

(Paper No. 2953.)

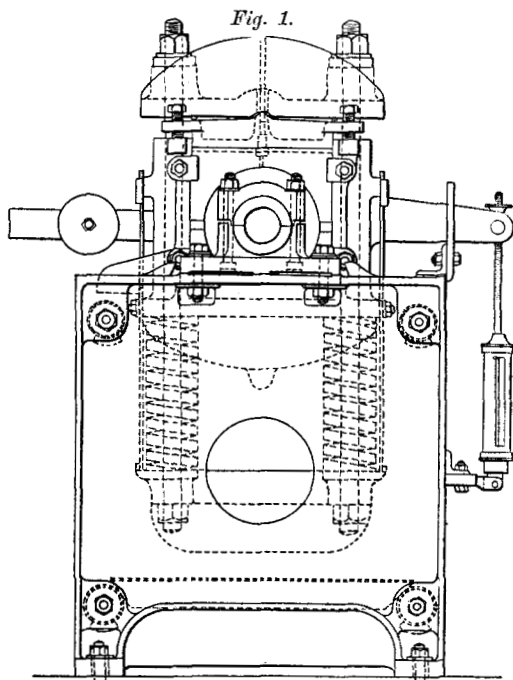
“Machinery Bearings.”

By JOHN DEWRANCE, Assoc. M. Inst. C.E.

IN this Paper are presented the results of a series of experiments undertaken by the Author to determine the frictional resistance to shafts revolving in bearings under varying loads, when subjected to different conditions. The testing-machine used is shown in *Figs. 1*. The journals upon which the experiments were made were 10 inches and 4 inches respectively in diameter, and formed part of a shaft supported at its ends in plummer-blocks. A saddle rests upon the upper bearing and is connected through four bolts compressing springs to a plate pressing upon the under side of the lower bearing. The load carried by the journal can therefore be varied with the degree of compression of the springs, the pressure upon the upper bearing being greater than that on the lower by an amount equal to the total weight of the saddle, spring and bearings, generally about 16 cwt. It was not convenient to apply a thermometer to the lower bearing, so the observations were usually confined to the upper one, which, having to support the greatest load and being at a disadvantage as regards lubrication, generally failed first. When the saddle had been fitted with a pair of bearings the four bolts were evenly screwed up, the amount of compression being indicated by scales and pointers on each side. When loaded, the saddle was free to turn with the shaft through a certain range, and a spring balance, fitted, as shown in the *Fig.*, to retain the saddle vertical, indicated the amount of the force required to turn the shaft.

The first experiments were made upon a pair of 10-inch by 16-inch bearings having two oil-holes on the top centre-line. They were carefully fitted about $\frac{1}{64}$ inch larger in diameter than the journal, but they would not run cool with the weight of the saddle alone. When a channel was made to connect the two holes and to lubricate the part of the bearing between them, the result was not greatly improved. When oil was thrown upon the shaft at the openings between the bearings they began at once

to sustain loads more or less satisfactorily, but the oil issued from the holes at the top. The pressure indicated on a gauge connected to the centre of the bearing was equal to the greatest load that could be applied to the large bearing; the experiments were, therefore, transferred to the 4-inch shaft with a bearing upon which the same load represented a greater pressure. A long series of experiments resulted ultimately in a pressure of 2,300 lbs. per square inch being recorded. The bearing was taken

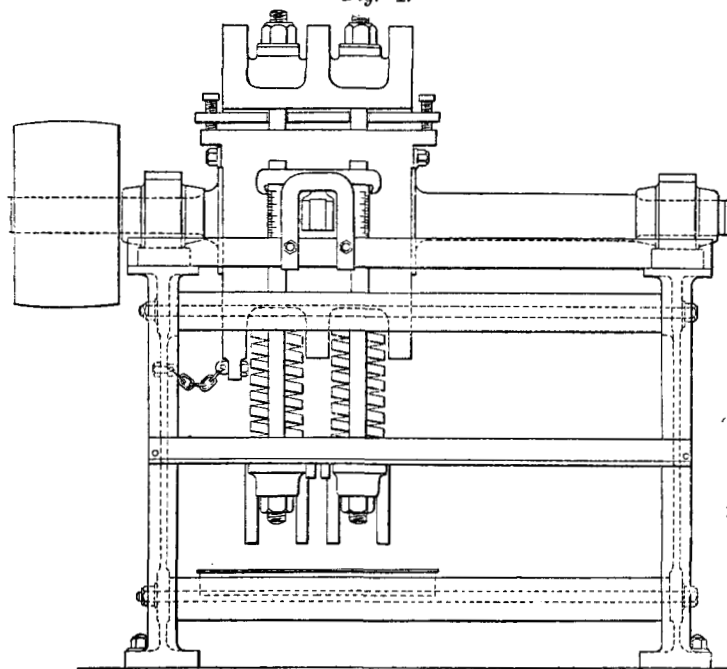


END VIEW.

