# JOURNAL TYPE BEARINGS FOR ELECTRIC MOTORS THEIR DESIGN, CONSTRUCTION, AND LUBRICATION

Thesis for the Degree of M. E.

Fred M. Hill

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THESIS

Bearings (Machinery)

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## I. INTRODUCTION

The aim of the author in preparing this thesis is to compile information gained from research and from personal experience that will assist the executive, engineer, designer, machinist, and the maintenance man in solving problems of both a technical and practical nature. A brief historical outline is included to broaden one's concept of the subject and to provide an interesting background. A considerable number of tabulations, diagrams, drawings, and photographs are used to illustrate various principles and facts and to guide one in the solution of everyday problems.

A subject of this nature cannot be covered by a few simple statements. To cover the subject in its entirety would be an endless task. There is a wealth of theoretical information available in technical publications, particularly of recent years. Engineering text books give brief and fundamental principles. Many handbooks give valuable tables and empirical formulas principally for the designer. There appears to be a decided need of correlation between the two, especially in the form of a single treatise. It is hoped that this work will be of value in fulfilling this need.

Many references will be given to form a basic network of the subject. The author's personal experiences and observations are given with the thought that they may prove of some worth in formulating more accurate conceptions. Acknowledgment is hereby given the Consumers Power Company of Michigan for the opportunities presented the author. During the period of four and one-half years, over 16,000 industrial motors were changed over from 30 to 60 cycle service. Many problems involving bearings and their lubrication naturally occurred, from which the author was privileged to gain first-hand experience. This experience has been a valuable aid in formulating the text of this work.

## II. HISTORICAL

Ancient machinery consisted of simple pulleys, windlasses, carts, and sleds roughly made from wood, and if lubricated at all, were packed with animal fat or tallow.[1] The chariot found in the Egyptian tomb of Yuaa and Thuiu about 1400 B.C. still had some of the original lubricant on the axle. Lucas, the Official Chemist of the Cairo museum, analyzed the lubricant and indicated it to be either a mutton or beef tallow, either one of which would have been suitable in such a warm climate.

On the inner wall of the Egyptian tomb Tehuti-Hetep appears a decoration inscribed about the time of Joseph, 1650 B.C., showing the method of lubrication employed in moving the great stone statues.[2] This appears to be the earliest historical record of the use of a lubricant. The Egyptians had available nearly every animal oil and fat, as well as at least thirteen vegetable oils of our time which were used for food and for the arts.

Hero of Alexander wrote about 150 B.C., illustrated and described a fire pump with bronze cylinders bored in a lathe and equipped with pistons and rods which were "rubbed with oil" according to a later writer. The oil was undoubtedly animal or vegetable. The manufacture of petroleum was indicated by Herodotus, 484-424 B.C., but its general use does not appear until Dr. James Young in 1847 found petroleum in Derbyshire, England, from which he obtained a heavy lubricating oil by destructive distillation. In 1763, James Watt became interested in the steam engine and among other things perfected a device for feeding melted tallow to the piston and cylinder walls. For over half a century tallows were used for this service.

About the year 1800, or shortly before, there was a general reawakening of mechanics and the use of machinery. In machine shops lard oil was the universal lubricant. Sperm oil held supremacy for lubricating light high speed mechanisms. For heavy machinery, beef and mutton tallows were found most suitable. At times chunks of pickled side pork with the hide and hair left on were found to be satisfactory. The author finds that there are still men living who used strips of pork rind with flax yarn in packing piston rods of water pumps, as late as thirty years ago.

In 1870, Gramme was given credit of producing the first commercial electrical machine for continuous operation. By the year 1885 electrical machines were being rapidly developed and the use of mineral oil was quite general. In 1886, Osborne Reynolds established the fact mathematically that a well-lubricated journal rotating at a fair speed becomes automatically separated from its bearing by a film of oil under pressure and that the functional resistance is then due entirely to the viscosity of the oil.[3]

## III. THE ELECTRIC MOTOR AND ITS BEARINGS

The electric motor today, some forty years after its introduction to industry, occupies a preeminent place among power drives. It has virtually supplanted all other devices except the gasoline and the steam engine in remote instances. This is due to several reasons, chief of which are its versatility, simplicity, dependability, and low maintenance cost. It also possesses good speed regulation characteristics, occupies little space, and its energy supply is comparatively simple. However simple and reliable it may be, it is not without its faults. Like all machinery it is subject to wear and the consequent depreciation. The quality of materials and workmanship is often quite variable, as well as the matter of designs. In examining the photographs in the appendix one will note in particular the wide variance in oil grooving. The proportions of length to width and other details of construction show a decided lack of standardization.

In the design of motor bearings there are certain fundamental requirements that cannot be fulfilled in the ideal manner. A number of compromises must be made. There is a decided gap between optimum conditions and actual running conditions. Laws governing optimum conditions do, however, serve as a guide or reference standard and aid ones conception of the hydrodynamic action taking place. Empirical formulas are very useful and for all practical purposes are sufficient. An understanding of the theory of lubrication is oftentimes a very valuable aid in solving bearing and lubrication problems.

## IV. THEORETICAL ASPECTS OF DESIGN

There are several most excellent articles covering the mathematical analysis of the hydrodynamic theory of lubrication. They can easily be found in our public libraries and will, therefore, be omitted here because of their great length. References to them, however, will be made and a rather complete list will be found in the appendix. A very complete mathematical analysis will be found in A.S.M.E. Transactions developed by Cardullo.[4] By the use of differential and integral equations he arrives at certain formulas of fundamental significance. His work is recommended where a very thorough analytical concept is sought. The first real explanation of the fact that oil is able to insinuate itself between the rubbing surfaces of a journal and its bearing while carrying a heavy load, is accredited to Reynolds.[5] However, Reynolds assumed the bearing to be of infinite width. In actual practice there is considerable side leakage. Michell developed the theory and solved the problem for finite width.

Kingsbury has written several articles of a highly technical nature covering both the bearing of infinite width and of finite width.[6, 7] In order to simplify the integration of mathematical formulas, Kingsbury resorted to an electrical integration method. The most recent development along this line comes from Needs, of the Kingsbury Machine Works, who also used the electrical integration method.[8] Needs lays particular emphasis on the effect of side leakage. His article is particularly valuable for the design of large turbine and generator bearings where mechanical construction and operating conditions more nearly approach the optimum. For graphical studies of journal bearings, Howarth has made valuable contributions.[9] Karelitz also shows bearing performances in graphical forms.[10] One of the most outstanding articles of a more practical nature was made by Hersey.[11] Hersey's work gives much experimental data from which empirical formulas are derived.

One of the most ingenious methods of obtaining actual pressure distributions was devised by McKee-McKee of the Bureau of Standards.[12] Their charts will be very useful in pointing out certain features in the construction and operation of electric motor bearings. Two sets have been photographed and are shown as Figs. 1 and 2. Reference will be made to them subsequently.

## V. PERFECT AND IMPERFECT CONDITIONS OF LUBRICATION

The foregoing references are all based on the assumption that the rubbing surfaces are completely separated by a film of oil. Such is not always the case, however, in practice. Because of misalignment, improper oil grooves, adverse loadings, inadequate supply of lubricant, and other conditions, there may be a partial metal-to-metal rubbing. Of course during starting and stopping there will always be metallic contact. The author has made tests on motors in operation using an electric circuit with bell alarm and has found the surfaces to be entirely separated by a film of oil even under heavy belt pulls. The tests were made on motors with bearings in good condition. The author has concluded that the bearing and journal of the modern electric motor, when constructed in accordance with scientific principles, does give results agreeing closely with conceptions of perfect lubrication.

Investigations have been made by McKee-McKee to determine frictional characteristics with loads sufficient to cause rupture.[14] The loadings used were many times greater than those imposed on an electric motor, being on the order of several thousand pounds per sq.in. of projected area. While it is possible to carry tremendous loads on a journal bearing under favorable conditions, it is advisable to use values somewhat under 100 lb. for electric motors.

## VI. PRACTICAL CONSIDERATIONS IN BEARING DESIGN

The oil is supplied to the bearing by means of the conventional oil ring in motors ranging from about one horse power upward. Above about 25 horse power, two rings are generally employed. Built-in type motors, those whose bearings are in reality a part of the driven machine, may be lubricated in a number of different ways. Those driving large air and ammonia compressors will ordinarily be of the forced feed type. Fractional horse power motors are almost universally oiled by means of a felt or wool pad, or wick. The quantity of oil required by these small motors comes well within the capacity of the wick. One advantage of wick oiling is that the motor can be operated in almost any position. Wick-packed bearings have proved to be entirely satisfactory in practice. Karelitz has investigated the performance of waste-packed bearings and his publication[15] should be of value to the designer. Ring-oiled motors must, of course, within close limits be operated in a given position and with the shaft horizontal.

Many designs of rings have been tried in an effort to increase the oil delivery. Fig. 3 shows six types of rings and their cross sections, four of which are in general use. They are ordinarily made of brass or bronze; however, one will occasionally find rings made of compositions resembling babbitt. Type (e) is made from two narrow conical-shaped washers of steel tack welded together. Types (a), (b), and (c) are commonly used; (e), seldom; and (d) and (f), rarely. (a) and (c) are universally used where jointing is necessary as in split type bearings. So far as the author is aware, no tests have been made to determine the oil delivery capacity of variously shaped rings. The performance of oil ring bearings has been investigated, however.[16] While ring oiling may not provide as copious a flow as some other methods, experience has shown it to be ample.

Once the lubricant has been supplied to the rubbing surfaces, the hydrodynamic action is depended upon to support the load. It is not at all rare to find motors that have been in operation for twenty years or more without bearing repairs. On the other hand, bearings have been known to fail after only a few hours of service. Bearing troubles and remedies will be discussed later.

## VII. LUBRICANTS AND LUBRICATING EFFICIENCY

The lubricant almost universally used for motor bearings is a petroleum oil having a viscosity of 135-140 Universal Saybolt at 100° F. If absolute viscosities are known or determined from calculation, they can be converted to commercial viscosities by the use of Fig. 4 which also gives relations between the various commercial systems. The oil should be well refined, free from suspended matter, water, acid, and vegetable or animal oils.

For low temperature operation, an oil having a viscosity of 105-110 Saybolt at 100° F. and a cold test of -20° F. is better suited. For the unusually high temperatures, a viscosity of 160-165 is more nearly correct. Practical knowledge and good judgment must be used at times when unusual conditions of operation prevail. In general the three grades just given will be adequate for temperature ranges from -20° to 120° F. Theoretically for maximum lubrication efficiency there is a certain film thickness which is dependent upon temperature of the film, the operating viscosity, the rubbing speed, bearing clearance, and width of bearing. Obviously it is impracticable to maintain maximum efficiency. An experienced person can, however, judge the operating viscosity by observing the action of the ring and oil. If the action is sluggish, due to the oil retarding the ring, then a lighter bodied oil should be used. If the action is lively, but the lubricant appears watery, a heavier oil is required.

There is a very important physical characteristic of mineral oils that should not be overlooked; namely, the difference in the rate of change of viscosity for given temperature changes. An asphalt base oil may have a viscosity twice as high as a paraffin base oil at 100° F., but when compared at a temperature of 150° F., they will be approximately the same value.

Let us now consider a number of conditions affecting the efficiency of lubrication. There must be reasonable tolerances in the diameters of the journal and bearing bore. Table I shows those used by one of the leading manufacturers of electrical machinery.[13] When the maximum clearances shown in the table have become doubled due to wear, the author considers new bearings to be necessary if good operating conditions are to be maintained. The accuracy of alignment of the bearings with the journal is oftentimes very imperfect in practice due to errors of machining and methods of manufacture. The error of misalignment is for the most part overcome in large motors by the use of self-aligning bearings. Misalignment causes concentration of bearing pressures and results in undue wear of the rubbing surfaces. The finish of the rubbing surfaces as left by the manufacturer is ordinarily good enough; however, grit in the bearing may leave the surfaces in such a roughened condition as to greatly impair the bearing efficiency. Fig. 5 shows a bearing and journal whose surfaces are in good condition. Fig. 6 shows the journal after grit had damaged the bearing. Fig. 7 shows a babbitt-lined bearing after scoring due to grit. Fig. 8 is a journal that has been badly pitted due to electric currents. Fig. 9 shows a very bad case of electrical pitting of a bearing. Fig. 10 shows a journal and its bearing that have become damaged due to slipping of the oil ring. Fig. 11 shows a bearing whose efficiency is greatly reduced by very bad oil grooving and electrical pitting. These cases just cited are for the most part extreme examples, but they are introduced here to show what one actually finds in practice. They will be discussed more in detail later on. The oil ring may perform badly due to being rough or out of round. Fig. 12 is a photograph of an oil ring taken from a large bearing showing the effect of dancing, which is detrimental to good oil delivery.

In addition to the above mentioned causes of poor efficiency, the author has found in bearing housings large quantities of such foreign matter as bronze or babbitt particles resulting from wear, dirt from the atmosphere and surrounding objects, water, sawdust, flour, etc. While linseed oil may not be considered a foreign matter, the author has actually found bearings in which this oil was used. It is not at all remarkable that motor bearings oftentimes run poorly but that they run at all.

As previously stated, a good motor lubricant should be a well-refined mineral oil, free from suspended matter, water, acid, vegetable oils, and animal oils. By suspended matter is meant, foreign substances as dirt and particularly grit. Water and acids cause rusting or otherwise attack the metallic rubbing surfaces. Vegetable and animal oils turn rancid, thereby causing gumming and the formation of free fatty acids. A most excellent book for one interested in making tests of lubricating oil is written by Battle and is obtainable in public libraries.[17]

## VIII. FACTORS AFFECTING LOAD CARRYING CAPACITIES

The ability of a bearing to carry loads depends upon ten generalized principles, some of which have already been touched upon. These principles have been laid down by such prominent men as Tower, Thurston, Goodman, Lasche, and Stribeck. Bierbaum has organized these principles and tabulated them as follows:[18]

1. The bearing surfaces are completely separated by a supporting film of oil.

2. The friction of operation is the fluid friction in the oil film, and adequate thickness of film is essential.

3. During construction, proper clearance or space should be provided for a normal thickness of oil film.

4. The advance edge of a bearing surface must be rounded or chamfered off in order to permit a supporting film of oil to form.

5. The oil film forms most effectively upon a bearing surface whose advance edge is at right angles to the direction of motion.

6. An increase in speed increases the thickness of film, all other conditions remaining constant and clearance permitting.

7. An increase in the viscosity of the oil increases the thickness of the film, all other conditions remaining constant and clearance permitting.

8. The larger the unbroken film of oil, the greater will be the average pressure supporting capacity per unit area, other conditions remaining constant.

9. Every unnecessary oil groove or interruption in the continuity of the oil film reduces the supporting capacity of the film.

10. For every bearing condition there is a film thickness corresponding to maximum lubricating efficiency.

It will be seen from this tabulation that load carrying ability is effected principally by four conditions; namely, rubbing speed, viscosity of lubricant, film thickness, and area of unbroken oil film. Of these four conditions the viscosity of the lubricant and the film thickness have already been decided upon for electric motor operation. In the matter of actually measuring the bearing clearance, the author has found the lead wire method to be the most reliable and most expedient. Fig. 13 shows a large bearing with a lead wire at the right before squeezing, and at the left after squeezing. By measuring the flattened wire, the exact clearance is determined. By this method the bearing and journal need not be removed from the motor. The bearing, of course, must be of the split type when this method is used. Due to irregularities of the rubbing surfaces, measuring the bore and journal diameters separately with micrometers may prove laborious as well as misleading.

Perhaps the greatest contributing cause to poor lubrication in electric motor bearings is found in the use of improper oil grooves. Considerable attention has been given this matter by the author who has had a great many bearing troubles to investigate. Figs. 14 to 28 incl. are photographs taken to show the various oil grooving systems actually found in practice. The most elaborate grooving is shown in Fig. 11. We have here a display of almost every conceivable oil groove, good and bad. No attempt will be made here to analyze the effects of each and every type shown. It is doubtful if such an analysis would prove more than the fact that most of them are bad. It is apparent that there is a very decided need for a better understanding of the principles of lubrication among those responsible for design and construction. Fig. 28 shows two die-cast bearings whose oil grooving conforms to correct principles. Fig. 29 shows the type adopted by the Frequency Change Department of the Consumers Power Company after a thorough study had been made of oil grooving systems. This type of bearing has proven satisfactory where severe loading conditions caused other bearings with bad grooves to fail. This bearing can be made in an ordinary lathe and does not require any special equipment other than a boring bar with suitably shaped end for cutting the longitudinal distributing grooves.

In order to show the effects of oil grooves running through pressure areas of the oil film, Fig. 30 has been constructed using lines of equal pressure similar to those shown in Fig. 1 (c) and Fig. 2 (c). It might be explained here that the difference in the pressure distribution lines between the two diagrams is caused by differences in clearance, speed, and viscosity. Large clearances and slow speeds cause localized distributions with elliptically shaped pressure lines, small negative pressure areas, and high maximum pressures. High speeds and small clearances give more extensive areas with lines that tend to become rectangular, large negative pressure areas, and lower maximum pressures.

It will be noted that the so-called "generalized operating variable" $ZN/P$ is nearly equal in both cases. In this expression $Z$ = absolute viscosity, $N$ = revolutions per minute, and $P$ = unit pressure. This expression is somewhat of a criterion for bearing design that has come into use in the last few years. Previously, the equation $PV = C$ was generally accepted as expressing the carrying capacity of a bearing. In this formula $V$ = velocity of rubbing surfaces, and $C$ = a constant. In the author's opinion, neither of these are sufficient to express the performance of a bearing, nor are they sufficiently inclusive for a logical design.

Turning now to Fig. 30, suppose an oil groove were cut circumferentially from A to B, as shown dotted in (A), pressures would then drop to zero on each side of the groove throughout its length resulting in a very great loss in load-carrying capacity. If curve E, in (B), represents the average axial pressure distribution before cutting the groove, and curves F and G after, then the shaded area represents the loss in capacity. The bearing then becomes equal to two narrow bearings of half width. Suppose now that a groove is cut across the bearing from C to D instead, if curve H, in (C), represents the distribution of the average circumferential pressures before cutting the groove, and curves I and J after, then the difference in area under H and the sum of areas under I and J represent the loss in capacity. The difference obviously is very great and is represented by the shaded area L.

Such a bearing as shown in Fig. 11 would have pressure distributions that are components of (B) and (C) in Fig. 30, resulting in very poor bearing efficiency in general. The edges of the grooves in Fig.11 are extremely sharp and naturally they act as scrapers.to remove the oil from, instead of aiding its delivery to the rubbing surfaces.

Even with the most favorable design of oil grooving as in Fig. 29, it is possible to have adverse conditions especially when the journal thrust is against one of the horizontal distributing grooves and the rotation of the journal is such that oil delivery is toward this groove. This condition is indicated in Fig. 31 although the journal is shown central. Better lubricating conditions would be had if for a journal thrust shown the rotation of the shaft were left hand instead of right. Oil would then enter where the clearance is nearly a maximum.

It is desirable to have the distributing grooves terminate a short distance before the drain grooves are reached to avoid direct loss of oil to the housing or well. The drain grooves should be cut close to the end of the bearing in order to give the maximum effective area for the lubricant. In calculations involving projected area, the width of the bearing should be considered as the width between grooves and not from end to end of the bearing. In solid type bearings the distributing grooves can be carried well above the center of the bearing as shown in Fig. 31 and thus be more effective against horizontal journal thrusts.

## IX. PRACTICAL BEARING DESIGNS

Having described operating conditions both good and bad, the question arises how best to arrive at a rational design. Solutions by higher mathematics are of little direct value. At best they serve as a guide only. Empirical formulas, if well chosen, are sufficient. The author has selected formulas based upon experimental data and has drawn curves by which one can readily solve for load-carrying capacity and for operating film temperatures both of which are limiting factors. If the ratio of the width of the bearing to the diameter of the journal is kept approximately three to one, and standard initial clearances as shown in Table I are employed, the design will come well within the limits of good practice. It is presumed that the shaft has already been designed for sufficient strength and rigidity. There is also the matter of critical speed to be investigated. Fig. 32 is included in the appendix for this purpose. It is self-explanatory and needs no explanation here except to say that the operating speed should be kept from 15 to 20% above or below the critical speed.[19]

The formulas selected for bearing loadings are based on tests made by Alford, whose work stands undisputed and is often quoted as authoritative.[20] The curve, Fig. 33, was constructed from three formulas after changing slightly the constants of two of them in order to make a smooth continuous curve. The changes made were 7 to 7.23 in the formula $W = 7\sqrt{V}$ and 30 to 29.4 in $W = 30\sqrt[4]{V}$ . The curve has a factor of safety of two. The critical breakdown values are twice those shown. By selecting factors of safety to suit operating conditions, the unit bearing load is readily obtained. It is understood that the General Electric Company uses these formulas in their designs.[19]

The next step is that of determining the temperature rise or the temperature of the oil film. Kimball & Barr gives two very useful formulas for the generation of heat, which are as follows:

For speeds up to 500 ft. per minute

$$uwV = \frac{2.3\ V^{1.5}}{t - 32} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (1)$$

For speeds above 500 ft. per minute

$$uwV = \frac{51.2\ V}{t - 32} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (2)$$

in which

u = coefficient of friction
w = pounds per sq.in. proj. area
V = rubbing speed in ft. per minute
t = temperature in degrees F. of oil film

The quantity uwV represents the heat generated in foot pounds per minute per sq.in. of proj. area. From page 126 of the same source we find curves giving the heat radiated from various types of bearings except that no mention is specifically made of electrical motor bearings. Leutweiler[21] gives K = 1150 for General Electric well-ventilated bearings in the formula

$$Q = \frac{(T_o + 33)^2}{K} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (3)$$

in which

Q = radiation in ft.lbs. per second per sq.in. proj. area
$T_o$ = difference in temp. between the bearing and cooling medium

Fig. 34 shows the curves reproduced but with slightly different terminology to correspond with this text. Curve 4 was constructed from formula (3) in which K = 1150.

