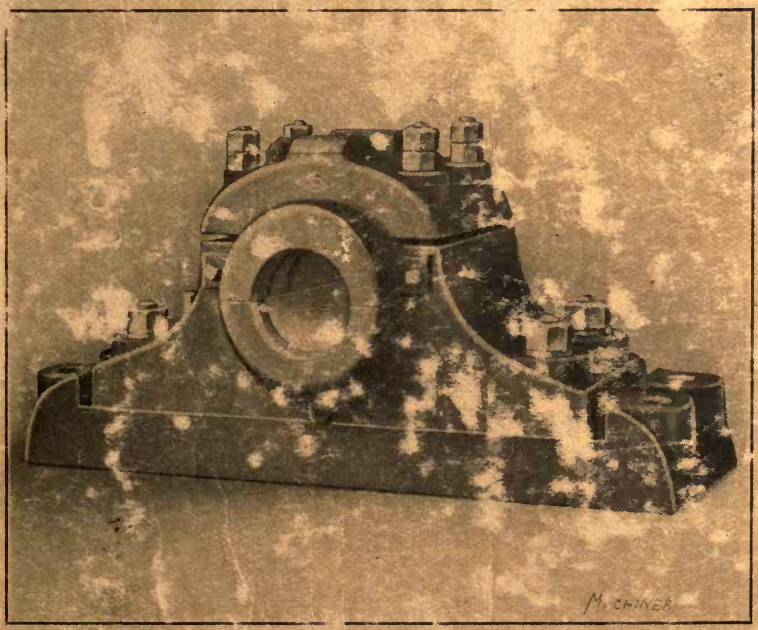
25 CENTS

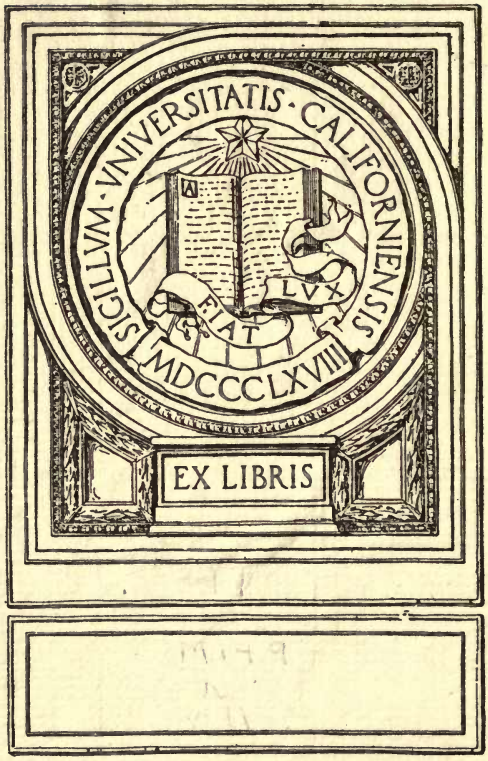
BEARINGS

DESIGN—FRICTION—LUBRICATION—
BEARING METALS

THIRD EDITION—REVISED AND ENLARGED



MACHINERY'S REFERENCE BOOK NO. 11
PUBLISHED BY MACHINERY, NEW YORK


SIGILLVM · VNIVERSITATIS · CALIFORNIENSIS
FIAT LVX
MDCCCLXVIII
EX LIBRIS

MACHINERY'S REFERENCE SERIES

EACH NUMBER IS ONE UNIT IN A COMPLETE LIBRARY OF MACHINE DESIGN AND SHOP PRACTICE REVISED AND REPUBLISHED FROM MACHINERY

NUMBER 11

BEARINGS

DESIGN—FRICTION—LUBRICATION—BEARING METALS

THIRD EDITION—REVISED AND ENLARGED

CONTENTS

The Design of Bearings, by FORREST E. CARDULLO - 3
Causes of Hot Bearings, by E. KISTINGER - - - 15
Thrust Bearings - - - - - - - - - 21
Friction and Lubrication - - - - - - - 25
Bearing Metals, by JOSEPH H. HART - - - - 35
Alloys for Bearings - - - - - - - - 40
Friction of Roller Bearings, by C. H. BENJAMIN - - 45

In the second edition of this Reference Series Book, a chapter on the principles of thrust bearings was introduced, together with a very important chapter on friction and lubrication of bearings. An additional chapter on bearing metals was also included in that edition. In order to provide space for this material, the chapter on ball bearings which was contained in the first edition, was eliminated. This chapter, together with a considerable amount of additional matter on the same subject, is included in MACHINERY'S Reference Book, No. 56, "Ball Bearings."

CHAPTER I

THE DESIGN OF BEARINGS*

The design of journals, pins, and bearings of all kinds is one of the most important problems connected with machine construction. It is a subject upon which we have a large amount of data, but, unfortunately, they are very conflicting. The results obtained from the rules given by different mechanical writers will be found to differ by 60 per cent or more. Many of our best modern engines have been designed in defiance of the generally accepted rules on this subject, and many other engines, when provided with what were thought to be very liberal bearing surfaces have proved unsatisfactory. This confusion has largely been the result of a misconception of the actual running conditions of a bearing.

Friction of Journals

A journal should be designed of such a size and form that it will run cool, and with practically no wear. The question both of heating and wear is one of friction, and in order for us to understand the principles upon which the design of bearings should be based, we must first understand the underlying principles of friction. Friction is defined as that force† acting between two bodies at their surface of contact, when they are pressed together, which tends to prevent their sliding one upon the other. The energy used in overcoming this force of friction, appears at the rubbing surfaces as heat, and is ordinarily dissipated by conduction through the two bodies. The force of friction, and hence the amount of heat generated under any given circumstances, can be greatly reduced by the introduction of an oily or greasy substance between the rubbing surfaces. The oil or grease seems to act in the same way that a great number of minute balls would, reducing the friction and wear, and thus preventing the overheating and consequent destruction of the parts. On this account, bearings of all kinds are always lubricated. Thus the question of journal friction involves the further question of lubrication.

For the purpose of understanding as far as possible what goes on in a bearing, and the amount and nature of the forces acting under different conditions, several machines have been designed to investigate the matter. In general they are so arranged that a journal may be rotated at any desired speed, with a known load upon the boxes. Suitable means are provided for measuring the force of friction, and also the temperature of the bearing. Provided with such an apparatus, we find that the laws of friction of lubricated journals differ very materially from those commonly stated in the text-books as the laws of friction. A comparison of the two will prove interesting.

---

* MACHINERY, December, 1906; January and February, 1907.

† Friction. * * * Resistance to motion due to the contact of surfaces.—*Standard Dictionary.*—Force. * * * Any cause that produces, stops, changes, or tends to produce, stop, or change the motion of a body.—*Standard Dictionary.*

Frictional Resistance in Lubricated and Unlubricated Bearings

It is generally stated in the text-books that the force of friction is proportional to the force with which the rubbing surfaces are pressed together, doubling, or trebling, as the case may be, with the normal pressure. This law is perfectly true for all cases of unlubricated bearings, or for bearings lubricated with solid substances, such as graphite, soapstone, tallow, etc. When, however, the bearing is properly lubricated with any fluid, it is found that doubling the pressure does not by any means double the friction, and when the lubricant is supplied in large quantities by means of an oil bath or a force pump, the friction will scarcely increase at all, even when the pressure is greatly increased. From the experiments of Prof. Thurston, and also of Mr. Tower, it appears that the friction of a journal per square inch of bearing surface, for any given speed, is equal to

$$f = kp^n \qquad (1)$$

where f is the force of friction acting on every square inch of bearing surface, p is the normal pressure in pounds per square inch on that surface, and k is a constant. The exponent n depends on the manner of oiling, and varies from 1 in the case of dry surfaces, to 0.50 in the case of drop-feed lubrication, 0.40 or thereabouts in the case of ring- and chain-oilers and pad lubrication, and becomes zero in case the oil is forced into the bearing under sufficient pressure to float the shaft.

The second law of friction, as generally stated, is that the force of friction is independent of the velocity of rubbing. This law also is true for unlubricated surfaces, and for surfaces lubricated by solids. In the case of bearings lubricated by oil we find that the friction increases with the speed of rubbing, but not at the same rate. If we express the law as an equation, we have

$$f = kv^m \qquad (2)$$

where f is the force of friction at the rubbing surfaces in pounds per square inch, k is a constant, v is the velocity of rubbing in feet per second, and the exponent m varies from zero in the case of dry surfaces to 0.20 in the case of drop-feed, and 0.50 in the case of an oil bath.

The third law of friction, as it generally appears in the text-books, is that the friction depends, among other things, on the composition of the surfaces rubbed together. This, again, while true for unlubricated surfaces, is not true for other conditions. It matters nothing whether the surfaces be steel, brass, babbitt, or cast iron, so long as they are perfectly smooth and true, they will have the same friction when thoroughly lubricated. The friction will depend upon the oil used, not on the materials of journal or boxes, when the other conditions of speed and pressure remain constant. Many people think that babbitt has less friction than iron or brass, under the same circumstances, but this is not true. The reason for the great success of babbitt as an "anti-friction" metal depends upon an entirely different property, as will appear later.

Combining into one equation the different laws of the friction of lubricated surfaces, as we actually find them to be, we have

$$f = kp^n v^m \qquad (3)$$

where f is the force of friction at the rubbing surface in pounds per square inch, k is a constant which varies with the excellence of the lubricant from 0.02 to 0.04, and the other quantities are as before. From this expression, we see that the friction increases with the load on the bearing, and also with the velocity of rubbing, although much more slowly than either.

Generation of Heat in Bearings

The quantity of heat generated per square inch of bearing area, per second, is equal to the force of friction, times the velocity of rubbing. All of this heat must be conducted away through the boxes as fast as it is generated, in order that the bearing shall not attain a temperature high enough to destroy the lubricating qualities of the oil. The hotter the boxes become, the more heat they will radiate in a given time. When the bearing is running under ordinary working conditions, it will warm up until the heat radiated equals the heat generated, and the temperature so attained will remain constant as long as the conditions of lubrication, load, and speed do not change. This rise in temperature above that of the surrounding air varies from less than 10 to nearly 100 degrees Fahrenheit, and is commonly about 30 degrees. We must keep either the force of friction or the velocity of rubbing, or both, down to that point where the temperature shall not attain dangerous values. As has been shown in the preceding paragraph, it was formerly believed that the force of friction was equal to a constant times the bearing pressure, and therefore, that the work of friction was equal to this constant times the pressure, times the velocity of rubbing. Now, since it is the work of friction that we are obliged to limit to a certain definite value per square inch of bearing area, it was concluded that a bearing would not reach a dangerous temperature if the product of the bearing pressure per square inch and the velocity of rubbing did not exceed a certain value. Accordingly, we find Prof. Thurston's formula for bearings to be

$$p\,v = C. \qquad (4)$$

where p is the bearing pressure in pounds per square inch, v is the velocity of rubbing in feet per second, and C has values varying from 800 foot-pounds per second in the case of iron shafts to 2,600 in the case of steel crank-pins. This has long been the standard formula for designing bearings, and while it is not satisfactory in extreme cases, it is very satisfactory for bearings running at ordinary speeds.

Turning our attention again to the results obtained from the machines for testing bearings, we find that while the results are very even and regular for ordinary pressures and temperatures, when we begin to increase either of these to a high point, the friction and wear of our bearing suddenly increases enormously. The reason is that the oil has been squeezed out of the bearing by the great pressure. This squeezing out of the oil, and consequent great increase in the friction, has three effects. The absence of the lubricant causes the parts to scratch or score each other, thus rapidly destroying themselves, the great increase in friction results in a sudden very high temperature,

in itself destructive to the materials of the bearing, and the heating is generally so rapid as to cause the pin and the interior parts of the box to expand more rapidly than the exterior parts, thus causing the box to grip the pin with enormous pressure. When the oil has been squeezed out in this manner, the bearing is said to seize.

Materials for Bearings

It is evidently of advantage to make the bearing of such material that the injury resulting from seizing shall be a minimum. If the shaft and box are of nearly equal hardness, each will tend to scratch the other when seizing occurs, and the scoring is rapid and destructive. This action will be especially noticed in case the shaft has hard spots in it, while the rest is comparatively soft, as is the case in the poorer grades of wrought iron. If, however, the shaft is made of a hard and homogeneous material, like the better grades of medium steel, and the bearing is made of some soft material, like babbitt, the bearing will not roughen the journal, and so the journal cannot cut the bearing. This is the first reason why babbitt bearings are so successful.

A second reason for the success of babbitt bearings lies in the fact that they cannot be heated sufficiently to make the bearing grip the journal. They will rather soften and flow under the pressure without actually melting away, just as iron and steel soften at a welding heat. The harder bearing metals, such as brass and bronze, do not have these advantages, and have been almost entirely replaced by babbitt in bearings for heavy duty, especially when thorough lubrication is difficult.

Babbitt is a successful bearing metal for still a third reason. The unit pressure on any bearing is not the same at all points. The shaft is invariably made somewhat smaller in diameter than the box. If there is a high spot on the surface of the box, that spot will have a very large proportion of the total pressure acting on it, and as a result the film of lubricant will be broken down at that point, and local heating and consequent damage result. In the case of babbitt bearings, before the damage can become serious the metal is caused to flow away from that point under the combined influence of the heat and pressure, the oil film is again established, and normal conditions restored.

Influence of Quality of Oil

The unit pressure which any bearing will stand without seizing depends upon its temperature and the kind of oils used. The lower the temperature of the bearings, the greater the allowable unit pressure. The reason for this is that oils become thinner and more free-flowing at the higher temperatures, consequently they are more easily squeezed out of the bearing, and it is more likely to seize. On this account, the higher the velocity of rubbing, the less the unit pressure that can be carried, but it does not follow that the allowable unit pressure varies inversely as the speed of rubbing, as was formerly thought.

The thicker and less free-flowing an oil is, the greater the unit pressure it will stand in a bearing without squeezing out. A watch

oil, or a very light spindle oil, will only run under a very small unit pressure; sometimes they are squeezed out of the bearing when the pressure does not exceed 50 pounds per square inch. On the other hand, a cylinder oil of good body will stand a pressure of over 2000 pounds to the square inch in the same bearing. There is a certain quality of oil which is best adapted to every bearing, and if possible it should be the one used.

A third cause influencing the pressure which may be carried is adhesiveness between the oil and the rubbing surfaces. Some oils are more certain to wet metal surfaces than are others, and in the same way some metals are more readily wet by oil than are others. It is evident that when the surfaces repel, rather than attract, the oil, the film will be readily broken down, and when the opposite is the case the film is easily preserved.

