Breakdowns of Stationary Engines*

Causes Originating in Various Parts of the Machinery

By Michael Longridge

"BREAKDOWNS of Stationary Engines" is rather a sensational title for a lecture, and it is possible that some may have come here expecting a more exciting story than I am prepared to tell. That such may not depart entirely unsatisfied, I will begin by referring to the bursting of the fly-wheel of a pair of horizontal compound engines, indicating about 400 indicated horse-power, and transmitting that power to a weaving-shed by a spur fly-wheel hung upon the crank-shaft. Both cylinders had Corliss valves. On the morning of the accident, the engine started at 6 A. M., and ran as usual till about 7 A. M., when it began to race. The operatives threw off their looms and ran. The engine-man and another man rushed to the engine-house, and with conspicuous bravery made for the stop-valve, which, as usual, was placed between the two cylinders, in a direct line with the fly-wheel. Before they could close the valve, possibly before they reached it, the fly-wheel burst, and both were killed. Some of the flying masses cut the traveling crane in two and unroofed the engine-house. Others cut through the wall of the engine-house and ploughed a path of destruction into the weaving-shed, unfortunately injuring four of the weavers who had been unable to escape. The fly-wheel arms were snapped off short, at their junctions with the boss, and a large piece was torn out of the boss itself, the pinion was broken, and the second-motion shaft—a heavy shaft, 8½ inches in diameter—was dragged from its fixings and bent. The engine foundations were dislocated, and the engines themselves moved bodily from their places. Fortunately the cranks and connecting-rods remained connected, and the cylinders were saved, but all the valve-gear was smashed, and the bed-plate of the condensing engine was badly cracked.

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When the wreckage had been cleared away sufficiently to allow access to the engines, I examined them. I found the governor, the wheels which drove it, and the knock-off gear in order. The front steam-valve of the high-pressure cylinder was open, and the spring in the front end of the dashpot, which should have closed it, was compressed

gesting this as an explanation of the breakdown, however, I should mention that the consolidation of the grease about the dashpot spring may have been due to cold, for, owing to the destruction of the engine-house roof and wall, the engines had been exposed to very cold weather for several days before I saw them. The cause of the breakdown, therefore, is uncertain.

In another mishap of a similar nature, the wreck was too complete to leave any clue to the cause. Fortunately, smashes so complete are the exception, not the rule. Of 71 breakdowns in 100 the cause can be discovered. The table herewith shows the number of breakdowns attributable to each of four groups of causes, as percentages of the total breakdowns during three consecutive periods of ten years.

Fig. 7.
CYLINDER BREAK-DOWNS
PER CENT OF TOTAL
BREAKDOWNS.

and cemented coil to coil, and to the walls of the dashpot, by a viscous brown cement consisting apparently of baked grease. The spring remained compressed when taken out. If it stuck fast when the engine was running, the steam-valve at the front end of the cylinder must have remained open and the racing would be accounted for. In sug-

Cause of Breakdown.	1881 to 1890. per cent*	1891 to 1900. per cent	1901 to 1910. per cent
1. Accidents and causes not ascertained.....	36	27	29
2. Negligence or ignorance of owner or attendants.....	19	22	19
3. Structural weakness, bad designing or workmanship.....	30	30	21
4. Old flaws, defects, or wear and tear.....	15	21	31
Total.....	100	100	100

The classification is not very satisfactory, because it is without clear lines of demarcation. It was begun many years ago and could not afterward be altered without much labor. But if the figures are not very accurate, their meaning is quite clear. Those in line 1 call for no remark, except that the high percentage in the first decennial period is principally due to want of opportunity for investigation. The constancy of the percentage in line 2 suggests that schools have not changed human nature overmuch; but the reduction in the last column of line 3

awakens hope that the vast sums spent on technical education have not been altogether wasted; and, if we take the figures for each of these ten years separately, hope changes to conviction, for we find the percentage falling more or less regularly from 23 in 1901 to 16 in 1910. If more evidence be wanted, it is given by line 4,

Fig. 17.—Piston-rod breaks occur at the cotter holes.

Fig. 18.—Another example of piston-rod break.

which by comparison with lines 2 and 3 reveals how work and age are year by year supplanting ignorance, neglect, bad workmanship, and bad design as causes of breakdowns of steam-engines. I do not mean to say that improvement in design only began ten years ago, but it was gradual. The substitution of the principles of mechanics and arithmetic for rule-of-thumb was slow, and the replacement of old engines by new was slower still, so that years elapsed before the effects of the change became appreciable. But if progress was slow, it was effective. When I began to collect statistics 34 years ago, engines running for the most part 50 to 60 hours per week broke down, on an average, once in $5\frac{1}{2}$ years; now, though speeds and pressures have been doubled, they break down once in $9\frac{1}{2}$ years. That is what we older men have done. Those who are taking our places are expected to do more; and if it will lighten your task to start it with some knowledge of the casualties from which the several parts of steam-engines most commonly suffer, I am here to tell you all I can in the time allotted to me.

Valves and Valve-Gears.—I begin with valves and valve-gears, because at the present time they fail more frequently than other parts of steam-engines. About one third of the total number of breakdowns originate in them. The proportion has nearly doubled in the last 35 years. It has increased concurrently with the use of Corliss and other complicated automatic trip-gears, made up of many pieces. Many of those gears are too weak for the abnormal stresses to which all valve-gears are occasionally exposed; but a large proportion of the breakages is due to failures of the screws, bolts and keys, by which the parts are held together. Many breakages of valve-gears cause little damage, but some have very serious results. Most "runaways" are due to failure of valve-gears or governor-gears controlling them. Every engine, therefore, ought to have a "knock-off" gear acting on the stop-valve or a special runaway valve. Electric gears which can be operated by pressing a button in any part of the works, as well as by the knock-off motion on the engine, are the best. Valves intended to be closed by springs should be actuated by gears which will close them if the springs should fail, as seems to have happened in the breakdown I first mentioned. Not a few breakdowns are caused by eccentrics slipping or becoming disconnected from valves, leaving the one or other of the cylinder ports permanently open or closed. When this port is a steam-port of a high-pressure cylinder, a runaway may follow; when it is an exhaust port, the compression curve may run up high enough to blow off the cylinder end. When the inlet to the low-pressure cylinder remains closed, the pressure in the valve-chest, or receiver, may rise sufficiently to burst one or the other. All these things have happened, and the only preventatives I can suggest are large safety-valves, ferruled to prevent overloading, not only on the cylinder ends, but on the receivers, or pipes, between the cylinders.