Equating the heat generation formulas (1) and (2) respectively to the heat dissipation formulas, we get for speeds up to 500 ft. per min.

$$60 \times \frac{2.3\ V^{1.5}}{t - 32} = \frac{(T_o + 33)^2}{K} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (4)$$

and for speeds greater than 500 ft. per minute

$$60 \times \frac{51.2\ V}{t - 32} = \frac{(T_o + 33)^2}{K} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (5)$$

Letting $T_o = t - t_r$ in which $t_r$ = room temp., and letting K = 1150, formulas (4) and (5) become

for speeds up to 500 ft. per minute

$$\frac{60\ (t - t_r + 33)^2}{1150} = \frac{2.3\ V^{1.5}}{t - 32} \quad . \ . \ . \ . \ . \ . \ . \quad (6)$$

for speeds above 500 ft. per minute

$$\frac{60\ (t - t_r + 33)^2}{1150} = \frac{51.2\ V}{t - 32} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (7)$$

It is a very common belief that the load carried by a bearing determines the heating. To the contrary, it has been proven that in perfectly lubricated bearings the generation of heat varies with the speed of rubbing. This significant fact is due to the reduction of the coefficient of friction with increase in load such that for a given velocity of rubbing the product of the coefficient of friction and the unit load is constant.

The running temperature of a bearing depends upon the quantity of heat generated and the quantity radiated. Obviously the temperature of the rubbing surfaces will reach an equilibrium when the rate of heat generation is equal to the rate of dissipation.

The generation of heat in a journal bearing is a function of the viscosity of the lubricant. In checking over the experimental data from which formulas (1) and (2) were derived, the viscosities were found to check closely with oil specifications given on page 5, so that the formulas have a direct application to electric motor bearings.

There appears to be no appreciable difference in the radiating surfaces of General Electric motors and those of other manufacturers in the same sizes. Therefore, curve 4 of Fig. 34 can be regarded as representative for electric motors generally. Equations (1) and (2) together with the curves of Fig. 34 can be used to solve problems involving bearing temperatures for electric motors, as well as for other types of journal bearings, but the process is one of cut and try. Equations (6) and (7) for electric motors only are difficult of direct mathematical solution. The author has constructed the curves of Fig. 35 by using equations (6) and (7) which greatly simplify the work. Having given the rubbing speed and the room temperature, the corresponding oil film

temperature is at once determined. If the film temperature is over 140° F., artificial cooling should be considered. Increasing the turbulence of the air around the bearing by adding fan capacity to the rotor may be sufficient.

Occasionally a vertical bearing of the journal type is encountered. The same general laws govern the design of this type also. In order to distribute the oil to the rubbing surfaces, it is common practice to cut a spiral groove in the journal in such a direction as to elevate the oil. Such motors are therefore unidirectional. Fig. 36 shows a vertical bearing of a grinding machine which happens to be belt driven. There is little difference, if any, between this bearing and the one used when the pulley is replaced by an electric motor. The clearance of this bearing is regulated by raising or lowering the bronze bushing. It is very important that the operator of this machine be thoroughly familiar with its construction. If the bushing is too loose, oil will run out of the bearing at the lower end. If too tight, severe damage may be done by overheating and seizing of the rubbing surfaces.

## X. BEARING MATERIALS

The question often arises whether bearings should be babbitt or bronze lined. At first thought it would seem advisable to use a material that would not allow the rotor to come in contact with the bore of the stator in case of failure. The writer has known of a large number of motors whose babbitt bearings have been overheated or otherwise worn so that rubbing of the rotor took place. In no instance was there any appreciable damage done. The motors simply became overloaded and blew their fuses or they would refuse to start. If the bearings gradually went down, some evidence of overheating of the stator or undue noises gave sufficient warning.

Bronze bearings due to their higher melting point do not fail quickly. Evidence of failure usually shows itself by overheating or by seizing to the shaft and stopping the motor. It is frequently found that the shaft has been scored badly, a result that is seldom found with babbitt unless abrasive material is present.

In so far as lubrication is concerned there is no appreciable advantage in either babbitt or bronze if the bearing is properly designed and lubricated so that the load is carried hydrodynamically. Generally speaking, bearings under two inches in diameter are made of bronze while those over that size are made of babbitt. Babbitt requires a steel or iron supporting shell and is, therefore, impracticable in the smaller sizes. Bronze sleeves require no extra support, but the quantity of metal necessary in the large sizes makes them too costly.

There are a great many varieties of babbitt metals on the market. The choice of a suitable grade becomes rather perplexing if the advertised claims for each brand are taken seriously. The most valuable guide found is shown in Table II which was taken from an

A.S.T.M. publication.[22] The range is sufficient to cover all practical purposes. The table gives all the physical properties necessary to characterize each kind, provided the babbitt has not been overheated and is poured at the correct temperature.

The range extends from high tin base alloys to those of high lead. The dividing line between the tin base and lead base babbitts is between No. 5 and No. 6. It will be noted that there is a decided changing over from a preponderence of tin to that of lead at this point. The antimony content remains virtually constant for alloys below No. 3. The physical properties for alloy No. 3 have the highest values almost throughout the entire range. Because of the high percentage of tin in this alloy, it must, of course, be comparatively expensive. It is understood that this alloy is the same as General Electric Company's babbitt No. 17. Where extreme conditions of service prevail, alloy No. 3 should give excellent service; however, there are lead base alloys which are perfectly suitable for services where resistance to shock, too high temperatures, and too high unit loads are not limiting conditions. Alloy No. 3 is particularly suitable for airplanes and automobiles where bearing loadings may run as high as 3000 lbs. per sq.in.;[23] where operating temperatures are high, and where a high degree of toughness is required to resist shock. It is a very good alloy for electric motor service, but a less expensive alloy such as No. 5 or No. 6 should prove entirely suitable considering the fact that loadings seldom exceed 100 lbs. per sq.in.

Westinghouse[24] has two grades of babbitts for electrical machinery. Their alloy No. 25 is of the lead base type while their No. 14 has a tin base. Their general catalog states that their lead base alloy has been used successfully for many years in the manufacture of motors, generators, turbines, and other electrical equipment. They recommend their tin base alloy for excessive pressure, vibration, high speed, and heavy duty.

Fig. 37 reproduced from Westinghouse instruction book shows that a lead base babbitt may be superior to certain tin base babbitts in resisting impact loads. This is very interesting because a tin base alloy is ordinarily considered to be superior. Another source[25] shows that the addition of lead in amounts of 1, 3, and 5% to a tin base babbitt actually increased the hardness over a temperature range of 25° C. to 80° C. It is also shown that a lead base babbitt may be superior to a tin base babbitt over the same temperature range and may closely approach alloy No. 3 in Table II. These facts show that a lead base alloy can be made that will compare favorably or even be superior to some of the tin base babbitts which are more expensive.

The question arises that if a lead base alloy has suitable physical qualities and only costs from 20 to 50% that of the tin base, why is it not universally accepted as the standard for use in electric motors. The answer, so far as the author can find, is because the lead base alloys may be greatly damaged by improper temperature control during the pouring of the bearing while the tin base babbitts are generally much less effected. This characteristic is clearly illustrated in Fig. 38. The proper temperature for pouring babbitts is shown in Table II.

To avoid overheating babbitts, close temperature control is essential. Whether of the lead base or tin base, they should not be heated over 490° C. at any time. The time honored use of the pine stick in hands of experienced and competent workmen to test proper pouring temperature may be a fair substitute for electrically controlled pots, but if any quantity of work is done of any importance, pyrometer control is necessary. The author has repeatedly observed babbitt heated in an open ladle to a dull red color (650° C.) during the process of babbitting. Such practice is, of course, detrimental to good results.

The effect of various constituents of babbitt are set forth in Materials Handbook[28] as follows: Antimony imparts hardness and smooth surface to soft metal alloys, it expands on cooling and unites with copper to form a crystalline alloy having valuable bearing qualities. Lead softens the mixture and raises the anti-friction qualities and increases fluidity. Copper hardens, toughens and raises the melting points. Arsenic, iron, zinc, and aluminum are generally considered objectionable.

Bronze bearing alloys do not appear so involved and complex. The physical properties of any of the bronze listed in Table III are sufficient for electric motor use. The author's personal experience in machining many hundreds of bearings of bronze has lead to the conclusion that a free turning rather tough bronze is sufficient. If the bronze is too hard, the sand cast scale is very detrimental to the cutting tool.

The composition of a suitable bronze for electric motor bearings would correspond to alloy, grade No. 2, of Table III composed of 80% copper, 10% tin, and 10% lead. The American Machinist[26] states that this alloy is suitable for high shaft speeds and where journals are not heat treated. The same source gives compositions of bronze alloys having tensile strengths as high as 33,000 lbs. per sq.in. and compressive strengths up to 28,000 lbs., but no mention is made of the machinability of such a bronze unless the Brinell numbers of 130-143 could be used as an index. Alloy 2 of Table III has a Brinell number of 55 and no doubt is readily machinable. The melting of bronze metals in the machine shop is not to be recommended because some understanding of metallurgy is essential. American Machinist states that laboratory control is necessary to insure uniformity of structure. A bronze bearing, like babbitt, is essentially a mixture of hard and soft crystals, the hard crystals being the supporting media for the load and the softer crystals being the matrix in which the hard crystals are embedded.

There is one problem concerning bronze bearing alloys or perhaps certain compositions, which the author has been unable to solve, that is, what causes the bronze to shrink onto the shaft and seize upon overheating. In several cases the entire bearing surface of bronze liners had to be scraped after seizure before the liners could be put back onto the journal. It was necessary to remove bronze from areas that had not seized and had previously run with sufficient clearance. One experienced motor maintenance man of the author's acquaintance said he had found certain makes of bronze would give this trouble and others would not. The author intended to conduct certain experiments to determine the cause, but time does not permit. A considerable amount of research has revealed no mention of such a characteristic.

## XI. BEARING TROUBLES AND REMEDIES

Bearing failures are not always caused by overload, lack of oil, or grit in the lubricant. There is one cause that is peculiar to the electrical industry and is known as electrical bearing currents. Mr. C. T. Pearce[27] gives a most comprehensive and complete discussion on the subject and one interested will find his publication very enlightening. Under certain conditions, bearing currents are induced in the shaft of electric motors and generators by alternating flux linkages surrounding the shaft. The flow of current through the bearing causes pitting that is extremely detrimental. Insulating the bearings will, of course, stop the flow of current and one will oftentimes find such insulation used in large and important electrical machinery. The pitting of the rubbing surfaces is characteristic. An examination of the pits reveals that they are minute craters having raised edges. A microscopic inspection of a pit well revealed that the metal was actually fused. It is not easy to find a pit in its original form due to wear that takes place immediately after its formation. Fig. 8 and 9 show a journal and bearing damaged by electrical bearing currents and will aid in the identification of such troubles. It will be seen that there is very little circumferential alignment as in the case of damage done by abrasives (Fig. 6 and 10). It would therefore appear that the pits were formed almost instantaneously. The raised edges of the pits act as effective cutting edges and destroy the bearing at a very rapid rate.

In remachining the journal, the author has found that the areas surrounding the pits are extremely hard and will dull a high speed cutting tool so rapidly that very little progress can be made. The areas are so hard that an ordinary file has very little effect in removing the metal. It is necessary to take a lathe cut of sufficient depth to get entirely beneath the pits. The hardening no doubt takes place by the metal cooling rapidly from fusion temperature to that of the bearing due to heat absorption of the shaft. Shafting materials have a sufficiently high carbon content (approximately .40) to take on hardness under such conditions. Insulation of bearings is not always sufficient. Large water-cooled bearings having insulating materials in the pipe connections have been known to develop pitting. In such instances contamination of cooling water has been found sufficient to be a good electrical conductor. The question sometimes arises whether static discharge from such sources as a belt drive may not cause pitting. It is believed not because of an insufficient amount of heat to cause fusion of the metals. If static from belt drives were to cause pitting, there undoubtedly would be much evidence to support it because of the large number of belt drives in everyday use.

Electrical pitting is not always caused by induced currents. In motors having wound rotors of the repulsion-induction type, pitting is very common. This is caused by defective short-circuiting mechanisms. Current generated in the windings, in these motors, flows through the short-circuiting device,through the shaft, through the bearing, thence to the frame and returns to the windings through the commutator brushes. Though the motor has a brush-lifting device, it is necessary for the short ciruiter and the brushes to be in contact with the commutator for a short period of time while the motor is accelerating during starting. It is during this interval that the damage is done. When ball bearings are used, the damage is much more severe and pronounced. Large slip ring motors are also subject to this trouble.

## XII. FITTING OF BEARINGS

There are several errors entering into the machining of bearings, particularly in the boring reaming or otherwise cutting the metal from the inside surfaces. The journal lends itself well to turning or grinding because its surface is external. The hole in the end bell of the motor and the hole in the bearing receiving the journal are quite apt to have various irregularities due to the inherent tendency of reamers to produce chatter marks and to cut a greater amount from one side than the other if the bore surfaces are unsymmetrical and if the end of the hole receiving the reamer is not square. For instance, if a bearing such as shown in Fig. 20, right, having a large cut-away section for oil wick packing were being reamed, the reamer would be crowded off center toward the hole. Unsymmetrical oil groving produces the same result. Considerable error due to both conditions acting together may be present in a reamed bearing and becomes greater with increased removal of metal. Holes in the end bells of motors are usually somewhat out of round, oftentimes tapered, and in many cases out of parallel with the center of rotation of the shaft. Self-aligning bearings in the larger motors are naturally free from the errors of misalignment, but not necessarily free from other errors just mentioned. In practice the errors of reaming are reduced to a practical minimum by reaming only such small amounts as to produce a smooth finish, usually just sufficient to remove boring tool irregularities.

To produce a perfectly round hole by reaming requires considerable attention to the finer points of reamer construction and the art of cutting metals. The usual run of expansion reamers are notoriously poor tools for precision work. They can, however, be made to produce reasonably satisfactory work after hand stoning and by careful manipulation.

It is a well-known fact that the cutting edge of a reamer must have a certain small land just behind the cutting edge to prevent the blade from "hogging in". The width of this land governs largely the stability or firmess of the reamer in cutting. If too wide, the reamer binds. If too narrow, it produces a haggled surface. When the width is right the operation is firm, cutting is free, and the hole has a burnished appearance. Different metals, of course, require different widths of lands.

The operation of boring in a lathe produces a hole whose average trueness leaves little to be desired; however, the surface finish is ordinarily too rough for bearing purposes. Reaming or scraping is resorted to for producing smoothness. For precise work the reamer is too treacherous. Hand scraping using prussian blue or other suitable indicator is to be preferred. It should be kept in mind that irregularities of surface contour may reduce the load-carrying ability and normal length of life considerably. Fig. 39 shows the cross section of a one-bladed reamer perfected by the author for reaming holes with the maximum degree of roundness and of predetermined diameter. The enlarged section of the blade shows the cutter having a land as described. The amount of clearance behind the land is determined by methods used in manufacture and has nothing to do with the cutting action. This reamer was designed to have maximum support opposite the cutting edge so as to avoid chattering. The reamer is of the adjustable type and is especially adaptable for line reaming because of its inherent stiffness. With conventional type multi-bladed reamers the fluting reduces the cross section so that the reamer loses the necessary rigidity for line reaming when the bearings are far apart as in motor construction.

The author has used the one-bladed reamer with marked success in the making of precision target rifle barrels and considers this type to be one of the most effective where precise reaming is desired.

There are a few fundamentals, however, that must not be overlooked in its construction and operation. Referring to Fig. 40, it will be noted that the pilot is not a tight fit in the hole being reamed either before or after the body makes contact. Fig. 41A shows the pilot supporting the cutter just prior to the body entering the hole. Fig. 41B shows the body supporting the cutter and being guided by the finished hole while the pilot is relieved of the duty of supporting after the reamer body has entered. The support of the cutter is directly opposite the cutting edge and the pilot only serves as a guide in the axial direction.

The cutting edge must overhand the reamer body slightly and its height H must be less than the clearance between the pilot and the unfinished hole. If there is any variation in heighth of the cutter, it must not be greater toward the middle of the body, otherwise the support of the body on the opposite side will be lost. It is desirable to have the front end of the blade a half-thousandths of an inch or so higher than the rear end.

Copious lubrication should be applied so as to keep chips from working under the reamer body and producing an oversize hole or the chip clearance groove can be partially filled with a rather sticky grease so as to hold the chips into the groove. This type of reamer is intended for removal of very small amounts of metal at a time and will produce excellent results in the hands of careful and competent workmen.

Line reaming is particularly desirable for fractional horse power motors because of the inaccuracies found in present manufacture probably due to circumstances resulting from competition. The mere removal of the old bushing and pressing in of a new one, even though the new one is perfectly concentric, usually results in a poorly aligned job due to the inaccuracies of the machining of the end bells. With small motors the line reaming method is the most practical. Large bearings are best scraped in because large reamers are unwieldly and their cost prohibitive unless large quantity production is being carried on.

The conventional type of bearing scraper does nicely in scraping babbitt but when used on bronze it lacks rigidity, does not have proper clearance, and cuts over too large a surface. The author has been called upon a number of times to scrape in bronze bearings that had seized to the shaft while running. The conventional type of scraper was found to be of little value. By experimenting, a design was worked out that gave very good performance. Fig. 42 shows a set ranging in size from 6 to 14 inches, which was found suitable for both small and large motor bearings. These were made from standard half round files without drawing their original temper. They are used in much the same manner as the conventional scraper. They differ in shape principally in that there is no offset in the handle, are much wider across cutting edges, and are more rounded at their points. Being straight in body they are very rigid because the cutting edge is in direct line with the points of support or operators' hands. The narrowness of the blade of the conventional type gives very little clearance to the cutting edge. On the other hand, with this type one can get almost any desired clearance by using various portions of the blade. Near the point the blade is narrow and the cutting clearance small, while toward the middle it becomes greater. If the cutting action is not quite correct for a given diameter hole and a given scraper, the next size larger or smaller scraper will undoubtedly be found satisfactory.

When the bronze sleeve is firmly supported as in a vice, one can literally cut shavings with remarkably little effort. The cutting edge is ground so as to have very little, if any, rake while cutting. Too much positive rake causes chatter and gives the user a sense of poor control.

## XIII. FIELD PROBLEMS

Three causes of motor bearing troubles have already been mentioned; namely, the seizing of bronze liners, shaft scoring due to foundry scale, and failure due to electrical bearing currents. There are also a number of other troubles. Loss of oil from a bearing is by no means a minor one. The cause oftentimes presents a baffling problem. The author has had first-hand experience with five causes as follows: rotor fan suction, foaming, ring throwing, vibrating shaft, and surface creepage. One is quite apt to suspect the fan of the rotor especially when rotor speeds are increased; however, it has been found that they seldom are the cause provided the bearing housing design is not at fault. Only one case has been found wherein the fan suction was proved to be the cause. Referring to Fig. 43, it will be noted that in the upper part of the housing there is a hole through the central wall designated as "pressure equalizing hole". When this hole is not present, or some other form of air passage between the two compartments, a slight vacuum on one side of the wall causes the oil level to rise on that side and to overflow through the opening around the shaft and into the motor.

The difference in pressure between the two sides need not be very great to give a difference of say 1" between the levels. Cases of this kind are easily taken care of by providing a pressure-equalizing hole. The only oil that can then be drawn from the bearing will be in the form of a mist carried out with air currents. Air currents are reduced to a minimum or completely stopped with external air by passes, as in the new Westinghouse design, or by sealing the opening around the shaft with felt or other substances.

Occasionally what would appear to be fan suction is nothing more than the use of too heavy a lubricating oil. High ring speeds are quite apt to carry air into the oil at a higher rate than the air will separate from the oil resulting in the formation of a foam which overflows the housing and becomes thrown into the motor windings by the rotor. An oil of lighter body is all that is required as a preventive.

Quite often the speed of the ring is sufficient to cause a veritable shower of oil within the housing and in such cases the oil is literally thrown out as a splatter. Globules of oil will be observed to come out of the housing between the shaft and the opening for the shaft. The use of a heavier bodied oil will oftentimes stop this trouble. In some cases barriers are necessary and in other cases a felt seal is required, depending upon the construction of the housing.

All motors ranging above the fractional horse power sizes have oil slingers designed to return oil to the housing and also to eliminate surface creepage toward the rotor. In cases where the oil ring is on the side of the wall next to the motor, drops of oil may fall from the upper overhanging portion of the housing and hit upon the rotating shaft. If it strikes on one of the slingers, it will, of course, be returned to the housing. If upon a straight portion, it will be partially thrown into the motor. It is, therefore, proper in such cases to have the slingers part in and part out of the housing as shown in Fig. 43.

A badly balanced rotor having a worn bearing has been known to throw oil out along the shaft in spurts as the shaft whipped from one side of the bearing to the other. Balancing the rotor, installing a new bearing, or providing a felt seal will stop the leakage. It is desirable to have the rotor in balance and to have a new bearing, but oftentimes a seal must be installed, at least temporarily.

Surface creepage is more or less prevalent in all motors. Quantities lost are usually insignificant except in cases where the oil level is carried too high in the oil inspection cup. The level is best carried at least 3/8" below the top. Oil cups are sometimes omitted and in their place a rough hole is cored in the casting or a small hole is drilled through the housing. Surface creepage is usually bad when such holes are provided. If the creepage is objectionable, standard oil cups can ordinarily be fitted to the housing

Another serious cause of bearing failure is that due to defective oil rings. Oftentimes a failure can be traced to a burr on the ring that happened to catch somewhere in the ring slot and stop rotating. The stopping is sometimes assisted by a cold viscous oil. The ring shown in Fig. 12 was taken from a 300 h.p. motor running 1800 r.p.m. Bearing trouble had not been experienced possibly because the bearing had two rings or perhaps the ring had never actually failed to rotate. On the right of center will be seen a bright strip where slippage had taken place. At the bottom will be seen a series of overlapping ripples similar to ruts formed in a gravel road by auto traffic. Apparently there had been severe slipping followed by combined slipping and dancing. The ripples actually overlapped each other by the surface metal flowing from one cavity into the other. It will also be noted that the inside edges of the ring had been left very rough from cutting off with a lathe tool. This ring had been made from a piece of brass tubing and was found to be considerably out of round. Rings should be round, of even cross section, free from burrs, and made of a material that resists wear. Bronze is the best material in common use at the present time.

From what has been written it is quite obvious that electric motor bearings like other commercial products may be well or poorly made from the standpoint of design, materials, and workmanship. One oftentimes gets the impression upon the inspection of bearings, and this applies usually to the whole motor, that there is always some one who can build a bearing a little worse and a little cheaper and still find a sale for his product.

In closing, the author wishes to express his earnest opinion that there is a need for standardized practice among manufacturers and a general enlightening of engineers and trades men toward bearing designs and excellence of workmanship. It is hoped that this thesis may in some manner be used to bring about this accomplishment.

