MACHINERY'S
REFERENCE SERIES

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

No. 11

BEARINGS

CONTENTS

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No. 11—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 surface, 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 textbooks as the laws of friction. A comparison of the two will prove interesting.

Frictional Resistance in Lubricated and Unlubricated Bearings.

It is generally stated in the textbooks 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 textbooks, 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 a 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

$$p\,v = C, \tag{4}$$

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 crankpins. 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 bearing 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 2,000 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 alway 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 distribu-

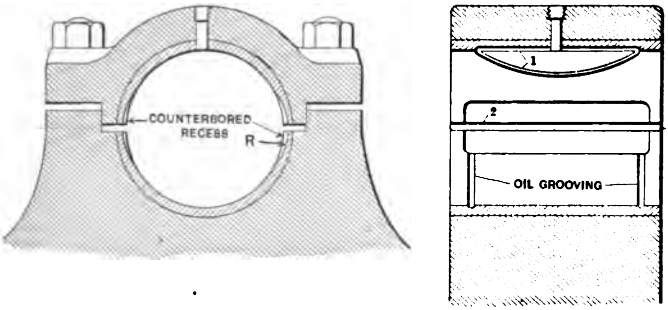


Fig. 1. Section of Outboard Bearing, showing Oil Grooving and Counterbored Recess.

tion 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 oil 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 in Fig. 5.

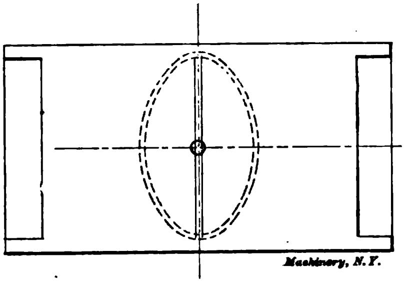


Fig. 2. Development of Cap, showing Oil Grooving and Counterboring.

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

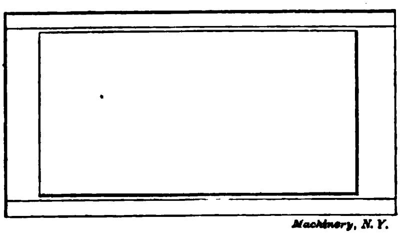


Fig. 3. Development of Lower Half of Outboard Bearing.

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 1,000 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

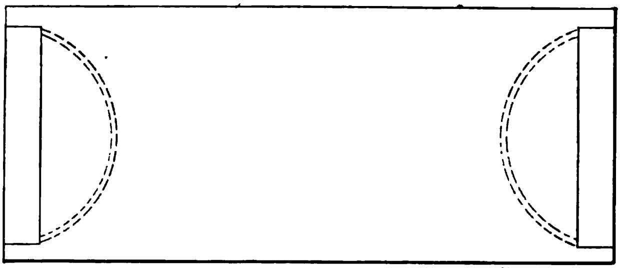


Fig. 4. Face of Cross-head Slipper.

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

$$p = \frac{PK}{DN + K}, \qquad (5)$$

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, 1,200 for crank-pins, and 1,600 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 above 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.

The above formula is in 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 a few cases where a unit pressure sufficient to break down the oil film is allowable.

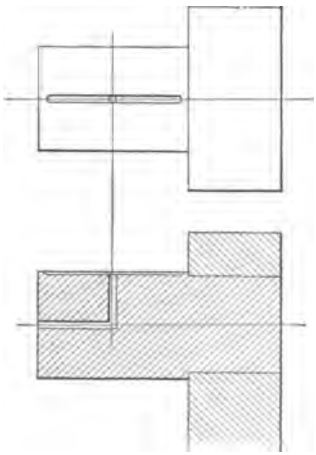


Fig. 5. Internally-oiled Crank-pin, showing Oil Passages and Grooves.

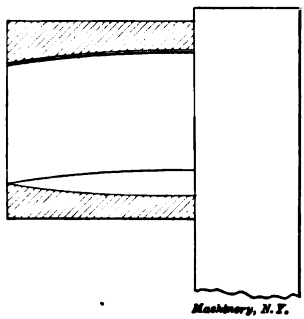


Fig. 6. Section Showing the Bending of a Crank-pin and Consequent Unequal Wear of the Box.

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 eough 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 crankpins. 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:

$$L = \frac{20\,W\sqrt{N}}{PK}, \tag{7}$$

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 $60{,}000 \div 56$, or 1,070 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 $1{,}070 \div 75$, 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 1,000 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 on 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 take $K = 1{,}500$. This gives us for the length of the bearing from formula No. 6:

$$L = \frac{80{,}000}{400 \times 1{,}500}\left(100 + \frac{1{,}500}{15}\right) = 26\tfrac{1}{2} \text{ inches (about).}$$

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 crankpin 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

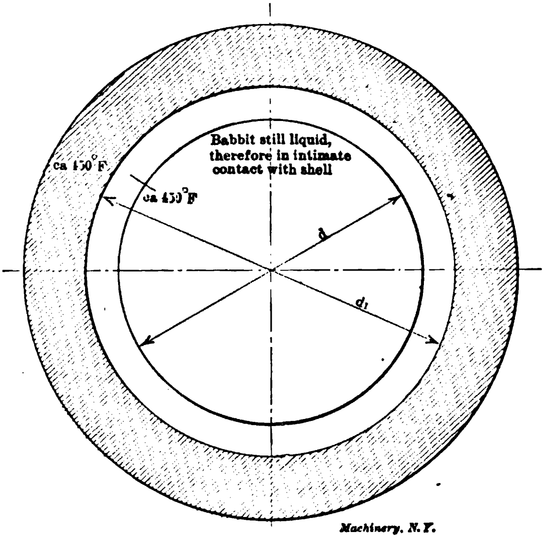


Fig. 7. One-piece Bearing; Babbitt just Poured. Both Shell and Babbitt at the Solidifying Temperature.