Air-Pump Motions, Valves and Buckets.—After valve-gears, the parts which fail most frequently are air-pump motions, valves and buckets. They accounted for 20 per cent of the engine breakdowns 30 years ago. They still account for about 16 per cent, and yet some trouble on the part of engine-tenders, and a little thought on the part of engine-builders, would enormously diminish the

number of mishaps. The cause of half of them is corrosion and neglect. Neglect is almost pardonable when one considers the dark, dirty, inaccessible holes in which air-pumps are so often placed. Corrosion is to be expected, and should be circumvented by putting neither bolts, nuts, screws, rings, valves, nor guards inside the pump. There are both vertical and horizontal air-pumps on the market which can be made without a single fastening inside. If you must use the old-fashioned vertical pump, put no rings in the bucket; water-sealed buckets need not be tight. I have known two plain buckets, $\frac{3}{4}$ -inch less diameter than the pumps in which they worked, giving 10 pounds vacuum in the cylinder.

Secure the bucket and guard against a collar or cone on the bottom of the rod, by a nut or cotter on top, as in Figs. 1 and 2, instead of tapering the cone downward and putting the nut below the bucket, as in Fig. 3, to drop off and allow the bucket to fall into the foot-box. Slide the foot-valve grid into grooves in the foot-box, and cast the guard upon the foot-box cover, as in Fig. 4, so that there may be no bolt to corrode and break. If the pump be horizontal, the bucket will not be water-sealed, and the piston must either be renewed when it has worn down, or fitted with a ring. The piston, being a plain disc, can be screwed and shrunk on to the rod. Do not use combinations of brass and iron. If the water be at all acid, a brass-sheathed rod will be cut through, where the sheathing ends, as with a knife. Avoid trunk motions. The screw of the eye at the bottom of the trunk will suffer from corrosion, and the pin from grit. Every trunk motion I have had to deal with has broken down at least once. Finally, pay great attention to all fastenings; put the air-pump in a light, accessible place, and if there be any doubt about the free flow of the water from the hot-well by gravitation, make the motion fit for a pressure of at least 50 pounds per square inch on the bucket.

Bed-Plates.—Breakages of engine-frames and bed-plates can be dismissed with a very few words, because

Fig. 27.

by far the largest number are caused by the pounding of the engines, or by wringing up holding-down bolts, after the seats below have been softened by oil. It is true that, with modern systems of lubrication and oil recovery, very little oil should reach the seat, but it is equally true that the modern practice of setting down an unplanned bed-plate in a rough lump of concrete, packing it level with

any bits of iron that may be handy, and grouting it with cement, provides so little material to resist the oil that the grouting ceases to give adequate support even sooner than did the old ashlar seats with gallons of oil poured over them each week. The consequence is that breakages of frames, bed-plates, and the like, still bear as large

Fig. 30.—Piece cut from steel shaft after break.

Fig. 31.—Piece from same shaft magnified four times.

a proportion to the total breakdowns as in the early eighties—namely, about 7 per cent. It is quite easy to keep engine-seats free from oil, and dry. All frames and bed-plates should have channels round the feet, as in Fig. 6, troughs under the ends of all bearings, or, when there is only a top and bottom brass, shields like those on the bearings for the shafts of dynamos and motors. The bed-plates of vertical engines of the marine type should have the bottoms of the crank-pits cast with them.

Where a bed-plate is to rest on ashlar, it should be planed on the underside, and rubbed to a perfect fit upon the dressed surface of the stone. When it is to rest on concrete, the bottom surface, which is to be supported by the cement grouting, should be as wide as possible, instead of a strip a few inches wide along each side; and holes should be arranged at intervals for running the cement into the space between it and the concrete seat. In my opinion, it is best to plane the underside at each holding-down bolt, and level the bed-plate upon a pair of broad machined folding-wedges at each bolt-hole, as in Figs. 5 and 6.

Cylinders should not be placed on concrete, but on iron frames or plating, according to their shape. This plating should have holes for running cement between it and the concrete seat, and there should be raised rims round these holes, round all openings for pipes, and round the edges, to retain oil and water; also machined facings for the cylinder feet to stand on, so that they may slide when the cylinders expand. Horizontal engines should be firmly held to the seat at the crank end, but the cylinder end should be free to slide. Never tighten a holding-down bolt till the superincumbent weight has been lifted from the bed-plate, and the space below it, if there be any, carefully packed.

Cylinders.—In old days, breakages of cylinders were generally due to water or stray piston-bolts; now the majority are caused by water or structural weakness. The diagram, Fig. 7, shows plainly what has happened. The abscissæ represent successive periods of five years from 1879 to 1913, the ordinates the breakages of cylinders, as percentages of the total number of breakdowns, in each quinquennial period. The year 1883 may be considered as about the end of the old time, when 80 pounds was a high pressure, and a tallow kettle simmered on the cylinder. About 1890, steam of 160 pounds pressure began to be used in mills, and many old cylinders, reduced in diameter by liners, were supplied with it regardless of weakness of ends and valve-chests. Also many of the new cylinders made about that time were too weak. The curve shows how breakages increased, and subsequently decreased as the new conditions were met by improved designs.