Scale, $\frac{1}{4}$ inch = 1 foot.**FRICTION-TESTING MACHINE.**

out and the surface dressed with a view to obtaining even greater pressures. When, however, it was replaced the following day, instead of continuing to deliver oil at the gauge connection as hitherto, the bearing would take oil rapidly. The pressure-gauge was removed and a vacuum-gauge substituted when it was found that a vacuum equivalent to 30 inches of mercury existed where previously a high pressure had been recorded. The bearing still ran cool and well, as shown in Table I of the Appendix.

When taken out there was nothing in its appearance that at first sight accounted for the change described, but further examination revealed that, although the back of the bearing had been planed, it was, from some cause, not quite flat, the centre of the bearing taking all the load. This had sprung the bearing slightly and had worn the central part of the surface. When the load was removed the centre sprang back, leaving that part of the surface round the hole separated from the shaft. The load

Fig. 1.

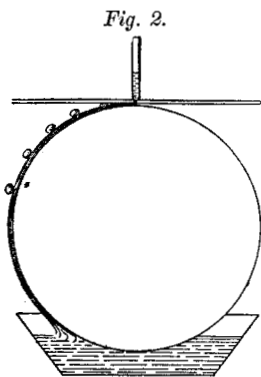
ELEVATION.

Scale, $\frac{1}{4}$ inch = 1 foot.

FRICTION-TESTING MACHINE.

applied during subsequent experiments had not been sufficient to spring the bearing flat again. The experiments on this point were continued until a bearing was produced that, when placed on to the shaft one way, gave a pressure in a hole at the centre of the bearing and, when reversed, gave a vacuum at the same point. As a result of the observations made during the foregoing experiments an arrangement was set up which is shown diagrammatically by *Fig. 2* to illustrate the way in which oil lubricates a bearing.

If oil is slowly poured on to a revolving shaft, it accumulates in a thick ring, most of which can be wiped off by the hand. The oil next to the shaft adheres more firmly than that which is further away. If the oil ring be imagined to be composed of films of oil, each of the thickness of a molecule of oil, the first film has the solid shaft to adhere to. The second is separated from the shaft, but it can adhere to the first, which is itself firmly held. Each successive film is less influenced by the force of adhesion that exists between the shaft and the oil, until the point is reached at which centrifugal force is stronger than the adhesion and the oil flies off the shaft. If a shaft be rotated in a bath of oil, the ring of oil is also formed. In *Fig. 2* the circle represents a shaft,



on the top of which a flat plate rests. However light this plate may be, it affects the ring of oil immediately it touches it. Unless the plate entirely stops all the oil from passing round with the shaft it must float upon that part of it which remains between it and the shaft. There must, in other words, be a film of oil at a sufficient pressure per square inch acting on the narrow surface of contact to support the total load put upon the plate. This pressure is the result of the multiplication of the force of adhesion of the oil to the shaft by the inclined plane formed by the

plate to the shafts.¹ The velocity of the oil is also increased in the same way. If the load on the plate is a light one the adhesive force is sufficient, when increased by the inclined plane, to produce the pressure required to lift the plate and carry several films round with the shaft; but, as the load is increased, fewer films are able to pass the plate until the point is reached at which the surfaces abrade one another for want of lubrication.

If a hole is made through the plate, this pressure can be observed, increasing as the hole approaches the point at which the plate rests upon the shaft. If the hole is pushed beyond this point a vacuum is produced. The oil and air in the hole adhere to the films and are carried round with the shaft by surface adhesion. A flat plate was employed in this experiment, as it was easier

¹ Another explanation of the phenomenon is given: see "On the Conversion of Heat into Work," by Dr. William Anderson, F.R.S., 2nd edition, p. 19.—SEC. INST. C.E.

to observe the relative position of the hole. The conditions are not greatly changed if the diameter of the shaft be reduced and the plate curved. A bearing that exactly fits the shaft all the way round will not run cool, and the well-known fact that a bearing must not be tight at the side indicates that the inclined plane must exist in some form or other. All successful bearings are constructed with an inclined plane in some form, and the load that a bearing will sustain is determined by the inclination. If the angle is sharp it will not multiply the adhesion of the oil so much as if it is more gentle. By suitably adjusting the inclination it has been found possible to pump oil between the surfaces to a pressure of as much as 3,000 lbs. per square inch.