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 any one 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 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

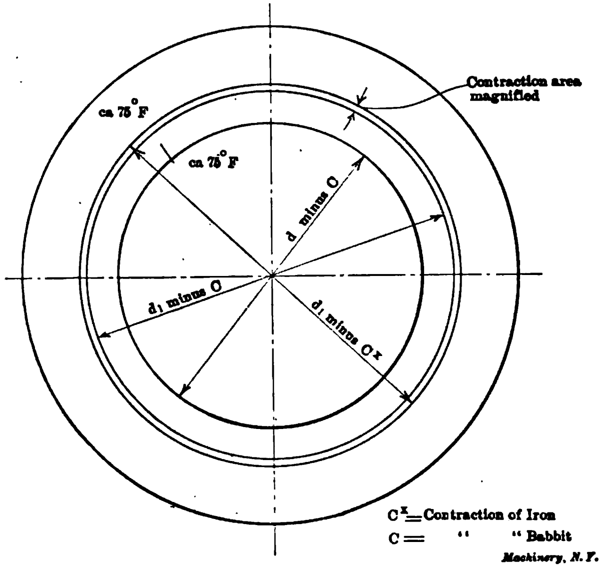


Fig. 8. Same Bearing as shown in Fig. 7 Cooled Down to Normal Temperature.

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 penetrate 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-

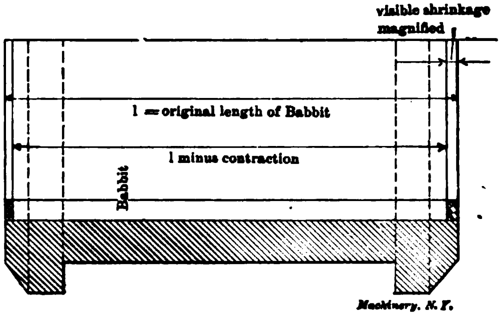


Fig. 9. Babbitt Bearing without Dove-tailed Grooves or other Retaining Device

age or loosening of the babbitt liner from the shell must absolutely take place after the solidifying 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

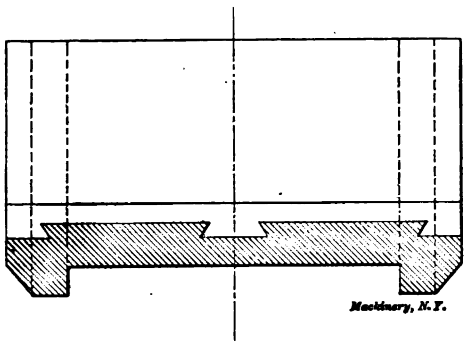


Fig. 10. Same Bearing as shown in Fig. 9, but with Dove-tailed Grooves. Visible Shrinkage Prevented, but Shrinkage Strains Produced.

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

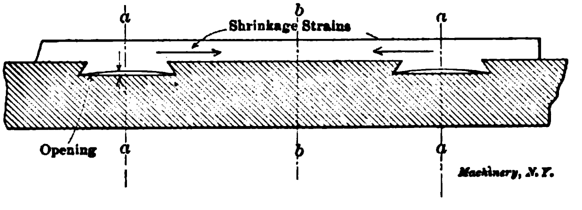


Fig. 11. Illustration of Shrinkage Strains produced by Unequal Distribution of Babbitt Metal.

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 1/200th 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

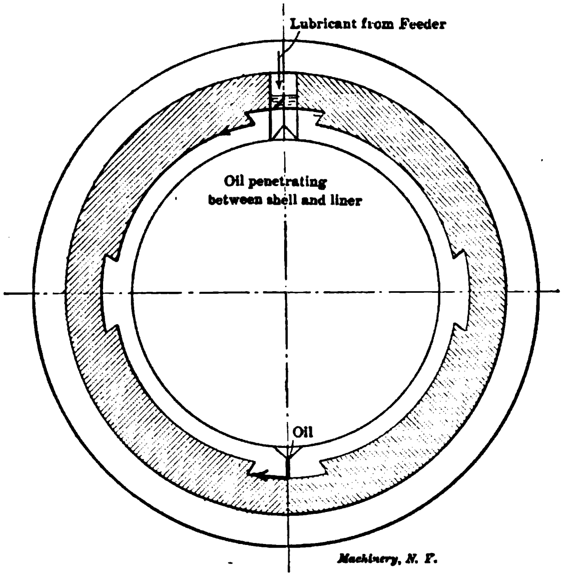


Fig. 12. Penetration of Oil between Shell and Liner.

bearing readily becomes hot, because the babbitt metal liner retains, instead of throwing off, the heat. The regular working pressure also sets up an 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.

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. 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 antifriction 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 running, 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 the above 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 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.

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

2. *Copper, Tin, and Lead:* This composition is now recognized the 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 per cent 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 above gives the results of tests on different compositions of bronzes.

CHAPTER IV.