Oil Grooving

The mechanical arrangement of the box and journal may tend either to preserve or destroy the lubricating film. Both should be perfectly round and smooth, the box a trifle larger in diameter than the journal. The allowance commonly made for the "running fit" of the box and shaft is about 0.0005 $(D + 1)$ inches, where D is the nominal diameter of the shaft in inches. Some manufacturers of fast-running machinery make the diameter of the box exceed that of the shaft by nearly twice this amount. The oil should be introduced at that point where the forces acting tend to separate the shaft and box. At this point grooves must be cut in the surface of the box, so as to distribute the lubricant evenly over the entire length of the journal. Having been so introduced and distributed, the oil will adhere to the journal, and be carried around by it as it revolves to the point where it is pressed against the box with the greatest force, thus forming the lubricating film which separates the rubbing surfaces. The supply of lubricant thus continually furnished, and swept up to the spot where it is needed, must not be diverted from its course in any way. A sharp edge at the division point of the box will wipe it off the journal as fast as it is distributed, or a wrongly placed oil groove will drain it out before it has entirely accomplished its purpose.

An important matter in the design of bearings is the cutting of these oil grooves. They are a necessary evil, and should be treated as such, by using as few of them as possible. They serve, first, to distribute the lubricant uniformly over the surface of the journal, and, second, to collect the oil, which would otherwise run out at the ends of the bearing, and return it to some point where it may again be of use. As generally cut, oil grooves have two faults; first, they are so numerous as to cut down to a serious extent the area of the bearing, and, second, they are so located as to allow the oil to drain out of the bearing. Let us take an ordinary two-part cap bearing such as the outboard bearing of a Corliss engine, and see how it is best to cut the grooves.

One of these bearings, as commonly made by good builders, is shown in Fig. 1. The oil is supplied, drop by drop, through a hole in the

cap. If there were no oil grooves, only a narrow band of the shaft revolving immediately under this hole would be reached by the oil. If now, we cut a shallow groove in the cap, lengthwise of the bearing, and reaching almost, but not quite, to the edges, the oil will be enabled to reach every part of the revolving surface. To this groove we sometimes add two, as shown by the dotted lines in Fig. 2, which show the inner surface of the cap as being unrolled, and lying flat on the paper. No series of grooves can be cut in the box which will distribute the oil as well or as thoroughly as those shown, and they should always be used in the caps of such bearings in preference to any others.

Having distributed the oil over the revolving surface, our next care must be to see that it is not wiped off before it reaches the point for which it was intended. Accordingly, we should counterbore the box at the joint in such a way as to make a recess in which the surplus oil may gather, and which will further assist when necessary in distributing the lubricant. This counterbore should extend to within ¼ or ½ inch of the ends of the bearing, as shown in Fig. 1.

When the oil is supplied through the cap, grooves for the distribution

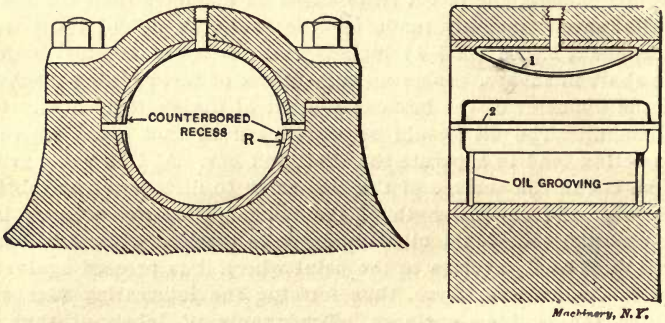


Fig. 1. Section of Outboard Bearing, showing Oil Grooving and Counterbored Recess

of the oil should not be cut in the bottom half of the bearing, since they will only serve to drain the bearing of the film of oil formed there. The old film is under great pressure at this point, and naturally tends to flow away when any opportunity is offered. If left to its own devices, part of it will squeeze out at the ends of the bearing and be lost. In order to save this oil, shallow grooves, parallel to the ends of the bearing, may be cut in the lower box, as shown in Figs. 1 and 3. Their office is to intercept the oil which would flow out at the ends, and divert it to the counterbored recesses, where it can again be made of use. These are the only grooves that should ever be used in the lower half of a two-part bearing, and they should only be used in the larger sizes.

Two classes of bearings which may well be made without oil grooves are, first, the cross-head slippers of engines, and, second, crank-pin boxes. The cross-head slipper should have a recess cut at each end, in the same way as the counterboring of the two-part box, as shown in Fig. 4. To this is sometimes added the semi-circular groove shown in

dotted lines, which does no harm, although it is unnecessary. The best way to oil a crank-pin is through the pin itself. In the case of overhung pins, a hole is drilled lengthwise of the pin to its center. A second hole is drilled from the surface of the pin to meet the first one. A shallow groove should now be cut in the surface of the pin, parallel to its axis, and reaching almost to the ends of the bearing, as shown

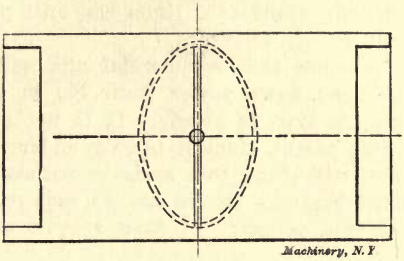


Fig. 2. Development of Cap, showing Oil Grooving and Counterboring

in Fig. 5. No grooves should be cut in the boxes, but the edges where they come together should be counterbored.

As much care and attention should be given to the oil grooving as to the size of a bearing, yet it is a matter often left to the fancy of the mechanic who fits it. The purpose of the grooves, to distribute the oil evenly, should ever be kept in mind, and no groove should be cut which does not accomplish this purpose, except it be to return waste

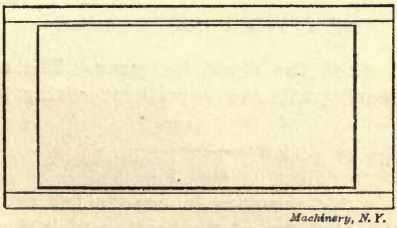


Fig. 3. Development of Lower Half of Outboard Bearing

oil to a place where it may again be of use. Most commonly, bearings have too many grooves. So far from helping the lubricants, they generally drain the oil from where it is most needed. Use them *sparingly.*

Calculating the Dimensions

The durability of the lubricating film is affected in great measure by the character of the load that the bearing carries. When the load is unvarying in amount and direction, as in the case of a shaft carrying a heavy fly-wheel, the film is easily ruptured. In those cases where the pressure is variable in amount and direction, as in railway journals and crank-pins, the film is much more durable. When the journal only rotates through a small arc, as with the wrist-pin of a

steam engine, the circumstances are most favorable. It has been found that when all other circumstances are exactly similar, a car journal, where the force varies continually in amount and direction, will stand about twice the unit pressure that a fly-wheel journal will, where the load is steady in amount and direction. A crank-pin, since the load completely reverses every revolution, will stand three times, and a wrist-pin, where the load only reverses, but does not make a complete revolution, will stand four times the unit pressure that the fly-wheel journal will.

The amount of pressure that commercial oils will endure at low speeds without breaking down varies from 500 to 1000 pounds per square inch, where the load is steady. It is not safe, however, to load a bearing to this extent, since it is only under favorable circumstances that the film will stand this pressure without rupturing. On this account, journal bearings should not be required to stand more than two-thirds of this pressure at slow speeds, and the pressure

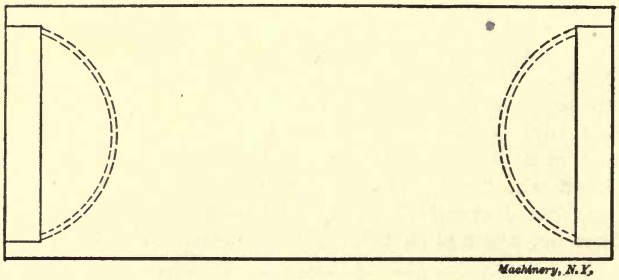


Fig. 4. Face of Cross-head Slipper

should be reduced when the speed increases. The approximate unit pressure which a bearing will endure without seizing is as follows:

$$p = \frac{P K}{D N + K} \qquad (5)$$

where p is the allowable pressure in pounds per square inch of projected area, D is the diameter of the bearing in inches, N is the number of revolutions of the journal per minute, and P and K depend upon the kind of oil, manner of lubrication, etc.

The quantity P is the maximum safe unit pressure for the given circumstances, at a very slow speed. In ordinary cases the value of this number will be 200 for collar thrust bearings, 400 for shaft bearings, 800 for car journals, 1200 for cank-pins, and 1600 for wrist-pins. In exceptional circumstances, these values may be increased by as much as 50 per cent, but only when the workmanship is of the best, the care the most skillful, the bearing readily accessible, and the oil of the best quality and unusually viscous. It is only in the case of very large machinery, which will have the most expert supervision, that such values can be safely adopted. In the case of the great units built for the Subway power plant in New York by the Allis-Chalmers

Co. the value of P in the formula given on page 10 for the crank-pins was 2,000—as high a value as it is ever safe to use.

The factor K depends upon the method of oiling, the rapidity of cooling, and the care which the journal is likely to get. It will be found to have about the following values: Ordinary work, drop-feed lubrication, 700; first-class care, drop-feed lubrication, 1,000; force-feed lubrication or ring-oiling, 1,200 to 1,500; extreme limit for perfect lubrication and air-cooled bearings, 2,000. The value 2,000 is seldom used, except in locomotive work where the rapid circulation of the air cools the journals. Higher values than this may only be used in the case of water-cooled bearings.

Formula No. 5 is in a convenient form for calculating journals. In case the bearing is some form of a sliding shoe, the quantity 240 V should be substituted for the quantity $D\ N$ in the equation, V being the velocity of rubbing in feet per second. There are few cases where a unit pressure sufficient to break down the oil film is allow-

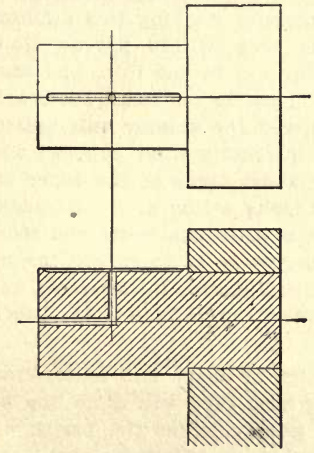


Fig. 5. Internally-oiled Crank-pin, showing Oil Passages and Grooves

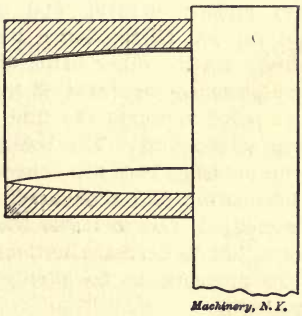


Fig. 6. Section showing the Bending of a Crank-pin and Consequent Unequal Wear of the Box

able. Such cases are the pins of punching and shearing machines, pivots of swing bridges, and so on. The motion is so slow that heating cannot well result, and the effects of scoring cannot be serious. Sometimes bearing pressures up to the safe working stress of the material are used, but better practice is to use pressures not in excess of 4,000 pounds per square inch.

In general, the diameter of a shaft or pin is fixed from considerations of strength or stiffness. Having obtained the proper diameter, we must next make the bearing long enough so that the unit pressure shall not exceed the required value. This length may be found directly by means of the equation:

$$L = \frac{W}{P\,K}\left(N + \frac{K}{D}\right) \qquad (6)$$

where L is the length of the bearing in inches, W the load upon it in pounds, and P, K, N, and D are as before.

A bearing may give poor satisfaction because it is too long, as well as because it is too short. Almost every bearing is in the condition of a loaded beam, and therefore it has some deflection. Let us take the case of an overhung crank-pin, in order to examine the phenomena occurring in a bearing under these circumstances. When the engine is first run, both the pin and box are, or should be, truly round and cylindrical. As the pin deflects under the action of the load, the pressure becomes greater on the side toward the crank throw, breaking down the oil film at that point, and causing heat. After a while the box becomes worn to a slightly larger diameter at the side toward the crank, in the manner shown in Fig. 6, which is a section showing an exaggerated view of the condition of affairs in the crank-pin box when under load.

It has already been noted that the box must be a trifle larger in diameter than the journal, and for successful working this difference is very strictly defined, and can vary only within narrow limits. Should the pin be too large, the oil film will be too thin, and easily ruptured. On the other hand, should the pin be too small the bearing surface becomes concentrated at a line, and the greater unit pressure at that point ruptures the film. This is exactly what happens when the pin is too long. The box rapidly wears large at the inner end, and the pressure becomes concentrated along a line as a consequence. The lubricating film then breaks down, and the pin heats and scores. The remedy is not to make the pin longer, so as to reduce the unit pressure, but to decrease its length and to increase its diameter, causing the pressure to be evenly distributed over the entire bearing surface.

The same principles apply to the design of shafts and center crank-pins. They must not be made so long that they will allow the load to concentrate at any point. A very good rule for the length of a journal is to make the ratio of the length to the diameter about equal to one-eighth of the square root of the number of revolutions per minute. This quantity may be diminished by from 10 to 20 per cent in the case of crank-pins, and increased in the same proportion in the case of shaft bearings, but it is not wise to depart too far from it. In the case of an engine making 100 revolutions per minute, the bearings would be by this rule from one and a quarter to one and a half diameters in length. In the case of a motor running at 1,000 revolutions per minute, the bearings would be about four diameters long. While the above is not a hard and fast rule which must be adhered to on all occasions, it will be found to be an excellent guide in all cases of doubt.

The diameter of a shaft or pin must be such that it will be strong and stiff enough to do its work properly. In order to design it for strength and stiffness it is first necessary to know its length. This may be assumed from the following equation:

$$L = \frac{20\,W\sqrt{N}}{P\,K}, \qquad (7)$$

where all the quantities are the same as in the preceding equations. Having found the approximate length by the use of the above equation, the diameter of the shaft or pin may be found by any of the standard equations given in the different works on machine design. It is next in order to recompute the length from formula No. 6, taking this new value if it does not differ materially from the one first assumed. If it does, and especially if it is greater than the assumed length, take the mean value of the assumed and computed lengths and try again.

Examples of Calculating Dimensions for Bearings

A few examples will serve to make plain the methods of designing bearings by means of these principles. Let us take as the first case the collar thrust bearings on a 10-inch propeller shaft, running at 150 revolutions per minute, and with a thrust of 60,000 pounds. Assuming that the thrust rings will be 2 inches wide, their mean diameter will be 12 inches. From equation No. 5 we will have for the allowable bearing pressure $\frac{200 \times 700}{12 \times 150 + 700}$, or 56 pounds per square inch. This will require a bearing of 60,000 ÷ 56, or 1070 square inches area. Since each ring has an area of 0.7854 $(14^2 - 10^2)$, or about 75 square inches, the number of rings needed will be 1070 ÷ 75, or 14. In case it was desirable to keep down the size of this bearing, the constant K might have had values as high as 1000 instead of 700.