Before referring to some of the faults of design which provoke fracture, I would like to remind designers of cylinders and other parts that deformation, or strain, induces stress. It is often an advantage to consider what deformation would occur in the thing they may be designing if it were made of India rubber. Figs. 8 and 9 show a cylinder made in 1894 for a pressure of 120 pounds per square inch. It worked for 8 years at that pressure, then for 18 months at 140 pounds, and finally for 6 months at 160 pounds, when the two longitudinal cracks AA, Figs. 8 and 9, were found. Regarding the cylinder as made of

India rubber, it would be seen at once that the two sides of the upper part of the jacket, Fig. 9, would be forced outward, and bending would occur just where the cracks appeared. Even if the outer cylinder were complete, as in Fig. 10, the steam-passage would be a source of weakness, because its bottom surface, having the same pressure on both sides, would flatten out and allow the same kind of deformation as before. The flat top of the steam-passage, Fig. 9, was also weak, and would probably have cracked along its longest edges in course of time, for the stress calculated by the formula for flat surfaces:

$$f = p \times \frac{b^2}{2l^2} \times \frac{\mu}{\mu + b^4}$$

given in Unwin's "Machine Design" was 16,000 per square inch.

In good designs, I have found f to lie between 3,000 and 8,000 pounds. The wide variation is not surprising when one considers how much the strength of castings is affected by shape and crystallization during cooling. In dealing with castings, experience is the only safe guide, though for purpose of comparison a formula may be useful as a rough criterion.

Figs. 11 to 13 show the high-pressure cylinder of a triple-expansion engine, made five or six years later, for a boiler pressure of 160 pounds, and a pressure of 70 pounds in the receiver between it and the I. P. cylinder. Soon after it was set to work, it cracked through the bars across the ports *A*, Fig. 13, and then from the corners *B* of the ports. It was replaced by another of the same design, which quickly failed in the same way. Here there was a load upon the end of 92 tons during admission and 52 tons during exhaust. If the cylinder had been rigid and the load distributed uniformly over the cross-section of the metal, the stress would nowhere have been excessive; but with a cylinder of India rubber, it is easy to see how a large part of the load would be concentrated on the bars across the ports, and how, on their failure, the deformation of the end would intensify the stresses at the corners of the ports. Also deformation of the barrel from a circular to an oval cylinder would be caused by the external pressure on the top and bottom, and the absence of an equal external pressure on the sides. This deformation is sometimes sufficient to make well-fitting pistons seize.

Finally, passages cast round cylinders, whether for live steam, as in these figures, or for exhaust, frequently originate circumferential cracks in the barrel, owing to the unequal expansion of the metal inside and outside them. They are also likely to cause sponginess where the walls join on to the barrel. The design is not one to be copied, and I cannot but suspect that if the draughtsman had kept the India rubber model in mind when he was drawing it, these cylinders would never have been cast.

The usual way of strengthening the ends of cylinders is to lengthen the distance between the ports and the end flanges, and then to lengthen the spigots on the cylinder-covers, to fill the increased clearance spaces. Unfortunately the clearance surface at each end of the cylinder is thereby increased by two rings, each equal in area to the circumference of the cylinder multiplied by the length of the spigot—rather an important disadvantage when saturated steam is to be used, and guarantees of steam consumption are required. Of course, it is possible to suppress the surface by making the spigot fit the cylinder, but then it is extremely difficult to get the cover off; or by making the joint at the surface of the spigot next the piston, but that may be as dangerous to the cylinder and cover flanges as a ring of jointing material placed entirely inside the bolt circle, unless special precautions, which I cannot now describe, be taken. The practical alternative seems to be to make the space so narrow that, when it is filled with water, the surface of the water presented to the steam in the cylinder will be so small that the condensation and evaporation will be negligible. The late Mr. Bryan Donkin made some experiments for me to determine how narrow the space would have to be, and he told me that under most conditions a difference of 1/16-inch in the diameter of cylinder and spigot would probably suffice to keep the space filled with water and the surface inactive. His experiments were made with clean steam; with oily steam, the space might possibly be increased, because the fluid filling it would be more viscous than pure water.

If the bell-mouth and spigot be bored and turned slightly conical, and to fit each other when there is no jointing material between the flanges, then, when the jointing material is put in, the water space will be very narrow, but will widen rapidly as the cover is being drawn off. When the bell-mouth is very short, deformation, and local stress at the corners of the ports, may be reduced by bolting a deep cover, like a short cylinder with its two flanges tied together by ribs between the bolt-holes, to the cylinder flanges. The difficulty of designing cylinders for high pressures and temperatures is much enhanced by attempting to combine valve-chest, steam and exhaust passages, and cylinder in a single casting. It is far better to make cylinders plain tubes, or plain tubes with nozzles

near the ends, and to carry the valves in suitable gable ends, cylinder-covers, or valve-chests bolted to the flanges on the barrels. The arrangement allows the barrel to be made of hard metal, giving an index number of about 40 when tested by Shore's scleroscope; while the more complicated castings, containing barrel, valve-chests, and passages, cannot be made with metal giving a higher number than about 36. Moreover, the distortions produced by temperature, and by the growth of gray cast irons under temperatures as low as 500 deg. Fahr., imperatively demand the very simplest forms.

Damage by Water.—Leaving cylinder design, I now wish to say a few words about breakdowns from water. Water is seldom carried over from boilers in sufficient quantity to cause damage when steam-pipes are properly laid, as shown in Figs. 14 and 15, with a continuous fall from the boiler junction-valve to a well-pipe or separator near the cylinder, and a continuous rise from the separator to the engine stop-valve, which should be at the highest point of the pipe, and as near to the cylinder as possible. Pipes with vertical bends, Fig. 16, are apt to harbor water when only a light breeze of steam is blowing through them, and this water is carried forward in gulps when the breeze becomes a hurricane. But even these gulps seldom do damage except when they get into the upper ends of vertical cylinders, the exhaust ports of which are shut before the end of the up stroke, or into horizontal cylinders with the exhaust-ports on the top. Generally the water causing breakdowns comes from jet-condensers, when an engine is running slowly, and the injection cock has not been shut, or the vacuum destroyed. The flow into a jet-condenser depends upon the vacuum therein, the flow out of it upon the speed of the air-pump; and it may easily happen, when an air-pump is running slowly, that more water may enter the condenser than the air-pump can discharge, in which case the excess first fills the exhaust-pipe and then runs into the cylinder. For this reason, when the air-pump is driven by the main engine, the vacuum should always be destroyed by letting air into the condenser before beginning to shut off steam; and when it is driven by an independent engine this engine should be started before, and stopped after, the main engine. It is often extremely interesting to discover how water has got into a cylinder, but time will not permit of going into this point now in further detail.