BALL BEARINGS.

The following discussion on ball bearings consists of an abstract of a paper by Mr. Henry Hess, presented before the American Society of Mechanical Engineers. It is of particular interest and value, because, although the field of usefulness of the ball bearing is as wide as the domain of mechanical engineering, but little information on ball bearings can be found in the engineering hand books or text books. Generally, the subject is dismissed with occasional reference. Sometimes a formula for carrying capacity is given, but this is usually wrong. There is, of course, much matter scattered through the technical press giving isolated experiences with a few bearings that happen to come within some one's observation. But, undoubtedly, the insufficient information on the elements of ball bearings is responsible for the directly contradictory statements to be found, and the generally accepted opinion that ball bearings are suitable only for comparatively light loads. This was the situation found by Prof. Stribeck, a German investigator, who has been experimenting with ball bearings for the German Small Arms and Ammunition Factories, of Berlin. The present chapter is a résumé of Prof. Stribeck's report, together with a number of notes based on Mr. Hess's own experience.

The Wear of Ball Bearings.

Sliding bearings wear out by abrasion of the carrying surfaces, but ball bearings do not give out on account of wear. In fact, they do not wear. They may be ground out by admitting grit, but that is an abnormal condition for ball bearings, the same as it would be for sliding bearings. The only normal cause for the giving out of ball bearings is the stress on the material, when this stress exceeds certain limits. Lightly loaded bearings can be so designed as to eliminate this cause altogether, and to insure practical indestructibility. In heavily loaded bearings this condition may not be possible to realize within practical dimensions, but the proportions may be so chosen that the stress will not cause a break-down within the lifetime of any mechanism to which the ball bearing is applied. An important principle in the design of ball bearings is that the balls may be subjected to loads, increasing as the shape of the supporting surface more nearly becomes complemental to that of the ball. A ball running between races having a flat or straight line cross section will not support as great a load as if the section were curved, or in other words, if the balls were running in a groove. The groove, of course, must never have a curvature equaling that of the ball, as that would substitute sliding for rolling contact.

Formulas for Loads on Ball Bearings.

The frictional resistance of a ball bearing is lower the less the number of balls. Bearings should be designed to have between 10 and 20 balls. For this number of balls the equation

$$P_o = \frac{5}{z} P_b \qquad (1)$$

should be used. In this equation

P_b = total load on the bearing consisting of one row of balls, in kilograms.

P_o = greatest load on one ball, in kilograms.

z = number of balls.

The load carrying capacity of one ball is $P_o = k\,d^2$, in which

d = the ball diameter, in eighths of an inch,

k = a constant depending upon the material and shape of the ball supporting surface. This constant should vary between 3 and 5 for common materials for ball bearings, and between 5 and 7.5 for improved steel alloys.

As balls are usually made to English measurements, one-eighth inch has been selected as unit for the ball diameters.

From equation (1) we have

$$P_b = P_o \frac{z}{5}, \text{ and substituting } k\,d^2 \text{ for } P_o, \text{ we have}$$

$$P_b = k\,d^2 \frac{z}{5}.$$

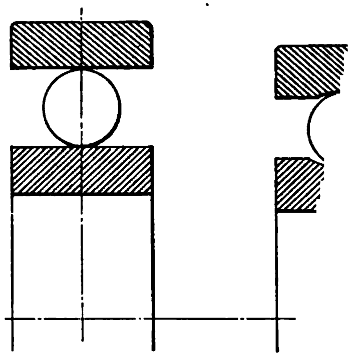
Speed rotation, in as far as it is uniform, does not affect the carrying capacity of a ball bearing. This applies, however, only to radial bearings, but not to thrust bearings of the collar type. In these, the carrying capacity decreases with the increase of speed. Variations in load reduce the carrying capacity, the effect increasing with the amount of load change and the rapidity of the change. Uniformity of ball diameter is very essential as the calculated carrying capacity can be realized only if each of the balls sustains its share of the load. High finish on both ball and ball sustaining surfaces is also essential. The presence of grinding scratches will very materially cut down the load carrying capacity.

Frictional Resistance of Ball Bearings.

The frictional resistance of ball bearings has, by actual measurement, been found to vary from 0.0011 to 0.0095. These are the coefficients of friction referred to the shaft diameter, thus permitting direct comparison with coefficients of sliding friction. Ball bearings having a coefficient of friction materially above 0.0015 under the greatest allowable load should not be recommended, because they are too short-lived. The high resistance indicates the presence of too large an element of sliding friction. The coefficient of 0.0015 for a good ball bearing under its greatest load, independent of the speed within limits, will, however, rise to approximately 0.0030 under a reduction of the load to about 1/10 of the maximum.

The Requirements of a Good Ball for Ball Be

The requirements for a good ball are, in the first shape and size. The permissible limit of error wil character of the material. It is evident that where than the other balls in the bearing, it must be capa tion sufficiently large so as to permit the others to load, and for that reason, the smaller the deformati rate to size must the ball be. In the second plac surface finish is essential. What is usually cons finish is in a ball bearing totally inadequate. Gri marks must not only not be recognizable by the r tected with an ordinary pocket lens, the balls s This, at least, is true of balls for bearings expec and to carry heavy loads under high speed. Tl balls is that the material out of which they ar



Figs. 13 and 14. Examples of Simı

limit as high as can be had. The unifo the mass of the ball is also very esse fact, is one of the most important factor because some balls in a lot are better t be based on the poorer ones. Such better balls carrying more than their r heavy a stress, and possibly breakage provided it is uniform, is better than error in regard to the truth and siz inch.

