes worn; the bearing area is increased, and the wear thereby reduced. A pair of fast and loose pulleys are shown in section in Fig. 46, with plate webs for neatness and cleanliness.

Machines which absorb much power, and therefore require heavy belts, or which are stopped on few occasions and for long periods, are provided with a different modification of the loose pulley. Instead of being placed on the shaft which is driven by the shifting belt as above, they are placed on a line shaft close to a bearing (shown in connection with an ordinary flat pedestal in Figs. 47 and 48, Plate 3). The pedestal, whether a flat or side one, is provided with top and bottom brasses and cast-iron cap. On that side of the pedestal which is nearest to the pulley, and cast along with the pedestal, is a semi-boss projecting sufficiently to pass through the boss of the loose pulley, a similar semi-boss on the cap completing the hollow cylindrical boss which on the outside fits the bore of pulley, and its inside is cast sufficiently large to clear the shaft. Obviously, the screwing down of the cap to take up the wear of brasses disturbs the cylindrical form of the projecting boss, but as the loose pulley never revolves, except during the transfer of the belt from one pulley to the other, there is no necessity for an accurate fit. For the same reason, a set screw with rounded point projecting into the bore of the pulley, and working in a groove of semicircular section, turned in the boss, furnishes means sufficient to prevent longitudinal motion. The advantages of this arrangement are: the belt and loose pulley are absolutely at rest when not required in actual work, thus saving much wear and tear. The shaft being relieved of the tension of belts does not encounter so much friction, and wear and tear of the bearings are reduced. A slight disadvantage exists in the fact that the belt must be set in motion by hand, when the machine requires to be started, in some cases the assistance of a second man being required; its advantages under the circumstances mentioned, however, overbalance its defects, and the arrangement is largely adopted.

The foregoing account is applicable to a large proportion of the best works in the district, and may be said to be a description of what has become the typical practice in such works; very little tendency to abandon the lines denoted being found at present or likely to be so for a long time. The chief modifications take the shape of heavier working parts, greater protection of moving parts, increased accessibility throughout, and better design. Under the last head is chiefly included the abolition of sharp corners, whether re-entrant or prominent, and more especially where great strains must be resisted, as in cast-iron framing and the journals of shafts, crank pins, &c. Another point, in which great improvement has been made, is in the foundations, of which an instance has already been given in upright shafts, and in towers carrying gearing.

In all works the desirability of a ready means of stopping the line shaft in any part of any room is strongly felt. Such a convenience would often provide the means of saving life and property by its prompt use. Clutches, and arrangements for throwing the wheels out of gear, have been adopted, but not largely. Apparatus for the prompt stoppage of the engine has been often adopted in small works for special purposes, and would probably extend greatly if it were not because of the fear of the abuse of such a system. An almost absolute system for the detection of the place from which a stoppage is effected, can be arranged simply by means of paper seals, and it is somewhat difficult to understand the apathy manifested on this point.

The question of hoisting and transit of materials may perhaps be suitably introduced here. The usual arrangement for conveying material from one floor to another is by a box hoist, working up and down a square brickwork isolated well, round which, in many cases, the staircase winds; with a view to safety, the two are sometimes arranged separately. Shafts for dust and ventilation must usually be provided, and by employing the staircase well for one of these purposes there is no necessity for a loss.

The hoisting motion is best obtained by a worm wheel and worm, driven by open and crossed belts for up and down direction. On the worm-wheel shaft are three grooved pulleys, over which are passed separate ropes, attached at one end to the hoist box, and at the other end to a balance weight. The use of three ropes is extending; the centre one is left slacker than the two outer ones, and as the whole are in sight from end to end, the arrangement is safe without being complicated. An important precaution to be observed consists in covering the top of the hoist box with stout boards, for protection from falling bodies.