Next, we will take the main bearing of a horizontal engine. We will assume that the diameter of the shaft is 15 inches, that the weight of the shaft, fly-wheel, crank-pin, one half the connecting-rod, and any other moving parts that may be supported by the bearings, is 120,000 pounds, and that two-thirds of this weight comes on the main bearing, the remainder coming on the outboard bearing. The engine runs at 100 revolutions per minute. In this case, $W = 80{,}000$ pounds, $P = 400$ pounds per square inch, and K depends upon the care and method of lubrication. Assuming that the bearing will be flushed with oil by some gravity system, and that, since the engine is large, the care will be excellent, we will let $K = 1500$. This gives us for the length of the bearing from formula No. 6:

$$L = \frac{80{,}000}{400 \times 1500}\left(100 + \frac{1500}{15}\right) = 26\tfrac{1}{2} \text{ inches (about).}$$

It is to be noted that, in computing the length of this bearing, the pressure of the steam on the piston does not enter in, since it is not a steady pressure, like the weight of the moving parts. The only matter to be noted in connection with the steam load is that the projected area of the main bearing of an engine shall be in excess of the projected area of the crank-pin.

For another example we will take the case of the bearings of a

100,000-pound hopper car, weighing 40,000 pounds, and with eight 33-inch wheels. The journals are 5½ inches diameter, and the car is to run at 30 miles per hour. The wheels will make 307 revolutions per minute when running at this speed, and the load on each journal will be 140,000 ÷ 8, or 17,500 pounds. Although the journal will be well lubricated by means of an oil pad, it will receive but indifferent care, so the value of K will be taken as 1,200. The length of the journal will then be

$$L = \frac{17{,}500}{800 \times 1{,}200}\left(307 + \frac{1{,}200}{5.5}\right) = 9\tfrac{5}{8} \text{ inches (about).}$$

As a last example, we will take the case of the crank-pin of an engine with a 20-inch steam cylinder, running at 80 revolutions per minute, and having a maximum unbalanced steam pressure of 100 pounds per square inch. The maximum, and not the mean steam pressure should be taken in the case of crank- and wrist-pins. The total steam load on the piston is 31,400 pounds. P will be taken as 1,200, and K as 1,000. We will therefore obtain for our trial length:

$$L = \frac{20 \times 31{,}400 \times \sqrt{80}}{1{,}200 \times 1{,}000} = 4.7, \text{ or, say, } 4\tfrac{3}{4} \text{ inches.}$$

In order that the deflection of the pin shall not be sufficient to destroy the lubricating film we have

$$D = 0.09 \sqrt[4]{W L^3}$$

which limits the deflection to 0.003 inch. Substituting in this equation, we have for the diameter 3.85, or say 3⅞ inches. With this diameter we will obtain the length of the bearing, by using formula No. 6, and find

$$L = \frac{31{,}400}{1{,}200 \times 1{,}000}\left(80 + \frac{1{,}000}{3\tfrac{7}{8}}\right) = 8.85, \text{ say 9 inches.}$$

The mean of this value, and the one obtained before is about 7 inches. Substituting this in the equation for the diameter, we get 5¼ inches. Substituting this new diameter in equation No. 6 we have

$$L = \frac{31{,}400}{1{,}200 \times 1{,}000}\left(80 + \frac{1{,}000}{5\tfrac{1}{4}}\right) = 7.1, \text{ say 7 inches.}$$

Probably most good designers would prefer to take about half an inch off the length of this pin, and add it to the diameter, making it 5¾ × 6½ inches, and this will be found to bring the ratio of the length to the diameter nearer to one-eighth of the square root of the number of revolutions.

CHAPTER II

CAUSES OF HOT BEARINGS*

In our modern high-speed steam and gas engines, turbines and the like, hot bearings are of more frequent occurrence than is generally supposed. Very often a new plant, just put into service, has to be shut down on this account. It not infrequently happens that the engine which has run "hot" is one of several, identical in design and construction, the bearings in the others having operated without trouble. Apparently there is no cause for this particular engine to give trouble, but in order to remove the difficulty, various makes of babbitt metals and bronzes are tried, sometimes with good results, sometimes without. Again, it occurs that a machine or engine operates at the beginning with perfect satisfaction, but after a time one or more of the bearings

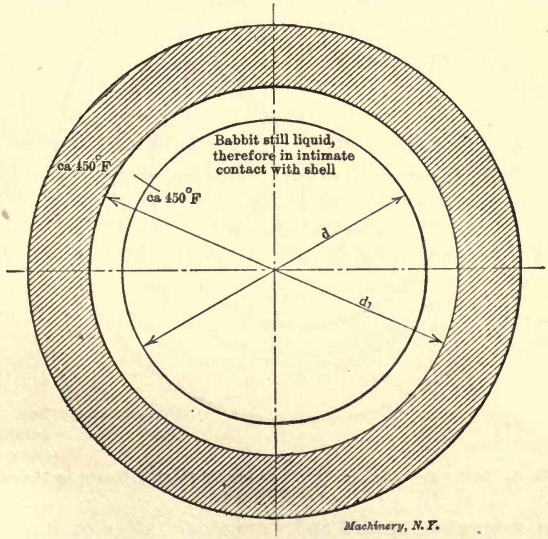


Fig. 7. One-piece Bearing· Babbitt just Poured. Both Shell and Babbitt at the Solidifying Temperature

begin to run "warm," and finally "hot," so that relining becomes necessary. As a general rule it is then simply accepted as a fact that the bearings "ran hot"; seldom does anyone think it worth while to seek out the fundamental causes for the trouble. That there is always the element of doubt in regard to bearings, is evidenced by the fact that our modern engine builders usually deliver an extra set of bearings with the engine, so that, in the event of trouble, a new set is at hand. The following may be of some assistance towards discovering and

*MACHINERY, November, 1907.

eliminating, in a scientific manner, and along technical and metallurgical lines, the real causes of hot bearings.

Investigation will show that the main reasons for hot bearings are:

1.—Shrinkage or contraction of the babbitt.

2.—Shrinkage strains set up in the babbitt metal liner by the unequal distribution of the babbitt metal over the shell.

3.—A lack of contact between the babbitt metal liner and the cast iron or cast steel shell.

4.—The lubricant becomes partially deflected into the wrong place.

Shrinkage or Contraction of the Babbitt

a. Shrinkage in a diametral direction. As an illustration of this point, one may take the simple example of an iron ball and ring. If this ball, when cold, will just pass through an iron ring, it will not

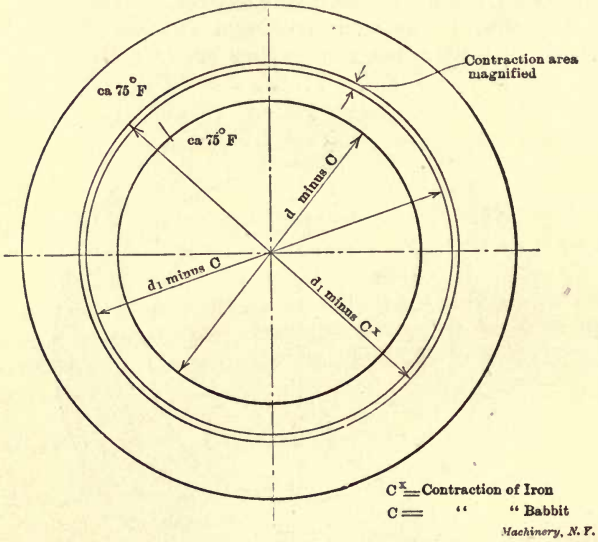


Fig. 8. Same Bearing as shown in Fig. 7 Cooled Down to Normal Temperature

do so when somewhat heated and expanded. After cooling down, however, it will again pass through the ring. A similar action takes place in a bearing.

In Fig. 7 of the accompanying illustrations the babbitt liner may be considered to have been just poured in, and the metal to be still liquid. At the exact solidifying point the babbitt will have filled all the interstices and be in good contact with the cast iron or cast steel shell, provided the babbitt itself has sufficient fluidity to enable it to penerate the smallest spaces. From this solidifying point on, the babbitt will contract according to its coefficient of contraction. Now, if the coefficient of contraction of the babbitt were the same as that of the material out of which the shell is made (usually cast iron or cast

steel), and provided that the shell had acquired the same temperature as the babbitt, the shell and the babbitt liner would then contract equally, and a fairly good contact would result, and there would be nothing to set up counter strains during shrinkage. But, as the coefficient of contraction of almost all babbitt metals is approximately two or three times higher than that of cast iron or cast steel, a shrink-

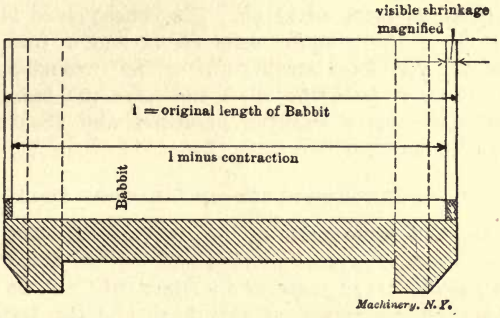


Fig. 9. Babbitt Bearing without Dove-tailed Grooves or other Retaining Device

age or loosening of the babbitt liner from the shell must absolutely take place after the solidfying point of the babbitt is reached. Fig. 8 shows this contraction as it would appear if magnified. The fact that most bearings are "split" does not, of course, change this result. If the babbitt is secured in the shell by means of dove-tailed grooves, or other anchoring devices, so that the actual visible contraction from the

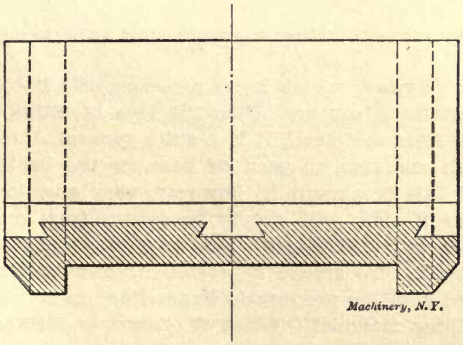


Fig. 10. Same Bearing as shown in Fig. 9, but with Dove-tailed Grooves. Visible Shrinkage Prevented, but Shrinkage Strains Produced

shell is lessened or minimized, then an unavoidable consequence of these grooves or other devices is *shrinkage strains*, set up while the babbitt cools down, as explained further on.

b. Shrinkage in an axial direction. With regard to shrinkage in the axial direction, it may be observed that the same results take place. Fig. 9 illustrates how the babbitt metal shrinks in a cast iron or cast steel shell in the axial direction, when there is no anchoring device

employed. In Fig. 10 may be seen the old-fashioned dove-tailed groove construction, prohibiting an actual visible shrinkage, but causing shrinkage strains.

Shrinkage Strains Produced by an Unequal Distribution of Babbitt Metal Liner

By referring to Fig. 11, it will be observed that the babbitt metal at *aa* is about twice as thick as at *bb*. The consequence is that, as the solidifying time of the greater mass *aa* is longer than that of the smaller mass *bb*, shrinkage strains are set up throughout the babbitt liner, which loosen it from the shell and have the tendency, in combination with the regular working pressures and shocks, to produce minute cracks in the liner.

Lack of Contact between Liner and Shell

In a bearing shell some parts of the liner are in close contact with the shell, as a result of careful pouring and the use of a properly made babbitt metal, while other parts of the liner will not be in good contact with the shell, by reason of shrinkage and the formation of air

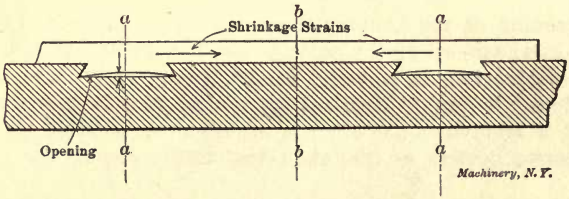


Fig. 11. Illustration of Shrinkage Strains Produced by Unequal Distribution of Babbitt Metal

bubbles and oxide gases, which latter are especially liable to be formed in babbitts containing copper. With the idea of filling up the hollow spaces between liner and shell, it is a quite general American practice, and an English one also, to peen or hammer the babbitt liner. The advisability of this treatment is, however, very questionable. By the peening process the air will simply be driven from one point to another, and be forced into places where at first a good contact existed, thus destroying it. To secure a permanent and intimate contact between liner and shell by peening is impossible, on account of the elasticity of the liner material. When the hammer strikes the metal, a contact may be formed, but as soon as the force of the blow is gone, the metal will spring away more or less by reason of its elasticity. Furthermore, the babbitt metal becomes more brittle by peening, and its strength diminished; this has been proved beyond doubt by a number of tests. Peening, unless performed with the utmost precaution, also produces minute cracks in the structure of the babbitt, which will constantly be enlarged by the regular working pressures. For these reasons, European continental practice has now practically abandoned the peening of babbitt metal liners. Summing up, in spite of

good pouring, or peening, or dove-tailed grooves and other similar anchoring devices, the liners are in a greater or less degree loose in the shells.

The Lubricant Penetrating the Hollow Spaces

When these loose bearings are in service, the hollow spaces between the liner and shell gradually become impregnated with an oil film from the lubricant employed, as shown in Fig. 12. Now, the coefficient of heat-conductivity of oil is only about 1/200 of that of an ordinary babbitt metal, or of cast iron. Therefore, the heat created in the liner by the working friction will not be conducted away to the shell, and thence to the engine frame, as quickly as though an intimate contact existed between shell and liner. The result is that the

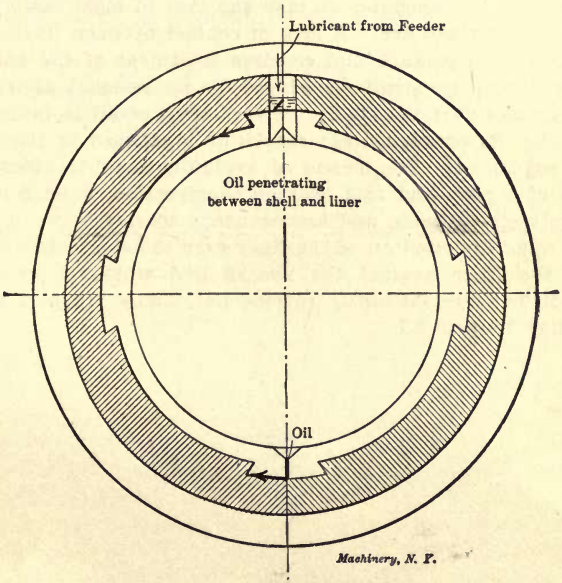


Fig. 12. Penetration of Oil between Shell and Liner

bearing readily becomes hot, because the babbitt metal liner retains, instead of throwing off, the heat. The regular working pressure also sets up a hydraulic pressure in the oil film, between the shell and the liner, which tends to produce breakages and cracks in the liner, as may sometimes be observed when removing bearings from gas engines, pumping engines and the like, subject to high pressures and shocks. A consequence of shocks is also that a liner which is somewhat loose will become distorted and "work"; this "working" produces additional friction and increased temperatures. All the facts mentioned above tend toward the one result, *viz.*, the increasing of the temperature in the bearings, even to the extent of melting down the babbitt liner.