These experiments were originally instituted to demonstrate whether with similar lubrication and conditions a bearing surface composed of one alloy would allow of a greater load than a bearing surface of another alloy, and they have proved that it will not. The composition of the metal of the bearing surface has little or no influence on the load that the bearing will support. Bearings composed of a metal that may, under certain circumstances, seize at a pressure of 20 lbs. per square inch, have been run with a load of more than 3,000 lbs. per square inch, and many different kinds of metal have been used in bearings that have run loaded to more than 1 ton per square inch. At these loads the lubrication becomes so uncertain and difficult that the point at which it fails is due to conditions that cannot be observed. In no case was there the slightest evidence that this point was reached sooner with one alloy than it was with another, so long as the metal itself would support the load. The experiments leave no doubt that so long as a bearing runs fairly cool the surfaces of the shaft and bearing are separated by films of oil. If the number of films is small it is possible to have considerable heating without actual seizing, but if the films of oil are entirely absent the surfaces adhere or seize at once. The simplest example of this kind of adhesion is afforded by the abrasion of an iron surface by a piece of brass. The crystals of the iron tear out crystals from the brass. By burnishing the iron surface this tendency is reduced to a minimum, and by corroding the surface chemically it is increased to a maximum. If the pressure and speed are low, a great deal of brass can be torn from the high places of a bearing, especially if there is a good supply of oil.

When a bearing that has worn to a surface is allowed to rest upon the shaft without the intervention of a film of oil, and the crystals of the bearing adhere to the shaft, they must make an

elevation on the shaft that would prevent it being turned except by a force sufficient to lift the load on the bearing to the height of the elevation due to the crystals. This would concentrate the whole load on this elevation, with the result that more crystals would be dragged out. If the load is very great, say more than 1 ton to the inch on an 8-inch by 4-inch bearing, seizure has occurred with a suddenness almost startling; but the presence of oil in some parts of the bearing, and the lower loads used in practice, generally make the seizing more gradual.

A piece of iron will not leave a mark upon a surface softer than itself; it becomes coated with the softer metal. If the bearing is of a material of which the crystals are individually stronger than those of the shaft, the crystals of the shaft adhere to the bearing, which, being stationary, causes the crystals to be heaped up in one place instead of being carried round and spread over the whole circumference of the shaft. This is the reason that the seizing of a cast-iron bearing is often attended with such disastrous results, and there can be no doubt that the softer the metal of the bearing the safer is the shaft from injury from seizing. Many hundreds of experiments were made upon the machine with soft-metal bearings without injury to the shaft, but when similar experiments were attempted with hard-bronze bearings the shaft was several times injured and had to be turned.

It is possible under some circumstances to provide sufficient lubrication without intermission. When this is the case, the shaft revolves in oil, and it is surprising with what a small power a heavy load can be supported, and how small the destruction of the surfaces may be under these conditions, as shown in Table II of the Appendix. It is well known that even when the lubrication has been continuous the surface of the bearing has suffered considerably, and in some cases the surface of the shaft. If it is accepted that so long as a bearing works cool and shows no sign of seizing, the surface and shaft are separated by films of oil, it is at first sight difficult to see how either metallic surface can wear away. In some cases the oil contains grit which is carried between the surfaces and scratches them. It has been proved on the testing-machines that dust that will float in a quiet atmosphere is usually less in bulk than the thickness of the film of oil. Had this not been the case the experiments could not have been conducted where they were, as the machine was exposed to a considerable amount of floating dust.

The corrosive effect of the oil itself on the surfaces does not appear to have been hitherto recognized. It was first observed