Radial Be

Other things being equal, it is surfaces at right angles to the load bearing such as shown in Fig. 13, be had from the modification sh races have curved cross section in races have the advantage of great been mentioned. Cutting a local as in Fig. 15, for the purpose races, is common practice, but it is not good practice. So long as the loads are low enough, a filling opening may be of no account, but at high speeds and loads this groove is objectionable, since then the catching of the balls at the junction of the filling groove results in damage to the balls, and through these to the race surface.

Thrust Bearings.

In thrust bearings, of the type shown in Fig. 16, the requirement that the sustaining surface should be at right angles to the direction

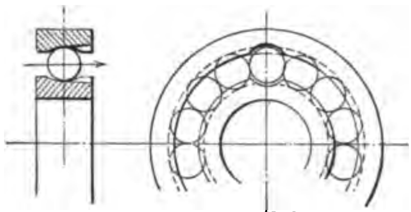


Fig. 15. Objectionable Filling Groove in Ball Bearing.

of the load is provided for. These bearings are frequently made with the surfaces *A* and *B* parallel. Providing that these surfaces are made truly parallel, that design is good, but in practice it is seldom possible to get these surfaces truly parallel, because even if such parallelism were possible of attainment, it could not be maintained under the deflections due to the load. It must be remembered that initial errors or deflections of a thousandth of an inch will cause the balls on one side to carry the entire load. For a given case this would demand bearings of larger size than otherwise necessary. By seating the lower

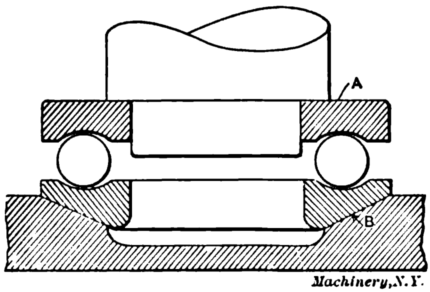
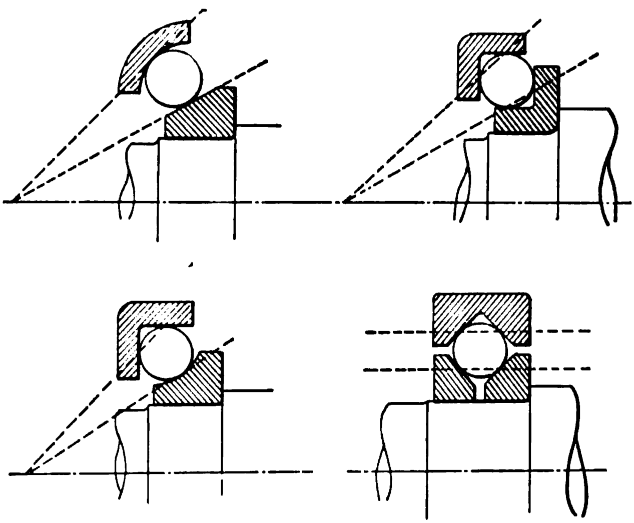


Fig. 16. Collar Thrust Bearing.

collar on a spherical surface as shown at *B*, Fig. 16, this collar can adjust itself in such a way as to distribute the load over the entire number of balls. The speed at which these bearings are run enters decidedly into the carrying capacity of this type of bearing. The utility of these bearings is greatly reduced when speeds exceed 1,500 revolutions per minute.

Angular Load Bearings.

The shapes and modifications of angular load bearings are innumerable. Figs. 17, 18, 19, and 20 may be taken as typical instances of these bearings, representing 2, 3, and 4-point contacts. In order to secure rolling contact, the contact points of the balls on the races should be points on a cone of rotation whose apex lies in the center line of the shaft, or they may be points on the surface of an imaginary cylindrical roller that would be parallel to the shaft. The defect in all these forms of bearings is their adjustable feature. This places them absolutely at the mercy of every one capable of handling a wrench. A bearing properly proportioned with reference to a certain load may be enormously overloaded by a little extra effort applied to the wrench, or, on the other hand, the bearing may be adjusted with too little pressure, so that the balls will rattle, and the results consequently be unsatisfactory. The prevalent idea that angular ball bearings can be adjusted to compensate for wear is erroneous. The wear will form a



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Figs. 17, 18, 19, and 20. Examples of Angular Bearings.

groove on the loaded side of the race, deepest at the point of maximum load, and adjusting the cone endwise will only cause the balls to be more tightly pinched between the sound portions of the races, which will most likely cause overload. The rough surface of the groove previously worn will attack the balls, and in due time the entire race and bearing will be destroyed.

Theoretically, it would seem that a radial bearing would be incapable of carrying thrust load, owing to the wedging of the balls between the races. In Fig. 21, however, is shown the condition of a ball bearing where the ball does not entirely fill the space between the races, if the bearing is not under load, and which, when under load, will assume the position shown. The ball does not come in contact with the race grooves where these are deepest, but so that the tangents to the race curvature at the contact points form angles with the line of thrust.