From various tests which have been made, the results of one may be

given here. A bearing with a perfect contact between liner and shell was tested under a constant load of 400 pounds per square inch and a constant sliding speed of 480 feet per minute. The same bearing was again tested under the same conditions, but with the liner not in intimate contact with the shell. As the tests were necessarily made under a slightly varying atmospheric temperature, the difference between the actual bearing temperature and the room temperature was taken as the basis of each, and in the former case the result was 60 degrees F., while in the latter 85 degrees F. When such differences are obtained in a testing machine, under the best operating conditions, how much worse must be the influence of the slightest lack of contact under usual working conditions, such as we have them in steam engines, air compressors, pumps, gas engines, etc.!

Summing up the foregoing we may say that in most cases the direct causes of hot bearings are: A lack of contact between liner and shell, caused, first, by shrinkage and careless treatment of the babbitt, and second, by shrinkage strains produced by an unequal distribution of the liner masses over the shell; the formation of an isolating oil film, together with its consequences; cracks or breakages in the liner produced as explained. The means of avoiding these troubles, and the principles of a good and safe bearing construction, must consequently be an absolutely intimate and homogeneous contact between liner and shell; an equal distribution of the liner over the shell; and a strengthening of the liner against the shocks and working pressures. If these conditions are faithfully carried out, many troubles and much expense may be avoided.

CHAPTER III

THRUST BEARINGS

Thrust bearings are, in general, of two kinds: step bearings and collar bearings. In the former the thrust is taken by the end of the supporting shaft, in the latter by projections or shoulders at some distance from the end of the shaft. The simplest kind of a thrust bearing is the pivot bearing, exemplified by the bearings for watch pinions and by a lathe center taking the end thrust of a cut on a piece held between the centers in a lathe. In general, however, the end thrust is taken by

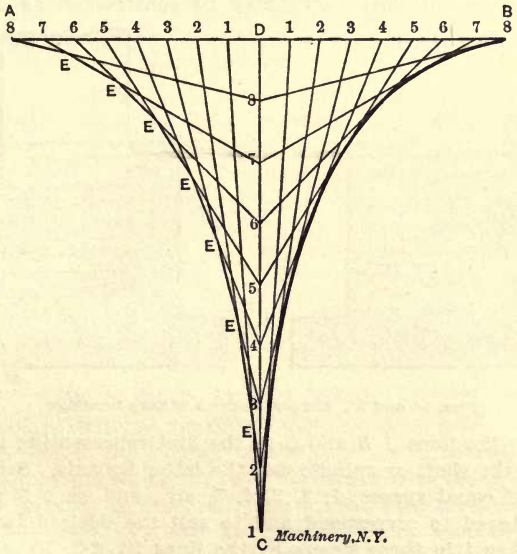


Fig. 13. Construction of the Schiele Curve

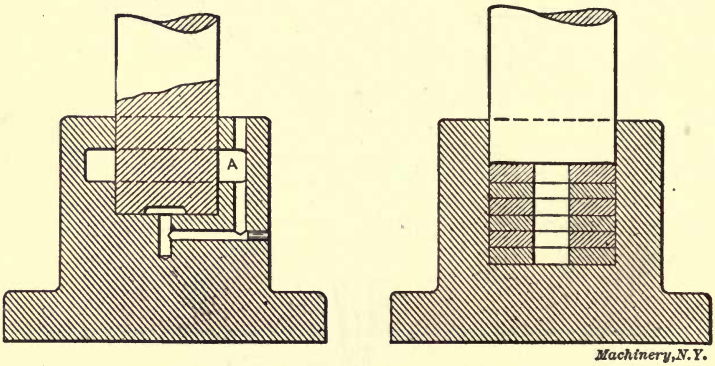
a large flat or nearly flat surface. When this is the case several considerations present themselves which must be given due attention by the machine designer.

Assume that the flat end of a vertical cylindrical shaft carrying a weight or otherwise subjected to pressure is supported by a flat surface. Then, if the shaft rotates, the velocities of points on its end surface at different radial distances from its axis, will vary. The velocities of the points near the outside will be, in comparison, very high, while the velocity of a point near the center will be low. On account of this variation in velocity, the wear on the end surface of the shaft and the thrust surface of the bearing will be considerably uneven. If the parts are well fitted together when new, so that a uni-

form pressure is produced all over the end of the shaft and bearing, then the outer parts of the bearing surfaces will wear away most rapidly. This again increases the pressure at the center, which sometimes may become so intense as to exceed the ultimate crushing strength of the material. The unequal wear of the surfaces of thrust bearings is one of the most difficult problems meeting the designer of machinery of which such bearings form a part.

Experiments carried out by Schiele show that the wear is theoretically along a curve called the *tractrix*, the construction of which will immediately be referred to. If an end thrust bearing is made of a form corresponding to the Schiele curve, then the wear in the direction of the axis of the thrust shaft will be uniform at all points; but while this curved form would be theoretically correct, it has been shown in practice that nothing is to be gained by the use of bearings having this complicated shape.

The tractrix or Schiele curve may be constructed as follows. In



Figs. 14 and 15. Simple Designs of Step Bearings

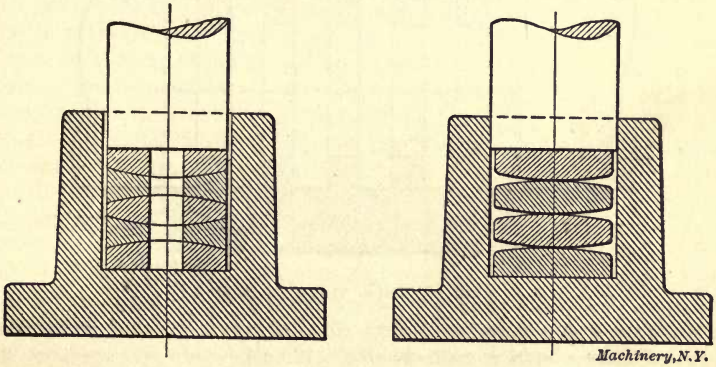
Fig. 13 draw the lines *A B* and *C D*, the first representing the extreme diameter of the shaft or spindle and the latter its axis. Set off on *A B* a number of equal spaces, 1, 2, 3, 4, 5, etc., and on *C D* other equal spaces numbered to correspond and to suit the desired length of the bearing. Then join these spaces by the lines 1-1, 2-2, 3-3, etc. The intersections *E* will be in the path of the curve to be constructed.

Simple Step Bearings for Light Duty

For light duty simple step bearings of the types shown in Figs. 14 to 17 answer the requirements well. The intense pressure at the center and the consequent unequal wear are partly avoided in the bearing in Fig. 14, by cutting away the metal at the center of the shaft, as shown, leaving an annular ring which takes the thrust. This procedure is advisable in all step bearings. Another difficulty met with in bearings of this type is the question of lubrication. If the speed of the shaft is high, the centrifugal force tends to throw the oil out from the center. Special provisions must then be made for again returning the oil to the center, as otherwise the bearing would wear down rapidly,

become heated, etc. In Fig. 14 a simple method is shown for automatically returning the oil to the bearing surfaces. An oil-passage is made from the chamber *A*, formed around the shaft, to the center of the shaft at the bottom. When the channel and chamber are once filled with oil, this oil will continue to circulate automatically; it will be drawn in at the bottom, be thrown outward by the centrifugal force, find its way into the chamber *A*, and finally, through the channel, return to the center of the bearing.

When a bearing for heavier duty is required, the design shown in Fig. 15 is quite commonly adopted. Here a number of disks or washers are placed between the end of the thrust shaft and the supporting bearing. The object of this is to introduce a number of wearing surfaces, instead of having the end of the shaft and the box take all the wear. Due to the fact that the series of washers introduced permits of a lower speed between each pair of washers, the wear is quite materially reduced. Should the pressure cause any two washers to heat



Figs 16 and 17. Improved Designs of Step Bearings

and bind, the frictional resistance between them ceases, as one washer is free to follow the motion of the other, and the oil will have an opportunity to get between the surfaces and cool them off.

A hole may be, and generally is, drilled through the centers of the washers, as shown in Fig. 15, and the same method for continual lubrication, as shown in Fig. 14, may be used to advantage. Every alternate washer is commonly made of hardened tool steel or case-hardened machine steel, while the others are made of bronze. This combination provides for good wearing qualities. If the thrust shaft is made of soft machine steel, and the box of cast iron, the top washer is often secured to the shaft, and the bottom washer to the box, so that all the wear may be concentrated upon the washers, which can easily be replaced.

In Fig. 16 is shown an improvement on the bearing in Fig. 15. This construction is recommended, in particular, in cases where the shaft and its bearing box cannot be properly aligned with one another. The washers have spherical faces, being alternately convex and concave. They are slightly smaller in diameter than the bearing box into which

they are inserted, so that they may have an opportunity to adjust themselves to a perfect bearing on each other, and thereby make up for the differences in the alignment of the thrust shaft and bearing box.

Another type of thrust bearing for loads which are not excessive is shown in Fig. 17. It is a well-established principle that it is better to take the thrust of a bearing as near the center of the shaft as the load to be carried will allow. The farther away from the center the support is, the greater is the motion and the greater is the retarding effect of the friction. The thin convex washers used are of tool steel, hardened, and although the bearing between them is very small, their strength and hardness is such that they are capable of standing a considerable pressure, though not as great a one, probably, as the other forms shown in Figs. 14, 15 and 16. In this bearing, also, there is no

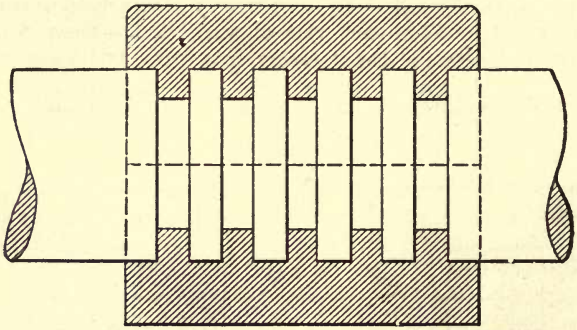


Fig. 18. Collar Thrust Bearing

difficulty in keeping the surfaces well oiled, since all that is necessary is to keep the chamber well flooded with oil.

Collar Thrust Bearings

When a considerable thrust is to be taken care of, or when the thrust is taken on the shaft at a distance from its end, collar thrust bearings are used. They are usually of the form shown in Fig. 18. In a well made bearing each of the collar surfaces takes its proportionate part of the load, and it is thus possible, without using excessive diameters, to properly distribute a very great thrust on a number of collars formed solidly with the shaft by cutting a number of grooves in the latter. One advantage of the collar bearing is that the difference between the outer and inner diameters of the bearing surface is not very great, and hence the velocities at the outer and inner edges do not vary appreciably; this, again, eliminates unequal wear on the thrust collar surfaces.

CHAPTER IV

FRICTION AND LUBRICATION*

Probably the most important and complete series of experiments on the friction of journals and pivot bearings yet undertaken was carried out by the late Mr. Beauchamp Tower for a Research Committee of the British Institution of Mechanical Engineers. In carrying out the experiments, as the result of an accidental discovery, an attempt was made to measure the pressure at different points of the bearing. A hole had been drilled through the cap and brass for an ordinary lubricator, when, on restarting the machine, oil was found to rise through the hole, flowing over the top of the cap. The hole was then stopped with a wooden plug, but this was gradually forced out on account of the great pressure to which the oil was subjected, and which on screwing a pressure gage into the hole was found to exceed 200 pounds per square inch, although the mean load on the journal was only 100 pounds per square inch. Mr. Tower proved by this and subsequent experiments that the brass was actually floating on the film of oil existing between the shafting and the bearing. By drilling a number of small holes at different points in the brass, and connecting each one of them during the test to a pressure gage, Mr. Tower was able to obtain a diagram showing the distribution of pressure upon the bearing. It appears that the pressure is greatest a little to the off side and at the middle of the length of the bearing, gradually falling to zero at each edge. The total upward pressure was found to be practically the same as the total load on the bearing, again showing that the whole of the weight was borne by the film of oil. Any arrangement which would permit the film to escape was found to result in undue heating, and the bearing would finally seize at a very moderate load. The oil bath lubrication was found to be the most perfect system of lubrication possible. In the table below the results obtained by Mr. Tower are specified for three different methods of oiling.

	Actual Load in Pounds per Square Inch	Coefficient of Friction	Relative Friction
Oil bath	263	0.00139	1.00
Syphon lubricator ...	252	0.00980	7.06
Pad under journal....	272	0.00900	6.48

With the needle lubricator and a straight groove in the middle of the brass for distributing the oil, the bearing would not run cool when loaded with only 100 pounds per square inch, and no oil would pass down from the lubricator. The groove, in fact, was found to be a most effective method of collecting and removing the film of oil. In the next place, the arrangement of grooves usual in locomotive axle boxes was adopted, the oil being introduced through two holes, one near each end and each communicating with a curved groove. This

* MACHINERY, March, 1907, and April, 1908.

bearing refused to take the oil, and could not be made to run cool, and after several trials the best results which could be obtained led to the seizure of the brass under a load of only 200 pounds per square inch. These experiments proved clearly the futility of attempting to introduce the lubricant at that part of the bearing. A pad placed in a box full of oil was therefore fixed below the journal, so as to be always in contact with it when revolving. A pressure of 550 pounds per square inch could then be carried without seizing, or very nearly the same load as in the case of oil-bath lubrication.

Results of Tower's Experiments

One important result was to show that friction is nearly constant under all loads within ordinary limits, and that it does not increase in direct proportion to the load according to the ordinary laws of friction. This is indicated by the result of the experiments recorded below.

Variation of Friction with Pressure.—Journal, 4 inches diameter, 6 inches long. Brass, 4 inches wide. Speed, 300 revolutions = 314 feet per minute. Temperature, 90 degrees F.