when experimenting with a pair of bearings of pure lead upon the 10-inch shaft. Olive oil was used, but after passing through the bearing several times, it became black and thick. This oil, after filtration, was composed of 16 per cent. of oleate of lead, 9.57 per cent. of oleic acid, and 74.62 per cent. of olive oil and glycerine. Oil of the same quality was then run through bearings composed of hardened tin, which were found to be but little affected. Disks of the metals used in the manufacture of bearings were immersed in oleic acid, and occasionally drawn up out of the acid so as to be exposed to the air. Lead and zinc rapidly corroded away; copper was corroded, but to a less extent. Tin and antimony were not appreciably affected. Oleic acid appears to attack lead, zinc and copper with great avidity. Even if the oil is free from acid it becomes charged with oxygen from the atmosphere which oxidizes the surfaces, the oxide itself being immediately carried away by the oil. A great number of experiments showed that a bearing composed of an oxidizable metal, such as hardened lead, could be worn and scraped to a surface corresponding to the shaft in a quarter of the time required to produce the same effect on a bearing composed of hardened tin. For this reason a number of the special forms of bearing were made of hardened lead. Hardened zinc was tried in one instance, the bearing being tested by hydraulic pressure after the usual pressure-gauge holes were drilled, and oleic acid being used as a lubricant. The acid attacked the surface so rapidly that instead of improving it became worse as time elapsed. Oil was then used, but when the surface had arrived at the point of delivering it at pressure, it was found that the oleic acid had soaked into the pores of the metal and so corroded it that the oil oozed out all over the bearing at very slight pressure. The alloys of zinc are probably the most crystalline used for bearings, and there seems to be no doubt that the size of the crystals greatly affects the rapidity of chemical corrosion.

This open grain or crystalline structure occurs more or less in all bronze castings, and renders them more subject to chemical corrosion than would otherwise be the case with an alloy of copper and tin. The chief recommendations of bronze as a material for bearings are its high melting-point, and its capacity of resisting high compressions. The melting-point of the tin alloys is 500° F. The alloys of lead and of zinc vary more than the tin alloys, but their melting-points are not much higher.

With suitable lubrication bearings should run cool; if the temperature rises above 200°, the viscosity of the oil is so much

reduced that the bearing will probably seize. There is considerable difference of opinion as to whether a bronze bearing will behave better than a tin bearing under such circumstances. The bronze bearing will, if allowed to run after it becomes heated, almost certainly injure the shaft, but the tin-alloy bearing will run till the temperature reaches 500° without injury to the shaft. Higher temperatures than 500° are dangerous, and with proper arrangements ought never to occur.

The compression test deserves more consideration than it has hitherto received. Many of the alloys of tin and lead now used to line bearings have so low a compressibility that they yield under the ordinary pressures applied to bearings with the result that the metal squeezes into the oil-inlets and stops lubrication. This circumstance is no doubt responsible for much of the trouble that has been experienced with the use of alloys of this class. It is suggested that no alloy should be used until it has been demonstrated satisfactorily that its point of first yield is considerably above the greatest load or shock to which it will be subjected in use.

The method of making such a test is very simple. A bush 4 inches in diameter by $3\frac{1}{2}$ inches bore, giving an area of metal of 5 square inches, is cast on a chill and is placed in a hydraulic press. A line is drawn on the side with a pair of compasses set to about 3 inches radius. The bush is loaded by successive increments of $\frac{1}{4}$ ton, the load being taken off each time. When it is found that the line drawn by the compasses thickens the previous line, the metal has yielded. Even after this point, different alloys behave very differently, some taking a large increase of load to cause a yield of $\frac{1}{8}$ inch, others continue to yield very fast after they first start. It is possible to make an alloy of tin that will not yield in this way until loaded to 8 tons per square inch.

The Author's experiments suggest the following rule. The oil should be introduced into a bearing at the point that has to support the least load and an escape should not be provided for it at the part that has to bear the greatest load.

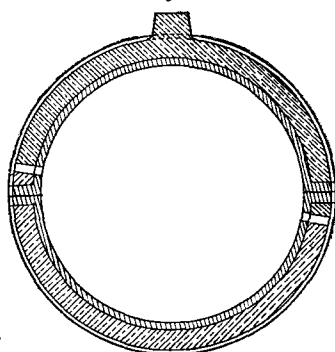
All the most important bearings belong to one of three classes—(A) those having a continuous load in one direction, (B) those having an alternating load in opposite directions, and (C), those with both a continuous load in one direction and an alternating load in opposite directions.

In class A is included the ordinary mill bearing or plummer-block used for supporting shafting. The oil is fed into the centre of the top bearing at the point that has to bear the least load, so that in this case the rule is conformed with. This class also

includes railway-carriage bearings. These were originally lubricated by holes through the crown of the bearing at the point of greatest pressure. It was, however, found that the lubricant would not enter at that point until the surfaces were more or less roughened. These bearings are now invariably lubricated according to the rule given. Footsteps of vertical shafts and thrust-blocks of marine engines belong to this class, but no experiments were made upon them.