A useful hoisting arrangement for bales, &c., requiring, however, a cat head, consists in providing a shaft which is kept in motion throughout the time occupied in hoisting operations. A shaft is provided parallel to this one, with rope barrel and large turned cast-iron pulley set opposite a similar but smaller one on the first shaft. The pulley on the barrel shaft rests against a wood brake block, so situated that the load prevents the pulley from leaving the brake block. When the large pulley rests against the brake block the shafts are about $\frac{3}{8}$ inch too far apart for the pulleys to come into contact; but the end of the barrel shaft which carries the pulley is arranged so as to be moved towards the running shaft by means of a bell-crank lever, which causes the two pulleys to come into contact, and the one drives the other. The lever is worked by a cord passing over a pulley, and is raised by pulling the cord, causing the pulleys to come into contact and the load to be hoisted. On releasing the cord the lever drops, and the pulley comes into contact with the brake block, which action is also assisted by the action of the load. By easing the pulley off the brake block the load is lowered more or less quickly according to the degree of separating force. A more elaborate form of this hoist is made by Messrs. J. Barker & Co., Limited, Oldham. It is arranged so that by variously placing one handle, the load may be raised, lowered, held, or traversed in either direction along a beam, or rather suspended beneath the beam. The great convenience of this arrangement is obvious.[1]

The use of traction engines is extending as local carriers, subsidiary to railways. In some cases, they are superseding existing railways where the markets are not more than 20 miles distant. Some complaints are made against the occasional production of smoke, and the discharge, after a period of idleness, of the soot which has accumulated in the chimney. Much of the country in this district is far from presenting a level surface. As a consequence the traction engines are often stopped on an incline, when it becomes difficult to take one wagon to the summit if heavily loaded. There is therefore much slipping, boulders and splinters of stone being freely used, in a manner similar to that in which sand is used on railways. The inevitable result is, that great wear and tear ensue, particularly with the wheels. The hauling apparatus recently devised and applied by Messrs. Aveling and Porter will probably be largely adopted in such cases.

The communication is accompanied by several drawings and tracings, from which Plates 3, 4, and 5 have been engraved.