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# APPENDIX ( FIGURES AND TABLES)

| Nominal diam | Journal: Max diam | Journal: Allowable variation below max diameter | Bearings, Horizontal: Min bore | Bearings, Horizontal: Allowable variation above min bore | Bearings, Vertical: Min bore | Bearings, Vertical: Allowable variation above min bore | Bearings, Step: Min bore | Bearings, Step: Allowable variation above min bore |
|---|---|---|---|---|---|---|---|---|
| 3/8 | 0.375 | 0.0005 | 0.377 | 0.001 | 0.377 | 0.001 | 0.3755 | 0.0005 |
| 1/2 | 0.500 | 0.0005 | 0.502 | 0.001 | 0.502 | 0.001 | 0.5005 | 0.0005 |
| 5/8 | 0.625 | 0.0005 | 0.627 | 0.001 | 0.627 | 0.001 | 0.6255 | 0.0005 |
| 3/4 | 0.750 | 0.0005 | 0.752 | 0.001 | 0.752 | 0.001 | 0.7505 | 0.0005 |
| 7/8 | 0.875 | 0.0005 | 0.877 | 0.001 | 0.877 | 0.001 | 0.8755 | 0.0005 |
| 1 | 1.000 | 0.0005 | 1.002 | 0.001 | 1.002 | 0.001 | 1.0005 | 0.0005 |
| 1 1/8 | 1.125 | 0.0005 | 1.128 | 0.001 | 1.128 | 0.001 | 1.126 | 0.0005 |
| 1 1/4 | 1.250 | 0.0005 | 1.253 | 0.001 | 1.253 | 0.001 | 1.251 | 0.0005 |
| 1 1/2 | 1.500 | 0.0005 | 1.503 | 0.001 | 1.503 | 0.001 | 1.501 | 0.0005 |
| 1 3/4 | 1.750 | 0.0005 | 1.753 | 0.001 | 1.753 | 0.001 | 1.751 | 0.0005 |
| 2 | 2.000 | 0.0005 | 2.003 | 0.001 | 2.003 | 0.001 | 2.001 | 0.0005 |
| 2 1/4 | 2.250 | 0.0005 | 2.253 | 0.001 | 2.253 | 0.001 | 2.251 | 0.0005 |
| 2 1/2 | 2.500 | 0.0005 | 2.503 | 0.001 | 2.503 | 0.001 | 2.501 | 0.0005 |
| 2 3/4 | 2.750 | 0.0005 | 2.754 | 0.002 | 2.754 | 0.001 | 2.7515 | 0.0005 |
| 3 | 3.000 | 0.0005 | 3.004 | 0.002 | 3.004 | 0.001 | 3.0015 | 0.0005 |
| 3 1/4 | 3.250 | 0.0005 | 3.254 | 0.002 | 3.254 | 0.001 | 3.2515 | 0.0005 |
| 3 1/2 | 3.500 | 0.001 | 3.504 | 0.002 | 3.504 | 0.001 | 3.5015 | 0.0005 |
| 4 | 4.000 | 0.001 | 4.005 | 0.002 | 4.005 | 0.001 | 4.002 | 0.001 |
| 4 1/2 | 4.500 | 0.001 | 4.505 | 0.002 | 4.505 | 0.001 | 4.502 | 0.001 |
| 5 | 5.000 | 0.001 | 5.006 | 0.002 | 5.005 | 0.002 | 5.0025 | 0.001 |
| 5 1/2 | 5.500 | 0.001 | 5.507 | 0.002 | 5.505 | 0.002 | 5.503 | 0.001 |
| 6 | 6.000 | 0.001 | 6.009 | 0.002 | 6.006 | 0.002 | 6.003 | 0.001 |
| 7 | 7.000 | 0.001 | 7.011 | 0.002 | 7.006 | 0.002 | 7.0035 | 0.001 |
| 8 | 8.000 | 0.001 | 8.012 | 0.003 | 8.006 | 0.002 | 8.004 | 0.002 |
| 9 | 9.000 | 0.001 | 9.013 | 0.004 | 9.006 | 0.002 | 9.0045 | 0.002 |
| 10 | 10.000 | 0.0015 | 10.014 | 0.005 | 10.007 | 0.003 | 10.005 | 0.002 |
| 11 | 11.000 | 0.0015 | 11.015 | 0.005 | 11.007 | 0.003 | 11.0055 | 0.002 |
| 12 | 12.000 | 0.0015 | 12.016 | 0.005 | 12.007 | 0.003 | 12.006 | 0.002 |
| 13 | 13.000 | 0.0015 | 13.016 | 0.005 | 13.007 | 0.003 | 13.0065 | 0.002 |
| 14 | 14.000 | 0.0015 | 14.016 | 0.005 | 14.007 | 0.003 | 14.007 | 0.002 |
| 15 | 15.000 | 0.0015 | 15.016 | 0.005 | 15.007 | 0.003 | 15.0075 | 0.002 |
| 16 | 16.000 | 0.0015 | 16.016 | 0.005 | 16.007 | 0.003 | 16.008 | 0.002 |
| 17 | 17.000 | 0.0015 | 17.018 | 0.005 | 17.007 | 0.003 | 17.008 | 0.002 |
| 18 | 18.000 | 0.0015 | 18.018 | 0.005 | 18.007 | 0.003 | 18.008 | 0.002 |
| 19 | 19.000 | 0.0015 | 19.018 | 0.005 | 19.007 | 0.003 | 19.008 | 0.002 |
| 20 | 20.000 | 0.0015 | 20.018 | 0.005 | 20.007 | 0.003 | 20.008 | 0.002 |
| 21 | 21.000 | 0.002 | 21.018 | 0.005 | 21.007 | 0.003 | 21.008 | 0.002 |
| 22 | 22.000 | 0.002 | 22.020 | 0.008 | 22.007 | 0.003 | 22.008 | 0.002 |
| 23 | 23.000 | 0.002 | 23.020 | 0.008 | 23.007 | 0.003 | 23.008 | 0.002 |
| 24 | 24.000 | 0.002 | 24.020 | 0.008 | 24.007 | 0.003 | 24.008 | 0.002 |
| 25 | 25.000 | 0.003 | 25.020 | 0.008 | .... | ... | ..... | .... |
| 26 | 26.000 | 0.003 | 26.020 | 0.008 | .... | ... | ..... | .... |
| 27 | 27.000 | 0.003 | 27.022 | 0.008 | .... | ... | ..... | .... |
| 28 | 28.000 | 0.003 | 28.022 | 0.008 | .... | ... | ..... | .... |
| 29 | 29.000 | 0.003 | 29.022 | 0.008 | .... | ... | ..... | .... |
| 30 | 30.000 | 0.003 | 30.022 | 0.008 | .... | ... | ..... | .... |
| 31 | 31.000 | 0.003 | 31.022 | 0.008 | .... | ... | ..... | .... |
| 32 | 32.000 | 0.003 | 32.024 | 0.010 | .... | ... | ..... | .... |
| 33 | 33.000 | 0.003 | 33.024 | 0.010 | .... | ... | ..... | .... |
| 34 | 34.000 | 0.003 | 34.024 | 0.010 | .... | ... | ..... | .... |
| 35 | 35.000 | 0.003 | 35.024 | 0.010 | .... | ... | ..... | .... |
| 36 | 36.000 | 0.003 | 36.024 | 0.010 | .... | ... | ..... | .... |

**Table I**

COMPOSITION AND PHYSICAL PROPERTIES[1] OF WHITE METAL BEARING ALLOYS

| Alloy Number | Specified Composition of Alloys | | | | Specific Gravity | Compositions of Alloys Tested | | | | Yield Point, lb. per sq. in.[2] | | Johnson's Apparent Elastic Limit, lb. per sq. in.[3] | | Ultimate Strength, lb. per sq. in.[4] | | Brinell Hardness[5] | | Melting Point | | Temperature of Complete Liquefaction | | Proper Pouring Temperature | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Copper, per cent | Tin, per cent | Antimony, per cent | Lead, per cent | | Copper, per cent | Tin, per cent | Antimony, per cent | Lead, per cent | 20° C. | 100°C. | 20° C. | 100°C. | 20° C. | 100°C. | 20° C. | 100°C. | Deg. Fahr. | Deg. Cent. | Deg. Fahr. | Deg. Cent. | Deg. Fahr. | Deg. Cent. |
| 1 | 4.5 | 91.0 | 4.5 | .... | 7.34 | 4.56 | 90.9 | 4.52 | none | 4400 | 2650 | 2450 | 1050 | 12850 | 6950 | 17.0 | 8.0 | 433 | 223 | ... | ... | 825 | 441 |
| 2 | 3.5 | 89.0 | 7.5 | .... | 7.39 | 3.1 | 89.2 | 7.4 | 0.03 | 6100 | 3000 | 3350 | 1100 | 14900 | 8700 | 24.5 | 12.0 | 466 | 241 | 669 | 354 | 795 | 424 |
| 3 | 8⅓ | 83⅓ | 8⅓ | .... | 7.46 | 8.3 | 83.4 | 8.2 | 0.03 | 6600 | 3150 | 5350 | 1300 | 17600 | 9900 | 27.0 | 14.5 | 464 | 240 | 792 | 422 | 915 | 491 |
| 4 | 3.0 | 75.0 | 12.0 | 10.0 | 7.52 | 3.0 | 75.0 | 11.6 | 10.2 | 5550 | 2150 | 3200 | 1550 | 16150 | 6900 | 24.5 | 12.0 | 363 | 184 | 583 | 306 | 710 | 377 |
| 5 | 2.0 | 65.0 | 15.0 | 18.0 | 7.75 | 2.0 | 65.5 | 14.1 | 18.2 | 5050 | 2150 | 3750 | 1500 | 15050 | 6750 | 22.5 | 10.0 | 358 | 181 | 565 | 296 | 690 | 366 |
| 6 | 1.5 | 20.0 | 15.0 | 63.5 | 9.33 | 1.5 | 19.8 | 14.6 | 63.7 | 3800 | 2050 | 3550 | 1800 | 14550 | 8050 | 21.0 | 10.5 | 358 | 181 | 531 | 277 | 655 | 346 |
| 7 | ... | 10.0 | 15.0 | 75.0 | 9.73 | 0.11 | 10.0 | 14.5 | 75.0 | 3550 | 1600 | 2500 | 1350 | 15650 | 6150 | 22.5 | 10.5 | 464 | 240 | 514 | 268 | 640 | 338 |
| 8 | ... | 5.0 | 15.0 | 80.0 | 10.04 | 0.14 | 5.2 | 14.9 | 79.4 | 3400 | 1750 | 2650 | 1200 | 15600 | 6150 | 20.0 | 9.5 | 459 | 237 | 522 | 272 | 645 | 341 |
| 9 | ... | 5.0 | 10.0 | 85.0 | 10.24 | 0.06 | 5.0 | 9.9 | 84.6 | 3400 | 1550 | 2400 | 950 | 14700 | 5850 | 19.0 | 8.5 | 459 | 237 | 493 | 256 | 620 | 327 |
| 10 | ... | 2.0 | 15.0 | 83.0 | 10.07 | 0.12 | 2.05 | 15.7 | 82.0 | 3350 | 1850 | 2250 | 1200 | 15450 | 5750 | 17.5 | 9.0 | 468 | 242 | 507 | 264 | 630 | 332 |
| 11 | ... | .... | 15.0 | 85.0 | 10.28 | 0.19 | 0.09 | 14.8 | 84.7 | 3050 | 1400 | 2750 | 1100 | 12800 | 5100 | 15.0 | 7.0 | 471 | 244 | 504 | 262 | 630 | 332 |
| 12 | ... | .... | 10.0 | 90.0 | 10.67 | 0.12 | 0.11 | 9.9 | 89.4 | 2800 | 1250 | 2250 | 950 | 12900 | 5100 | 14.5 | 6.5 | 473 | 245 | 498 | 259 | 625 | 329 |

[1] The compression test specimens were cylinders 1½ in. in length and ½ in. diameter, machined from chill castings 2 in. in length and ¾ in. in diameter. The Brinell tests were made on the bottom face of parallel machined specimens cast in a 2-in. diameter by ½-in. deep steel mold at room temperature

[2] The values for yield point were taken from stress-strain curves at a deformation of 0.125 per cent reduction of gage length.

[3] Johnson's apparent elastic limit is taken as the unit stress at the point where the slope of the tangent to the curve is 1½ times its slope at the origin.

[4] The ultimate strength values were taken as the unit load necessary to produce a deformation of 25 per cent of the length of the specimen.

[5] These values are the average Brinell number of three impressions on each alloy using a 10-mm. ball and a 500-kg. load applied for 30 seconds

**Table IIa**

| Alloy Grade No. | Tin, per cent. | Antimony, per cent. | Lead, per cent. | Copper, per cent. | Iron, max., per cent. | Arsenic, max., per cent. | Zinc, per cent. | Aluminum, per cent. |
|---|---|---|---|---|---|---|---|---|
| 1........ | 91 | 4½ | 0.35[a] | 4½ | 0.08 | 0.10 | none | none |
| 2........ | 89 | 7½ | 0.35[a] | 3½ | 0.08 | 0.10 | none | none |
| 3........ | 83⅓ | 8⅓ | 0.35[a] | 8⅓ | 0.08 | 0.10 | none | none |
| 4........ | 75 | 12 | 10 | 3 | 0.08 | 0.15 | none | none |
| 5........ | 65 | 15 | 18 | 2 | 0.08 | 0.15 | none | none |
| 6........ | 20 | 15 | 63½ | 1½ | 0.08 | 0.15 | none | none |
| 7........ | 10 | 15 | 75 | 0.50[a] | .... | 0.20 | none | none |
| 8........ | 5 | 15 | 80 | 0.50[a] | .... | 0.20 | none | none |
| 9........ | 5 | 10 | 85 | 0.50[a] | .... | 0.20 | none | none |
| 10........ | 2 | 15 | 83 | 0.50[a] | .... | 0.20 | none | none |
| 11........ | .. | 15 | 85 | 0.50[a] | .... | 0.25 | none | none |
| 12........ | .. | 10 | 90 | 0.50[a] | .... | 0.25 | none | none |

[a] Maximum.

**Table IIb**

TABLE SHOWING PHYSICAL PROPERTIES OF BRONZE BEARING METAL ALLOYS.

| Alloy, Grade No. | Desired Composition. | | | Ultimate Tensile Strength, lb. per sq. in.[1] | Elongation, in 2 in. per cent.[1] | Brinell Hardness (500 kg. for 30 sec.). | Shrinkage, in. per ft. | Weight, lb. per cu. in. | Compression Deformation Limit, lb. per sq. in.[2] |
|---|---|---|---|---|---|---|---|---|---|
| | Copper, per cent. | Tin, per cent. | Lead, per cent. | | | | | | |
| 1........... | 85 | 10 | 5 | 28 000 | 12.5 | 60 | 0.25 | 0.31 | 18 000 |
| 2........... | 80 | 10 | 10 | 25 000 | 8 | 55 | 0.25 | 0.31 | 15 000 |
| 3........... | 80 | 10 | 10 | 22 000 | 8 | 50 | 0.25 | 0.32 | 12 500 |
| 4........... | 77 | 8 | 15 | 20 000 | 10 | 48 | 0.25 | 0.33 | 12 000 |
| 5[a]......... | 73 | 7 | 20 | 18 000 | 7 | 45 | 0.25 | 0.33 | 11 000 |
| 6[a]......... | 70 | 5 | 25 | 15 000 | 5 | 40 | 0.25 | 0.33 | 10 000 |

[a] More difficult to handle in the foundry than the other grades on account of the segregation of lead. Can be improved by the addition of 1 per cent of nickel.

[1] The tension tests were made on "sand cast-to-size" test specimens.

[2] The compression tests were made on machined test specimens (sand castings) of 1 sq. in. sectional area, 1 in. high. The compression deformation limit is taken as the load producing a compression in the specimen of 0.001 in.

**Table III**

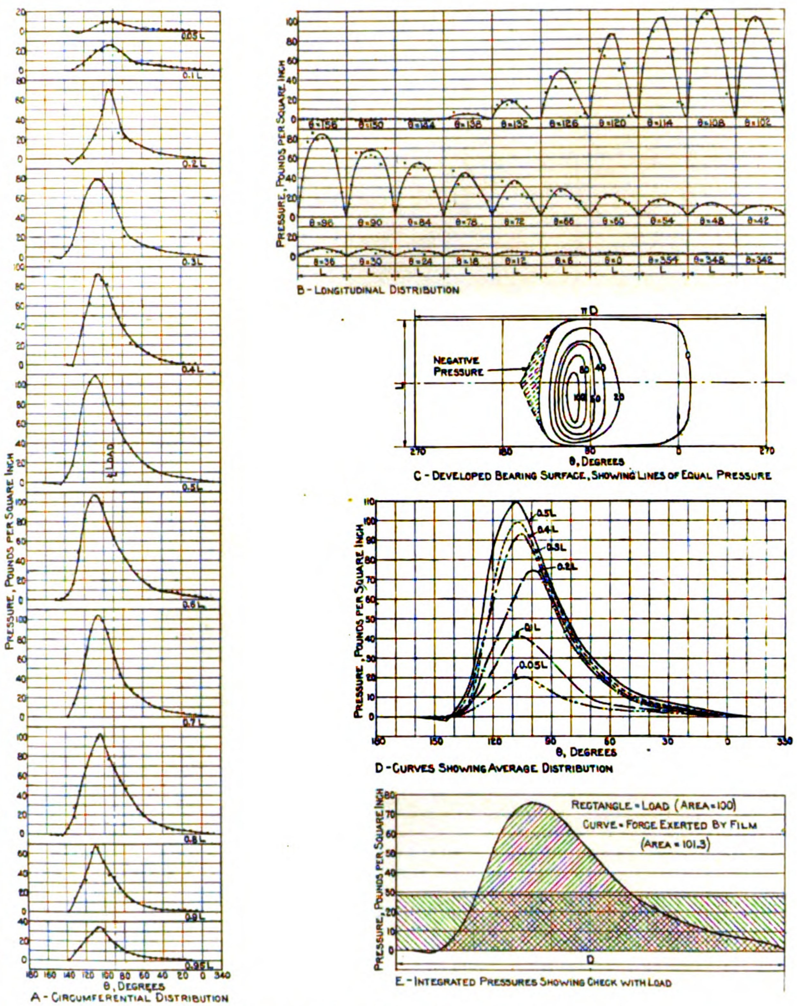



Fig. 6 Curves Showing Distribution of Pressure in Oil Film of Journal Bearing

(Diameter of journal, 0.8703 in.; length of bearing, 1 in.; clearance, 0.005 in.; load, 25 lb.; speed, 244 r.p.m.; viscosity of oil, 36.5 centipoises; $ZN/P$ = 310.)

**Fig. 1.**



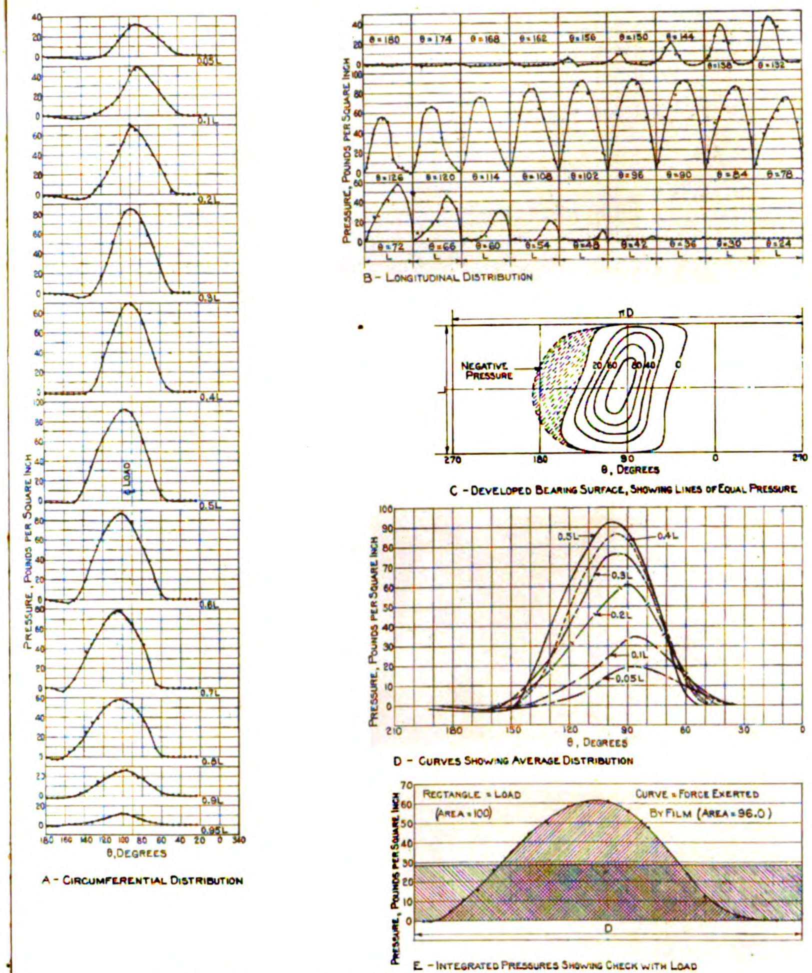



Fig. 10 Curves Showing Distribution of Pressure in Oil Film of Journal Bearing

(Diameter of journal, 0.8733 in.; length of bearing, 1 in.; clearance, 0.002 in.; load, 25 lb.; speed 531 r.p.m.; viscosity of oil, 18.4 centipoises; $ZN/P$ = 342.)