Pistons and Piston-Rods.—Previous to 1899, breakdowns originating in pistons averaged 2.5 per cent of the total breakdowns. Since then, the average has been 4.8 per cent. The rise came suddenly soon after the rise of steam pressure to 160 pounds, but I do not know that there is any connection between the two. During the last three years, 47 per cent of these breakages have been caused by stray screws, or bolts, or broken rings, 17 per cent by water, and the rest in various ways. When the substitution of plain pistons, with cast-iron rings sprung into grooves, for pistons with junk-rings and fancy packings is complete, breakdowns from piston breakages will be halved.

Breakages of piston-rods occur nearly always at the cotter-holes at the crosshead end, and are due to concentration of stress at the edges of the cotter-holes, as explained by Dr. Unwin on page 223 of Part II of his "Machine Design," 1912 edition. I endorse his explanation, and Figs. 17 and 18 afford convincing proof that cotters do bend in the way indicated by the drawing in his book. These figures are from photographs of two pairs of folding-wedges from the piston-rods of two old beam-engines after exhibitions of water in the cylinder. I would have shown single cotters, such as are generally used, but I, unfortunately, have none by me at the moment; moreover, I am not sure that folding-wedges are not preferable to single cotters, because a piece with parallel sides is easier to fit accurately than a tapering wedge; but if they be used, it is clear that the position of the pieces shown on the screen should be reversed, the broad piece should bear on the rod, and the narrow on the crosshead.

Connecting-Rods.—Fractures in connecting-rods occur generally at the ends which couple the rods to the crank-pins, or crosshead centers; sometimes in the forks, when the crosshead ends are forked; and rarely in the bodies of the rods.

The commonest types of end are: the box or closed eye, Fig. 22; the open strap secured by gib and cotter, Fig. 21; and the marine end with cap and bolts, Figs. 19 and 20. Of the three, the marine end, Figs. 19 and 20, is, in my experience, the least safe, because of the frequent breakages of the cap-bolts, usually at the junction of the head and shank, or at the screw thread. The principal causes of these breakages are: repetitions of stress or blows, abrupt changes of form, cross-bending stresses produced by the inertia of the rods. The antidotes are: for the first, probably spare bolts, to give alternate periods of rest and service. Annealing is risky in unskilled hands. For the second, large fillets at junctions of heads and shanks, gradual reductions of diameters in shanks, and threads with rounder roots than the British standards; but fillets, cones, etc., at changes of section

must be very neatly finished where they merge into the cylindrical parts, and threads very carefully chased, especially in steel, to leave nothing in the nature of cuts or scratches to initiate cracks. Also stiffer palms and caps are often needed to prevent deformations, which would be apparent on reference to an India rubber model, and consequent concentration of stress on one side of the bolt.

For the third, bolts with *areas* instead of *diameters*, approximating to half those of the rods and the substitution of jaws, Fig. 20, long enough to hold at least one of the brasses for palms, Fig. 19. Rods like Fig. 19 should be taboo, for it is easy to imagine the effect of a cross-bending stress if there be any wear or slackness. Bolts of small engines are more heavily strained by this stress than large, because the ratio of their length to their diameter cubed is greater. For connecting-rod bolts wrought iron is a safer material than steel. Open straps, Fig. 21, secured by gibs and cotters are but little safer than marine ends. The jaws of the cotters wear, the straps open under the cross-bending stress produced by the inertia of the rod, and break, generally through the oil-hole. Fracture almost invariably begins on the inside surface, and when it has extended a certain distance, often not far enough to be visible above the flange of the brass, the strap breaks and opens out. Never trust a strap till you have seen the inside surface. Open straps should be kept closed on the butts of the rods by bolts, in place of, or in addition to, the gibs.

Of the three forms, the box end, Fig. 22, is undoubtedly the most reliable. Fractures, when they occur, usually commence, as in the straps, on the inside surface at the hole for the lubricator, but extend slowly enough to be discovered before the rod gets free. Among more than 6,000 breakdowns of engines of various kinds I can only find one where fracture of a wrought-iron or steel box-end caused damage to an engine, and this was the fault of the engine-tender, for the fracture had been visible on the outside surface of the end as far as the lubricator-hole some time before the rod gave way.

Crank-Pins.—The history of crank-pins has been very like that of cylinders, showing a rising percentage of breakages, culminating in the five years 1894 to 1898, and then falling. The rise was due to bad designing, improper methods of fixing, and over-loading; the fall to the more general adoption of the right method of fixing, combined with reduction of stress. In my early days, stresses of 12,000 to 14,000 pounds per square inch were common, and stresses of 50 per cent higher were frequently met with. Samples of the old designs and methods of fixing are shown in Fig. 23. In Fig. 23, *A*, *B*, *C* and *D* are examples of pins with conical or cylindrical shanks fitted into the cranks and secured by riveting, screws and nuts, or cotters. They usually broke inside the crank-eyes, because the fit was not accurate enough to prevent slight movements, which wore and enlarged the ends of the crank-eyes next the journals, thus throwing back the point of support and increasing the bending moments. It will be noticed that the pins were usually put in from the faces of the cranks, so that they could come out and wreck the engine when they broke. *E* and *F* are also typical examples. They have not sharp corners at the junctions of the journals with the collar, though such were not unusual; but the fillets provided, 1/16-inch to 1/8-inch radius, are too small to be effective. Both pins were of steel. The fractures were due to bending stresses. They commenced at the parts of the circumference nearest to, and farthest from, the crank-shaft, and extended inward as shown on the shaded part of *G*, Fig. 23. The lives of these pins were 14½ and 104 million revolutions, and the bending stresses 17,500 and 11,500 pounds per square inch respectively.