BATH OF LARD OIL

Pressure in pounds per square inch of bearing $p = \frac{W}{d \times l}$

Pressure per sq. in.	Coefficient of Friction $= \mu$	Product $p \times \mu$
520	0.0013	0.676
415	0.0016	0.664
310	0.0022	0.682
205	0.0031	0.635
153	0.0041	0.627
100	0.0067	0.670

BATH OF OLIVE OIL

Pressure in pounds per square inch of bearing $p = \frac{W}{d \times l}$

Pressure per sq. in.	Coefficient of Friction $= \mu$	Product $p \times \mu$
520	0.0013	0.676
468	0.0015	0.702
415	0.0017	0.705
363	0.0019	0.689
310	0.0021	0.651
258	0.0025	0.645
205	0.0030	0.615
153	0.0044	0.673
100	0.0069	0.690

The coefficient of friction with bath lubrication varies inversely as the pressure, or, in other words, the friction of the bearing is altogether independent of the pressure upon it; the first law of friction should therefore read: *Temperature and velocity remaining constant, the friction coefficient is proportional to the nominal pressure, and the work done against friction is independent of the load, provided this does not exceed from* 400 *pounds to* 600 *pounds per square inch.* From this it follows that the work done in overcoming friction is

independent of the load upon a machine, and that there is no appreciable increase in the loss due to friction from no load to full load. Under a load of 300 pounds per square inch and with a surface speed of 300 feet per minute, Mr. Tower found the coefficient of friction to be 0.0016 for oil-bath lubrication, and 0.0097 for a pad.

In the next place it was found that the coefficient of friction is inversely proportional to the temperature, other conditions remaining the same, as shown below.

Variation of Friction with Temperature.—Journal, 4 inches diameter, 6 inches long. Brass, 4 inches wide. Speed, 300 revolutions = 314 feet per minute. Load, 100 pounds per square inch of nominal area.

BATH OF LARD OIL

Temperature Degs. F.	(Degs. F. − 32) = t	Coefficient of Friction = μ	Product $t \times \mu$
120	88	0.0044	0.387
110	78	0.0050	0.390
100	68	0.0058	0.394
90	58	0.0069	0.400
80	48	0.0083	0.398
70	38	0.0103	0.391
60	28	0.0130	0.364

The second law of friction should therefore be stated: *Nominal pressure and velocity remaining constant, the coefficient, and therefore the work done against friction, is inversely proportional to the temperature of the bearing.*

This has also been very neatly demonstrated by a recent experimenter, Mr. Dettmar, whose machine is electrically driven, and therefore the consumption of current could be very accurately measured during a five hours' run at constant speed and voltage. As load and velocity remain constant throughout the test, a decrease in the loss due to friction could only occur with a diminution in the coefficient. The current fell off in the same ratio as the temperature increased.

The results of Tower's experiments seem to indicate that friction increases with the velocity, although not nearly in proportion to the square of the velocity as observed by Dettmar. As the result of the more exact determination possible with his machine, Dettmar found that friction increases very nearly as the 1.5 power of the velocity.

The mean values of the coefficient of friction for different lubricants, and with different methods of lubrication as observed by Mr. Tower, are given in the following table:

Variation of Friction with Different Lubricants.—Journal, 4 inches diameter, 6 inches long. Brass, 4 inches wide. Speed, 300 revolutions = 314 feet per minute. Temperature, 90 degrees F.

Lubricant	Coefficient of Friction	Max. Safe Pressure in Pounds per sq. Inch of Nominal Area
Olive oil	0.00172	520
Lard oil	0.00172	570
Sperm oil	0.00208	570
Mineral oil	0.00176	625
Mineral grease	0.00233	625

Nicolson's Experiments on Friction and Lubrication

The remainder of this chapter consists of an abstract of a paper read by Dr. J. T. Nicolson before the Manchester (England) Association of Engineers. The chief aim of the Nicolson experiments was to give some definite ideas about the resistance offered to the relative motion of lubricated surfaces, and they, in particular, related to journals and bearings as used in engineering practice. Experimental results obtained by Stribeck, Dettmar, Heimann, Lasche, and others have been utilized for framing rules which indicate that some views commonly held in regard to bearings are not correct. In particular, the idea that the length of the bearing should increase in proportion to the speed is shown to be erroneous.

Dry Friction

When one solid rubs upon another without any lubricant, the resistance offered to relative motion is due either to actual abrasion or to molecular interference between the two surfaces. Even though a metallic surface may appear to be perfectly smooth to the eye, its real condition, if viewed with a powerful microscope, resembles that of a rugged mountain system. When one surface is slid upon another, these surfaces exercise a resisting force. The following laws may be considered as generally covering the question of dry friction:

1. Within certain limits, the frictional resistance may be said to be proportional to the load, and to be independent of the extent of the surface over which the load is distributed; but when the pressure or load per unit area is large, the friction inceases at a greater rate than the load, or, in other words, the coefficient of friction increases with the pressure.

2. The coefficient of friction varies with the speed of motion. It is greatest when the motion is slowest, and when one body is just commencing to move relative to another, we have what is called friction of repose. This friction has been found by experiments to be from 0.3 to 0.4 for iron upon iron; for moderate speeds the friction varies from 0.15 to 0.25 for the same material; and for speeds from 10 to 90 feet per second, coefficients of from 0.10 to 0.20 have been found by experiments.

3. The friction of solids with no lubricant interposed has been found to diminish as the temperature increases. This is due to the fact that abrasion is easier at high temperatures.

Friction and Lubrication

When some lubricant is placed between moving bodies, the valleys or the uneven surfaces are leveled up, and the intensity of the molecular action is diminished. For the frictional work when a shaft rotates in a well lubricated bearing, we may state the following formula, expressing the frictional work done per revolution:

$$\text{Frictional work per revolution} = \frac{\pi\, d\, u\, W}{12} \text{ foot-pounds.}$$

In this formula,

d = diameter of shaft in inches,

u = coefficient of friction (0.15 on an average),

W = load on the bearing in pounds.

This formula holds true when there is plenty of oil, so long as the speed is small. If we take as an example the case of the spindle for a 10-inch lathe, running slowly, with a weight of 3000 pounds carried by the front bearing, which is 3½ inches in diameter, then the friction work per revolution is

$$\frac{\pi \times 3.5 \times 0.15 \times 3000}{12} = 412 \text{ foot-pounds per revolution.}$$

If a cut were 1/4 inch × 1/16 inch on soft steel, the cutting force would be, say, 3500 pounds, and on a 20-inch face-plate diameter the work spent in cutting per revolution would be

$$3500 \times \frac{20\,\pi}{12} = 18{,}300 \text{ foot-pounds.}$$

The work lost in friction by the journal is therefore 2.26 per cent of the useful work. A similar calculation for a 48-inch lathe would show a loss of about 10 per cent. These great frictional losses constantly occur with lathe spindles or other rotating shafts, revolving slowly, even when abundantly fed with oil, and indicate the necessity for using measures to preserve a separating film of oil between the shaft and bearing, and not to allow them to run in metallic contact. This is more difficult to accomplish at slow than at high speeds.

Automatic Lubrication

The following rules for supplying bearings with oil will give the best results in practice: If the oil is fed in by the ordinary cup and syphon, or by a ring or centrifugal method of supply, it should be made to flow onto the journal at the place where the pressure is least. The oil should therefore be fed from a point situated in the top rear quadrant of the bearing when the journal is loaded by gravity only, and the point should be further back the slower the speed. This applies, then, especially to the large lathes. If the loading of the journal is principally due to cutting force acting upward upon it, the feed should be placed in the bottom front quadrant, and nearer the front, the slower the speed of rotation. This meets the case of the smaller sized lathes.

The compromise ordinarily effected to enable the lubricant to enter, whatever may be the direction of the loading, is the simple one of fitting the oil cup on the top of the bearing. This seems almost the only thing to do in the case of automatic lubrication, but it is the correct position only when the resultant force upon the journal, due to gravity and cutting force, etc., acts nearly horizontally and from front to rear.

Forced Lubrication

When the lubricant is supplied by mechanical means at a fixed rate and at any required pressure, it must be fed in at the points of greatest

oil pressure in the bearing. For large lathes, where gravity is more important, the region of greatest pressure lies in the rear bottom quadrant. For small lathes, on the other hand, in which the force on the spindle acts upward, owing to the cutting force being relatively greater, the maximum oil pressures occur in the front top quadrant. To meet all contingencies, it would appear on the whole best, in the case of forced lubrication, either to force the oil in at the back of the bearing, well below the center, or preferably to fit three alternative branches from the oil pressure supply pipe to the back, top, and front, any one of which may be turned on at will to suit the conditions of working.

Frictional Resistance Due to Viscosity

In describing the phenomena occurring when a journal rotates in a bearing, we have, so far, not alluded to the nature or magnitude of the frictional resistance experienced when there is an abundant supply of lubricant completely separating the former from the latter, and preventing any metal-to-metal contact. It is frequently stated that "there is no friction without abrasion," or, in other words, that unless two metals rub against each other there can be no resistance due to relative motion. This, however, is not the case. When a film of lubricant is interposed between two metallic surfaces there is a resistance to relative motion of these surfaces due to the shearing or transverse distortion of the oil film.

This resistance does not depend on the load. It is governed only by the area of viscous fluid to be sheared and the viscosity of the oil, *i. e.*, the kind of oil and its temperature (with which the viscosity greatly alters), and it also gets greater the smaller the thickness of the film, so that if the shaft is a close fit within its bearing the resistance to motion will be greater than if the fit is an easy one.

There are very few cases in engineering practice where a journal rotates with a uniform thickness of oil around it, and it is only at very high speeds that this takes place. At moderate and low speeds the shaft moves to one side an amount depending on the speed of the load, the eccentricity for any given load becoming less the greater the speed. We have already said that the frictional resistance depends on the thickness of the oil film. Experiments have shown, however, that the thickening of the film on one side of the shaft is more than counteracted by the thinning of the film on the other, so that, in general, the friction gets greater when the journal becomes more eccentric.

Considering, therefore, the bearing running slowly, in which a lubricant has just formed a complete film all around the shaft, it will have its maximum amount of eccentricity, and the frictional resistance will, on this account, be large. As the speed increases, the eccentricity diminishes. The friction increases with the speed, but it diminishes, on the other hand, with the eccentricity. Experiments show that at first there is a decrease and then an increase, so that the coefficient of friction attains a minimum value which depends on the circumstances in each case. With further increase in speed, the diminishing

of friction, due to the lessening eccentricity, becomes insignificant, and after a certain interval the simple law of friction is followed, whereby friction increases in proportion to the velocity of rubbing.

For speeds greater than at from 20 to 80 feet per minute, the temperature of the oil film also exerts its influence. This temperature rises above that of the bearing, and its viscosity becomes reduced. The frictional resistance then increases less rapidly than in exact proportion to the speed. The faster the journal runs, the more the temperature of the oil film rises above that of the bearing, and the thinner or less viscous becomes the oil. Thus, for speeds from 50 to 90 up to about 450 feet per minute, the coefficient of friction is proportional to the square root of the speed of rubbing. For speeds between 450 feet and 800 feet per minute the friction increases more slowly, and varies as the fifth root of the velocity. For speeds as high as 3,600 feet per minute and upward, the influence of the speed disappears altogether, and the conclusion is arrived at that for bearings of high-speed generators, for instance, driven by steam turbines, whose rubbing speeds are nearly a mile a minute, the coefficient of friction is the same, whatever be the speed.

Application of Results of Experiments to the Design of Bearings

In endeavoring to apply the theoretical explanations and the experimentally found formulas, the question arises: What is the proper proportion of length to diameter, under any given condition, as to load, speed and kind of lubrication? According to hitherto accepted rules, the length of the bearing should increase with the load and with the number of revolutions. The experiments and formulas arrived at by the author indicate, however, that the heat developed in the bearing depends only upon the rubbing velocity, and is quite independent of the length of the journal. We cannot, therefore, hope to lower the temperature by lengthening the bearing. The heat generated increases as fast as the area for dissipating it increases, and, although by lengthening the journal the bearing pressure is diminished, the frictional resistance and the heat generated are increased. On the other hand, we know from experience that journals must be made long for high speeds, and the above calculations seem, at first sight, to be in conflict with accepted practice. The explanation of this is as follows: While it is true that the final temperature to which the bearing will rise after a long run, under a given load, and with a given lubricant, depends only on the diameter of the spindle and the speed of revolution, that is, only upon the rubbing velocity, and not at all upon the length of the journal, we have to remember that if the finally attained temperature be too high, the lubricant will be squeezed out unless the bearing pressure is low.

Another conclusion arrived at by these experiments, contrary to the view usually accepted, is that the length of the bearing must be greater, the slower the speed. This, however, is clearly correct, for the slower the speed, the greater difficulty has the shaft in dragging

in its supply of oil to meet the required demand, in opposition to the bearing pressure which is squeezing it out, and consequently the unit bearing pressure should accordingly be lower in order to enable the journal to maintain its oil film unbroken.

Journals for Heavy Loads at Slow Speed

One kind of bearing which presents special conditions, and which is frequently met with and has to be dealt with in practice, is that in which a journal has to run under a heavy load at a very slow speed. What we have here to guard against is the entire collapse or tearing asunder of the film of lubricant, owing to the slow speed at which the bearing is being worked; and when once the tearing of the oil film begins, the journal is unable to bring up a fresh supply, owing to its small surface speed.

Calculations and experiments show that it is impossible to give the large dimensions to the front bearing of a heavy lathe that would be necessary to prevent the oil film from being broken at such slow speeds; and, as a matter of fact, lathe spindles turning at the slow speeds used for heavy cuts inevitably run metal-to-metal with their bearings, giving rise to the high frictional resistance corresponding to the coefficient of friction of 0.15 for greasy metals. The work thus spent and wasted on friction and wear may amount to from 2 per cent to 10 per cent of the total useful work expended on cutting. From ¼ to 9 (according to size) horsepower is, therefore, wasted on the friction of the front journal alone when the lathe is running at these slow rates with a heavy job between centers. Even if the working pressure is light, and the thrust on the front journal is due to the standard cut only, it can be shown that 2½ per cent of the useful work is spent on friction on any size of lathe when the speeds are so low as to squeeze out the oil film.

We are here face to face with a very serious loss of power, and a correspondingly large amount of wear of the spindle and in the front bearing, not at all due to high speeds of rotation of the spindle; and it is owing to this that the elaborate arrangements for adjustment of the spindle in a lathe head-stock have to be provided.