**Fig.2.**

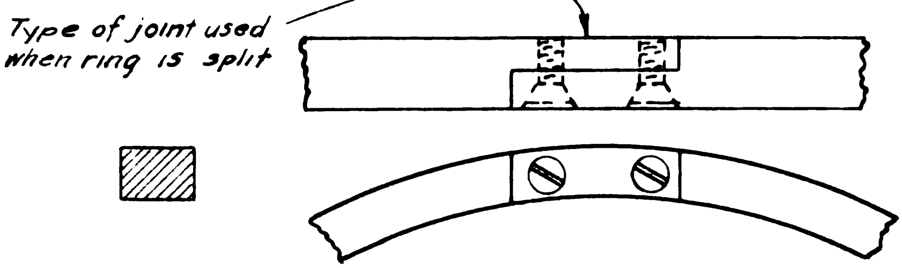



FIG. 3 (a)

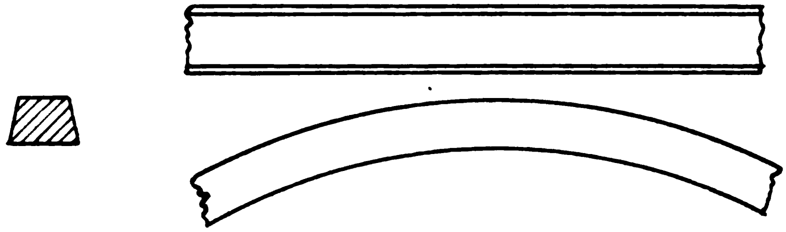


FIG. 3 (b)

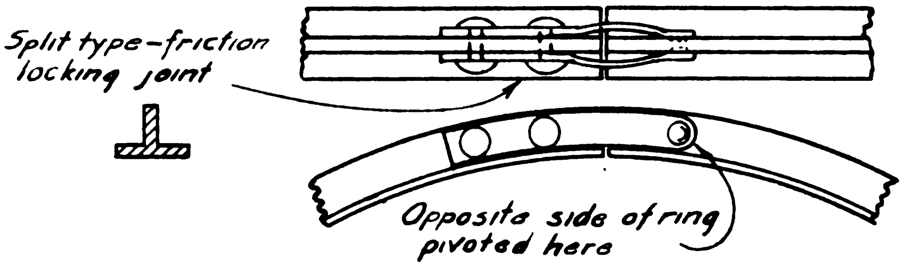



FIG. 3 (c)

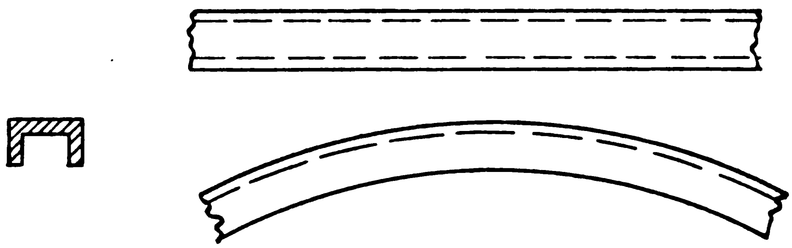


FIG. 3 (d)

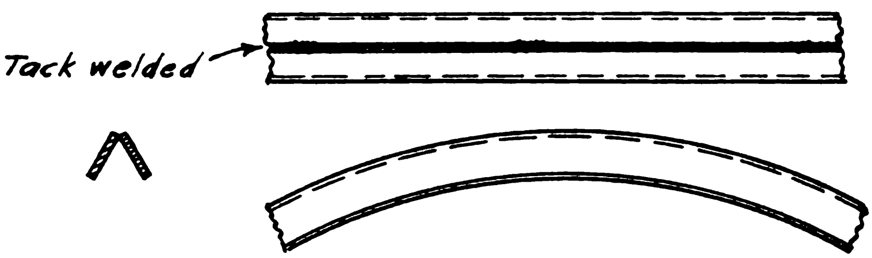



FIG 3 (e)

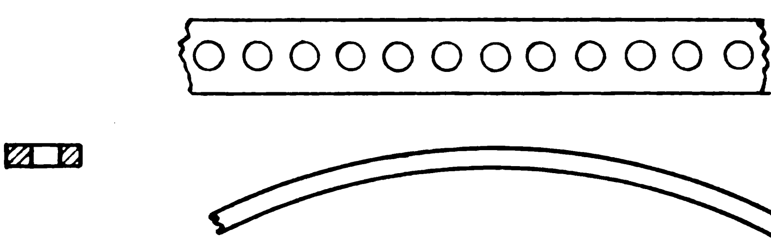


FIG. 3(f)

MACHINE-SHOP PRACTICE MSP-56-2 899

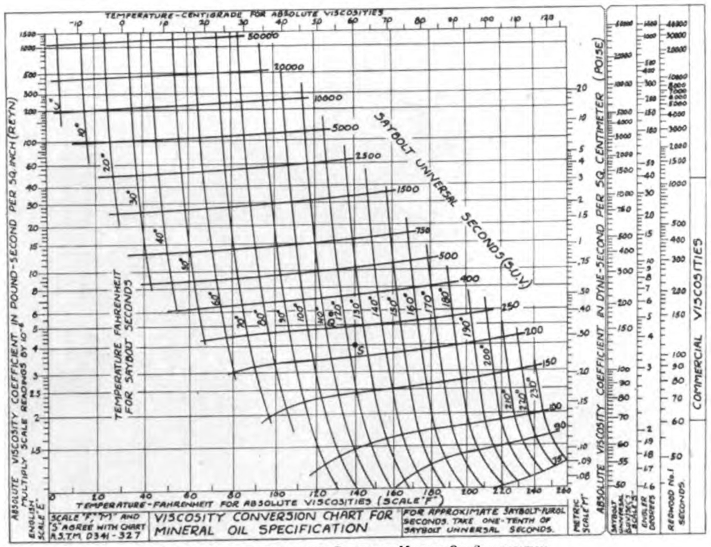



Fig. 6 Viscosity Conversion Chart for Mineral-Oil Specification

**Fig. 4.**

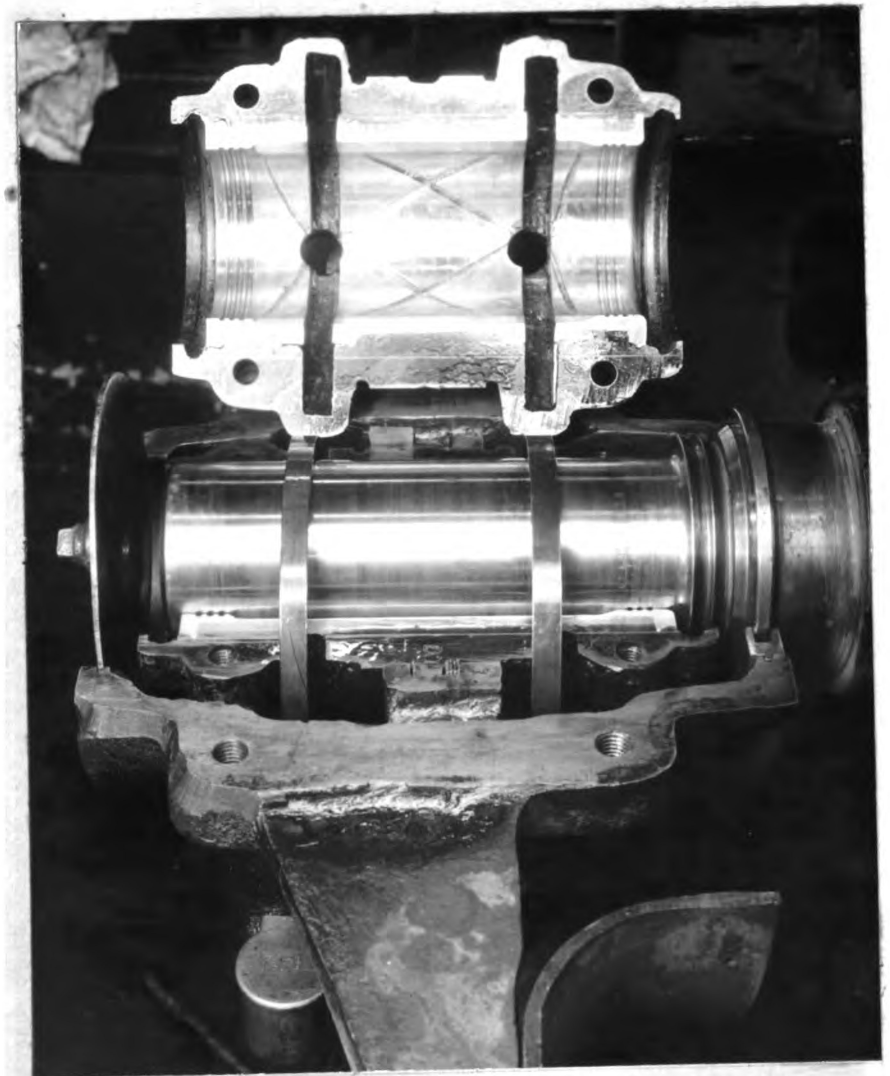


**Fig. 5. Babbitt Lined, Split Type, Self-Aligning, Sealed Bearing. 3½" Journal.**

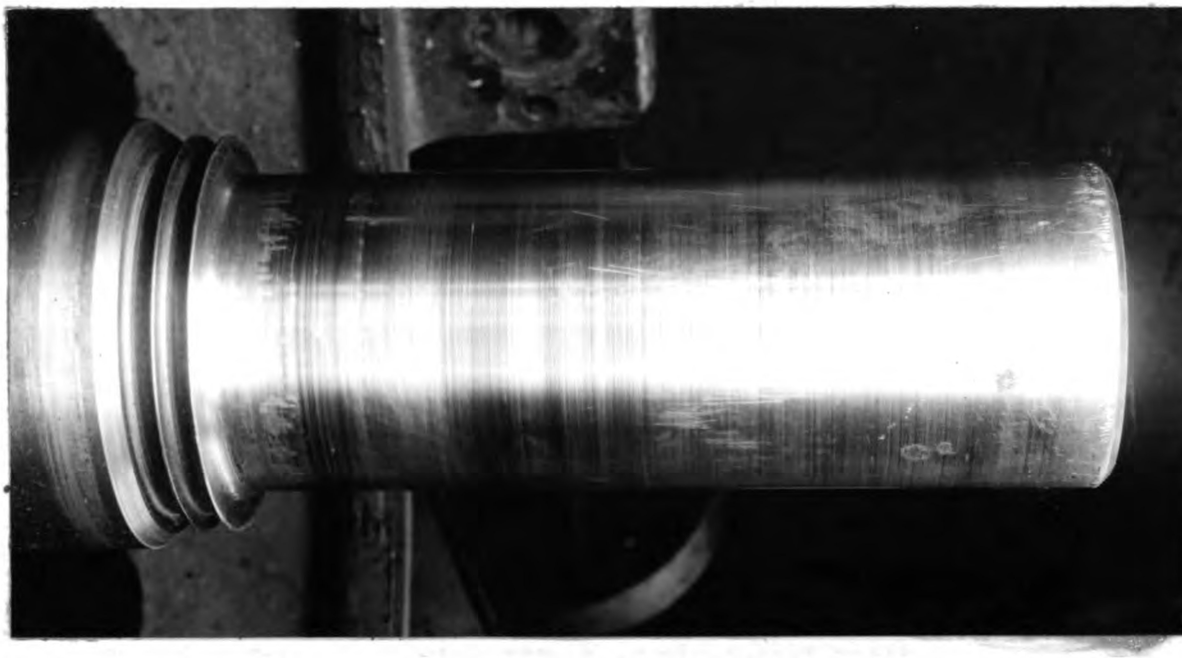


Fig. 6. Journal Badly Scored by Grit.

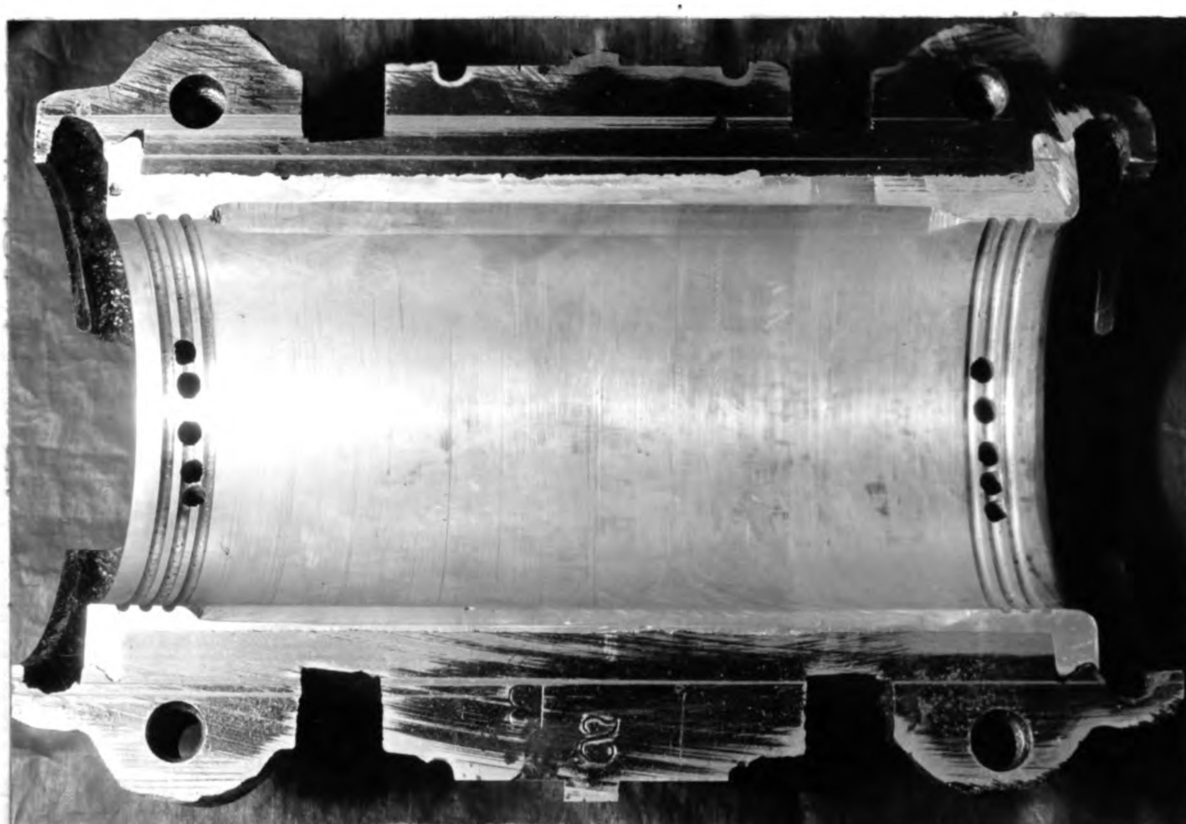


Fig. 7. Babbitt Lined Bearing Scored by Grit.

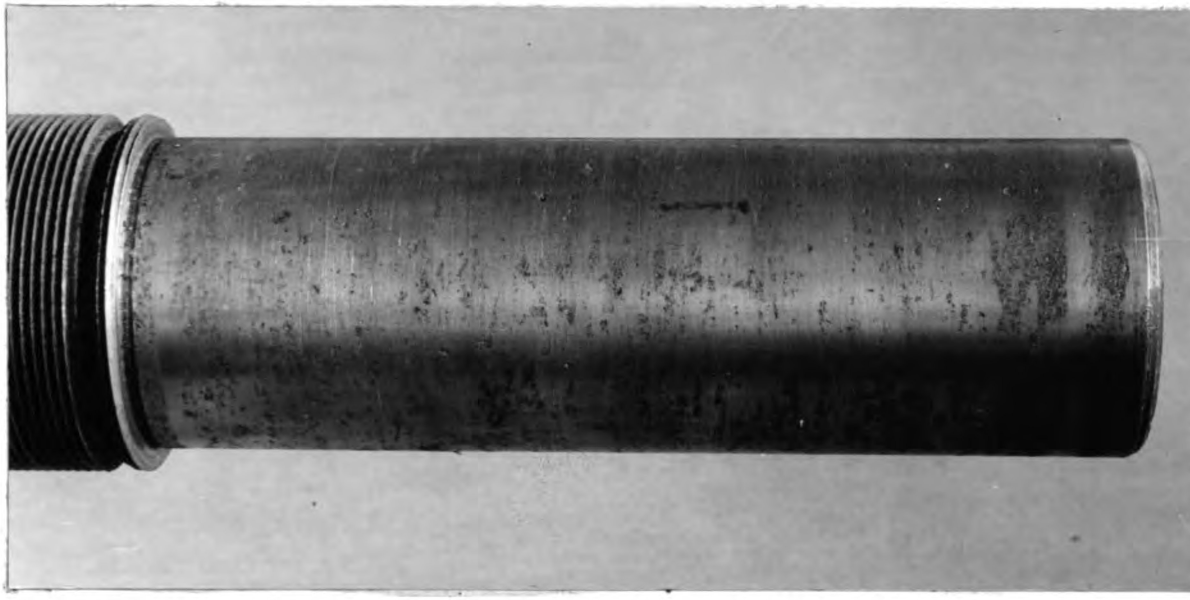


Fig. 8. Journal Showing Effect of Electrical Pitting.

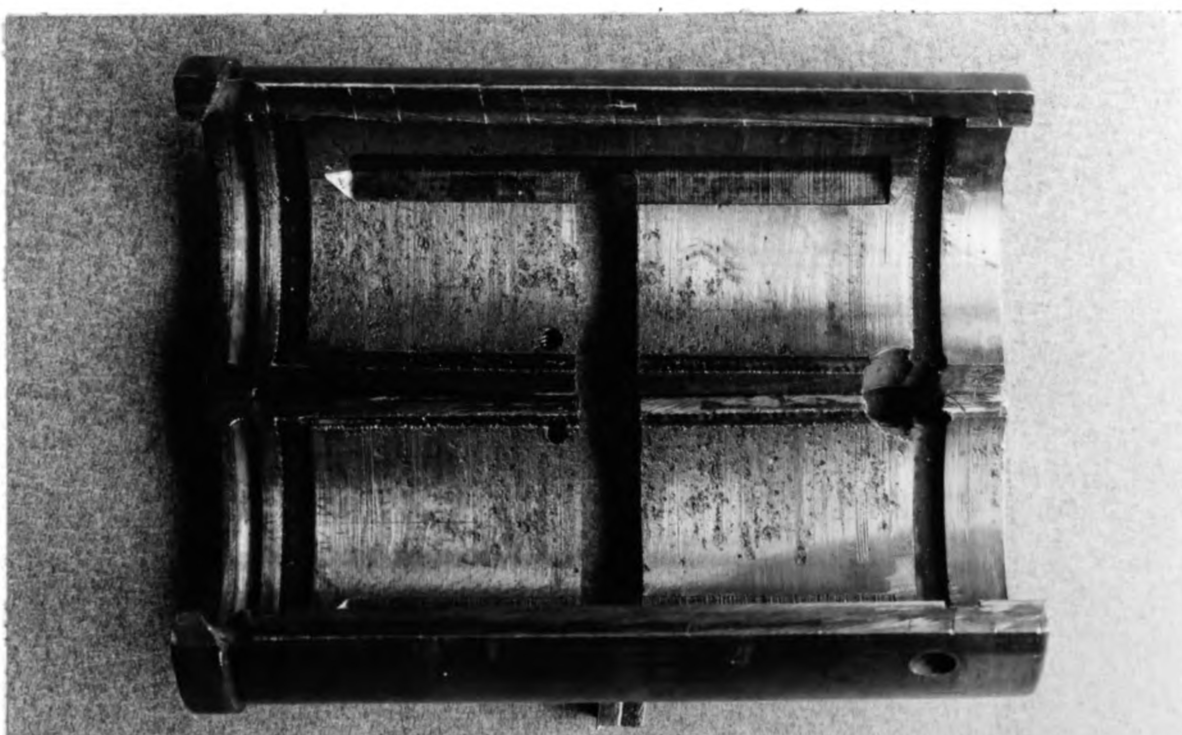


Fig. 9. Bronze Bearing Showing Effect of Electrical Pitting.

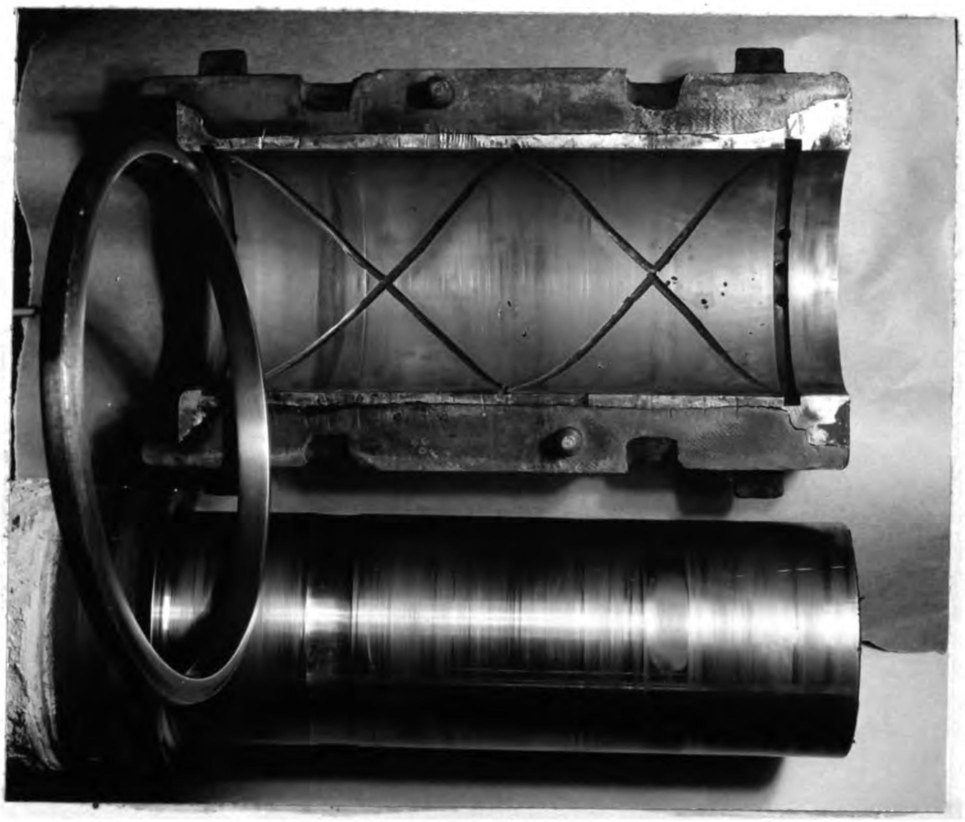


Fig. 10. Damaged Journal and Bearing Due to Oil Ring.

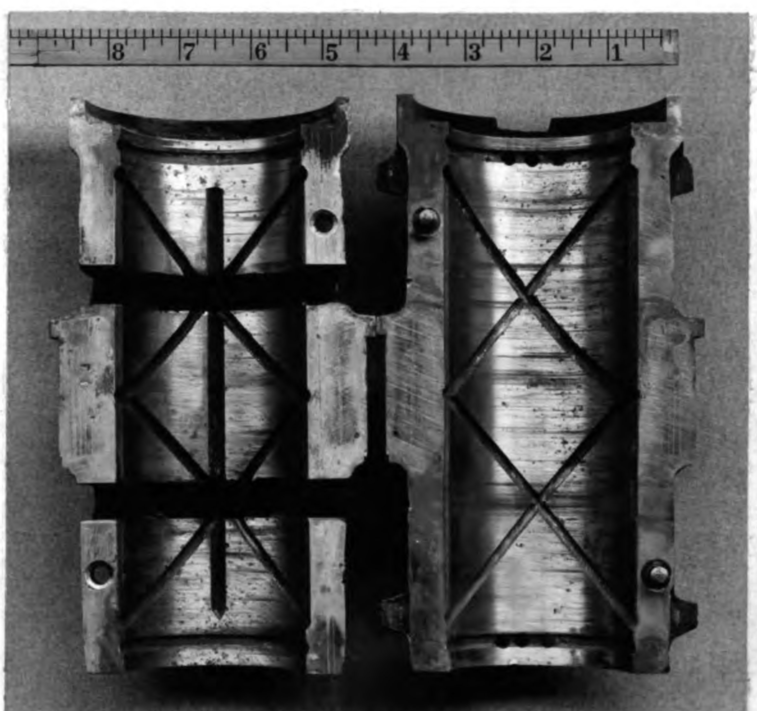



Fig. 11. Excessively Grooved Bronze Bearing.