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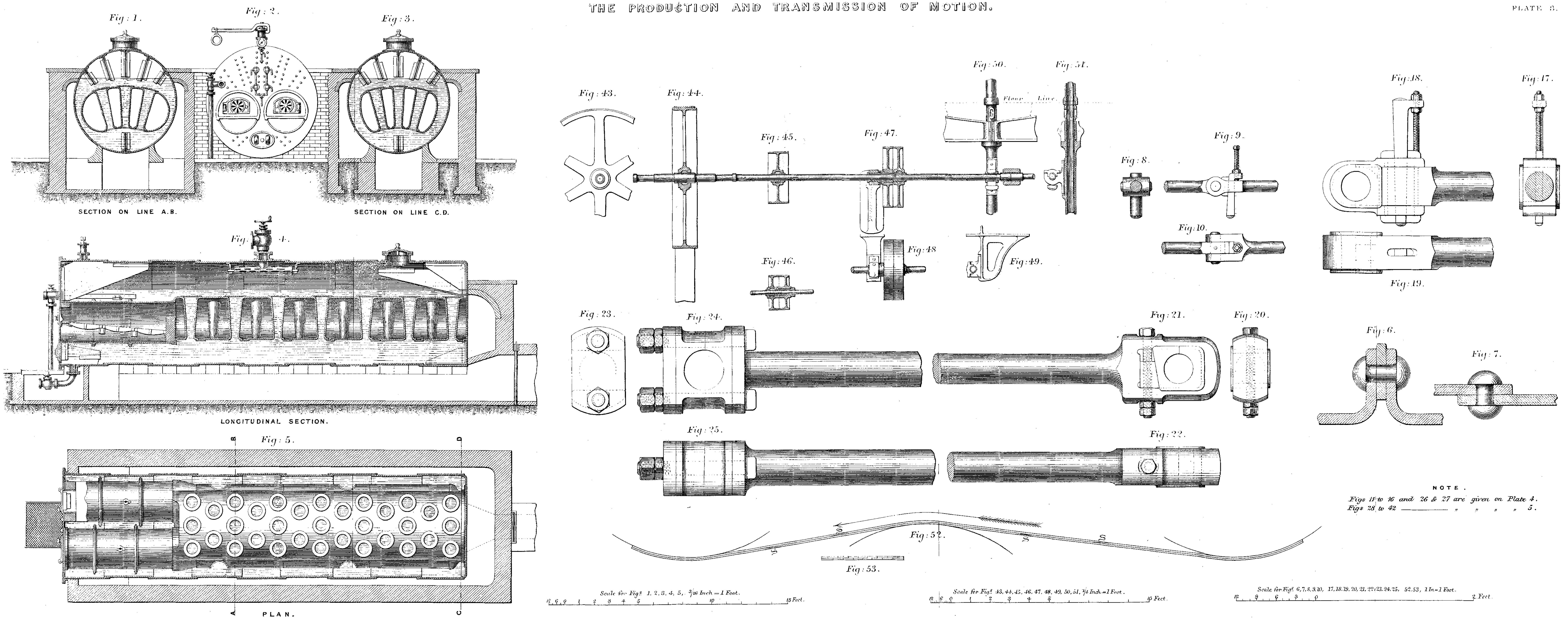
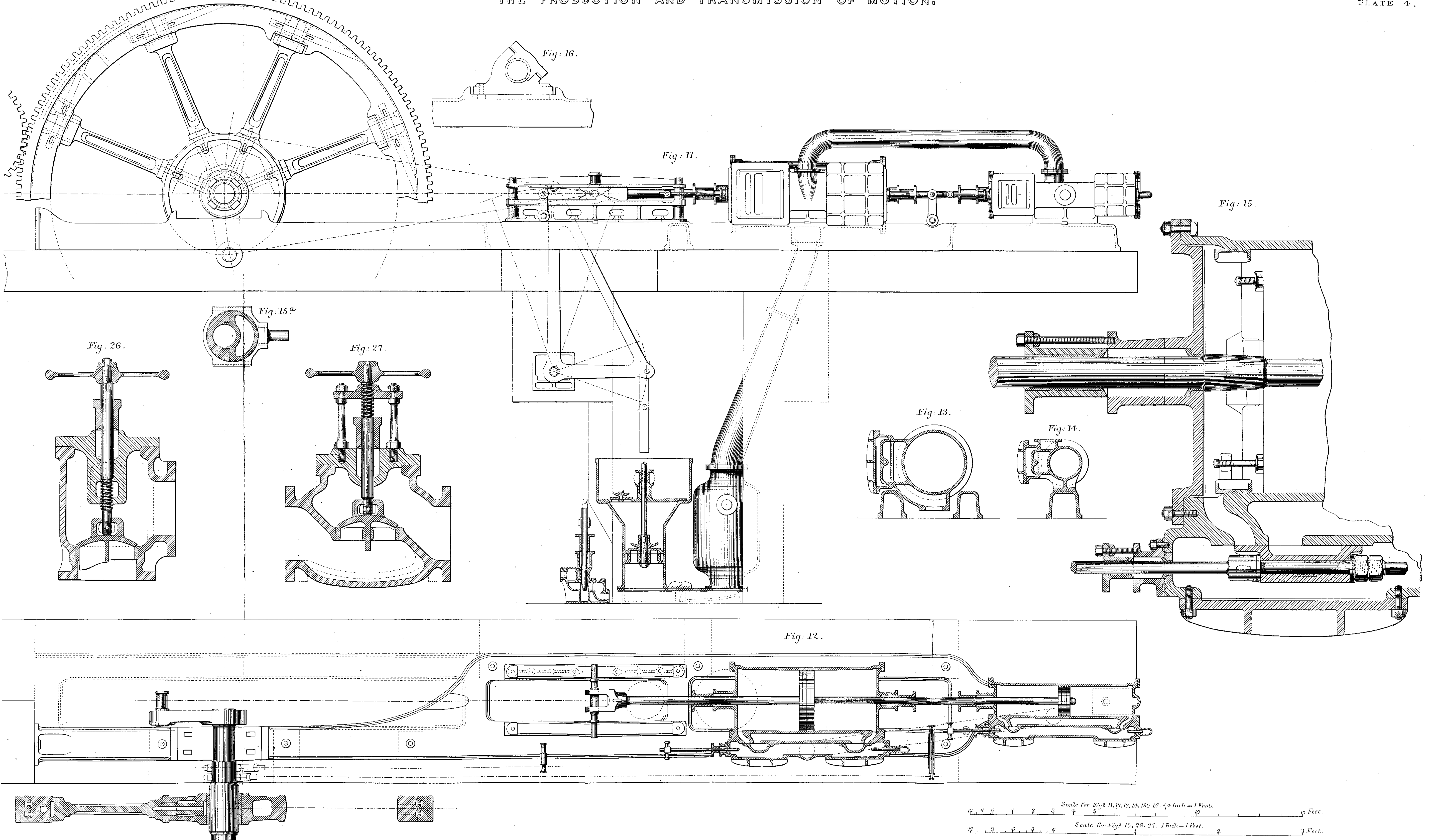
[1] *Vide* "Engineering," vol. xxi., p. 389.




G. W. SUTCLIFFE, DELT.



THOS KELL, LITH 40 KING ST COVENT GARDEN.