A calculation of the amount of the wedging action in Fig. 21, with the radial freedom of ball permissible in these bearings, indicates an inadvisably large amount of wedging. Actual running tests, however, as well as a large fund of accumulated experience, have already proved that these bearings will carry much more thrust load than the calculation of the theoretical wedge angle indicates as possible. It is probable that the deformation which occurs at the point of ball contact, which results in small surface *areas* of contact, instead of mere *points*, has a mean tangent to the compression surface of greater inclination, and that the wedge is therefore more blunt. It has been experimentally determined that the thrust carrying capacity of the uninterrupted type of annular bearings is to the radial capacity as 0.1 is to 1, and may be as great as 0.25 to 1, the variation depending upon the ball diameter, race curvature, and number of balls. It has been experimentally found that speed has but a slight influence on the thrust carrying capacity of this class of ball bearings, and for speeds above

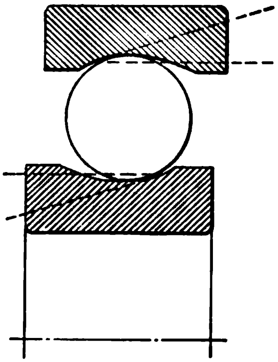

Fig. 21. Radial Bearing used as Thrust Bearing.

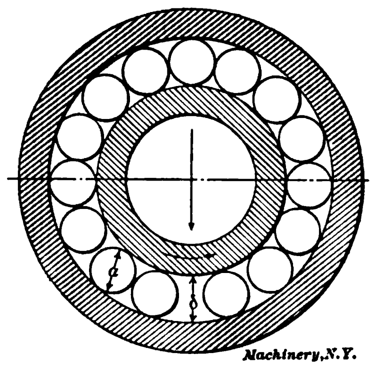

Fig. 22. Diagram used in Analyzing Condition of Sliding Friction.

1,500 revolutions per minute, these radial bearings are more efficient thrust carriers than the collar type.

The Supposed Sliding Friction in Ball Bearings.

Many designers have supposed that in ball bearings adjacent balls press against one another with considerable force. With the inner race in Fig. 23 running as indicated, the balls will also rotate as shown. The surfaces of the balls run in opposite directions, and therefore rub against one another. This is assumed to be a serious defect by those who reason that these surfaces are in contact under pressure. The same general cure in innumerable forms, as shown in Fig. 24, has been proposed time and again. This cure consists in the provision of smaller balls interposed between the larger ones, so that all the contacting surfaces roll in the same direction relative to one another. This remedy is, however, fallacious, in that it brings about the very condition it seeks to avoid. If two large balls, Fig. 25, compress a smaller one between them, and the three have their centers connected by a straight line, they will retain their relative positions, but if the interposed ball has its center to one side, as in Fig. 26, then this ball will be forced outward. The resort to a cage for retaining the interposed roller or ball results in the latter being pressed against the sides of the cage, and keeps the ball in forcible sliding contact, the very thing that it was intended to avoid. In another design, Fig. 27, the interposed member is brought into contact with the race, and while the various balls have a rolling contact in relation to one another, the interposed member has a wrong direction with reference to the race against which it is forced, and thus a sliding contact is produced. All these designs are based on a failure to recognize that axiom in mechanics according to which a force whose direction is normal to the supporting surface has no component in any other direction.

Analyze the conditions in a ball bearing, and, referring to Fig. 22, suppose that the shaft is loading the inner race, and that the latter is fallaciously assumed to act as a wedge, forcing the balls at the bottom apart and consequently producing pressure between the balls at the

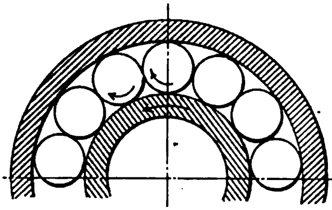

Fig. 23. Diagram showing Direction of Rotation of Balls, indicating Sliding Friction.

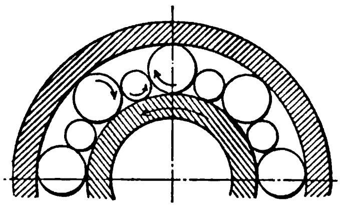

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Fig. 24. Fallacious Means for avoiding Sliding Friction between Balls.

top. In that case the space δ must be smaller than the ball diameter a. The rotation of the inner race, however, will carry the balls around the bearing, and the diameter a is therefore forced through the smaller space δ. To do this the ball must lift the inner race. The force to do this is imparted by the load and is equal to the rolling friction, and can, therefore, amount to but a fraction of that load. We would then have the absurd condition of this smaller force overcoming the larger original force. Were we to assume that the inner race is not raised by a ball in passing, but that the ball is compressed sufficiently to get through, it would mean the absurdity that the small force represented by the rolling friction would be sufficient to deform the ball.

Correct Ball Bearing Mounting.

The following requirements are placed on correctly mounted ball bearings:

a. The proper size must be selected for the load and conditions in question. Rated capacities are usually for steady loads and speeds, but variations from these conditions demand a cutting down of the listed capacity.

b. Bearings must be lubricated. The often repeated statement that ball bearings can run without a lubricant is not correct.

c. Bearings must be kept free from grit, moisture, and acid. No lubricants developing free acids should be permitted.

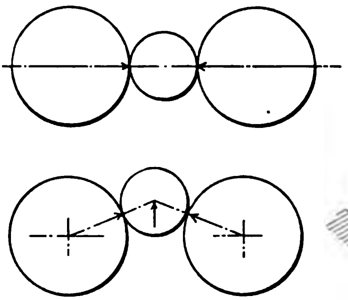
d. The inner race of the bearing should be firmly secured to the shaft. This can be done by a light driving fit, reinforced by binding the race between a substantial shoulder and a nut.

e. The outer race must have a sliding fit in its seat.