The breakage of *H* may have been entirely due to the abrupt change of diameter from 6¾ inches to 7 inches, for piston-rods in direct tensile stress will break with a smaller change than is shown in *I*, Fig. 23; but probably it was started by the shearing action of the edges of the crank-eye, when cooling and contracting upon the pin. The failure of the pin *J* can be attributed to nothing else but shear. It was made of steel, of 60,000 pounds tensile strength, 29½ per cent extension in 6 inches, 60 per cent contraction of area, and a strip, cut out of it after the breakdown, was bent cold nearly double without cracking. It broke under a stress of 9,400 pounds per square inch, after 5¾ million revolutions. I think the crank-shafts *K*, Fig. 23, and *L* are similar cases. The stresses and lives were: for *K*, 8,200 pounds per square inch and 17 million revolutions; and *L*, 5,000 pounds per square inch and 218 million revolutions; the longer life of *L* was due to the lower stress, and probably to a lower pressure exerted by the crank in cooling. I do not think the effect of the pressure of sharp edges, or even of scratches in starting skin-cracks in steel, or the almost inevitable extension of such cracks under recurrent stresses, is sufficiently appreciated. About scratches, I shall say a few words presently. Of shrinkage, I say, without hesitation, that the portion of the internal piece upon which an external piece is to contract should be of larger diameter than the adjacent portions, and that the external piece

should entirely cover the enlarged portion of the internal. *M* Fig. 23, shows how a crank-pin should be put in, while *N* and *O* show the proper method of shrinking a piece upon a shaft. It is to the adoption of this principle by many engine-makers, combined with reduction of working stress to 8,000 pounds or 9,000 pounds per square inch, that the reduction in the number of crank-pin breakages must be attributed. The failure depicted in *P* is one of those things "no feller can understand."

Crank-Shafts.—The majority of mill-engines have plain shafts with overhung cranks carried in two bearings. In 1879, these shafts, except a few old ones of cast iron and a few new ones of steel, were of wrought iron. Now nearly all are of mild steel. I think most of the breakages may be ascribed to gradual disintegration by wear and tear, and vibration of spur-gearing, or to the extension of small flaws or cracks existing in the interior of the shaft when new, or developed upon the surface afterward by chilling it when hot. The duration of life measured in revolutions of the engine undoubtedly bears some relation to the stress, but it is uncertain what this relation is. I have been able to get fairly reliable information about a certain number of broken shafts, and from it I have plotted the points shown on the two diagrams, Figs. 24 and 25. Fig. 24 refers to wrought iron, Fig. 25 to mild-steel shafts. In both, the abscissæ represent stresses calculated by Rankine's formula, and the ordinates length of life in millions of revolutions of the engine. Beyond a general tendency to abbreviation of life by increase of stress, the spots show nothing, and I fear the effects of repetitions of stress are here, and in similar statistics always will be, obscured by the other causes of breakage I have mentioned, and the difficulty of getting accurate information about shafts which have worked for many years.

I would like to describe a few breakages of crank-shafts. The first case is one of the disintegration of a cast-iron shaft by wear and tear. The dimensions of the shaft are shown in Fig. 26. It belonged to a pair of beam engines, and was put in, with them, in 1850. In 1900, there were several slight longitudinal cracks in both necks, and in the left-hand neck one 14 inches long, which was said to have been visible for 20 years. On April 7th, 1910, this neck got hot, and the mill had to be stopped to allow it to cool. On April 8th, it ran hot again, and was examined in the evening, when, besides the 14-inch crack, 24 other well-defined cracks of various lengths, up to 5½ inches, were visible, all longitudinal in direction and situated near the middle of the bearing. It then transpired that the neck had been hot in 1908, and had been cooled with water. It was therefore doubtful whether the cracks were more than surface cracks caused by the water; but having regard to the age of the shaft, its replacement was advised. The decision was justified when the shaft was broken under the tup. The fracture surfaces, shown in Fig. 27, did not coincide with any of the longitudinal superficial cracks, but were transverse and diagonal. The new and white fractures were caused by the tup, while the dark old fractures, blackened by oil, were very considerable in extent, and showed the disintegration of the interior of the shaft by age and wear. The right-hand neck showed similar black and white areas when broken. Some particulars of the life of the shaft, so far as they could be ascertained, may be of interest:

	1850 to 1873.	1873 to 1897.*	1897 to 1910.
Millions of revolutions.....	70	76	41
Initial load on piston (in pounds).....	90,000	56,500	51,000
Bending stress on neck (pounds per square inch).....	3,650	2,270	2,050
Load on piston at cut-off.....	76,500	36,800	33,960
Bending stress equivalent to bending and torsion by Rankine's formula (pounds per square inch).....	3,750	1,930	1,580

* Mill stopped one year during this period.

Total life, 187,000,000 revolutions. Average stress, 2,700 pounds per square inch.

Fig. 28 shows the gradual destruction of a wrought iron shaft 28 years old, during the last 4 years of its life. The first crack, probably the opening of a cinder mark or imperfect weld in the forging, was seen in the left-hand neck in October, 1888. The development during the following 3 years is shown in the figure. In 1892, it had farther extended, and there was also another crack in the right-hand neck, of the development of which I have no record. The cracks shown by the full lines are on the part of the circumference farthest from the crank-pin, those shown by the dotted lines on the part nearest the crank-pin. The stress in the shaft was 8,500 pounds per square inch, and the life 123 million revolutions.

When speaking of crank-pins, I referred to the effect of scratches or skin cracks upon steel, and I want to illustrate this point. Fig. 29 shows the crank-shaft of a pair of McNaught beam-engines running at 32½ revolutions per minute. After the shaft had worked 9 years, a very interesting crack appeared in one of the necks, as shown in Fig. 29. For a length of 5½ inches, it was absolutely straight, without the jagged edges a crack usually presents, and followed the line of a circumferential scratch, such as might have been made by a knife held against the revolving shaft. At either end it

branched off diagonally for a length of 2½ inches, each extension having the normal appearance of a crack. I marked it with a center-punch, and let the engine run on. It ran for 4 months, and then the crack began to extend so rapidly that the shaft was thought to be unsafe, and was replaced. The stress on the neck was about 7,500 pounds per square inch, and the life of the shaft only 51½ million revolutions.