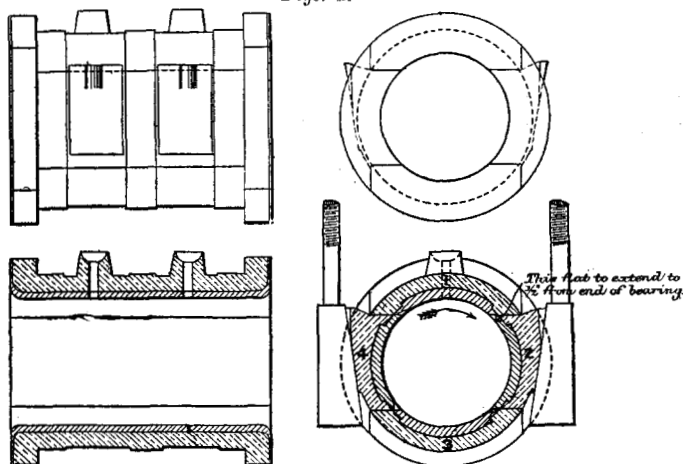
In class B are included the bearings of vertical engines. The bearings of a marine-engine resemble the ordinary plummer-block, and in the shaft-tunnel their duty is the same; but those near the connecting-rod have an entirely different duty to perform, which is the same as the connecting-rod bearing. The oil is generally applied at the centre of the top bearing in defiance of the rule. If the oil-hole is left plain it is found that no oil will enter during that part of the stroke when there is pressure on the top half of the bearing. To meet this difficulty oil-channels are cut. It must be evident that as the pressure of oil between the surfaces gradually increases from the point of least pressure to a pressure equal to the load at the point of greatest pressure, channels that run circumferentially around the shaft must be bad unless they are confined to the part of least pressure. Their effect is to scrape off the oil at the point of greatest pressure, and deliver it unused at a point of less pressure. The result in the ordinary marine-bearing is that the oil delivered into the bearing runs down the channels as far as possible, and is not used on what might be called its first journey through the bearing. It is then taken up by the shaft and is carried to the second half of the bearing. The proper point to introduce the oil is just above the joint of the bearing at the side. There the oil is distributed over the shaft and carried to the point of greatest pressure. As there are no channels from this point the oil cannot escape, and will support almost any load. There is no reason why one bearing only should be lubricated in all large bearings; it is desirable that each half should have its own supply of oil (*Fig. 3*).

Fig. 3.



In class C are included the bearings of horizontal engines, espe-

cially the main bearing next the connecting-rod. This is the most troublesome kind of bearing, as it has double duty to perform. To facilitate the taking up of the wear the device shown in *Figs. 4* has been used. The bearing is in four parts, three of which can be drawn towards the centre. The oil is introduced at the top at a point of least pressure, but before it arrives at the first point of greatest pressure it has to jump a joint. At some parts of the stroke it cannot do this, as the load is too great, but at others it can, and it is then carried into bearing No. 2. Before it can reach No. 3 it has another jump greater than the last, because when the oil escapes here it cannot return. No. 3 receives very little oil, but No. 4 gets even less, as the pressure is never off No. 3, and

Figs. 4.

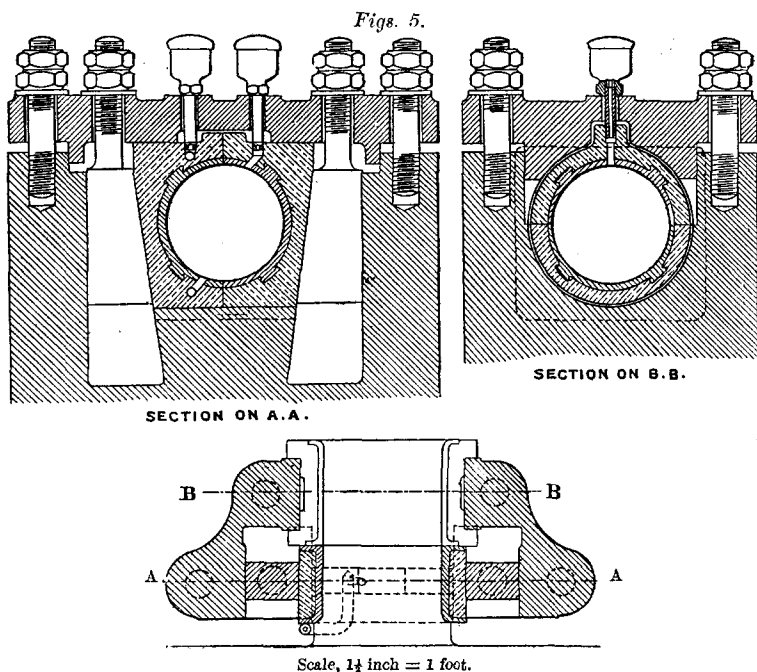
Scale, $1\frac{1}{2}$ inch = 1 foot.