These conditions should under all circumstances be adhered to, and a failure to do so will result in very unsatisfactory bearings. The two following conditions are frequently disregarded, and while the disregard of these conditions is not so serious as of those mentioned before, it is safe engineering to follow them, and a disregard of them is a standing invitation to trouble.

f. Thrusts should always be taken up whether in the same or opposite directions by the same bearing.

g. Bearings should never be dismembered, or at least never more than one at a time. That will avoid all danger of mixing the balls



Figs. 25 and 26. Analysis of Sliding Friction.

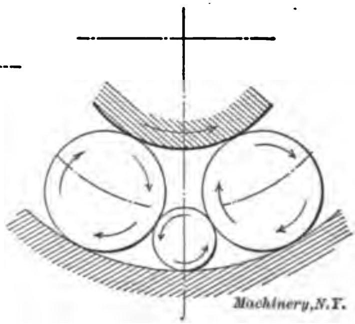


Fig. 27. Fallacious Means for Avoiding Alleged Sliding Friction.

from different bearings; such balls from different bearings are very apt to vary more than is permissible for the individual bearing.

Illustrations of Correct Mounting.

The development of ball bearings being so recent is probably the cause that so little information as to correct mountings for various conditions is available. The experience of those best informed has been that faulty mountings are very general, and for this reason illustrations have been given of correct mountings, going more into details than would be considered necessary for a more familiar mechanical subject.

Fig. 28 shows a bearing in which the inner race has a light driving fit on the shaft, and is securely clamped between a shoulder on the shaft and a nut. The shoulder on the shaft should be high enough to get a firm grip on the surface of the side of the race. It is good practice to make this shoulder about one-half as high as the race thickness, perhaps a little less for large bearings and a little more for small bearings. The outer race has a tight sliding fit in the housing, so that the bearing as a whole may be able to respond to relative shifting of the shaft and housing without being subjected to end thrust through the balls.

Fig. 29 shows a radially loaded shaft held against endwise motion in either direction. This bearing is capable of carrying thrust load in either direction, but never more than one bearing on the same shaft should be held in this manner. This bearing differs from the preceding mounting only therein that it has the outer race secured between shoulders in the housing. This arrangement and the preceding one are usually found combined on the same shaft, which is then held endwise at one point only, so that temperature changes, or deflections of the shaft, can cause no cramping.

Fig. 30 shows separate radial bearings for radial and thrust loads. This type of bearing is used when it is desirable to take thrust load on bearings of the radial type, although the space available does not permit of a single radial bearing of sufficient diameter to take both

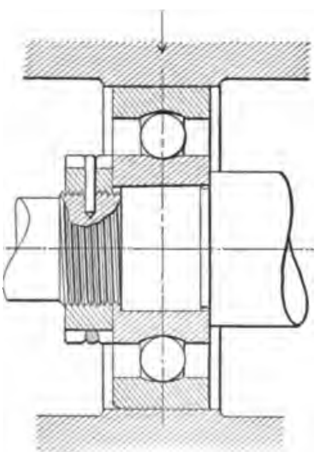


Fig. 28. Free Mounting for Radially Loaded Bearing.

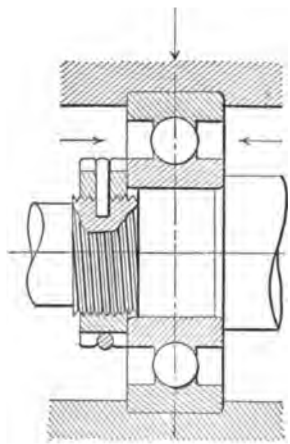


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Fig. 29. Radially Loaded Bearing held against Longitudinal Motion.

loads. One bearing is then mounted entirely free circumferentially so as to take the radial load, while the other bearing is mounted between shoulders, and takes the thrust load.

Figs. 31 and 32 show thrust loads on a collar bearing in one direction only. Here the stationary race is provided with a spherical seat so that it will distribute the load over the complete circle of balls. In order to permit compensating shifting, the fixed seat must be radially free of the shaft and of the housing. The shoulder on the shaft should be large enough so as not to permit any bending strains on the rotating race. When it is inconvenient to provide a large enough shoulder on the shaft, a washer can be inserted between the shoulder and the race, as shown in Fig. 32. In Fig. 33 is simply shown a modification of the bearings in Figs. 31 and 32. This bearing takes thrust loads in two directions on two collar bearings. Fig. 34 shows an arrangement by which thrust load in two directions may be taken up by a single collar bearing. This arrangement is one which economizes space, cost of bearings, and number of parts. Fig. 35 shows an arrangement where a radial bearing is used for taking the radial thrust, and a collar bearing is used for taking the end thrust. Fig. 36 shows an arrangement for taking up radial load as well as thrust load in two directions, these loads being carried on one radial bearing and two collar bearings. In this design attention may be called to the distance