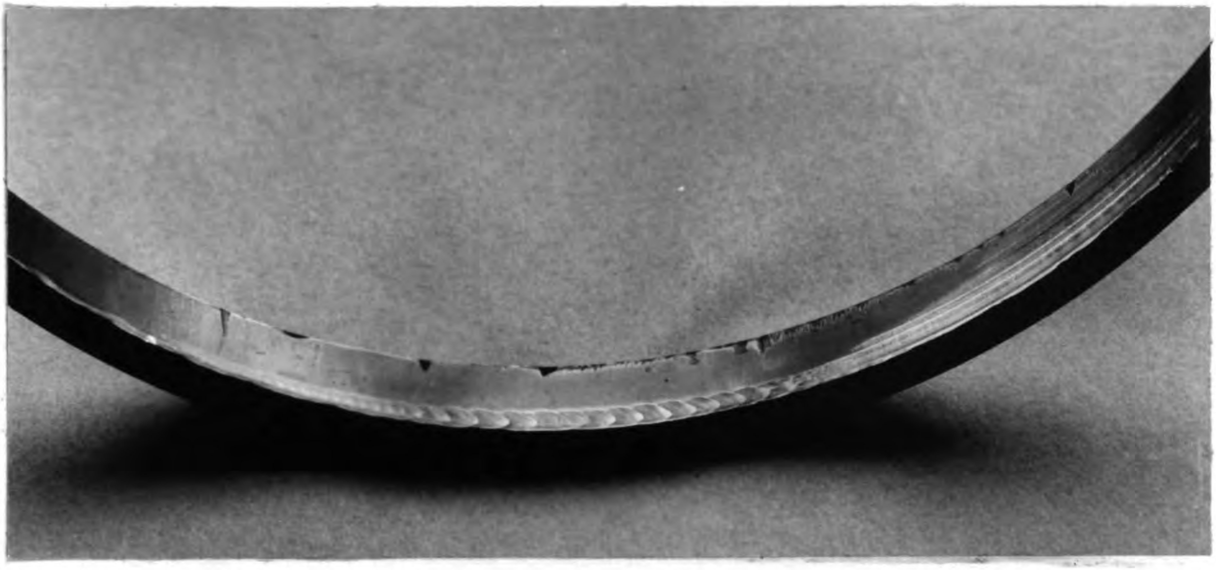


Fig. 12. Rough Oil Ring Showing Effects of Slipping on Journal.

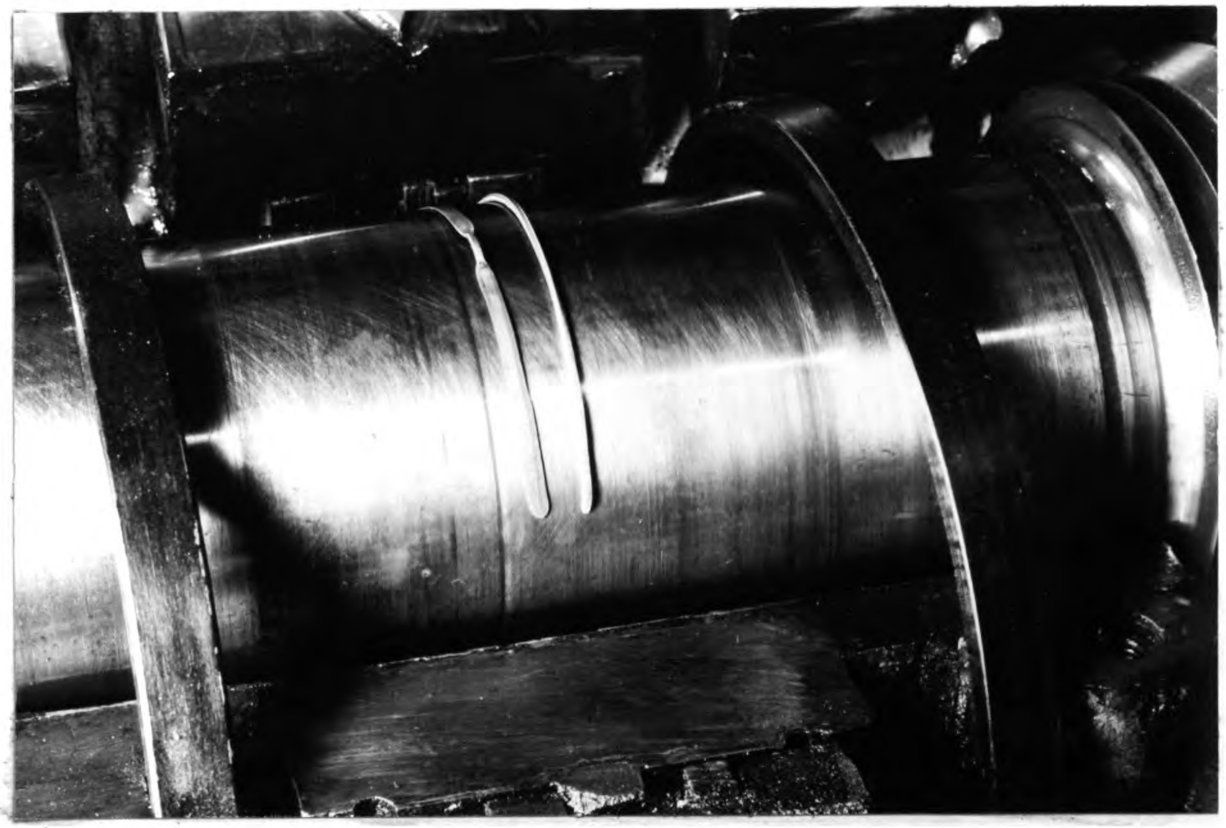


Fig. 13. Lead Wires Before and After Crushing for Clearance Measurement.

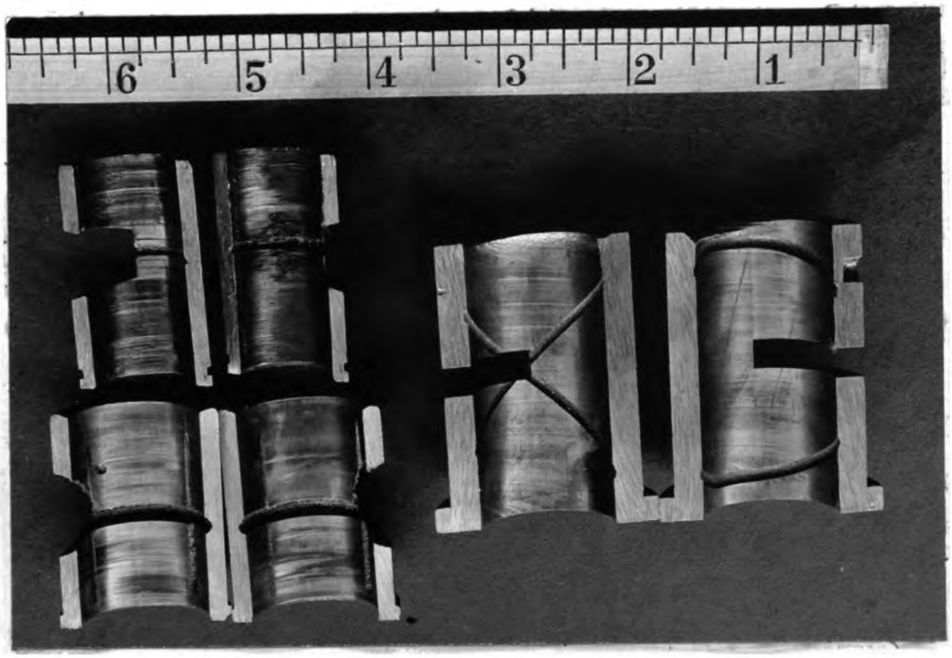



Fig. 14. Small Bronze Bearings Having Very Poorly Designed Oil Grooves.

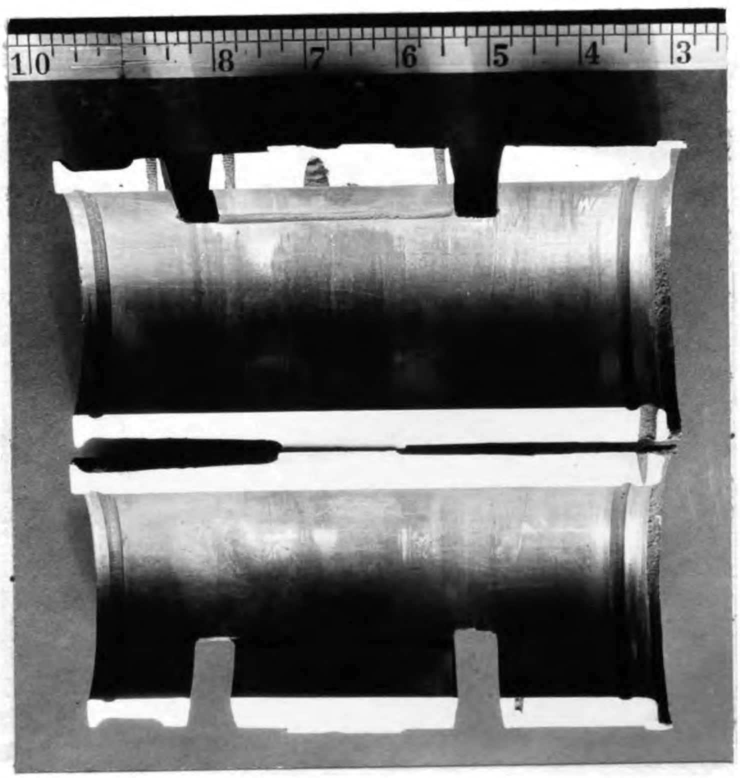



Fig. 15. Bronze Bearing Having Inadequate Oil Distributing Groove.

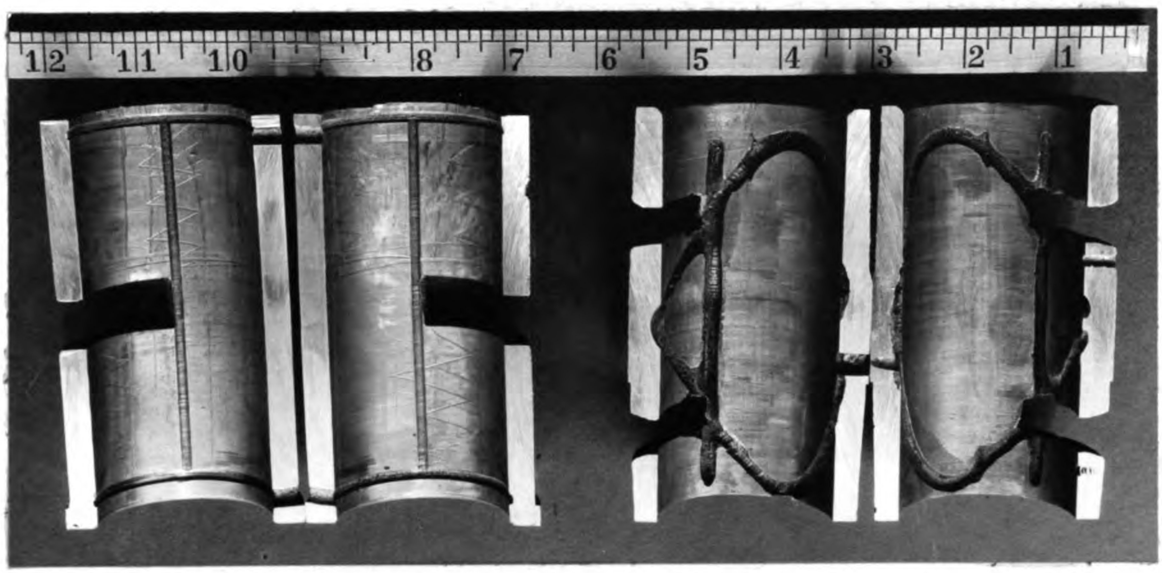



**Fig. 16. Lft.-Narrow Sharp Edged Dist. Groove Running Into Drain Grooves. Rt.- Bad Pear Shaped Groove Connecting High and Low Pressure Areas. Bronze Bearings.**

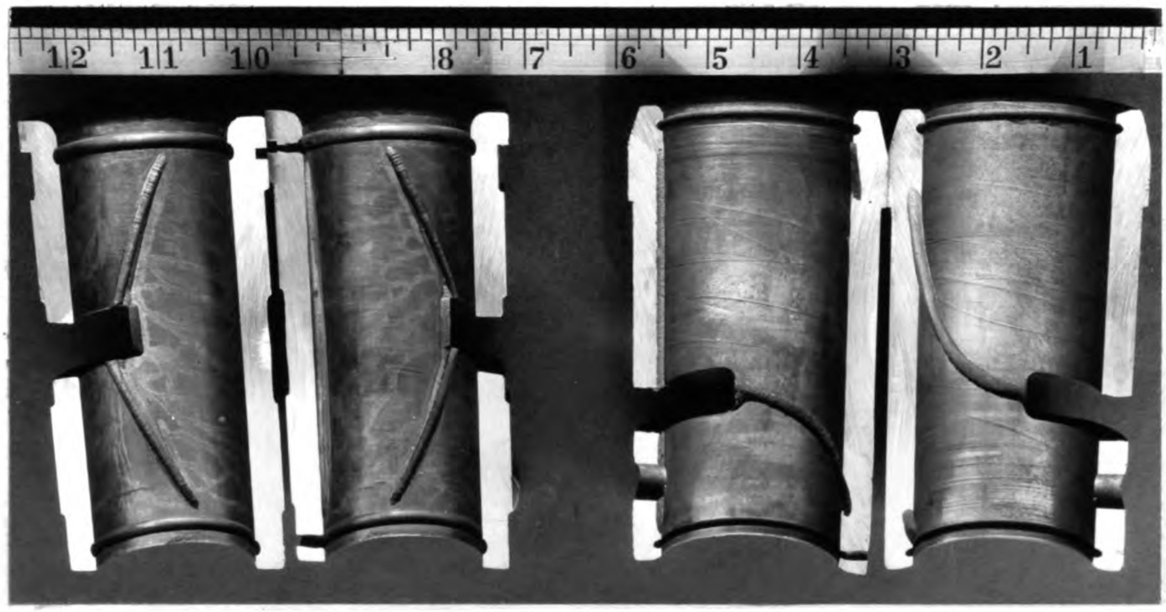



**Fig. 17. Lft.- Slanting Grooves Run Into Ring Groove Too High. Rt. - Grooves Connect High and Low Pressure Areas. One Adds Oil While Other Robs.**

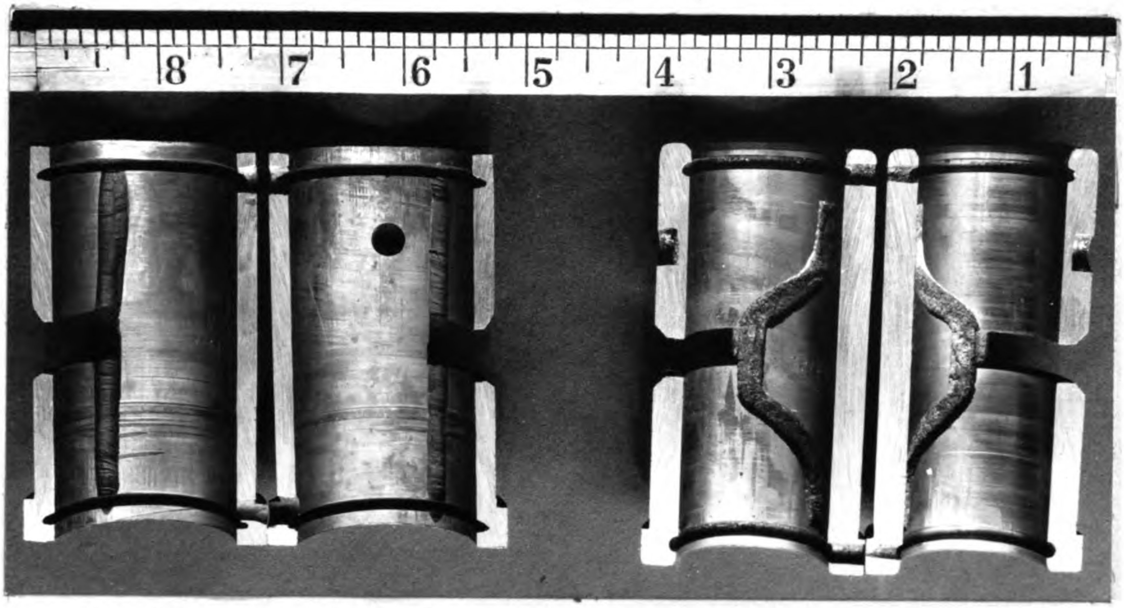



Fig. 18. Lft.- Hand Cut Distributing Grooves Running Into End Grooves. Rt.- Good Oil Scoops, but Very Bad Dist. Grooves.

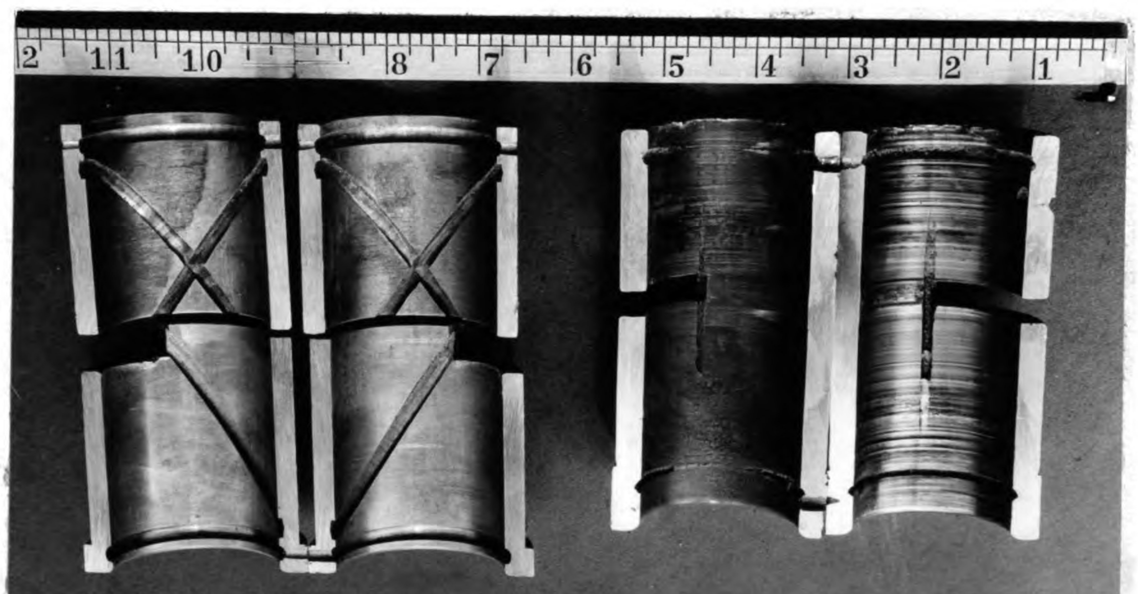



Fig. 19. Lft.- Extremely Sharp Edged Grooves Connecting High and Low Pressure Areas. Rt.- Badly Scored Bearing with Dist. Grooves Filled with Deposits.

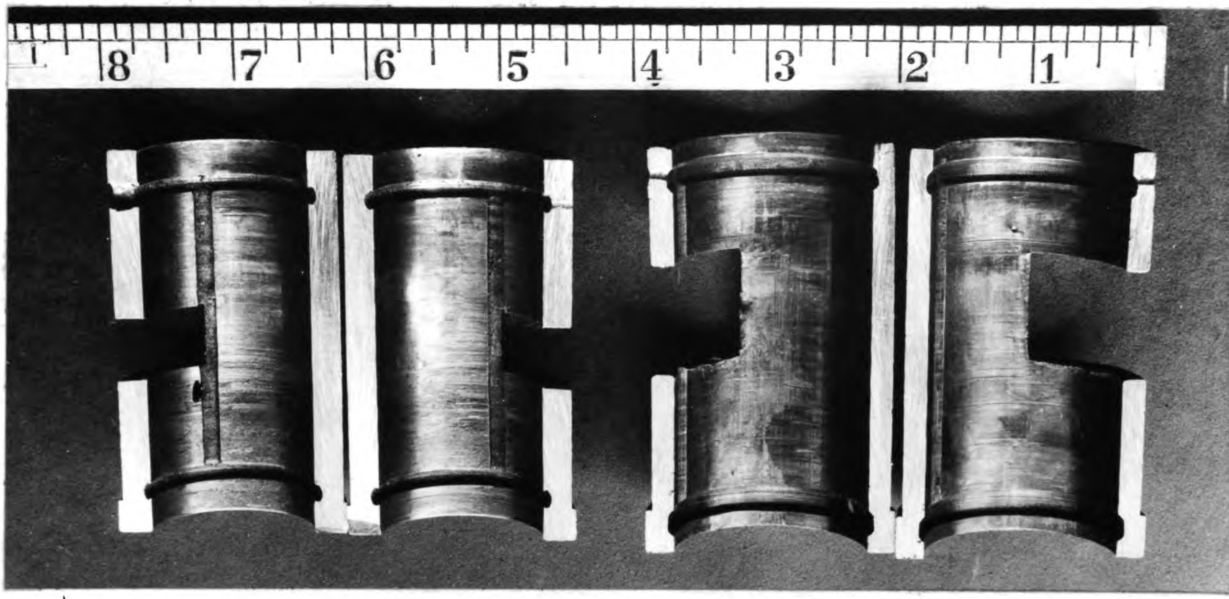



Fig. 20. Lft.- Good Oil Scoops. Sharp Edged Dist. Grooves Run into End Grooves. Rt.- Dist. Grooves in Wrong Part of Bearing.