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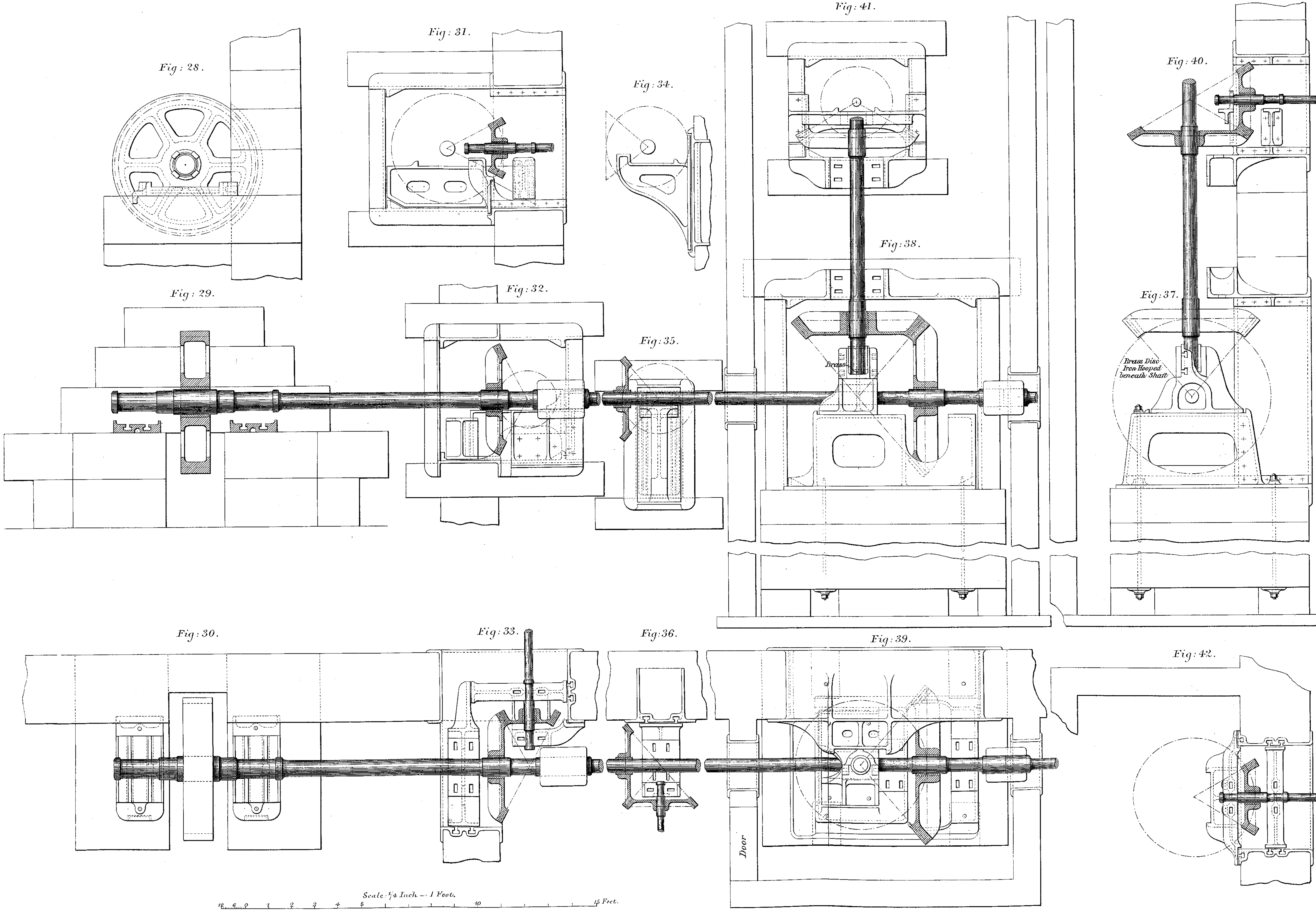
Fig: 28.
Fig: 31.
Fig: 34.
Fig: 41.
Fig: 40.
Fig: 38.
Fig: 29.
Fig: 32.
Fig: 35.
Fig: 37.
Brass
Brass Disc
Iron Hooped
beneath Shaft
Fig: 30.
Fig: 33.
Fig: 36.
Fig: 39.
Fig: 42.
Door
Scale: 1/4 Inch = 1 Foot.
12 6 0 1 2 3 4 5 10 15 Feet.