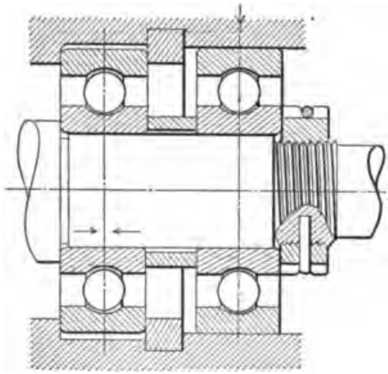
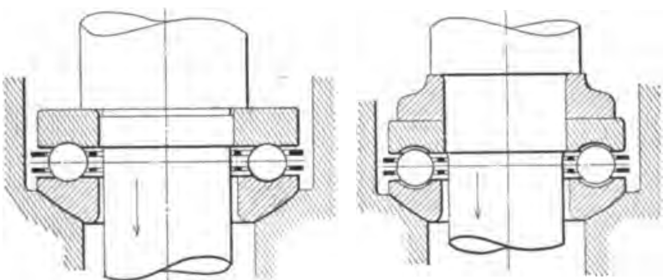


Fig. 30. Separate Radial Bearings for Radial and Thrust Loads.

piece inserted for binding the inner race of the radial bearing against the shoulder of the shaft.

It is occasionally inconvenient to arrange the bearing so that the parts of the inner races can be clamped to the shaft, or it may be desirable to have a shaft sliding through the bearing. In such cases a sleeve may be introduced on which the inner race of the bearing is firmly clamped endwise, the shaft simply resting in this sleeve. This



Figs. 31 and 32. Thrust Load in One Direction on Collar Bearings.

gives a long bearing to the shaft, which would not be possible if the shaft was directly mounted in the ball race, because the peening effect of the vibrating loads would, even if the race itself was prolonged, be concentrated on a narrow zone of the shaft. A bearing of this kind is shown in Fig. 37.

In Fig. 38 is shown a bearing which is intended for shafting which may not be fully to standard size. The inside of the ball race is tapered, and a split bushing, tapered on the outside as shown, will, when tightened, bind the shaft as well as the race to it, and will compensate for all variations in size. The nut should not be used to draw the bushing in, but should merely act as a lock to hold it in place after it has been driven home with a soft hammer.

Ball bearings should always be enclosed so that lubricant will not

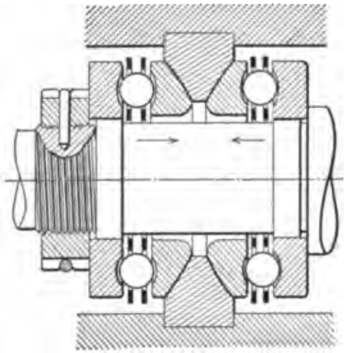


Fig. 33. Thrust Load in Two Directions on Two Collar Bearings.

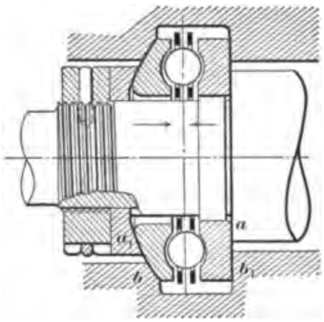


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Fig. 34. Thrust Load in Two Directions on One Collar Bearing.

be lost by leakage, and so that foreign matter will be excluded. Fig. 39 shows an efficient way of enclosing a ball bearing without using any packing. At the end where the shaft passes out of the enclosure, a flange should be bored out about 0.020 inch larger in diameter than the shaft. This flange should be separated into two lips by an angular

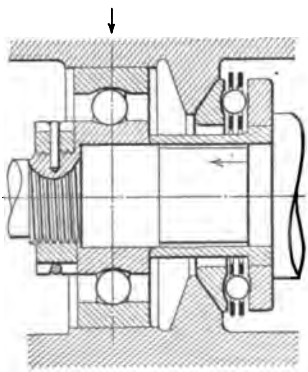

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Fig. 35. Radial Load, and Thrust Load in One Direction.

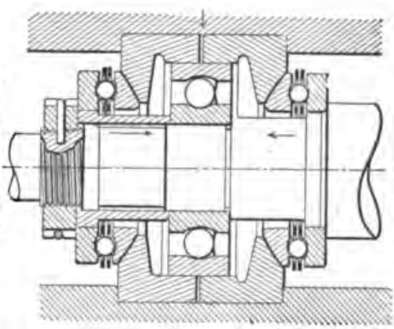

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Fig. 36. Radial Load, and Thrust Load in Two Directions.

groove, either cored or bored, as shown at *A*. These lips should not be less than ⅛ inch wide and should have sharp edges. The groove should be provided with a hole or narrow slot *B* at its lowest point to communicate with the bearing oil space. The groove itself should have a width of not less than 3/16 inch, and a depth of about 5/16 to ⅜ inch, and should not be filled with packing material. Fig. 40 shows an arrangement of a similar kind, excepting that here is introduced a second groove and a third lip. This arrangement is employed where water may be occasionally encountered, and will prevent its entrance. What little may find its way past the outer lip into the outer groove is soon drained out again through the holes provided.

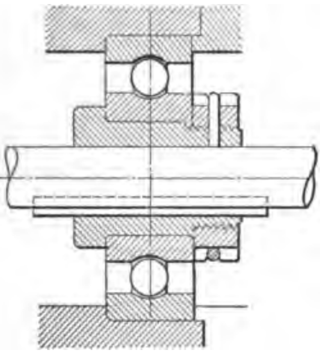

Fig. 37. Shaft Free in Longitudinal Direction in Inner Race.