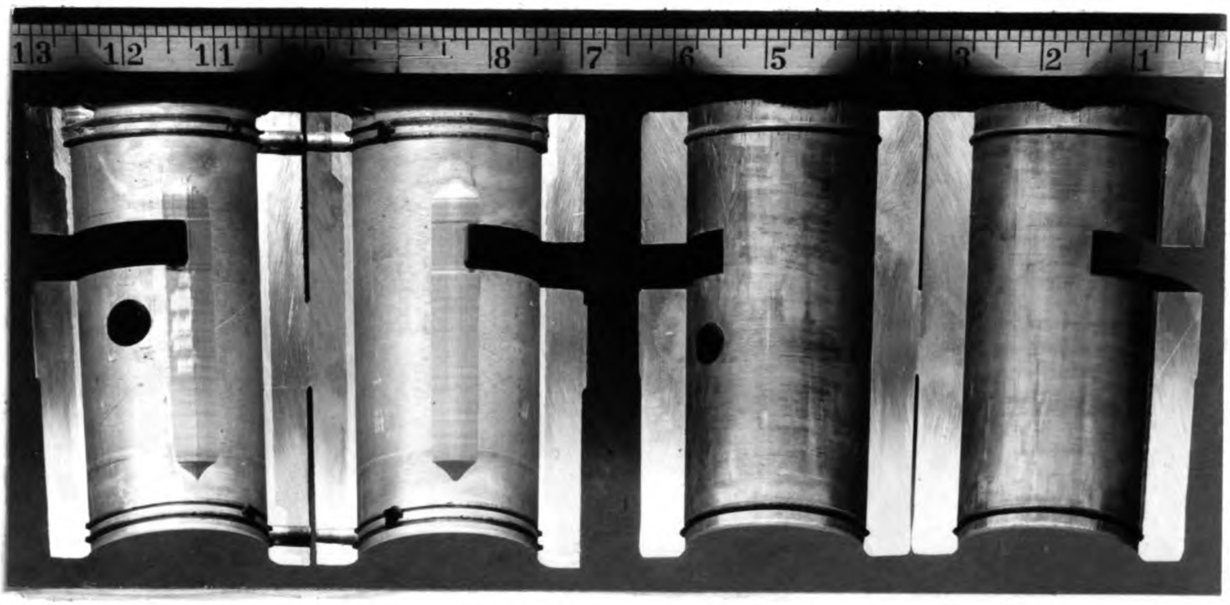


Fig. 21. Lft.- Babbitt Lined Bearing With Good Dist. Grooves, but Slightly Short. Rt.- Bronze Bearing Without Dist. Grooves.

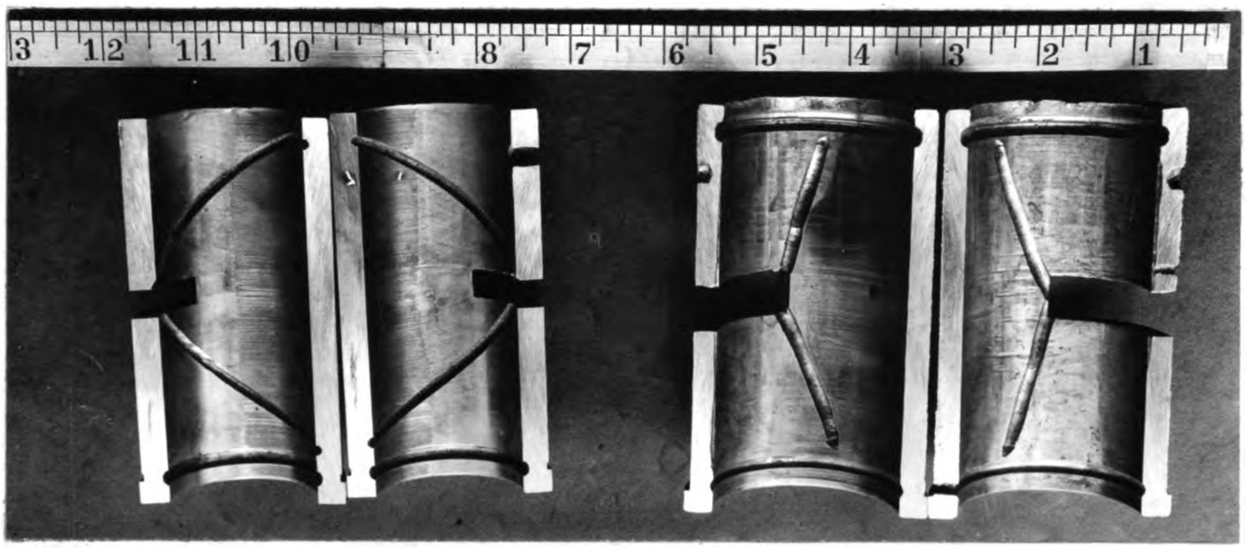



Fig. 22. Lft.- No Oil Scoops. Dist. Grooves Running To Top of Bearing. Rt.- Oil Scoops Indefinite, Grooves too Sharp.

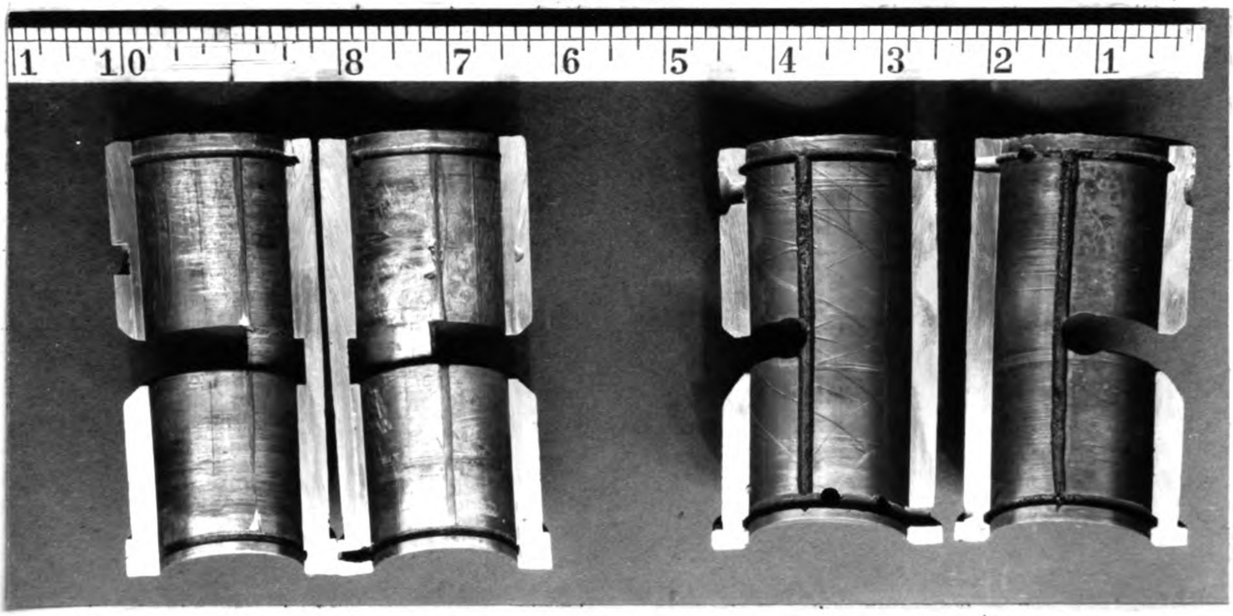



Fig. 23. Lft.- Dist. Grooves too Small. Broad Central Groove Very Bad. Rt.- Scoring Caused by Burr on Shaft.

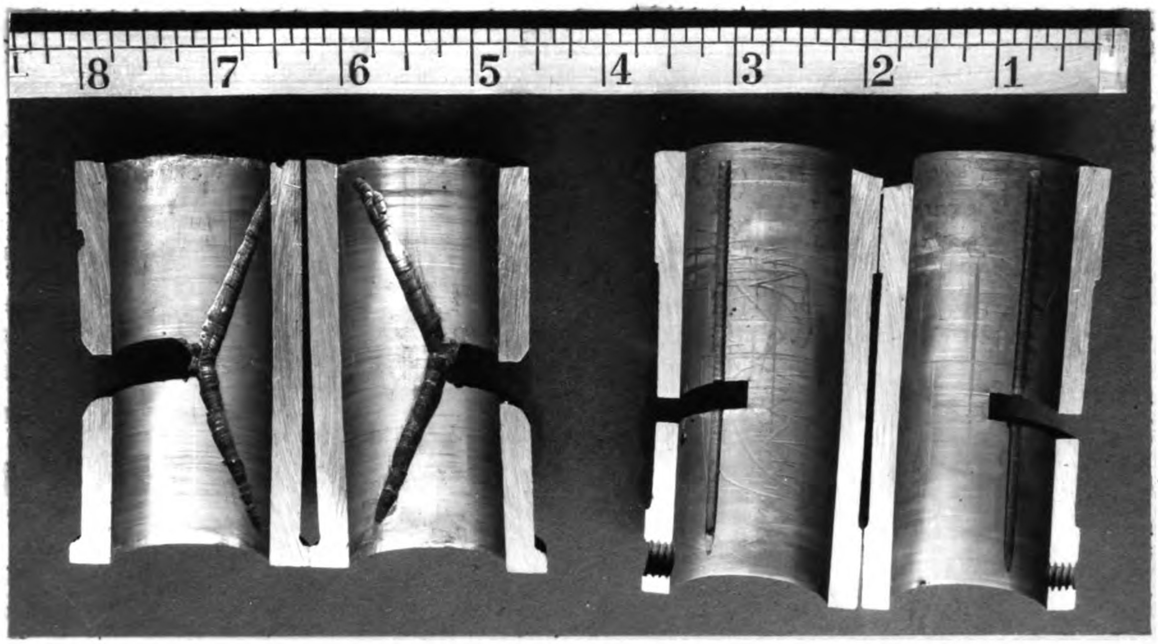



Fig. 24. Lft.- Adequate Oil Scoops. Hand Cut Grooves, Haggled but Fair. Rt.- Sharp Narrow Dist. Grooves Too High.

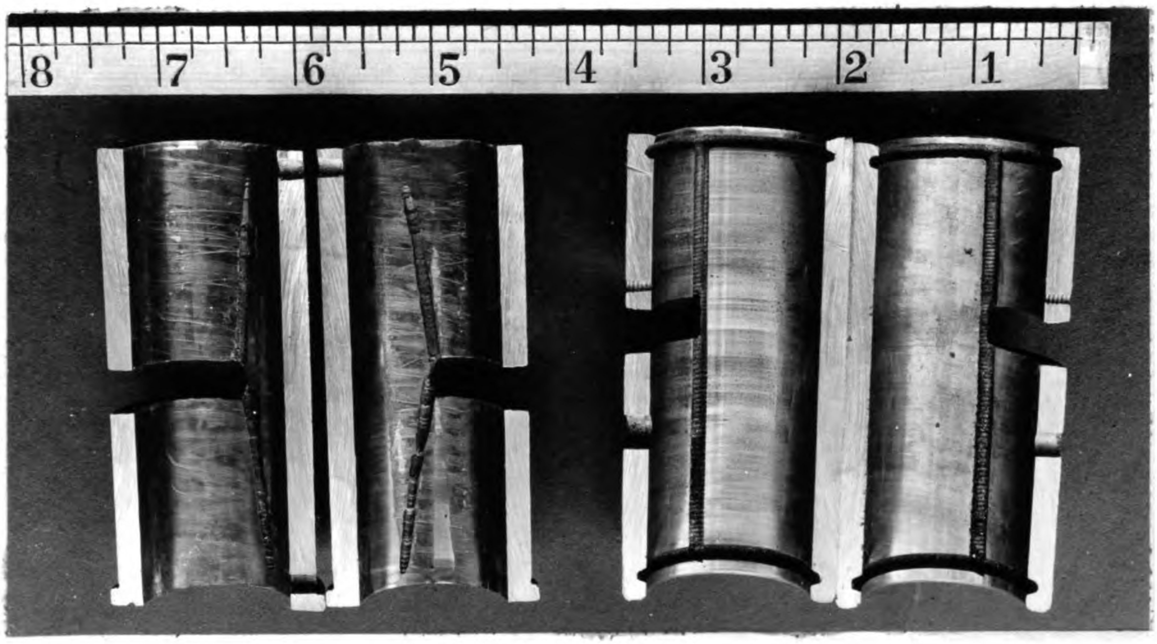



Fig. 25. Lft.- Hand Cut Grooves Too Shallow and Narrow. Rt.- Sharp Edged Dist. Grooves Running into End Grooves.

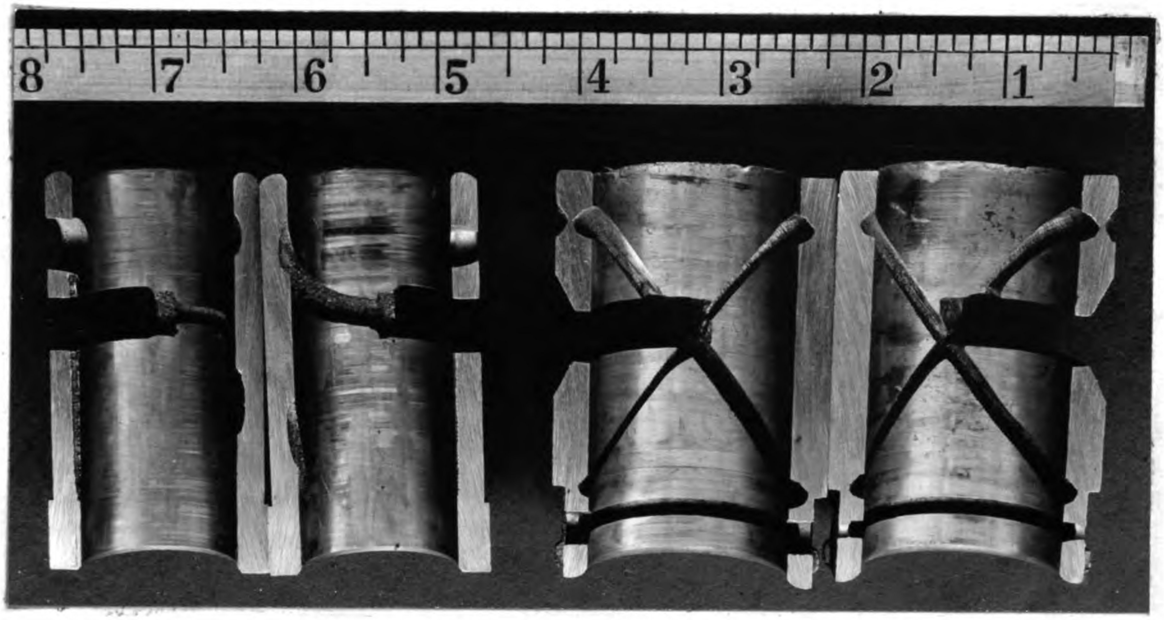



Fig. 26. Lft.- Good Oil Scoops, but Very Bad Dist. Grooves.
Rt.- Extremely Bad Grooves Having Very Sharp Edges.

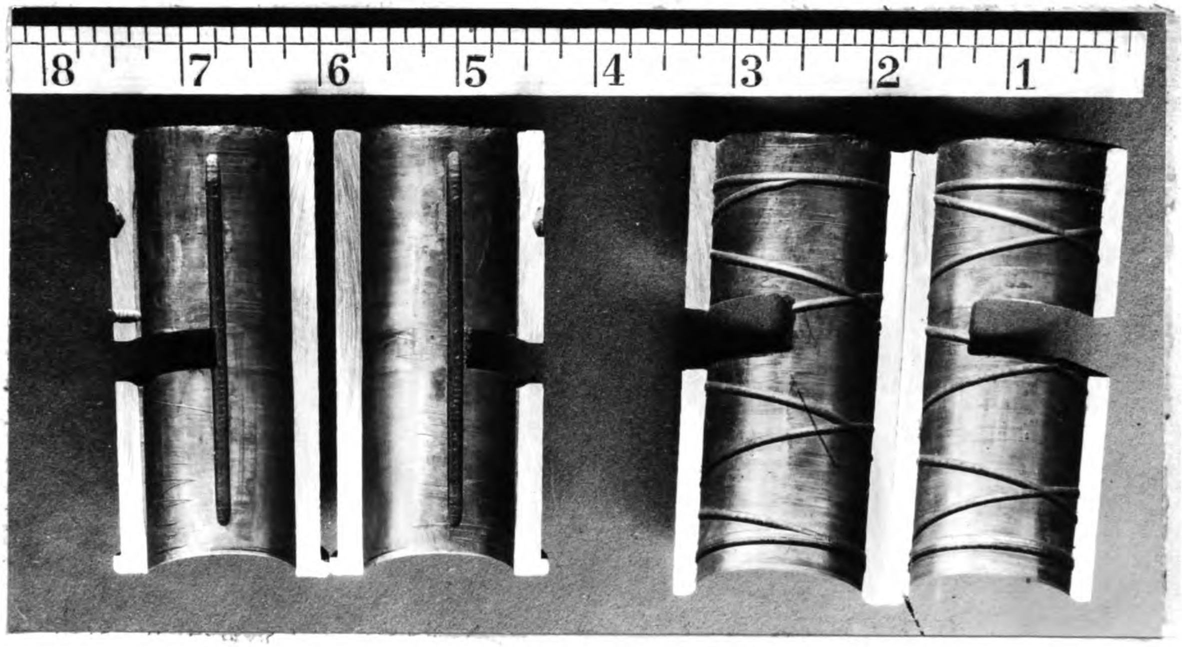



Fig. 27. Lft.- No End Grooves. Only one Scoop.
Rt.- Elaborate Spiral Grooves of Little Value.

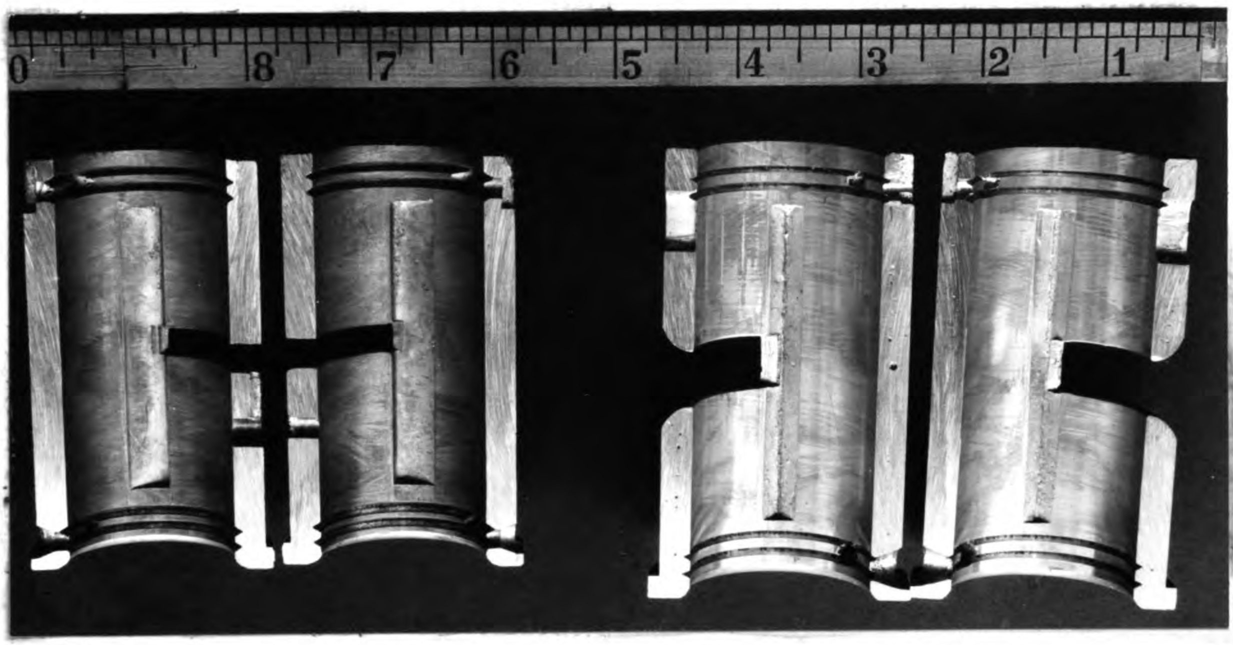



Fig. 28. Lft.- Die Cast Babbitt Bearing Having Excellent Oil Grooves. Rt.- Another Die Cast Babbitt Bearing Having Ratchet Type Dist. Grooves of Excellent Design.