there is still another joint to jump. The only way of meeting the difficulty is to have two complete bearings. One of these would be jointed at the top and bottom, and would take the thrust of the piston, the other would be jointed in the opposite direction, and would take the weight of the shaft and fly-wheel. Such a bearing is shown in *Figs. 5*. It is very important that the oil should be fed at a point exactly central in the length of the bearing. If it is fed a little to one side the oil separates the surfaces and tilts the bearing, the consequence being that the oil is forced out at one end too fast, and at the other end the bearing is probably running dry.

The sides of a bearing should be carefully eased off, and should

slope toward the centre to bring the oil that escapes from the other half towards the centre. Channels in the bearing or shaft-surface can do no good except as receptacles for the débris when the bearing seizes. Such an advantage is very doubtful, but if they are used for this purpose they should be in a line with the shaft, and not pass round it. They must not extend more than three-quarters the length of the bearing.

When it is fully recognized that a bearing will sustain a load



exceeding 1 ton per square inch, and that most costly and serious troubles are experienced with bearings that are loaded only to a twentieth of that load, it must be admitted that the subject requires investigation. The Author hopes that this Paper may in some small degree have the effect of suggesting new channels of thought and experiment that may add to knowledge on a subject of such great importance.

The Paper is accompanied by five drawings and two photographs, from which the *Figs.* in the text have been prepared.

APPENDIX.

TABLE I.—RESULTS OF EXPERIMENT LVI.

Time.	Total Load on Top Bearing.	Temperature.	Total Tangential Pull.	Time.	Total Load on Top Bearing.	Temperature.	Total Tangential Pull.
Minutes.	Lbs.	° F.	Lbs.	Minutes.	Lbs.	° F.	Lbs.
..	5,376	98	..	80	32,906	230	470·0
10	5,376	113	80·0	90	37,632	242	400·0
20	7,706	116	115·0	100	42,470	245	350·0
30	12,723	128	155·0	115	42,470	242	400·0
40	16,934	134	157·5	125	42,470	262	400·0
50	19,264	154	215·0	135	42,470	264	400·0
60	23,834	220	400·0	145	42,470	258	350·0
70	27,552	182	350·0	150	47,846	280	470·0

Area of bearing-surface of top bearing = $2\frac{1}{2}$ inches by 8 inches = 20 square inches; bearings lubricated with sperm oil. Diameter of shaft, 4 inches; speed, 266 revolutions per minute.

After having run for 150 minutes, bearing was taken out; its bearing-area was highly polished at the ends, and seemed to have seized at the middle.

EXPERIMENT LVIII.

The same bearings were used as in Experiment LVI, and they were lubricated with sperm oil and pads. The total load on the bearing was 15,132 lbs. = 756·6 lbs. per square inch. A vacuum gauge connected to the centre of the bearing showed a vacuum of 28·4 inches of mercury, the barometer standing at 30·2 inches.

TABLE II.

Time.	Temperature.		Load.	Tangential Pull.	Time.	Temperature.		Load.	Tangential Pull.
	L. H.	R. H.				L. H.	R. H.		
7.30	°F. 127	°F. 127	Tons. 18·96	Lbs. 115·5	12.0	°F. 145	°F. 139	Tons. 18·96	Lbs. 95·0
8.0	136	132	18·96	115·0	12.30	146	139	18·96	95·0
8.30	138	134	18·96	115·0	1.0	146	140	18·96	95·0
9.10	96	96	4·6	60·0	2.10	98	98	4·6	95·0
9.30	128	125	18·96	110·0	2.30	130	130	18·96	105·0
10.0	135	131	18·96	102·5	3.0	134	134	18·96	100·0
10.30	138	133	18·96	100·0	3.30	137	137	18·96	95·0
11.0	140	137	18·96	100·0	4.0	138	138	18·96	95·0
11.30	143	138	18·96	95·0	4.30	140	140	18·96	92·5

Area of bearing-surface of top bearing = 18 square inches; bearings lubricated by pads hung on either side of the shaft and feeding neat's-foot oil by capillary attraction. Diameter of shaft, 4 inches; speed, 266 revolutions per minute.