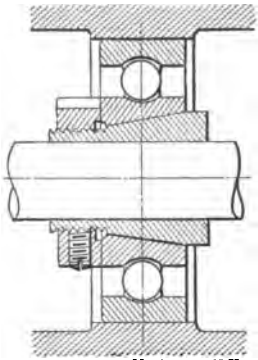

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Fig. 38. Adapter Bearing for Shafting varying from Standard Size.

Where much grit is encountered, as in grinding machinery, a packing may be necessary, and filling the outer groove with a fairly consistent grease will provide such a packing without introducing friction. A bearing of this kind is shown in Fig. 41. A grease cup of the spring loaded piston type will automatically maintain the integrity of

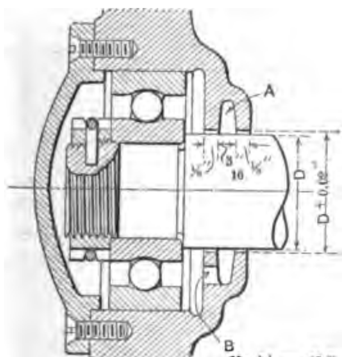

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Fig. 39. Example of Enclosed Bearing.

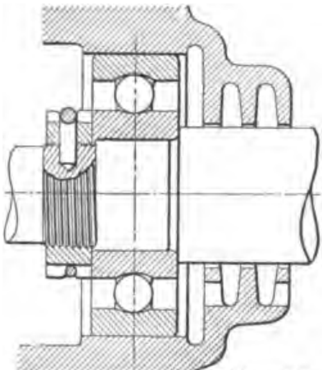

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Fig. 40. Bearing Enclosure with Double Groove.

this packing. In some cases felt ring packings may be used, but these ought to be soaked in good soft paraffine, and a spring wire ring should be placed around the felt washer so as to force the outer edges of the washer outward, which will cause the felt to come into more intimate contact on the sides. Felt washers may also be applied as shown in Fig. 43. Here the washer is tapered on one or on both sides, and the sealing of the enclosure is made entirely against the sides of the surfaces against which the felt washer bears. The felt washer is pressed inwards by means of springs on the outside. The two modifications

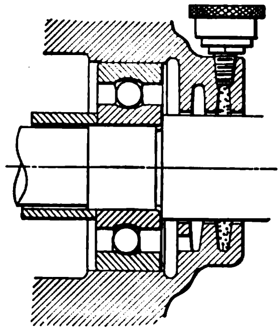

Fig. 41. Enclosed Bearing with Grease Packing.

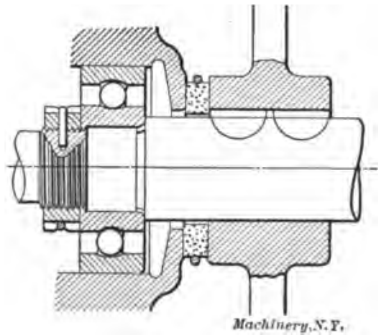

Fig. 42. Felt Ring Packing.

shown in Figs. 42 and 43 are intended to be inserted between the faces of a stationary boss and a rotating hub. The modification in

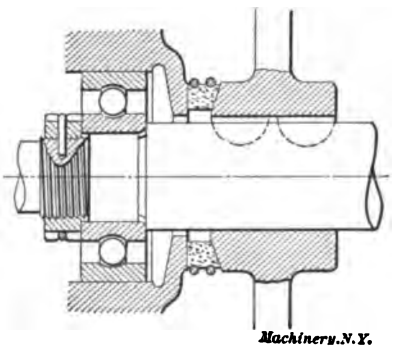

Fig. 43. Angular Felt Ring Packing.

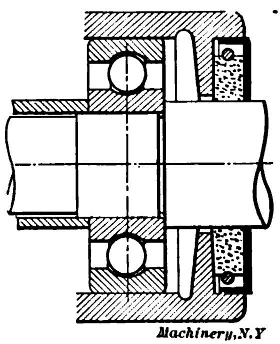

Fig. 44. Modification of Felt Ring Packing in Fig. 42.

Fig. 44, however, is enclosed entirely within one or the other. A felt ring is set into a counterbore and held in place by a light metal cap sprung into position.

CHAPTER V.

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

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	2.84	10.24	.036	.160
114.2	3.27	12.10	.029	.106
164.2	4.21	19.10	.026	.116
214.2	4.78	22.85	.022	.104
264.2	5.15	26.10	.019	.099
Average	....	....	.026	.117

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.

In the first set of experiments Hyatt roller bearings were compared with plain cast iron sleeves at a uniform speed of 480 revolutions per

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

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.

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 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.

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	6.78	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

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. Table III shows the detailed results of experiments on one size of journal, and is similar to Table I.

Under a load of 358.3 pounds the solid roller bearing showed an end thrust of about 20 pounds, which would account for the difference in friction between that and the Hyatt. Table IV gives a summary

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	.088	.096
$2\frac{15}{16}$	.029	.022	.025	.039	.019	.027	.125	.089	.107

of the tests in this series and may be compared with Table II. The relatively high values for the Hyatt 2 7-16 bearing must be due to a slight cramping of the rolls due to too close a fit, as was noted in some of the former experiments. Under a load of 470 pounds, the Hyatt bearings developed an end thrust of 13.5 pounds and the McKeel one of 11 pounds. This end thrust is due to a slight skewing of the rolls and would vary, sometimes even reversing in direction.

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