Fig. 30 shows a piece cut from the neck of a steel shaft, after the shaft had been broken by a tup. The shaft was probably new in 1893. The neck was 13½ inches in diameter and 26 inches long. Test-pieces cut from the broken shaft gave a tensile strength of 25.7 tons per square inch, a yield-point of 14.3 tons per square inch, an elongation of 40½ per cent in 2 inches, and a contraction of area of 60.3 per cent. In 1905, the neck was free from cracks, but in 1908, when the next examination was made, there were a number of fine marks like scratches, principally in a circumferential direction, as clean as if cut with a knife. One in particular, about 7 inches from the fly-wheel end of the neck, was 18 inches long. As the marks were so fine and clean, it was hoped that they were nothing more than scratches, or, at most, skin cracks brought out by heating, and as the shaft was very lightly stressed, only about 4,500 pounds per square inch, it was allowed to run on. A few days later, however, the engine had to be stopped, as black oil appeared inside the bore. The long crack was then found to have extended almost continuously round the neck, following one or other of the circumferential scratches. Fig. 31 shows a piece broken off the shaft and magnified about 4 times, which clearly shows how the small surface scratches were eating into the steel. The cause of the scratches could not be ascertained.

Spur Gearing.—About 1879, when horizontal engines of 800 to 1,200 indicated horse-power began to replace beam-engines of half that power in cotton-mills, when pitch-line speeds were rising from 1,800 feet to 2,300 feet per minute, loads on teeth from 10,000 pounds to 18,000 pounds, and gear ratios from 2¼ to 1 to 3 or 3½ to 1, an epidemic of breakages of spur-gear set in. Some 20 per cent of the engine breakdowns were breakages of spur-wheels on engine crank-shafts, and the pinions driven by these wheels broke almost as frequently. Many of these breakdowns were due to irregularities of pitch, twisted teeth, sinuosities of pitch circles, concentration of pressure on the end of teeth through movement of the crank-shaft, back-lash from the momentum of rope-drums on second-motion shafts, and other faults of workmanship and use common to old-pattern molded wheels and to old engines. But the damage was not confined to these. Many new wheels, as good as mechanical skill could make them, with teeth molded by machinery, spur-runs cast in complete circles, and machined internally, carried on crank-shafts held in pedestals, with brasses adjustable horizontally as well as vertically, broke also.

These breakages were usually attributed to "vibration," "wear and tear," "teeth bearing up to points," and similar effects of a cause ignored or not discerned. That cause, in my opinion, was the use of teeth too long and too wide to "correspond with their environment," the most important fact of that "environment" being the unavoidable motion of the engine crank-shafts on which the driving-wheels were hung.

I will try with aid of Figs. 32 to 35 to make this clear. Fig. 32 represents 3 teeth *A, B, C* of a spur-wheel geared with 3 teeth *D, E, F* of a pinion. The teeth are supposed to be cycloidal, for involute teeth have not found favor with makers of main driving gear. Their length is exaggerated to make this illustration clear. Similar profiles can be set out, according to the rules in the books, to provide 3 points of contact *x, y, z*, and 3 teeth to carry the load. They look very pretty upon paper, but they will not work if the spur-wheel *A, B, C* be forced toward the pinion *D, E, F* by the motion of the shaft. The contact at *y* ceases. The pressure is taken off *B* and concentrated upon the points of the teeth *C* and *D*, and there intensified enormously by the forcible retardation of the driving-wheel or acceleration of the pinion. And even if the crank-shaft could be held absolutely steady, the same effect might be produced if the teeth came from the mold full at the points, or if the pitch-line had proud places, both common defects in uncut wheels. If, however, the faces of the teeth were cut away sufficiently to limit the contact to the immediate neighborhood of the pitch-point *y*, the motion of the wheel toward the pinion would simply cause the tooth *B* to slide on *E* without strain or change of angular velocity. And of all intermediate positions of the point of contact it may be said that the farther it moves from the pitch-point the more destructive become the effects of shaft displacements, incorrect tooth forms, or sinuosities of pitch lines, and conversely the shorter the path of contact the less the mischief resulting from these unavoidable irregularities. Experience has led millwrights to the same conclusion, for they invariably set out wheel teeth so fine at the points and roots that they bear on only near the pitch line when new, and when through wear the bright bearing surfaces extend to the points of the teeth, they chip the points to

avert the damage which we have just referred to.

The question is, why do they leave surfaces which are not used when the teeth are new and which have to be chipped off when the teeth are old? I never could get a satisfactory answer. Neither, as far as I know, would wheelmakers cut short their teeth till 1888, when a firm of engineers consented to make a pair of wheels with short teeth, Fig. 33, for their own shop engine. These wheels must have felt like the first lady who appeared in public minus a crinoline, so greatly did they differ from the fashion of their day. They had 49 and 47 teeth 2½-inch pitch, 15/16-inch long and 4½ inches wide, the ratio of length to pitch being 0.36. With fixed centers and accurate molding teeth ¼-inch long would have kept contact. These wheels held their grease well and ran smoothly till the engine was replaced in 1892. They were but small, and running at a low speed (1,080 feet per minute), but they were more convincing than arguments or drawings, because they were made of cast iron and could be seen at work.