It is impossible to give enough area in the front bearing of a lathe head-stock to prevent metallic contact of journal and brass at the slower speeds, if dependence is placed upon the lubricant being carried in by the ordinary action of the shaft's rotation, the supply being automatic. By using a force pump, however, and injecting a stream of moderately heavy oil into the bearing at the place where the pressure is greatest, it is possible to raise the journal off the brass even when at rest, and to keep it floating with a film of oil interposed between itself and the bearing when in motion, be that motion as slow and the load as high as it may. If metal-to-metal contact can in this way be prevented at slow, and by the ordinary methods at high speeds, there seems to be a possibility that wear may be entirely eliminated. If this be so, it follows that adjustments for wear are unnecessary, and instead of the elaborate and expensive designs of front and back bear-

ing which are now used, we may expect that a simple solid bush of ample thickness will meet every requirement. Such a solid bush, of hard bronze round the steel spindle, has a great deal to recommend it from the point of view of accuracy of fit, solidity, and stiffness, as compared with the intricate methods of adjustments now common.

Modern Practice for Lubricating Bearings

The chief distinction between the modern and the older methods of lubricating bearings lies in that the oil is no longer supplied drop by drop, as formerly, but in an abundant stream, the oil serving the purpose not only of lubrication, but of carrying away the heat.

For high speed bearings, the principle most often adopted is that of the "closed circuit"; that is, the oil is used over and over again; after dropping off the journal into a collecting reservoir it is filtered and used anew, being automatically supplied to the journal at any suitable point. A cooling arrangement is sometimes fitted in the reservoir, so as to remove the heat from the oil, and consequently also from the bearings. The system of forced lubrication is also adopted to a great extent. The oil is then, by means of a pump or other suitable device, pressed in between the rubbing surfaces so that the journal floats on the heavy film of lubricant.

Lubricating Horizontal Bearings

The most common method of lubrication for horizontal journals running at high speed is the ring-oiled bearing, in which a loose ring, resting on the shaft, turns with it, dipping into the oil reservoir at the lower side, and bringing up the oil to the top surfaces of the journal, from where it flows over into the oil grooves. No ribs or other projections should be fitted on the rings, as such arrangements produce a resistance to their passage through the oil bath, and bring them to a standstill. At high speeds, the centrifugal force renders the flow of oil from the ring to the journal difficult, and scrapers are used for diverting the oil into the oil channels. These, however, should never touch the ring, as they will then stop its motion.

Self-oiling bearings having rings fast on the shaft are not much used. The fast ring cannot stick, but it requires a longer design of bearing. The ring may act as a collar where endwise motion is to be prevented; but as such motion is usually an advantage, the ring should ordinarily be attached to the shaft so that it can slide on its key. For high speeds the scraper may be used with fast rings, to overcome the centrifugal force.

Forced Lubrication

By the use of a pump to force the oil drawn from the reservoir into the bearing to the point of maximum pressure, the length of the bearing can be very much diminished even for the slowest speeds, especially for journals whose load and rotation direction do not change. For such bearings the length need, in all probability, not be more than equal to the diameter of the shaft. With such bearings there ought

hardly to be any wear at all. The system is extensively used in high speed steam engines and gas engines.

Grease as a Lubricant

Grease has certain advantages as a lubricant which make its use advisable in many places, but it should not be expected that its lubricating value is ever as good as that of the best oil, although it may give better results in some places. For example, grease is particularly valuable for bearings exposed to dust, for when it is forced into the bearings with compression grease cups, the grease flows outward around the journals, forming a perfect dust protector, both because it seals the bearing and because the outward flow of the grease repels the intrusion of dust and abrasive particles. In such places the best oil would not give nearly as good results as grease, although its lubricating quality is generally considerably greater. On the other hand, the use of grease for lubricating machinery of a mill would not be advisable where the power factor is important in the cost of production. For example, some tests were made several years ago in the lubrication of the machinery of a flour mill that was run by two water wheels of the same size, as stated by Mr. W. F. Parish, Jr., in a paper read before the North Eastern Coast Institute of Engineers and Ship Builders. In making the trial of grease the section driven by No. 1 water wheel was fitted up first. As the grease displaced the oil it was noticed that the speed of the mill decreased with a consequent decrease of production. At first no one thought that the grease was responsible for the slowing down, but as the second part of the mill slowly decreased in speed as the use of the grease was extended, a consulting engineer was called in, who suggested that, in view of the fact that speed had decreased with the introduction of grease, it was responsible for the loss of production. Upon the resumption of the use of oil the speed of the machinery again rose to its original figure, proving conclusively that the lubricating value of the grease was inferior to that of oil and that the difference was an important factor in the mill's production. The relative value of different oils in the lubrication of textile mills has long been known to be important in influencing the cost of production.

CHAPTER V

BEARING METALS*

By conservative estimate the value of the bearing metal in actual use in the United States exceeds $50,000,000, of which fully one-half is used on the locomotives and rolling stock of the railroads of this country. In view of the increase in the amount of machinery and rolling stock steadily going on, and the constant wearing out and replacement of bearings, the value and importance of this product cannot be overestimated. The life of a machine is largely dependent upon its bearings, and in view of this the fact that knowledge in regard to bearing metals and alloys is not more general, is remarkable. Again, the nature of the production of these alloys is such that, while in some cases they have been patented and are manufactured under trade names, in many others they are made up of scrap, with widely varying proportions of the different metals incorporated in their structure; on this account, probably no phase of engineering progress in machinery construction and operation is the subject of more difficulty and dissatisfaction.

The fact that bearing metals have to be taken largely on faith or else tested by more or less complicated processes for their chemical constituents, and the further fact that trade conditions in this field are such that the properties of metals are apt to vary greatly in different shipments, is a matter of grave import to the average machinery manufacturer and operator. Only the largest consumers can afford to make the necessary tests and investigations of a given consignment in order to test its quality, and, in addition, a definite amount of special knowledge is requisite for this purpose, in view of the often wide variations in properties of the alloy, with a comparatively small variation in the proportion of its constituents. Under these circumstances the average small machine shop and consumer in this field accepts bearings on faith alone and is dependent largely upon the commercial reputation of the firm furnishing the material. That this should not be so is a foregone conclusion, but in view of this condition of affairs the rapid progress of the firm whose standing can be relied upon in this field is readily explained.

Bearings are usually composed of alloys of copper, lead, tin, antimony and zinc, and are known as babbitt metal (after the name of the discoverer of this material), white metal, brass, phosphorous bronze, and various other trade names. Quite a number of these are patented, such as "plastic bronze," etc., but many are sold merely under trade name, and in some instances are of uncertain composition.

The principal qualities which a good bearing metal should have are good anti-frictional properties, so as to withstand heavy loads at high speed, without heating, and, second, sufficient compressive strength so

*MACHINERY, August, 1909.

as to neither be squeezed out of place under high pressure, nor crack or break when subjected to sudden shocks. In addition to these, many other properties must be considered in a choice of bearing metals depending upon the special purpose for which the material is to be utilized. Temperature variation is often an important factor, especially in refrigerating plants, and the coefficient of expansion should be considered to prevent undue binding, with consequent destruction of the bearing and the possible variation in other properties, such as brittleness, ductility, etc., under various temperature conditions. In addition, many bearings must operate under conditions where they are subject to chemical action, whether that of brine or ammonia in refrigerating plants, or acids, alkalies, etc., in chemical establishments, and in dynamo and motor construction and operation, the electrical conductivity must be considered as well. This statement applies equally to all bearings incorporated in electrical machinery, where these must serve as electrical conductors such as the bearings for the wheels in trolley cars, etc.

The chief properties to date which have been developed to a greater extent than others in machine design are those of friction elimination and resistance to compressive loads. Theoretically, all metals have the same friction, according to Thurston, and the value of the soft white alloys for bearings lies chiefly in their ready reduction to a smooth surface after any local impairment of the surface, such as would result from the introduction of foreign metal between the moving surface and the bearing. Under these circumstances the soft alloys flow or squeeze from the pressure into the irregularity, forming a larger area for the distribution of the pressure, thus diminishing its amount per unit of area. Further, the larger area over which the pressure is extended the less becomes the liability to overheating and consequent binding. Under these circumstances the frictional properties of a bearing are in inverse ratio to their compressive resistance, and invariably the best bearing alloys, from a high speed standpoint, are unsatisfactory for utilization in heavy machinery. The recent introduction of an iron or steel grid to form the base of the main bearing, and to be filled with much softer bearing metals than could ordinarily be installed, or in some cases even graphite, is a step in the right direction and presents possibilities of great importance in this field of machine development.

Lead flows more easily under pressure than any of the common metals, and hence it has the greatest anti-frictional properties. Of course, a number of metals exceed lead in this property, but their cost or some other factor render them unavailable. Lead is the cheapest of the metals, except iron, and in comparison to the other metals used in the formation of bearing alloys their relative prices are somewhat in the following order per one hundred pounds: Lead, $4; zinc, $5; antimony, $9; copper, $13; and tin, $30 or more. It can thus be seen that the more lead that is used in a given bearing, the softer it is, the less friction it possesses, and the cheaper it can be furnished. It is, however, too soft to be used alone, as it cannot be retained in the recesses of the

bearing even when used simply as a liner and run into a shell of brass, bronze or gun-metal or some other alloy. Various other metals have been alloyed with it, such as tin, antimony, copper, zinc, iron and a number of non-metallic compounds, such as sodium, phosphorus, carbon, etc., and the effect of the different ingredients is to-day fairly well understood.

If antimony is added to the lead it increases its hardness and brittleness, and if tin is added as well it makes a tougher alloy than lead or antimony alone. Nearly all of the various babbitt metals on the market are alloys of lead, tin and antimony in various proportions, with or without other ingredients added. In such babbitts the wear increases with the antimony as a general thing, and the price with the

COMPOSITION OF BEARING METALS.

Alloys	Lead.	Tin.	Antimony.	Copper.	Zinc.	Other Constituents.
Babbitt 1	80.00	20.0	.....	.....	.....	.....
Babbitt 2	72.0	21.0	7.0	.....	.....	.....
Babbitt 3	70.0	10.0	20.0	.....	.....	.....
Babbitt 4	80.5	11.5	7.5	0.5	.....	.....
Babbitt 5	0.5	68.0	.....	1.0	31.5	.....
Babbitt 6	....	20.0	.....	.....	80.0	.....
Babbitt 7	.....	86.0	10.0	4.0	.....	.....
White metal	82.0	.....	12.0	6.0	.....	.....
White Brass		64 0	.....	2.00	34.0	.....
Magnolia metal	80.00	4.75	15.0	trace		$Bi = 0.25$
Car brass lining	80.5	11.5	7.5	0.5	.....	.....
Ajax plastic bronze	30.0	5.0	.....	65.0	.....	.....
Ajax metal	11.5	11.5	.....	77.0	.....	.....
P. R. R. car brass, B.	15.0	8.0	.....	77.0	.....	$P = 0.80$
S bearing metal	9 5	10.0	.....	79.7	.....	.....
Delta metal	5.1	2.4	.....	92.4	.	$Fe = 0.1$
Camelia metal	14.8	4.3	.....	70,2	10.2	$Fe = 0.5$
Tempered lead	98.5	0.08	.....	0.11	.....	$Na = 1.30$

Bi = bismuth; P = phosphorus; Fe = iron; Na = sodium.

tin. The higher antimony babbitts are used in heavy machinery, as they are harder, while those low in antimony are used in high speed machinery. The steady increase in speed at which various operating units are maintained is responsible for a wide deficiency in this field in the duty performed by the bearing metal. The chief difficulty to-day in the operation of the modern turbine is undoubtedly the maintenance of satisfactory bearing surfaces. Soft babbitts have never sufficient strength to sustain the weight and shock of heavy machinery bearings and can only be used as liners. The tendency to increase in speed as well as weight or size of machinery is limited to-day simply by the satisfactory operation of the bearing metal itself.

Undoubtedly, in investigations in this field, sufficient attention has not been paid to the effect of temperature on the bearing properties of the alloys used for these bearings. More rigid investigation in this field and limitations in regard to the temperatures permissible, with

means for maintaining these within fairly close limits, will undoubtedly result in a great increase in the possibility of improvements in speed and weight of various types of machinery. More or less extensive experiments along these lines are being conducted in regard to the bearings used in turbine construction, since the speed here has rendered the problem an acute one and is necessary for efficient operation of the turbine itself.

The accompanying table will doubtless prove interesting as showing the various constituents of the more or less common bearing metals now on the market. The original babbitt metals were very expensive materials, on account of the proportions of the more expensive metals found in them, and have been much modified in actual practice. A wide deviation in the composition of babbitt is readily shown in the first part of the table. The first babbitt is a fairly good alloy for high speed machinery, but is not very hard. Its melting point is about 500 degrees F.; in fact, the properties of all allows or bearing metals can be very widely deduced from their melting point. The second babbitt is somewhat harder and melts at a higher point. Both of these are used largely for lining purposes. The fourth babbitt is used very widely for heavy machinery. All of the babbitts mentioned have been fairly successful.

Babbitt 6 has good wearing properties, but cannot be used for high speeds. Most of the other metals included in the table where copper is not used in excess can be regarded as in the same class as babbitts. The "white" class has a fairly good electrical conductivity, much greater than that of ordinary babbitt, and is used in the bearings of generators, motors, electric cars, etc. A rather interesting thing about the alloys containing sodium is based upon the fact that sodium by oxidation produces a material which will saponify with the oil used in the bearing and produce soap, thus assisting lubrication. The extent and amount of such action is scarcely as yet understood, and practically no experiments have been made with this investigation in view. Possibilities along this line, however, are great, not only for this particular alloy, but for many others not as yet considered.

The other alloys included in the table consist to a very great extent of copper, tin, and lead, and usually have a thin liner of lead or some soft babbitt, and hence wear much better than an entire bearing of the soft babbitt. The tendency to wear decreases with increase of lead and increase of tin. Increase of lead, of course, affects the frictional quantities of the alloy and hence its heating properties. A certain amount of other metal, however, is necessary to keep the lead from separating from the copper. A study of the table itself, with a knowledge of the various properties of the metals themselves, will show conclusively the bearing properties of the different alloys. Pure copper is so tenacious that it is practically impossible to work it with any tools whatever without preliminary treatment, and this same property extends into and influences its bearing properties.

The structure and treatment has more to do with the production of suitable bearing alloy than is generally considered. The tensile

strength of solder and, in fact, all alloys, decreases very greatly with the pressure or tension at the time of solidification, and in general the cooling process, and the influence on tempering, affect the structure and consequently compressional resistance to a much greater extent than is generally considered. The same properties which influence the hardening and tempering of steel by heat, extend to a greater or less degree to all metals and are much more pronounced in alloys than in the simple elements.