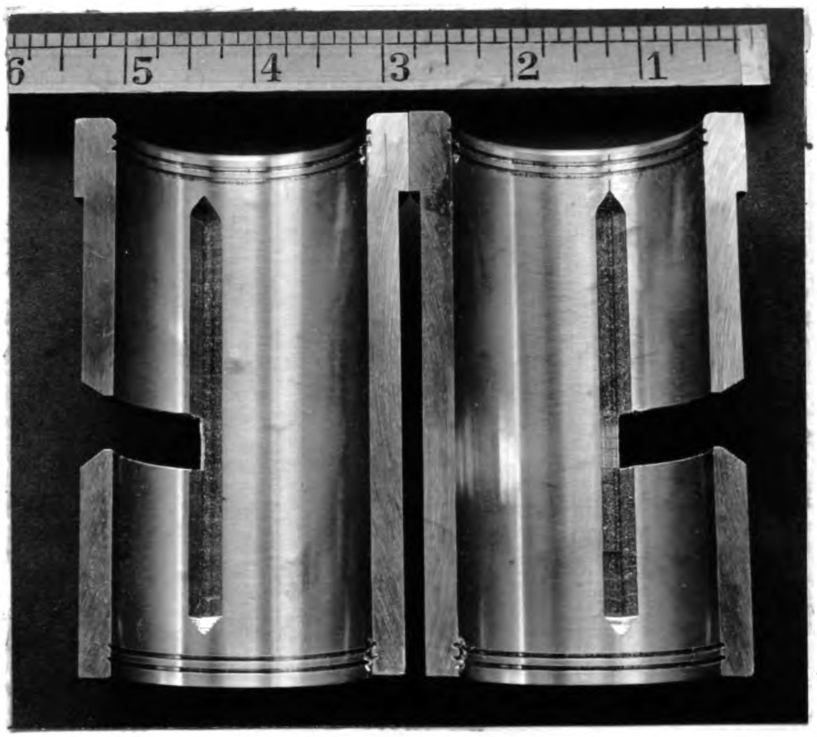
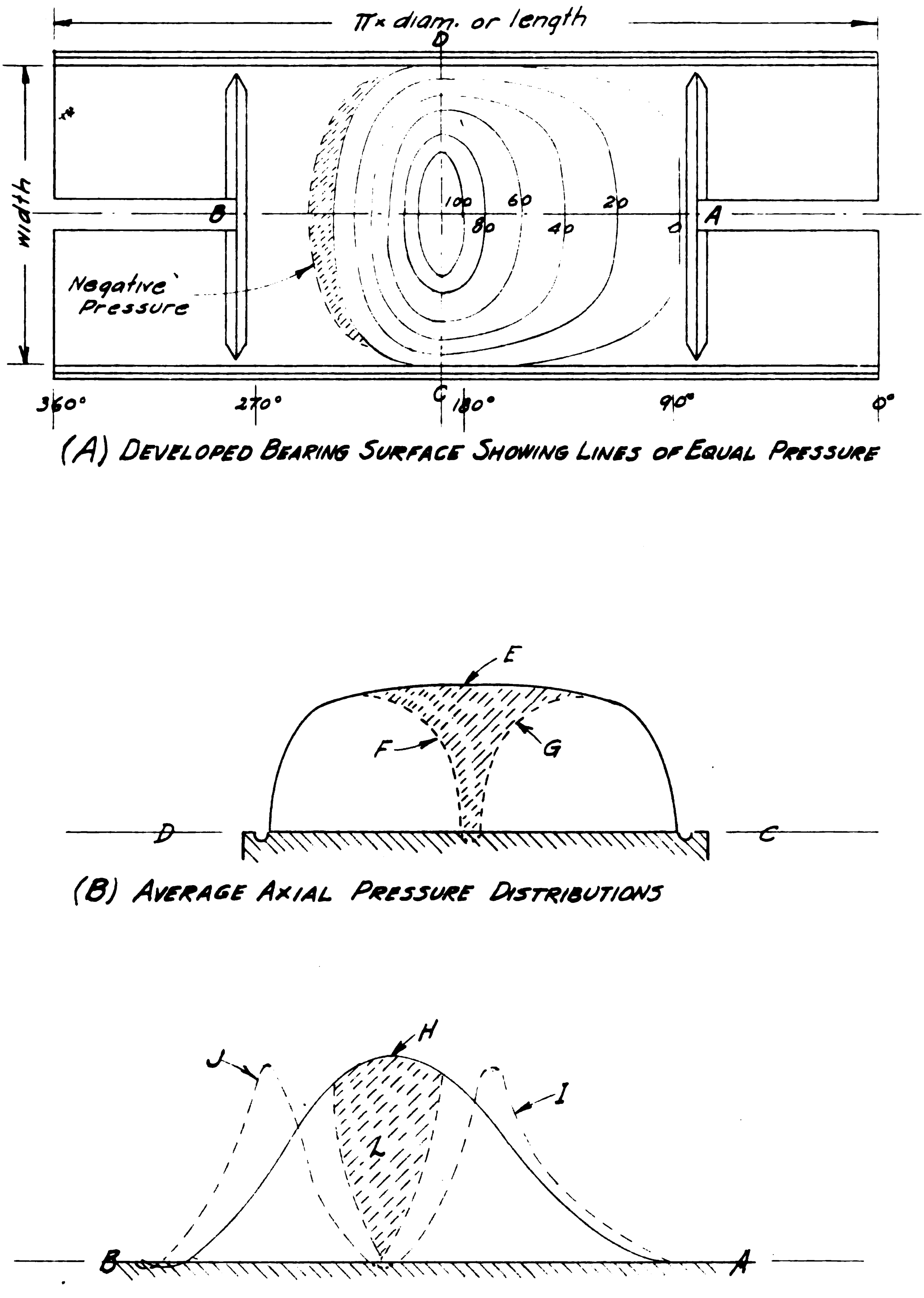



Fig. 29. Bronze Bearing Showing Oil Grooving System Adopted by C.P.Co.



π × diam. or length
D
width
B
A
100
80
60
40
20
0
Negative
Pressure
C
360°
270°
180°
90°
0°
(A) DEVELOPED BEARING SURFACE SHOWING LINES OF EQUAL PRESSURE
E
F
G
D
C
(B) AVERAGE AXIAL PRESSURE DISTRIBUTIONS
H
J
I
L
B
A
(C) AVERAGE CIRCUMFERENTIAL PRESSURE DISTRIBUTIONS


FIG. 30

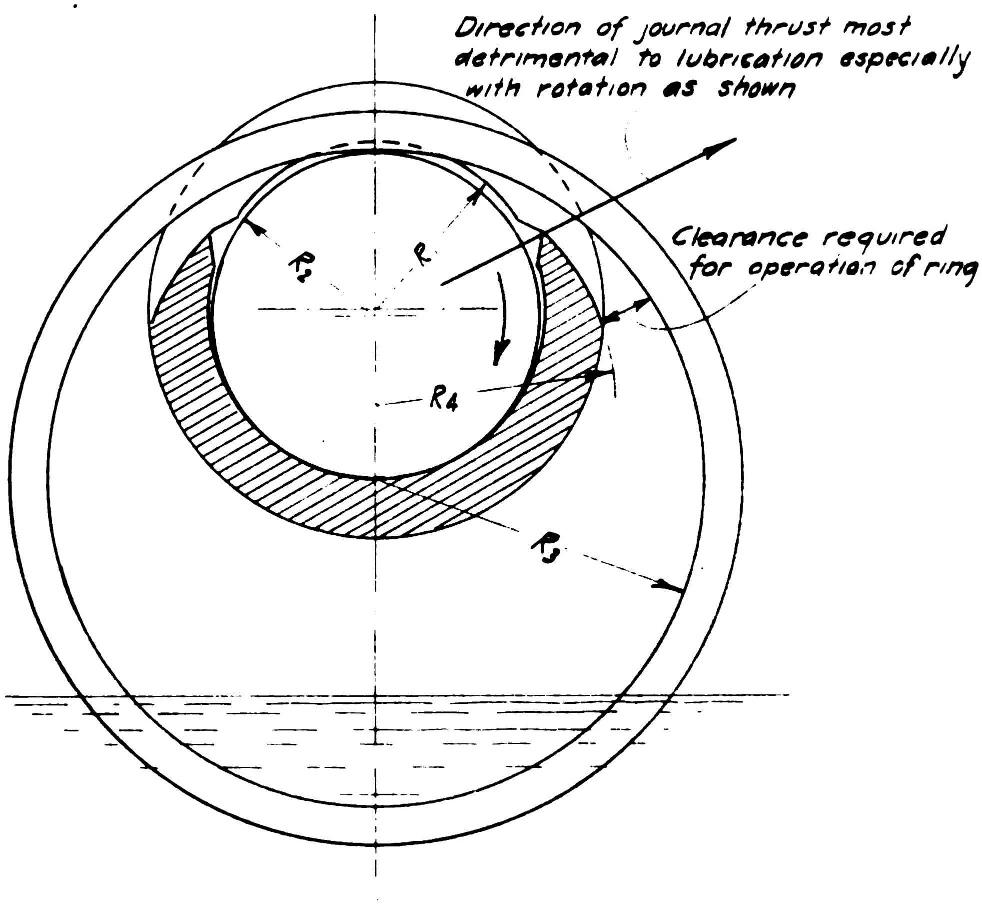
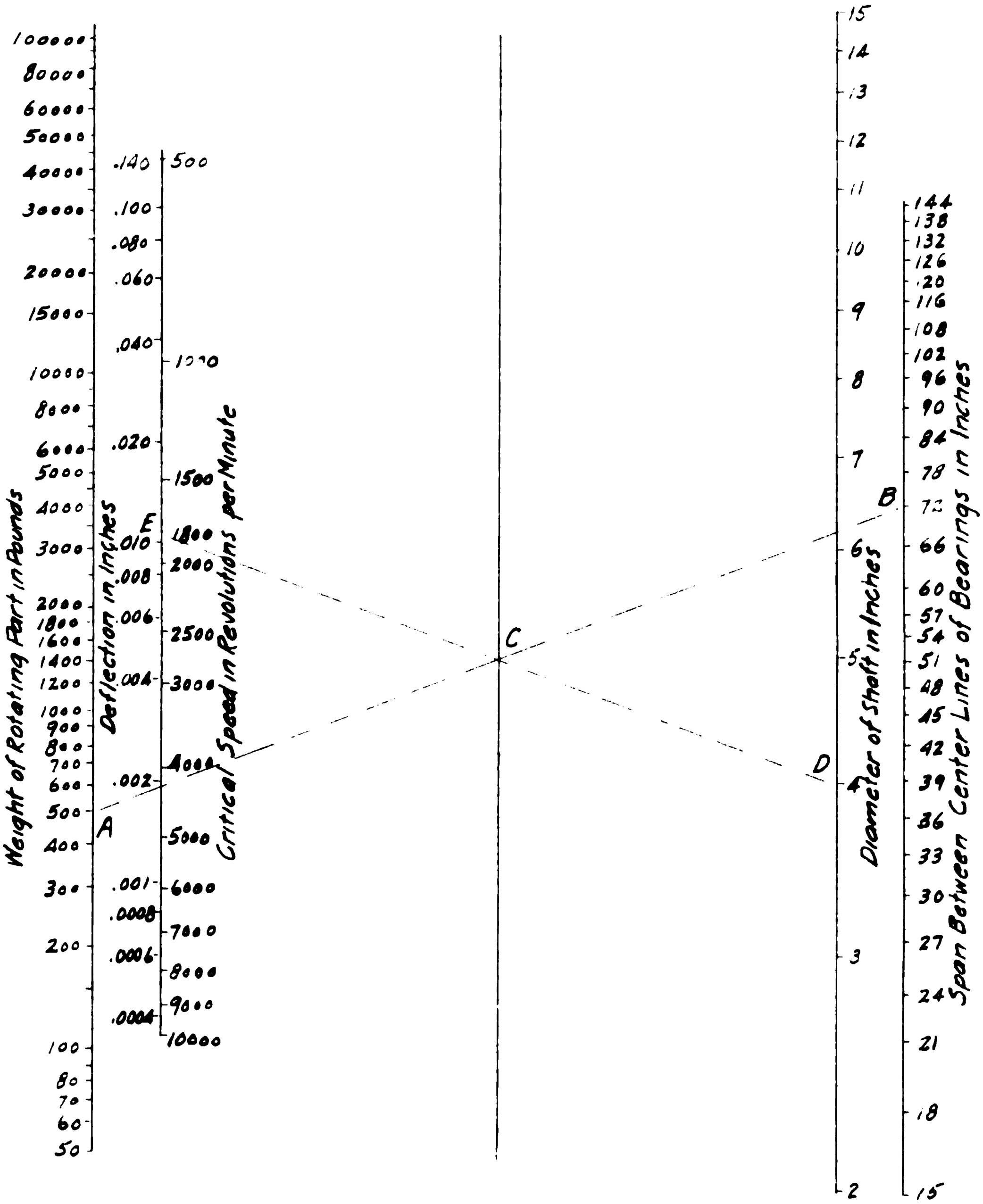

Direction of journal thrust most detrimental to lubrication especially with rotation as shown
Clearance required for operation of ring
R
R2
R4
R3

Fig. 31

APPROXIMATE CRITICAL SPEED AND DEFLECTION




Locate the weight point, A and the span point, B.
A straight line gives the common point, C, for any shaft.
For any shaft diameter, as point D, a line through point C, gives point E, the critical speed and deflection.

CRITICAL SPEED CHART

FIG. 32

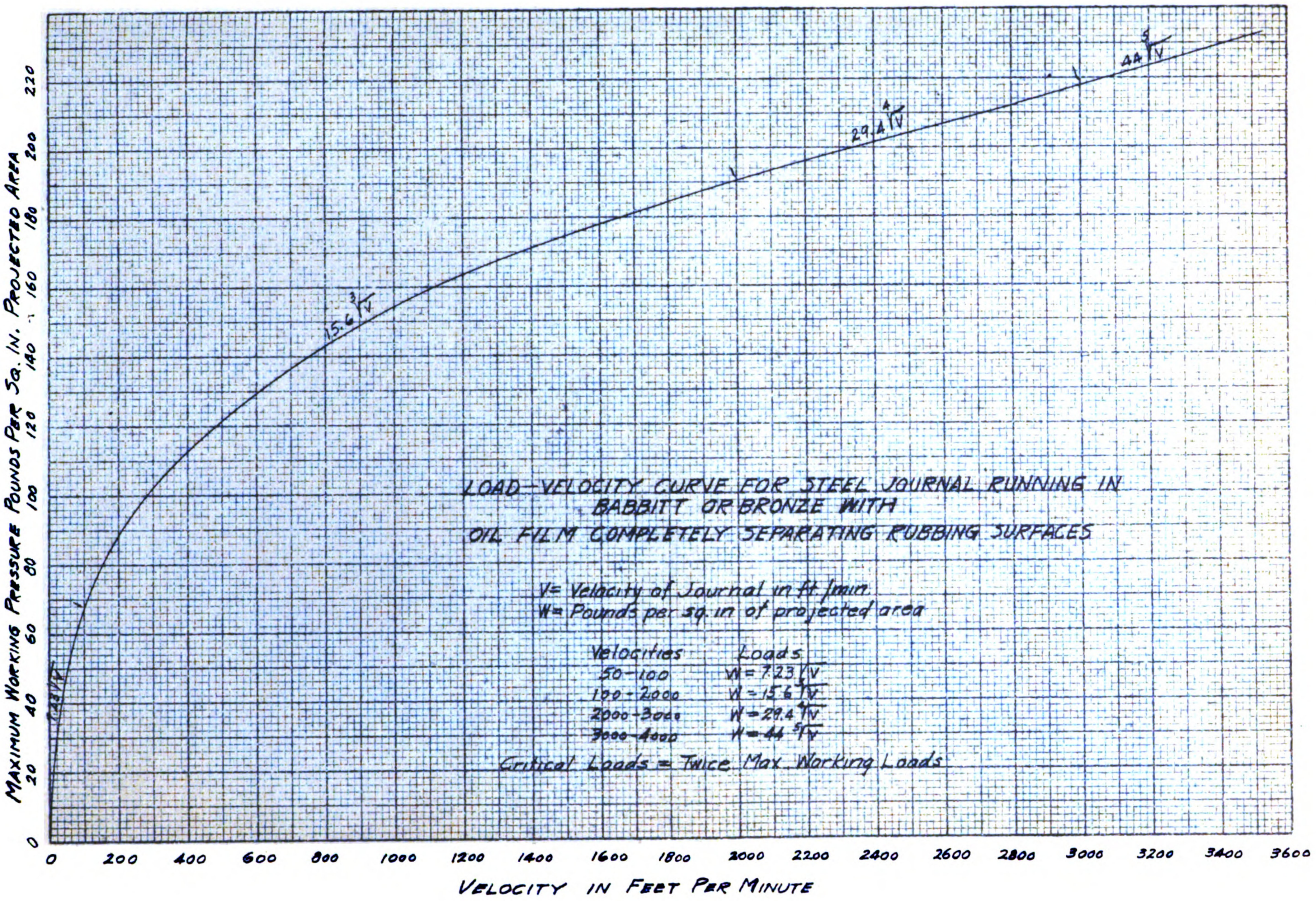
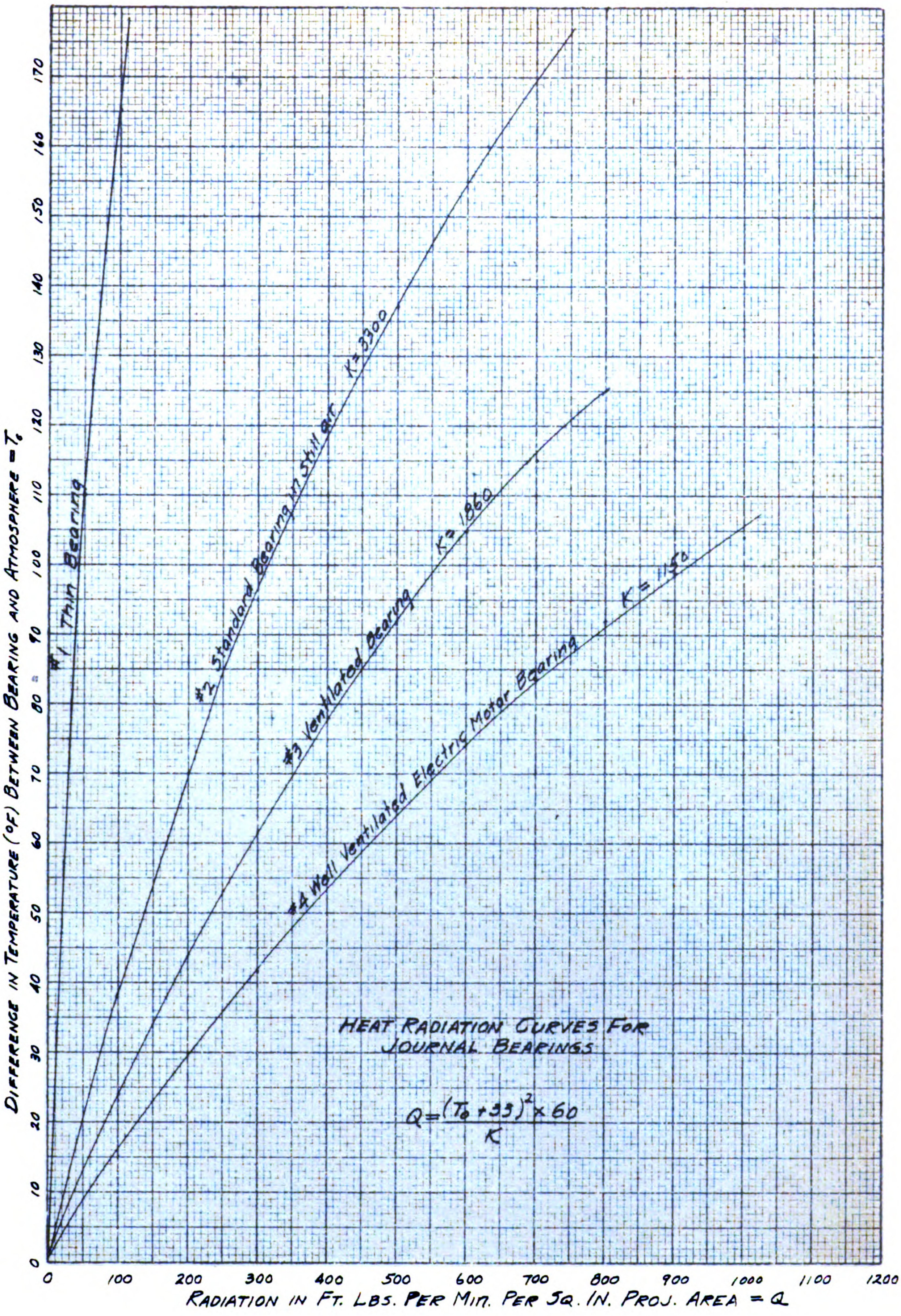
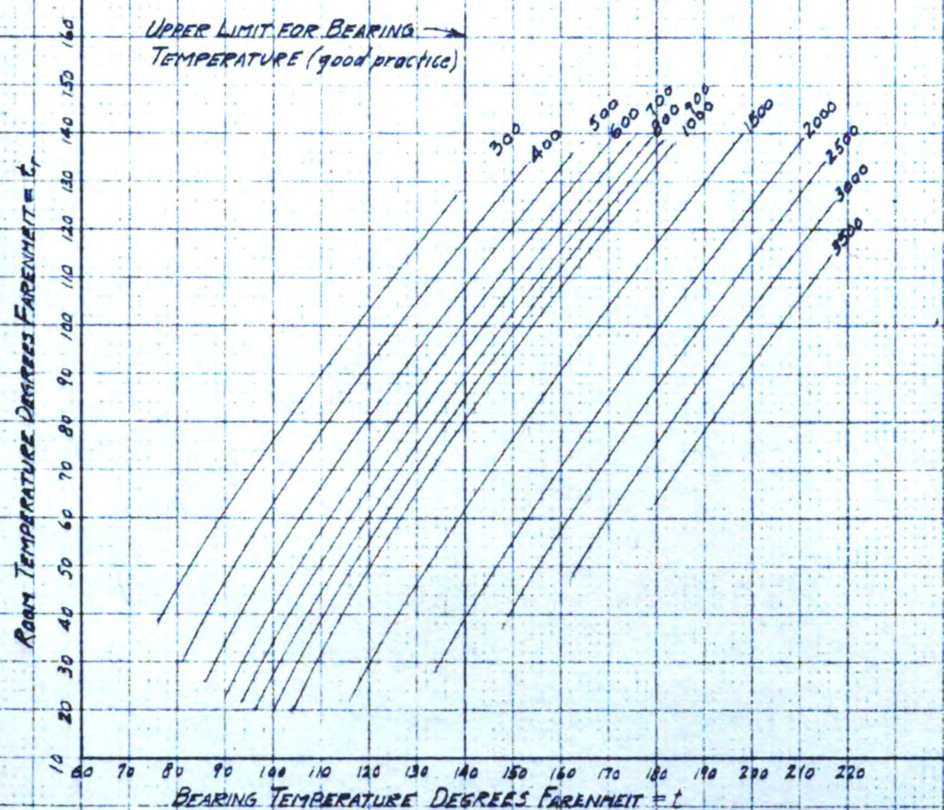

LOAD-VELOCITY CURVE FOR STEEL JOURNAL RUNNING IN
BABBITT OR BRONZE WITH
OIL FILM COMPLETELY SEPARATING RUBBING SURFACES
V= Velocity of Journal in ft./min.
W= Pounds per sq. in of projected area
Velocities
Loads
50-100
W = 7.23 √V
100-2000
W = 15.6 ∛V
2000-3000
W = 29.4 ∜V
3000-4000
W = 44 ⁵√V
Critical Loads = Twice Max. Working Loads
15.6 ∛V
29.4 ∜V
44 ⁵√V
7.23 √V
Maximum Working Pressure Pounds Per Sq. In. Projected Area
0
20
40
60
80
100
120
140
160
180
200
220
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
2600
2800
3000
3200
3400
3600
Velocity in Feet Per Minute

FIG. 33



Difference in Temperature (°F) Between Bearing and Atmosphere = $T_o$
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
#1 Thin Bearing
#2 Standard Bearing in Still Air K = 3300
#3 Ventilated Bearing K = 1860
#4 Well Ventilated Electric Motor Bearing K = 1150
Heat Radiation Curves For Journal Bearings
$Q = \frac{(T_o + 33)^2 \times 60}{K}$
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
Radiation in Ft. Lbs. Per Min. Per Sq. In. Proj. Area = Q


FIG. 34




TEMPERATURE – SPEED CURVES
– for –
PERFECTLY LUBRICATED
– & –
WELL VENTILATED
JOURNAL TYPE
ELECTRIC MOTOR BEARINGS

FORMULAS

FOR V UP TO 500 FT. PER MIN.