Figs. 34 and 35 show what they accomplished in a few months. The former shows the teeth of a wheel and pinion put in in November, 1888, after two other sets of gearing had been destroyed in the space of 11 years. The teeth were 413/16-inch pitch, 3¾ inches long, and 19 inches wide, while the distance between the centers of the shaft-bearings was only 66½ inches. With these dimensions and a pitch-line speed of 2,410 feet per minute, the motion of the crank-shaft threw the whole load of 16,400 pounds upon the corners of the teeth, and drove the points of the pinion-teeth into the flanks of the wheel-teeth with jar and vibration sufficient to break one of the segments in the following March. Then more teeth were cracked and more segments broken, and it became necessary to replace the entire rim. The new rim was made of steel, with teeth of the same pitch and breadth as the old, but 25/16 inches instead of 3¾ inches long, Fig. 35. They would have been better 1½ inches long. The tooth of an engine driving-wheel is a cantilever whose strength varies not as pitch² × width ÷ length, but as pitch ÷ length, because, with a short and moving crank-shaft, very little, if any, strength is gained by making the width of a tooth more than one pitch. Thus a reduction of length to 1½ inches would have permitted a reduction of pitch to 3¼ inches, an increase in the number of teeth in both wheels, a shortening of the necessary path of contact, and, if three pitches be considered a sufficient width for the teeth of one engine driving-wheel, as I think it is, less concentration of pressure on the corners from the lifting of the shaft.

Some people consider double-helical teeth stronger than straight teeth. This is a delusion. Width for width of wheel-face, they are much weaker, because, owing to the lifting and end-long motion of crank-shafts, the pressure never acts on both treads at the same time unless provision be made for allowing the pinion-shaft to move end-ways, and even then the force required to overcome the inertia of the pinion, and its shaft, adds seriously to the pressure on the teeth. There is another cause of destructive vibration—namely, irregularity of pitch and inequality of pitch of wheel and pinion. The first is seldom important in machine-molded wheels, and can generally be rendered innocuous by chipping and filing. The second, depending upon the contraction of the castings, can only be remedied by gearing the wheels more or less deeply than was intended by the designer. In my opinion, both wheels of a pair should have pitch circles turned on both sides of the rims, with radii proportional to the number of teeth in each, whether they be the radii marked on the drawing or not. The wheels should be geared to these circles and kept there. The change of the position of the pitch-point on the profile will not materially affect the velocity ratio if the teeth be short. The fashionable length of teeth is now 0.45 to 0.55 of the pitch, instead of 0.66 to 0.75 as formerly, and it might be reduced still more with advantage in many cases.

I should like to have said something about the breakages of spur-wheels other than breakages of teeth, but as most of the breakdowns of these wheels originate in the teeth, it seemed best to devote the time available exclusively to them. The breakages of spur-gearings are now 10 per cent of the total breakdowns. The reduction in the percentage from 20 to 10 is principally due to the supersession of gearing by ropes. The improvements in design and workmanship have done little more than enable wheels to live under conditions which the old wheels could not have faced. But they have done something, and the substitution of steel for iron, of cut or milled for molded teeth, and of shorter teeth and correspondingly finer pitches, will do a little more, but I doubt if it will ever be possible to transmit more than, say, 1,200 horse-power from the crank-shaft of a steam-engine by spur-gearing.

I have now touched briefly, I could not do more, on the principal breakages of most of the important parts of a steam-engine except the beams, gudgeons and parallel motions, and other parts peculiar to beam-engines; and as these are becoming obsolete, I have left them out. I will, however, conclude with an anecdote relating to a

beam-engine. Beams, "walking beams" as they are sometimes called, are usually broken by water in the cylinder or some other extraneous cause, but sometimes they fail gradually from old age, the signs of failure being the appearance of small cracks sometimes in the flanges, sometimes in the webs. A beam in northeast Lancashire began to develop such cracks; they were pointed out to the owner of the engine, their import was explained to him, and he was advised to replace the beam. After listening attentively and in silence to the explanation, he got up, put on his hat, and said: "Mister, them cracks be nowt; I shan't put in no new beam. Why't has nowt to do but wag up and doon."

The Moon's Influence on the Earth's Magnetism*

By S. Chapman, Chief Assistant at the Royal Observatory, Greenwich

SCHUSTER has proved that the varying field which causes the ordinary daily changes in the magnetic elements is produced by electric currents flowing mainly above the earth's surface, and located in the rarefied upper strata of the atmosphere, perhaps where the Auroræ have their origin. The air is rendered electrically conducting in these tenuous regions presumably by the ultra-violet radiation from the sun. The electromotive forces which impel the currents are supposed to be produced by the motions of the air, as indicated by the barometric variation, across the earth's permanent field of magnetic force. A not unreasonable value for the electric conductivity of the atmosphere was indicated by Schuster's calculations, based on his theory.

If the atmosphere oscillates similarly in all layers, from the relative amplitudes of the 24 and 12 hourly terms in the barometric variation, it is possible to calculate the relative amplitudes of the corresponding terms in the magnetic variation, provided that the electric conductivity is constant throughout the day and night. There is a fairly large discrepancy revealed between the ratio thus calculated and the observed ratio of the diurnal and semi-diurnal components of the magnetic variation. This points to error in either or both of the assumptions mentioned, viz., that the atmosphere oscillates similarly in all layers and that the electrical conductivity of the air is constant. The latter is almost sure to be incorrect—as Schuster remarks, the conductivity probably falls to a low value at night. On this assumption the theory and observation can be reconciled much more closely, but there still remains the uncertainty as to whether the amplitudes of the 24 and 12 hourly oscillations of the atmosphere preserve the same ratio one to another at all levels.

The moon also influences both the barometer and the earth's magnetic field—the former by the production of a simple semi-diurnal gravitational tide. Both effects are of minute amount, and therefore laborious to evaluate from the observations, but otherwise there is no difficulty in discussing them. It is natural to suppose that the connection between the lunar-diurnal variations at atmospheric pressure and of magnetic force is similar to that between the corresponding solar-diurnal variations, and this is borne out by many features of the lunar-diurnal variations, as, for instance, that the magnetic variations are greater at times of perigee than at apogee—there being some evidence, moreover, that the ratio of the amplitudes at the two periods is equal to the ratio of the corresponding values of the moon's tide-producing force. Again, when the lunar-magnetic variation is determined as the mean from a number of whole lunar months, it should be of the same simple semi-diurnal type with the lunar barometric variation; for although the electrical conductivity of the upper air is a function of the solar time—and therefore also of the lunar time, at any particular phase of the moon—in the mean of one or more whole lunar months (during which time each solar hour will have occurred equally often at all lunar hours) the electrical conductivity of the upper air is a constant as regards lunar time. Thus the semi-diurnal atmospheric oscillation due to the moon must produce a corresponding purely semi-diurnal magnetic variation in the mean of a month; this is what is actually observed.