Sufficient has been said to show the importance of the bearing metals in machine design to-day, and to give a brief outline of the situation in regard to the character and type of the metals available, with a few of the properties of the same. The possible combinations of alloys for this purpose are very great. Comparatively little progress has been made along investigations covering all possible alloys of different materials in different proportions. The recent introduction and placing

COMPOSITION OF BRONZES

	Parts.
White Metal:	
Tin	7.6
Copper	2.3
Zinc	83.3
Antimony	3.8
Lead	3.0
Hard Bronze for Piston Rings:	
Tin	22.0
Copper	78.0
Bearings—Wearing Surfaces, etc.:	
Copper	6
Tin	1
Zinc	¼
Naval Brass:	
Copper	62.0
Tin	1.0
Zinc	37.0
Brazing Metal:	
Copper	85.0
Zinc	15.0
Anti-friction Metal:	
Copper—(best refined)	3.7
Banca tin	88.8
Regulus of antimony	7.5
Well fluxed with borax and rosin in mixing.	
Bearing Metal—(Pennsylvania Railroad):	
Copper	77.0
Tin	8.0
Lead	15.0

on the market of a large number of metals, such as calcium, etc., very common in nature, and ultimately bound to be furnished at a very low rate, and many of them possessing very suitable properties for bearing alloys, is undoubtedly bound to influence the situation; and various engineering devices, such as the steel grid, recently developed, will undoubtedly receive attention in the immediate future with consequent increase in efficiency in this field. The development is but at its inception along this line, and standardization of the alloys at hand should

be at once insisted upon and maintained by the various machine manufacturers. This latter is the chief difficulty to-day in commercial development. The scientific end will largely take care of itself. The effect of different metals upon alloys by their presence in various proportions can be foretold to-day largely from theoretical considerations; but that the commercial situation to-day, however, is unsatisfactory, is a foregone conclusion.

The table on the preceding page, giving the composition of bronzes used by the U. S. Navy Department, was contributed to MACHINERY'S Data Sheets by Mr. F. W. Armes, and is reproduced from Data Sheet No. 31, April, 1904.

CHAPTER VI

ALLOYS FOR BEARINGS*

In an important article, in the *Journal of the Franklin Institute* for July, 1903, Mr. G. H. Clamer discussed the advantages and disadvantages of various compositions and alloys for bearings, and especially alloys for railway journal brasses. He also quoted the results of many tests on various compositions made on an Olsen testing machine designed by Prof. Carpenter of Cornell University. The present chapter is devoted to an abstract of Mr. Clamer's article, and contains all the most important features of his discussion on a subject on which not so much is generally known as would be desirable.

Upon close examination we find that there are but few metals available for bearings. As mentioned in the previous chapter, they are copper, tin, lead, zinc and antimony. While other metals may be introduced in greater or less proportions, the five mentioned must constitute the basis for the so-called anti-friction alloys. The combinations of these metals now used may be grouped under the two heads of white metal and bronze. Bronze is the term which was originally applied to alloys of copper and tin as distinguished from alloys of copper and zinc; but gradually the term "bronze" has become applied to nearly all copper alloys containing not only tin, but lead, zinc, etc., and no sharp lines of demarcation exist between the two.

Principal Requirements of Bearing Metals

White metals are made up of various combinations of lead, antimony, tin, copper and zinc, and may contain as few as two elements, or all five. Bronzes are made up of combinations of copper, tin, lead and zinc, all of them containing copper and one or more of the other elements. The essential characteristics to be considered in any alloy for bearings are composition, structure, friction, temperature of run-

* MACHINERY, October, 1903.

ning, wear on bearing, wear on journal, compressive strength, and cost.

It is utterly impossible to have one alloy reach the pinnacle of perfection in all of these requirements, and so it is important to study the possible compositions and determine for what purpose each is adapted. It has been shown that a bearing should be made up of at least two structural elements, one hard constituent to support the load, and one soft constituent to act as a plastic support for the harder grains. Generally speaking, the harder the surfaces in contact, the lower the coefficient of friction and the higher the pressure under which "gripment" takes place. It would seem for this reason that the harder the alloy the better; and it was with this idea in mind that the alloys of copper and tin were so extensively used in the early days of railroading. A hard, unyielding alloy for successful operation must, however, be in perfect adjustment, a state of affairs unattainable in the operation of rolling stock. For this reason the lead-lined bearing was introduced and the practice of lining bearings has now become almost universal in this country.

General Comparison between Hard and Soft Alloys for Bearings

While the harder the metals in contact the less the friction, there will also be the greater liability of heating, because of the lack of plasticity, or ability to mold itself to conform to the shape of the journal. A hard, unyielding metal will cause the concentration of the load upon a few high spots, and so cause an abnormal pressure per square inch on such areas, and produce rapid abrasion and heating.

The bronzes will, generally speaking, operate with less heat than softer compositions, while the softer metals will wear longer than the harder metals. In the matter of wear of journal, however, the soft metals are more destructive. Particles of grit and steel seem to become imbedded in the softer metal, causing it to act upon the harder metal of the journal like a lap. High-priced compositions are being used that have but little resistance to wear compared with cheaper compositions, and low-priced alloys are in service that are not cheap at any price. It is generally conceded that soft metal bearings cause a marked decrease in the life of the journal, and yet they have many marked advantages, as we shall presently see.

Alloys Containing Antimony

1. *Lead and Antimony:* These metals will alloy in any proportion. With increase in antimony the alloy becomes harder and more brittle. It has been determined that when it is made of 13 parts antimony and 87 parts lead, the composition will be of homogeneous structure. If there is a greater proportion of antimony, free crystals of antimony appear, imbedded in the composition; and if less than 13 per cent, there appear to be grains of the mixture itself imbedded in the lead as the body substance.

According to one writer, an anti-frictional alloy should consist of hard grains, to carry the load, which are imbedded in a matrix of plastic material, to enable it to mold itself to the journal without undue

heating. Such a condition would be met in a lead and antimony alloy having above 13 per cent antimony, but it is not advisable to use in any case more than 25 per cent antimony, as the composition would be too brittle. The same writer claims that alloys having from 15 to 25 per cent antimony are the best adapted for bearings.

Mr. Clamer, however, does not agree with this, and says that alloys containing below 13 per cent antimony can likewise be said to consist of hard grains consisting of the composition itself, imbedded in the softer material, lead, as mentioned above. He says: "It has been my experience that, although the friction may be higher in such alloys, the wear is greatly diminished, and where pressures are light, causing no deformation, this is a great advantage. I have seen many instances in service where alloys between 15 per cent and 25 per cent were greatly inferior to alloys between 8 per cent and 12 per cent, owing to their frequent renewal due to wear. It will perhaps be interesting to hear that the Pennsylvania Railroad Company, at the suggestion of Dr. Dudley, their chemist, have adopted the 13 per cent antimonial lead alloy as a filling metal for bearings in order to obtain the best results. In a general way my own work in the subject has confirmed the opinion that lead is the best wear-resisting metal known, and that with increasing antimony, or increasing hardness and brittleness, the wear becomes more marked. This is due to the splitting up of the harder particles."

The friction, as we may naturally expect, becomes less with increase of antimony, and the temperature of running likewise diminished when running under normal conditions; but the harder the alloy, the more difficulty is experienced in bringing it primarily to a perfect bearing, and the greater the liability of heating through aggravated conditions. The wear on the journal one would naturally expect to be decreased with increasing hardness; but this journal wear is in all probability not due so much to the alloy directly as it is to the fact that the softer metals collect grit, principally from the small particles of steel from the worn journal, and, acting as a lap, cause rapid wear. With the harder metals these particles are worked out without becoming imbedded.

The cost of the lead and antimony alloy is the least which can be produced. It can be used in many services where higher priced alloys are being relied upon mainly for their high cost. It is one of the greatest extravagances of large industrial establishments to use materials that are too good for certain uses, and even perhaps unsuited, under the supposition that they must be good because they paid a good price for them. This fact has no greater exemplification than in the purchase of babbitt metal, and is due to the great uncertainty which exists, not only among consumers, but among manufacturers, many of whom carry on their business much the same as the patent-medicine man.

2. *Lead, Antimony, and Tin:* It should not be assumed that antimony-lead is the cheapest alloy to use under all circumstances; not so, for when high pressures are to be encountered, tin is a very desirable

adjunct. Tin imparts to the lead-antimony alloy rigidity and hardness without increasing brittleness, and can produce alloys of sufficient compressive strength for nearly all uses. The structure of a triple alloy of this nature is quite complicated, and not yet sufficiently defined.

The cost of the alloy increases with increase of tin; but for certain uses, where sufficient compressive strength cannot be gotten by antimony, because of its accompanying brittleness, it is indispensable, and will answer in nearly every case where the tin basis babbitts are used.

3. *Tin and Antimony:* These are seldom used alone as bearing alloys, but are extensively used for so-called Brittania ware, and in equal proportions for valve seats, etc.

4. *Tin, Antimony, and Copper:* This combination is what is known as genuine babbitt, after its inventor, Isaac Babbitt, who presumably was the first man to conceive the idea of lining bearings with fusible metal. The formula, which for no arbitrary reason he recommended, is as follows:

Tin	89.1
Antimony	7.4
Copper	3.7

This formula is still considered the standard of excellence in the trade, and has been adopted by many of the leading railroads, the United States Government, and many industrial establishments. It is used in the majority of cases where cheaper composition would do equally as well. It is the most costly of all bearing alloys because of the high content of tin.

5. *Tin-Antimony-Lead-Copper:* This quadruple combination of metals cannot be satisfactorily described, as it would no doubt take years of study to fathom the complicity of the metallic combinations here represented. Suffice it to say that lead, although of itself a soft metal, renders this alloy, when added in but small proportions, harder, stiffer, more easily melted and superior in every way to the alloy without it, and yet consumers will raise their hands in horror when a trifling percentage of lead is found in their genuine babbitt. This is one of the instances where cheapening of the product is beneficial.

The foregoing represents the more important combinations of alloys of tin and lead basis. These are of far more importance in the arts than the white metals, the main portion or basis of which is zinc.

At various times new combinations of zinc have been proposed, but, with very few exceptions, they have not come into popular use for two reasons: First, because of the great tendency of zinc to adhere to iron when even slightly heated. What is technically known as galvanizing the journal is effected under these conditions. Second, because of the brittleness produced under the effects of heat, such as is produced by friction when lubrication is interfered with, and consequent danger of breakage.

Bronzes

Bronze is the term which originally was applied to alloys of copper and tin as distinguished from alloys of copper and zinc.

1. *Copper and Tin:* This, according to our general conception of the word, is a bronze only when the copper content exceeds that of the tin. According to the proportions in which the metals exist, it has widely different properties. In general, the alloy hardens when tin is present up to proportions of 30 per cent or a little over, and when this limit is exceeded, it takes on more and more the nature of tin until pure tin is reached. From a scientific point of view this alloy is one of the most interesting, and has attracted the attention of many investigators, who have spent years of study on it, to learn its various properties and explain its constitution.

The alloys which interest us most, however, are those which are so constituted as to be adapted for bearing purposes. These would be said to contain from 3 to 15 per cent tin, and from 85 to 97 per cent copper. The alloy of tin containing a small percentage of copper is often used as a babbitt metal, but this comes under the class of white metals, which have already been discussed. Bronze containing above

COPPER AND TIN, AND COPPER, TIN, AND LEAD SERIES

	Copper	Tin	Lead	Friction	Temp. above Room	Wear in Grams
1	85.76	14.90	.....	13	50	.2800
2	90.67	9.45	.....	13	51	.1768
3	95.01	4.95	.....	16	52	.0776
4	90.82	4.62	4.82	14	53	.0542
5	85.12	4.64	10.64	18½	56	.0380
6	81.27	5.17	14.14	18½	58	.0327
7	75?	5?	20?	18½	58	.0277
8	68.71	5.24	26.67	18	58	.0204
9	64.34	4.70	31.22	18	44	.0130

15 per cent of tin has been recommended at various times for bearings, owing to its hardness, but very unwisely, for such a bearing demands mechanical perfection and perfect lubrication. It has no plasticity of its own, and as soon as the oil film is interrupted, rapid abrasion and "gripment" take place, with hot boxes as the result. The very erroneous idea is still held by many, that to resist wear and run with the least possible friction, a bearing alloy must be as hard as possible. It is true that hard bodies in contact move with less friction than soft ones; but the alloy which is the least liable to heat and cause trouble is the one which will stand the greatest amount of ill use; by this is meant an alloy which has sufficient plasticity to adapt itself to the irregularities of service without undue wear.

The alloys of copper and tin were used extensively some twenty or twenty-five years ago, and were considered the standard for railroad and machinery bearings. The old alloy, known as "Cannon Bronze," containing 7 parts copper and 1 part tin, is still being specified by some few unprogressive railroad men and machinery builders.

2. *Copper, Tin, and Lead:* This composition is now the recognized standard bearing bronze, its advantage over the bi-compound coming from the introduction of lead. The bronze containing lead is less liable to heat under the same state of lubrication, etc., and the rate of wear is much diminished. For these reasons and the additional fact that

lead is cheaper than tin, it seems desirable to produce a bearing metal with as much lead and as little tin as possible. The metal known as Ex. B. composition (tin 7 per cent, lead 15 per cent, copper 78 per cent) is stated to be the best that can be devised. This alloy contains the smallest quantity of tin that will hold the lead alloyed with the copper. By adding a small percentage of nickel, however, to the extent of one-half to 1 per cent, a larger proportion of lead may be used, and successful bronzes have been made by this process, which contained as much as 30 per cent lead. Such bronzes, containing a large amount of lead, through the addition of nickel, are known in the trade as "Plastic Bronzes" and are a regular commercial article. The table on page 44 gives the results of tests on different compositions of bronzes.

CHAPTER VII

FRICTION OF ROLLER BEARINGS*

During the years 1904-05 a series of tests on roller bearings was conducted at the Case School of Applied Science, Cleveland. A complete report of these tests was published by Professor C. H. Benjamin in the October, 1905, issue of MACHINERY, of which the following is an abstract. An attempt was made in these experiments to compare roller bearings with plain cast iron bearings and with babbitted bearings under similar conditions. Four sizes of bearings were used in the tests, measuring respectively 1 15-16, 2 3-16, 2 7-16 and 2 15-16 inches in diameter. The lengths of journals were four times the diameters.

The bearings were in two parts and were held in a circular yoke by setscrews. This yoke carried two vertical spindles, one above and one below, on which were placed the weights for loading the bearings. The friction was measured either by the deflection of the compound pendulum thus formed, or, as in most of the experiments, by weighing its tendency to deflect by means of an attached cord running over a pulley and carrying a scale pan, as shown in Fig. 19. The shafts or journals used were of ordinary machinery steel, carefully turned to size and having a smooth finish. These shafts were rotated at the speeds shown by means of a belt and pulley. The cast-iron bearings used for comparison were cast whole and bored to size, but the babbitted ones were in halves and were held the same as the roller bearings.