$$\frac{60(t-t_r+33)^2}{1150} = \frac{2.3\,V^{1.5}}{t-32}$$

FOR V ABOVE 500 FT. PER MIN.

$$\frac{60(t-t_r+33)^2}{1150} = \frac{51.2\,V}{t-32}$$

FIG. 35

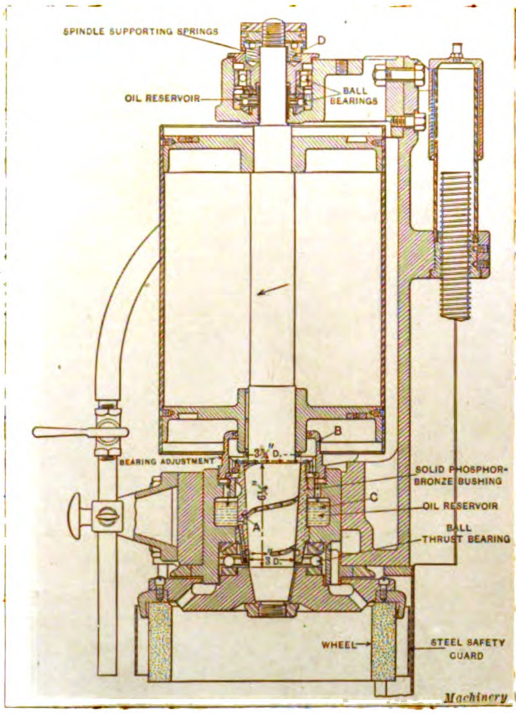



Fig. 36.
Vertical Type Grinder Bearing

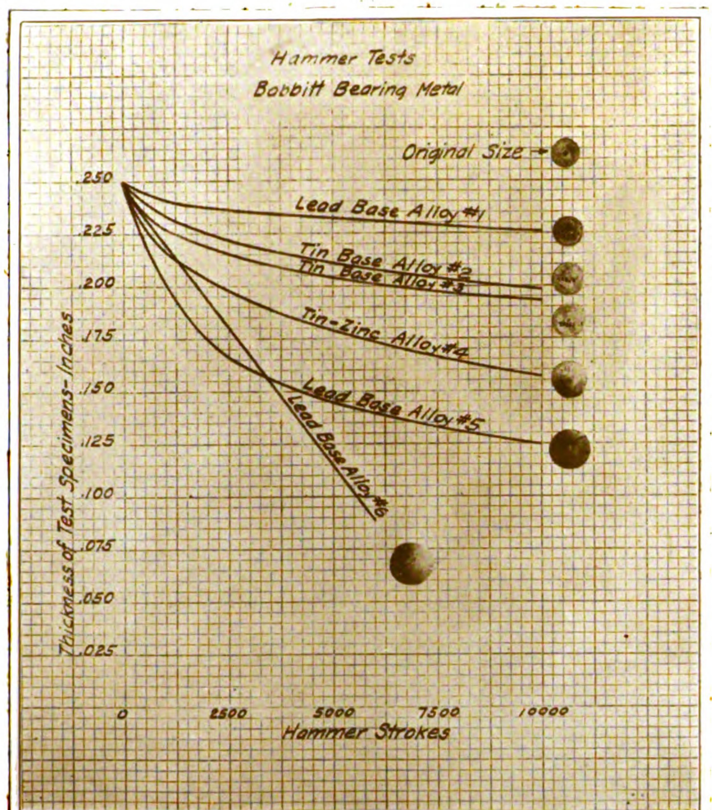



Fig. 1—Results of Comparative Test of Samples of Tin and Lead Base Alloys Subjected to Intermittent Hammer Blows

**Fig. 37.**

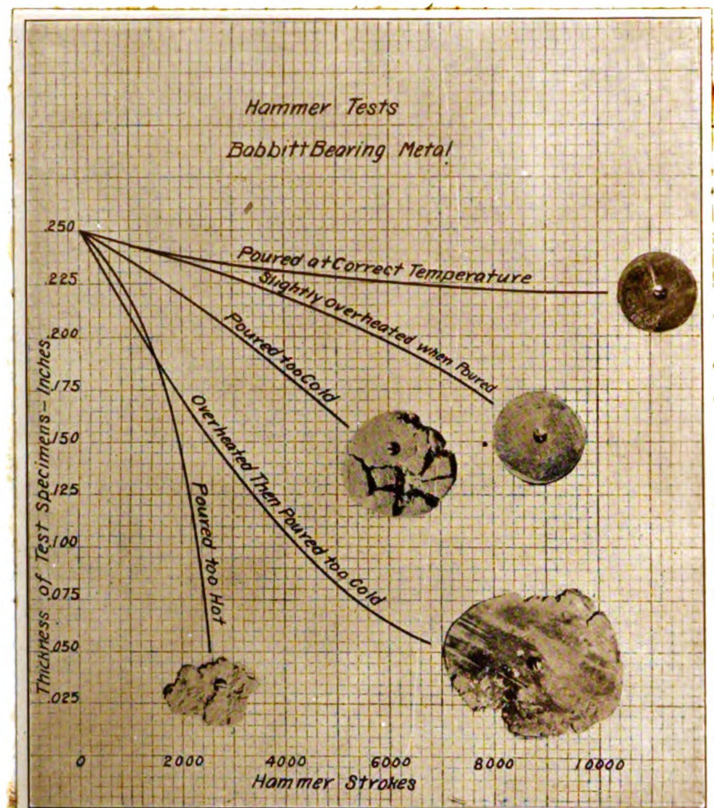
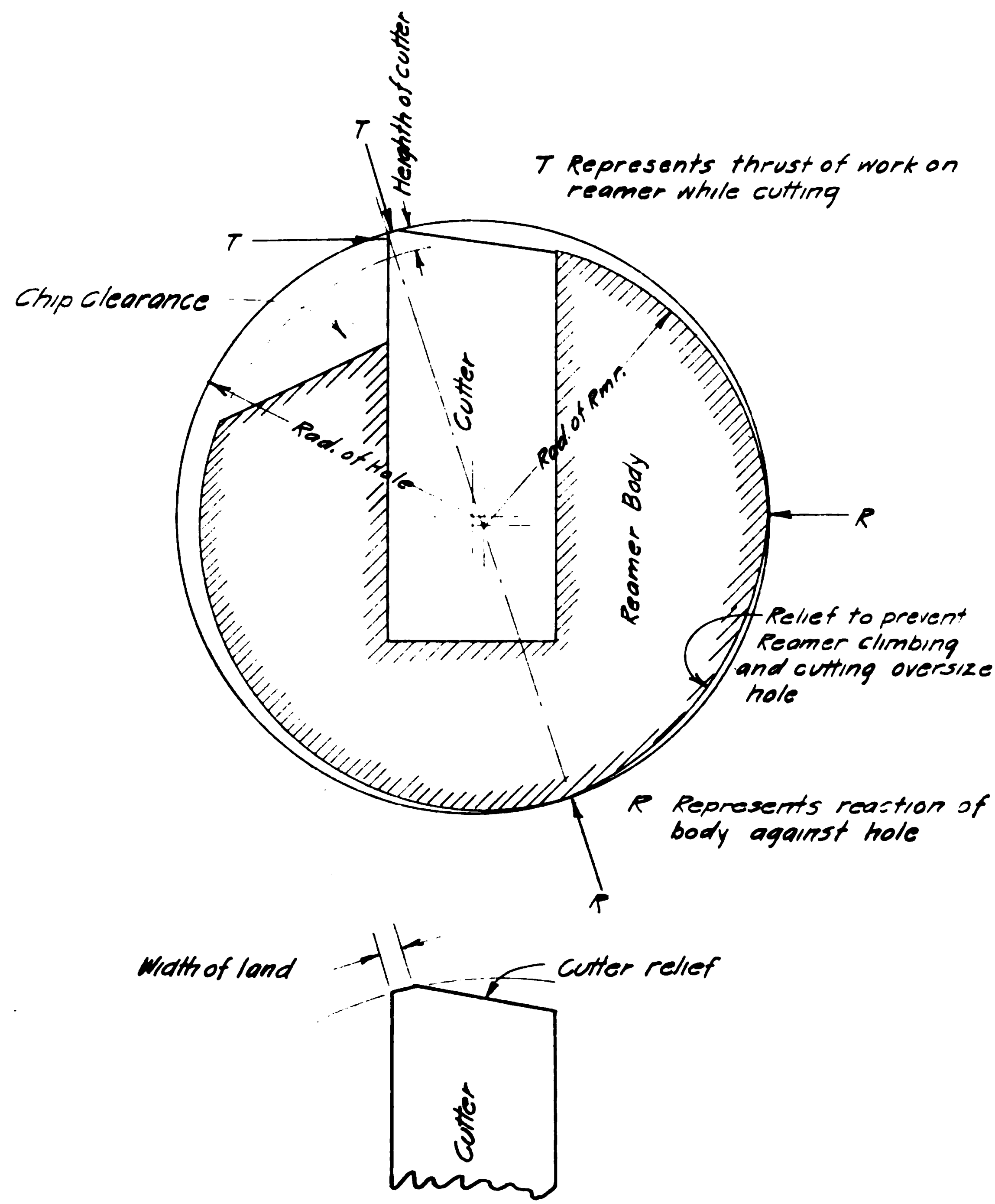


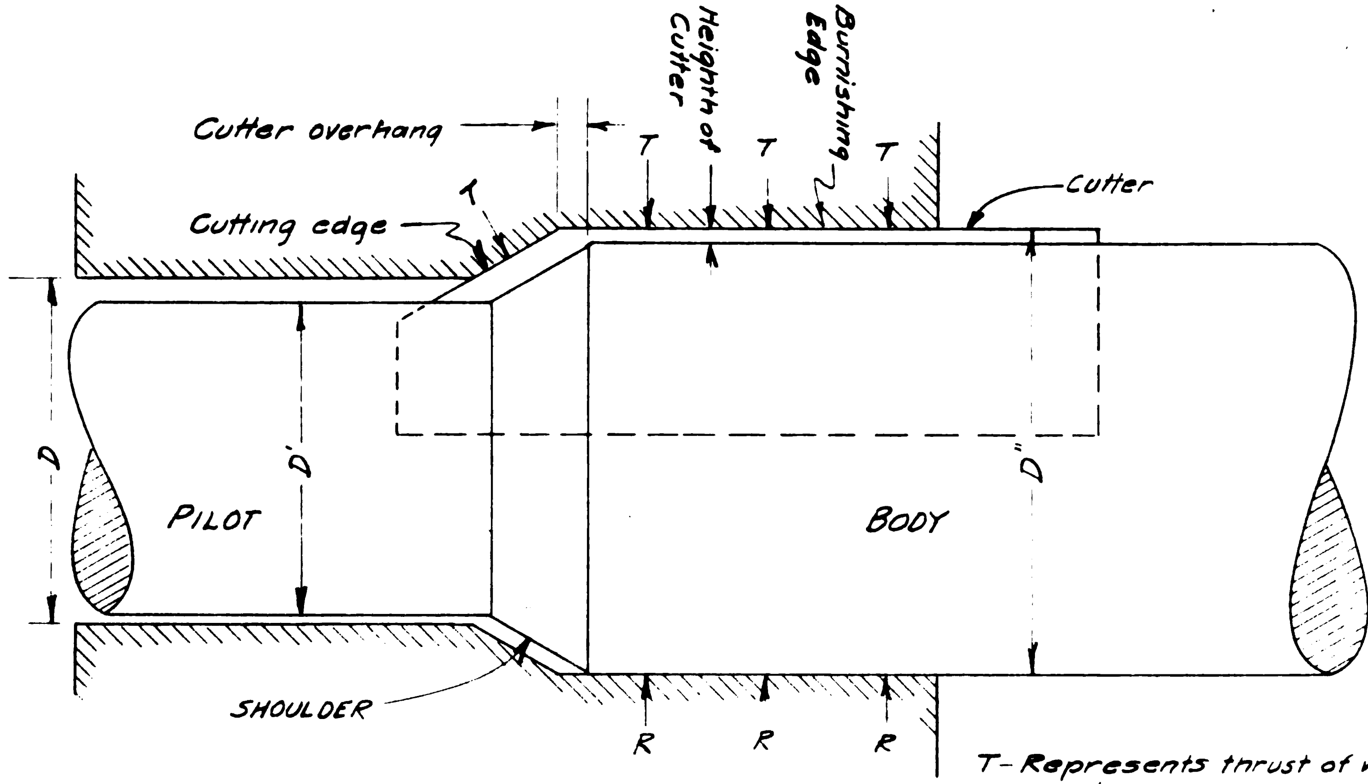

Fig. 4—Test Pieces of Lead Base Alloy Poured Under Different Temperature Conditions and Subjected to a Series of Hammer Blows

**Fig. 38.**




CROSS SECTION OF SINGLE BLADED ADJUSTABLE LINE REAMER

FIG. 39

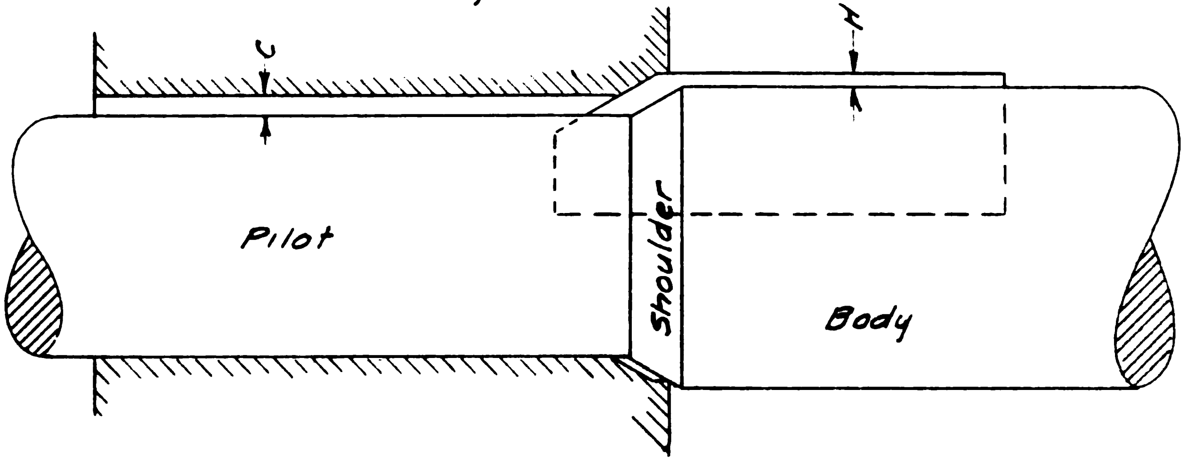



T-Represents thrust of work against cutter.

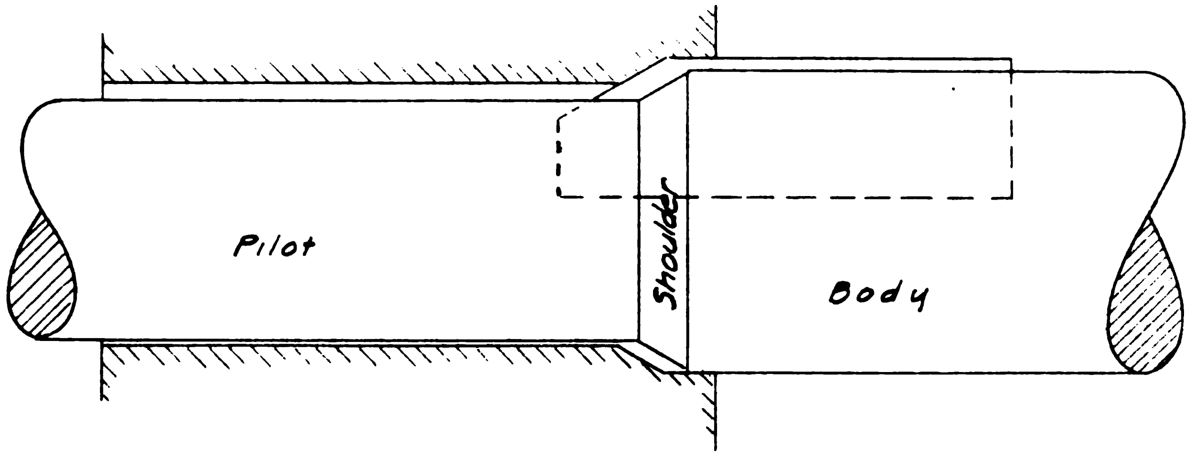
R-Represents reaction of body of reamer against hole

SIDE VIEW OF SINGLE BLADED ADJUSTABLE LINE REAMER

FIG. 40




(A) - SHOULDER OF REAMER JUST ENTERING REAMED HOLE




(B) - BODY OF REAMER AFTER ENTERING REAMED HOLE
CUTTER NOW SUPPORTED BY BODY

FIG. 41

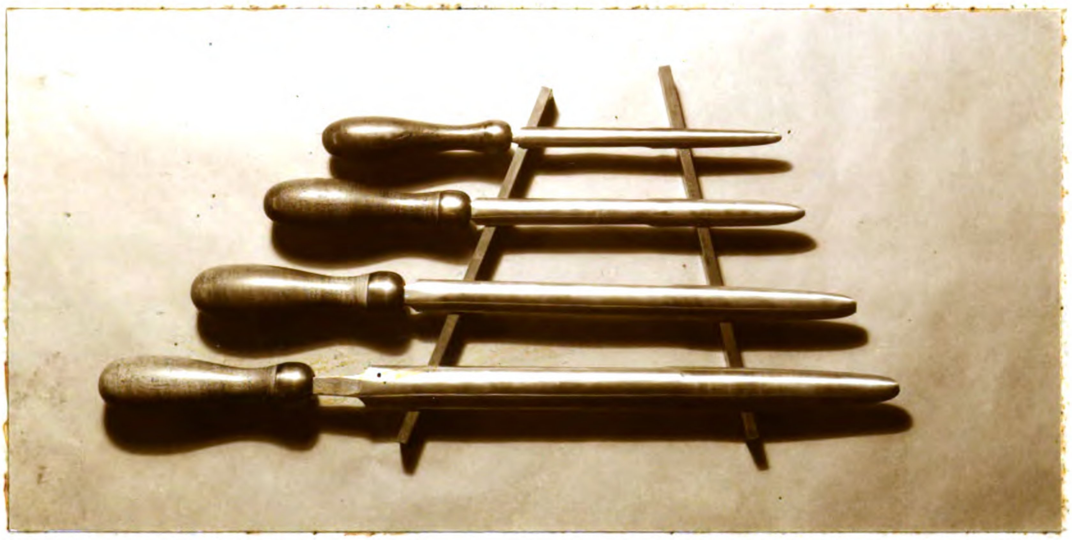
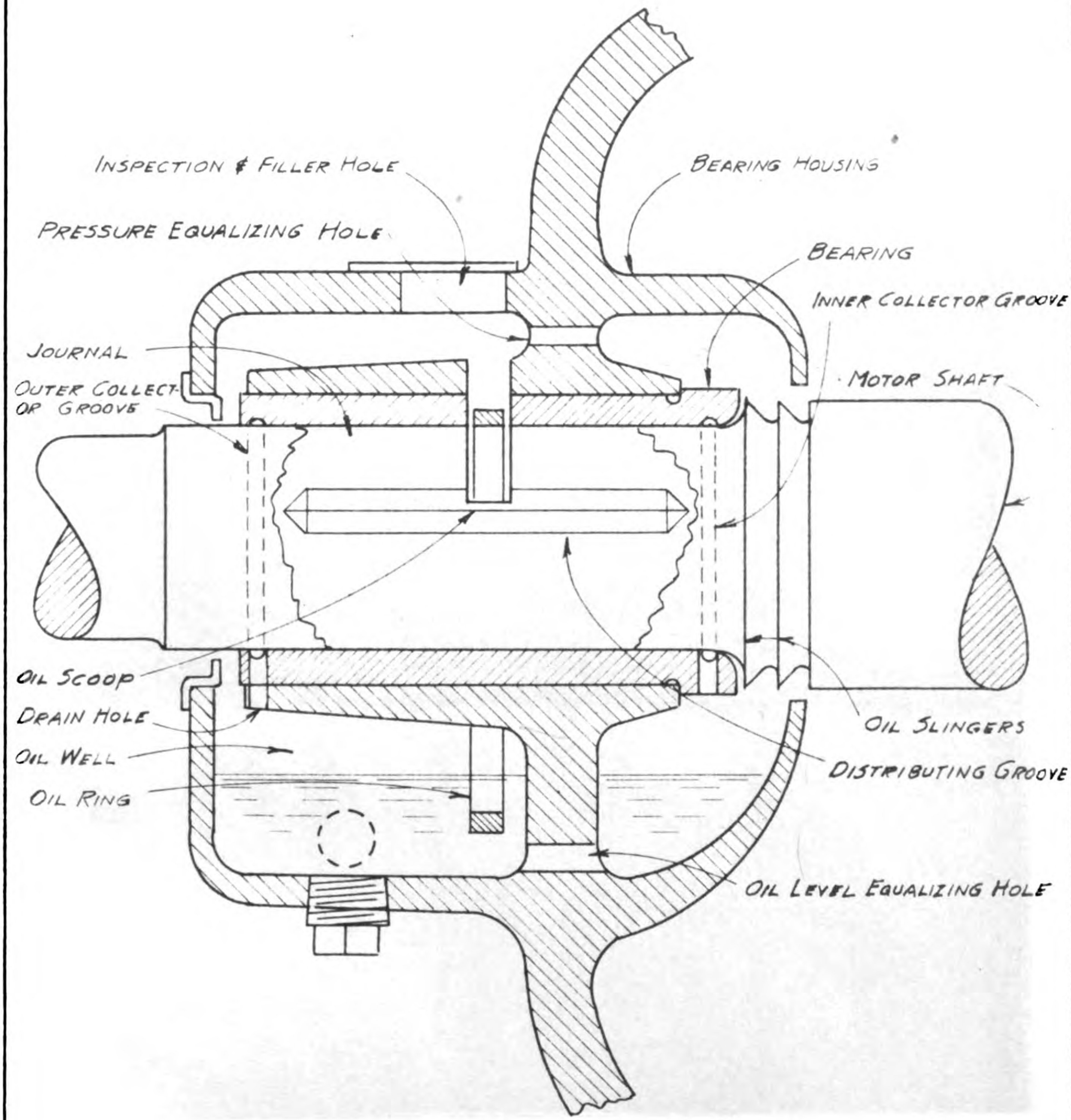


Fig. 42. Scrapers for Bronze Bearings.




CROSS SECTION OF TYPICAL JOURNAL BEARING

FIG. 43