But, as already hinted, at any particular phase of the moon the effect of the variable electrical conductivity of the upper air, in conjunction with the simple semi-diurnal atmospheric oscillation, should be to produce magnetic variations, not only of frequency 2 periods per day, but also of frequencies 1, 3, 4, etc. This the author has actually found to be the case, and it appeared further that the epochs of the various components undergo regular changes during the course of the lunar month, the change of epoch per month for the components of frequencies 1, 2, 3, 4, being respectively -2π , 0, 2π , 4π . At new moon, when the sun and moon are on the same meridian, the epochs of all the components

are nearly equal. The changes of epoch throughout the month clearly result in the disappearance from the variation for the mean of the whole month of all components except the second, which is of constant epoch. These changes with the position of the sun greatly confirm the hypothesis of a variation in the electrical conductivity of the upper air, depending on the solar hour angle.

The author has also proved mathematically¹ that the law of change of the epochs of the various components during the course of the month is precisely what was to be expected from the combination of a semi-diurnal atmospheric oscillation depending on lunar time and a variable conductivity depending on solar time. In fact, the change of epoch per month of the component of the magnetic variation, of frequency n , is $2(n-2)\pi$, which is verified (*loc. cit.*) from the observations for $n=1$ to $n=4$. A consideration of the amplitudes of the several harmonic components in the magnetic variation will, moreover, when sufficiently good material is available, enable the dependence of the atmospheric conductivity upon solar time to be determined; a beginning has already been made with this work. The investigation is possible with the lunar-magnetic variation, and not with the solar-diurnal magnetic variations, because in the former case, unlike the latter, the atmospheric oscillation which is responsible for the magnetic effect is of a single period (12 hours), so that any other periodic terms appearing in the magnetic variation must be due solely to the variable electrical conductivity of the air. When the law of change of the latter has been evaluated from the lunar-magnetic variations, it may be possible to work back from the solar-diurnal magnetic changes and to determine the relative amplitude of the atmospheric oscillations on which they depend; thus the similarity of atmospheric oscillations at widely different levels might be tested.

A natural test to apply to this theory of the lunar-diurnal magnetic variations is the examination of their dependence on sun-spottedness. The solar-diurnal magnetic variations show a marked periodicity in amplitude, corresponding to the sun-spot periodicity, and this points, on Schuster's theory, to a corresponding change in the electrical conductivity of the upper air. This should affect the lunar-diurnal variations in a similar way. Different opinions have been held as to whether or not the observations do show this. The uncertainty is due mainly to the large accidental error in the variations as determined from a single year's observations. In the curve accompanying this abstract all the evidence upon the point which is at present available is summarized; unfortunately it does not suffice to settle the question at issue.

An Orchid With Explosive Flowers

DR. H. N. RIDLEY ("Straits Settlements Gardens Bulletin," vol. i., 1913, pp. 191-3) describes an orchid from Sarawak which shows a remarkable floral mechanism. This species (*Plocoglottis porphyrophylla*) is widely distributed in Malaya; but, being inconspicuous and a lover of deep shade, is but little known. It bears only one flower open at a time, but remains in flower for over three months, producing a fresh flower every few days until the raceme is more than two feet long and has borne about fifty flowers. Unfortunately, the author's account is not accompanied by illustrations, but the following description may be followed if the reader examines any ordinary orchid flower. In the young flower the ovary begins to twist, as usual in orchids, until it has overtopped the bract; it carries the swelling bud through about 75 degrees and then stops twisting. During this twisting the dorsal sepal outgrows the other two sepals, and pushes over the apex of the bud. All the sepals at this stage are similarly narrowly ovate, the lateral ones asymmetrically so. The lateral petals are linear, and curved round the column to meet at their tips. The lip is about as broad as long, cuspidate above its broad shoulders, with the margins in the lower part frilled and turned under; if these margins are uncurved, it is seen that they are the lateral lobes of the lip. Under each broad shoulder a wart has begun to form.

Between this stage of the bud and maturity the following changes occur: The contiguous halves of the lateral sepals thicken from the middle upward; the cuspidate tip of the lip turns back, its shoulders enlarge, and the warts become sharp little upstanding cones, while the side lobes increase along their margins so that they are too full for the space they have, and toward the base of the lip tend to form an upstanding rounded crest. Two very fleshy staminodes lie within the curve of this crest, one on each side. In opening, which occurs in late afternoon or early evening, a slit appears between the lateral sepals; then these sepals break away and slowly take up a position at right angles to the ovary, and their thickened areas become convex

¹ On the diurnal variations of the Earth's Magnetism produced by the Moon and Sun, *Phil. Trans.*, A. 213, pp. 279-321.

inward and throw the thin parts back; then the lateral petals rapidly elongate, curving over strongly, so that their points pass between the bases of the lateral sepals, and in this curious action they deflect the lip on to its base holding it down against a certain amount of resistance in contact with the lateral sepals. Thus the flower gapes somewhat. During the night the dorsal sepal turns back and the lateral petals straighten. The upper lateral sepal no longer held away from its fellows by the lateral petals now moves down to be in contact with it and is thus almost median as regards the lip, and as the lateral sepals move away the lip is caught against its convex swelling and held folded down as the lateral petals placed it. A touch now frees the lip and causes it to spring up against the column.

It is apparently fertilized by rather small insects, which, attracted to the flower, are trapped by the upspringing of the lip against the column, and, in struggling to free themselves, effect pollination. The mechanism is very curious; the lip is a trigger put into place by the lateral petals, and held there by one of the lateral sepals. This alone makes it of unusual interest, but this is heightened by the angle at which the flower stands, by the movement out of the median line of the column and by the movement toward it of a lateral sepal. The flower has apparently no scent and no free honey; its colors are lemon-yellow to yellowish-green, with deep crimson markings on the lip and the swollen parts of the lateral sepals are maroon.—*Knowledge*.

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