In beginning an experiment, a pointer on the lower end of the pendulum was brought to a zero mark vertically beneath the center of the shaft by adjusting the screws in the yoke. After the shaft began to

* MACHINERY, October, 1905.

revolve the pointer was held to the zero mark by putting weights on the scale pan. The product of the force thus applied to the pendulum by the distance of the point of application from the center of shaft gave the moment of friction, and dividing this by the radius of journal gave the friction at the surface of the journal. Dividing this again by the total weight on the journal gave the coefficient of friction.

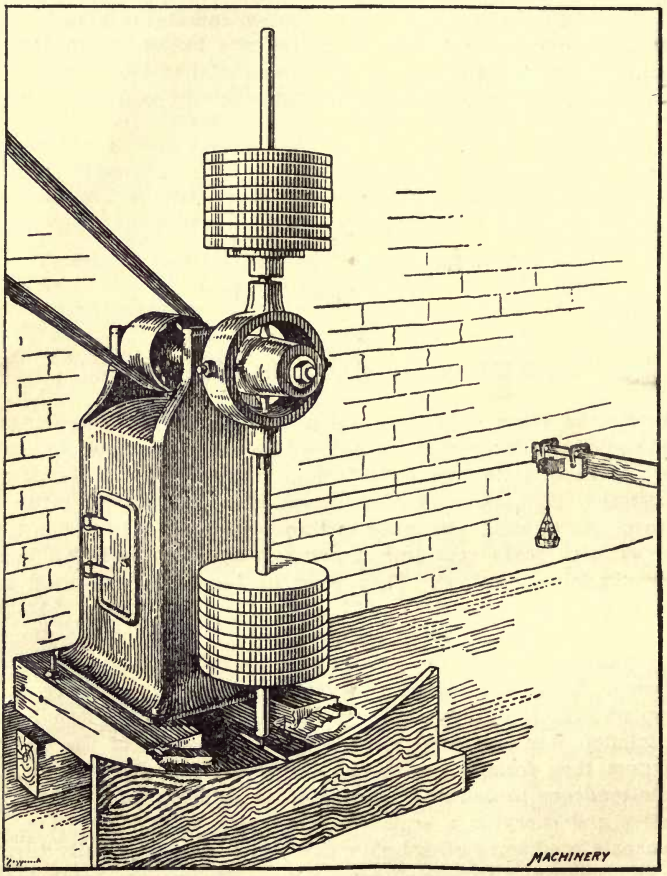


Fig. 19. Apparatus for Testing Roller Bearings

In the first set of experiments Hyatt roller bearings were compared with plain cast iron sleeves at a uniform speed of 480 revolutions per minute, and under loads varying from 64 to 264 pounds. The cast iron bearings were thoroughly and copiously oiled, the lubrication being rather better than would be the case in ordinary practice. Table I shows the results of the test on one bearing in detail, and from this it is seen that the value of f, the coefficient of friction, diminishes as

the load increases, or in other words, the friction did not increase as fast as the load. This holds true as a general rule in all the roller bearings, but not generally in the plain bearings, either cast iron or babbitt.

Table II gives a summary of this series of experiments for the different sizes of journals, the different loads being the same as in Table I. The relatively high values of f in the 2 3/16 and 2 15/16 roller

TABLE I.

Journal 1 15-16 inches in Diameter. 480 Revolutions per minute.

Total Load, pounds.	Friction.		Values of f.	
	Hyatt	Plain.	Hyatt.	Plain.
64.2	2.34	10.24	.036	.160
114.2	3.27	12.10	.029	.106
164.2	4.21	19.10	.026	.116
214.2	4.78	22.35	.022	.104
264.2	5.15	26.10	.019	.099
Average	.....	.....	.026	.117

bearings were due to the snugness of the fit between the journal and the bearing, and show the advisability of as easy a fit as in ordinary bearings.

The same Hyatt bearings were used in the second set of experiments, but were compared with the McKeel solid roller bearings and with plain babbitted bearings freely oiled. The McKeel bearings contained rolls turned from solid steel and guided by spherical ends fitting re-

TABLE II.

Values of Coefficient of Friction f. Speed 480 Revolutions per minute.

Diameter of Journal.	Hyatt Bearing.			Plain Bearing.		
	Max.	Min.	Ave.	Max.	Min.	Ave.
$1\frac{15}{16}$	.036	.019	.026	.160	.099	.117
$2\frac{3}{16}$	.052	.034	.040	.129	.071	.094
$2\frac{7}{16}$	.041	.025	.030	.143	.076	.104
$2\frac{15}{16}$	.053	.049	.051	.138	.091	.104

cesses in cage rings at each end. The cage rings were joined to each other by steel rods parallel to the rolls. The same apparatus was used as in the former tests, but heavier loads were used and the machine was run at a slightly higher speed. Table III shows the detailed results of experiments on one size of journal, and is similar to Table I. The last value given for the Hyatt bearing shows distortion of the roller due to the load and indicates the limit for this size. This is omitted in getting the averages. There is the same indication as in Table I of a decrease of f with increase of load, and this was noticed in all the tests. The results for the babbitt metal are not as uniform as the others on account of the difficulty of balancing.

Under a load of 358.3 pounds the solid roller bearing showed a end thrust of about 20 pounds, which would account for the differenc in friction between that and the Hyatt. Table IV gives a summar of the tests in this series and may be compared with Table II. Th relatively high values for the Hyatt 2 7-16 bearing must be due to slight cramping of the rolls due to too close a fit, as was noted in som of the former experiments. Under a load of 470 pounds, the Hyat

TABLE III.

Journal 1 15-16 inches in Diameter. Speed of 560 Revolutions per minute.

Total Load.	Friction.			Value of *f*.		
	Hyatt.	McKeel.	Babbitt.	Hyatt.	McKeel.	Babbitt.
113.3	3.64	3.77	8.38	.032	.033	.074
162.3	3.77	4.24	8.97	.023	.026	.055
211.3	4.04	5.24	8.97	.019	.025	.042
260.3	4.31	5.37	8.97	.016	.021	.034
309.3	4.57	6.46	10.15	.015	.021	.033
358.3	4.71	6.73	10.75	.013	.019	.030
407.3	4.84	7.27	11.98	.012	.018	.029
456.3	37.70	7.81	20.90	.....	.017	.046
Averages	.....	.....	.....	.0186	.0225	.043

bearings developed an end thrust of 13.5 pounds and the McKeel on of 11 pounds. This end thrust is due to a slight skewing of the roll and would vary, sometimes even reversing in direction.

The babbitt bearing is a slight improvement over the cast-iron sleeve but the difference is quite as apt to be due to improved lubricatior (Notice the variation in the averages for the various sizes in Table IV)

In conclusion it may be said that the friction of the roller bearin

TABLE IV.

Values of Coefficient of Friction f. Speed 560 Revolutions per minute.

Diameter of Journal.	Hyatt Bearing.			McKeel Bearing.			Babbitt Bearing.		
	Max.	Min.	Ave.	Max.	Min	Ave.	Max.	Min.	Ave.
$1\frac{15}{16}$	.032	.012	.018	.033	.017	.022	.074	.029	.043
$2\frac{3}{16}$	.019	.011	.014	.....	.....	.....	.088	.078	.082
$2\frac{7}{16}$	.042	.025	.032	.028	.015	.021	.114	.083	.096
$2\frac{15}{16}$	.029	.022	.025	.039	.019	.027	.125	.089	.107

is shown to be from one-fifth to one-third that of a plain bearing a moderate loads and speeds. It is also noticeable that as the load on roller bearing increases the coefficient of friction decreases. It wa found by the experimenters that a slight change in the pressure du to the adjusting nuts was sufficient to increase the friction consider ably. In the McKeel bearing the rolls bore on a cast-iron sleeve an in the Hyatt on a soft steel one. If roller bearings are properly ad justed and not overloaded, a saving of from 2/3 to 3/4 of the frictior may be reasonably expected.

CONTENTS OF DATA SHEET BOOKS

No. 1. Screw Threads.—United States, Whitworth, Sharp V- and British Association Standard Threads; Briggs Pipe Thread; Oil Well Casing Gages; Fire Hose Connections; Acme Thread; Worm Threads; Metric Threads; Machine, Wood, and Lag Screw Threads; Carriage Bolt Threads, etc.

No. 2. Screws, Bolts and Nuts.—Button-head, Square-head, Headless, Collar-head and Hexagon-head Screws; Standard and Special Nuts; T-nuts, T-bolts and Washers; Thumb Screws and Nuts; A. L. A. M. Standard Screws and Nuts; Machine Screw Heads; Wood Screws; Tap Drills; Lock Nuts; Eye-bolts, etc.

No. 3. Taps and Dies.—Hand, Machine, Tapper and Machine Screw Taps; Taper Die Taps; Sellers Hobs; Screw Machine Taps; Straight and Taper Boiler Taps; Stay-bolt, Washout, and Patch-bolt Taps; Pipe Taps and Hobs; Solid Square Round Adjustable and Spring Screw Threading Dies.

No. 4. Reamers, Sockets, Drills and Milling Cutters.—Hand Reamers; Shell Reamers and Arbors; Reamers; Taper Pins and Reamers; Cowell & Sharpe, Morse and Jarno Taper Sockets and Reamers; Drills; Wire Gages; Milling Cutters; Setting Angles for Milling Teeth in End Mills and Angular Cutters, etc.

No. 5. Spur Gearing.—Diametral and Circular Pitch; Dimensions of Spur Gears; Tables of Pitch Diameters; Odontograph Tables; Rolling Mill Gearing; Strength of Spur Gears; Horsepower Transmitted by Cast-iron and Rawhide Pinions; Design of Spur Gears; Weight of Cast-iron Gears; Epicyclic Gearing.

No. 6. Bevel, Spiral and Worm Gearing.—Rules and Formulas for Bevel Gears; Strength of Bevel Gears; Design of Bevel Gears, Rules and Formulas for Spiral Gearing; Tables Facilitating Calculations; Diagrams for Cutters for Spiral Gears; Rules and Formulas for Worm Gearing, etc.

No. 7. Shafting, Keys and Keyways.—Horsepower of Shafting; Diagrams and Tables for the Strength of Shafting; Forcing, Driving, Shrinking and Running Fits; Woodruff Keys; United States Navy Standard Keys; Gibs, Keys, Milling Keyways; Duplex Keys.

No. 8. Bearings, Couplings, Clutches, Crane Chain and Hooks.—Pillow Blocks; Babbitted Bearings; Ball and Roller Bearings; Clamp Couplings; Plate Couplings; Flange Couplings; Tooth Clutches; Crab Couplings; Cone Clutches; Universal Joints; Crane Chain; Chain Friction; Crane Hooks; Drum Scores.

No. 9. Springs, Slides and Machine Details.—Formulas and Tables for Spring Calculations; Machine Slides; Machine Handles and Levers; Collars; Hand Wheels; Pins and Cotters; Turn-buckles, etc.

No. 10. Motor Drive, Speeds and Feeds, Change Gearing, and Boring Bars.—Power required by Machine Tools; Cutting Speeds and Feeds for Carbon and High-speed Steel; Screw Machine Speeds and Feeds, Heat Treatment of High-speed Steel Tools, Taper Turning; Change Gearing for the Lathe; Boring Bars and Tools, etc.

No. 11. Milling Machine Indexing, Clamping Devices and Planer Jacks.—Tables for Milling Machine Indexing; Change Gears for Milling Spirals; Angles for Setting Indexing Head when Milling Clutches; Jig Clamping Devices; Straps and Clamps, Planer Jacks.

No. 12. Pipe and Pipe Fittings.—Pipe Threads and Gages; Cast-iron Fittings; Brass Fittings; Pipe Flanges; Pipe Bends; Pipe Clamps and Hangers; Dimensions of Pipe for Various Services, etc.

No. 13. Boilers and Chimneys.—Flue Spacing and Bracing for Boilers; Strength of Boiler Joints; Riveting; Boiler Setting; Chimneys.

No. 14. Locomotive and Railway Data.—Locomotive Boilers; Bearing Pressures for Locomotive Journals; Locomotive Classifications, Rail Sections; Frogs, Switches and Cross-overs; Tires; Tractive Force; Inertia of Trains; Brake Levers; Brake Rods, etc.

No. 15. Steam and Gas Engines.—Saturated Steam; Steam Pipe Sizes; Steam Engine Design; Volume of Cylinders; Stuffing Boxes; Setting Corliss Engine Valve Gears; Condenser and Air Pump Data; Horsepower of Gasoline Engines; Automobile Engine Crankshafts, etc.

No. 16. Mathematical Tables.—Squares of Mixed Numbers; Functions of Fractions; Circumference and Diameter of Circles; Tables for Spacing off Holes, Solution of Triangles; Formulas for Solving Regular Polygons; Geometric Progression, etc.

No. 17. Mechanics and Strength of Materials.—Work; Energy; Centrifugal Force; Center of Gravity; Motion; Friction; Pendulum, Falling Bodies; Strength of Materials; Strength of Flat Plates; Ratio of Outside and Inside Radii of Thick Cylinders, etc.

No. 18. Beam Formulas and Structural Design.—Beam Formulas; Section Moduli of Structural Shapes; Beam Charts; Net Areas of Structural Angles; Rivet Spacing; Splices for Channels and I-beams; Stresses in Roof Trusses, etc.

No. 19. Belt, Rope and Chain Drives.—Dimensions of Pulleys; Weights of Pulleys; Horsepower of Belting; Belt Velocity; Angular Belt Drives; Horsepower transmitted by Ropes; Sheaves for Rope Drive; Bending Stresses in Wire Ropes; Sprockets for Link Chains; Formulas and Tables for Various Classes of Driving Chain.

No. 20. Wiring Diagrams, Heating and Ventilation, and Miscellaneous Tables.—Typical Motor Wiring Diagrams; Resistance of Round Copper Wire; Rubber Covered Cables; Current Densities for Various Contacts and Materials; Centrifugal Fan and Blower Capacities; Hot Water Main Capacities; Miscellaneous Tables: Decimal Equivalents, Metric Conversion Tables, Weights and Specific Gravity of Metals, Weights of Fluids, Drafting-room Conventions, etc.

MACHINERY, the monthly mechanical journal, originator of the Reference and Data Sheet Series, is published in three editions—the *Shop Edition*, $1.00 a year; the *Engineering Edition*, $2.00 a year, and the *Foreign Edition*, $3.00 a year.

The Industrial Press, Publishers of MACHINERY,

49-55 Lafayette Street, New York City, U. S. A.