A SURVEY OF CYLINDRICAL BEARING PRACTICE OF THE UNITED STATES

by

Chapin Winston Yates

A Thesis Submitted for Partial

Fulfillment for the Degree of

MASTER OF SCIENCE

in

MECHANICAL ENGINEERING

Approved:

In charge of Major work

Head of Major Department

Dean of Engineering

Chairman of Graduate Programs and Degrees

Virginia Polytechnic Institute

May, 1940

TABLE OF CONTENTS

TABLE OF CONTENTS

ACKNOWLEDGEMENTS

I am especially indebted to Professor James I. Clower for his timely advice, for his willingness to aid in the solution of the many and varied problems encountered in the preparation of this thesis, and also for the use of his office, his drawings and supplies. Without the cooperation and inspiration afforded by Professor Clower, the completion of this thesis would have been impossible.

I also wish to express thanks to Dean Earle B. Norris, to Professor W. T. Ellis, and to Professor J. B. Jones for their splendid cooperation.

Many others aided in this survey, the number being too many to name here, and some who rendered assistance prefer that their identities not be disclosed. I thank these people who aided in assemblying the most recent data presented in this thesis.

INTRODUCTION

W. J. Pelo defines a bearing as "that part of a machine that bears the friction." (1)[1] This definition of a bearing is essentially correct, but it is hardly complete, especially as applied to the journal-type bearing. A journal-type bearing could be more completely defined as that part of a machine that constrains the motion of the journal about a definite axis. Without bearings there could be no constraining motion; therefore, machines are entirely dependent upon bearings.

According to an estimate made by Ainsworth and Schwartz, the total number of automobiles in use in the United States on October 31, 1938 was 23,350,152. (2) Conservatively, estimating the number of main bearings in each automobile engine as four and the number of connecting-rod and piston-pin bearings as twelve, there are over three hundred and seventy-three millions of these bearings in automobile engines. This formidable number of bearings includes only a small per cent of the bearings in the automobile and totally ignores those of all other machinery. A study of this apparently simple machine element--one that so greatly affects civilization--appears to be well worth while.

---

[1]Numbers in parentheses refer to the Bibliography at the end of the thesis.

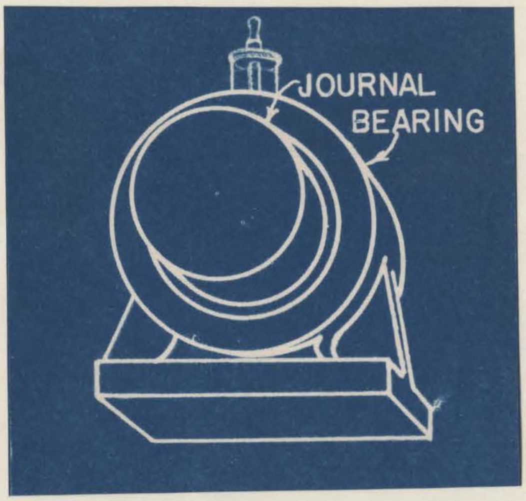


Fig. 1 - A Simple Journal Bearing

The bearing can best be understood by referring to a sketch, as that shown in Fig. 1, which is a simple journal bearing. There appears to be nothing very complex, or nothing very difficult to understand about this machine element. A study of the formation and action of the oil film in the bearing, however, reveals the fact that it is very complex and difficult to analyze rationally.

Figure 2 shows a journal stationary and resting upon the bearing. In this case, it is assumed that the clearance space is filled with a lubricant, which has been squeezed from between the bearing and journal at the point of contact at the bottom. Figure 3 shows the journal rotating slowly; and since metallic contact exists between the two surfaces, the journal rides up on the bearing. Due to the high adhesive force between the oil and the metal and the low cohesive force of the oil, some oil is drawn in between the two metal surfaces. As the journal speed increases, more oil is drawn into the clearance space until finally enough oil is forced into this space to build up an oil film sufficient to separate completely the two metal surfaces and to force the journal into the positions shown in Figs. 4 and 5. The journal rotates at a higher speed in Fig. 5 than in Fig. 4, thus giving a different position for the journal center.

With the bearing operating under conditions of perfect film lubrication its life is practically infinite. For ultimate

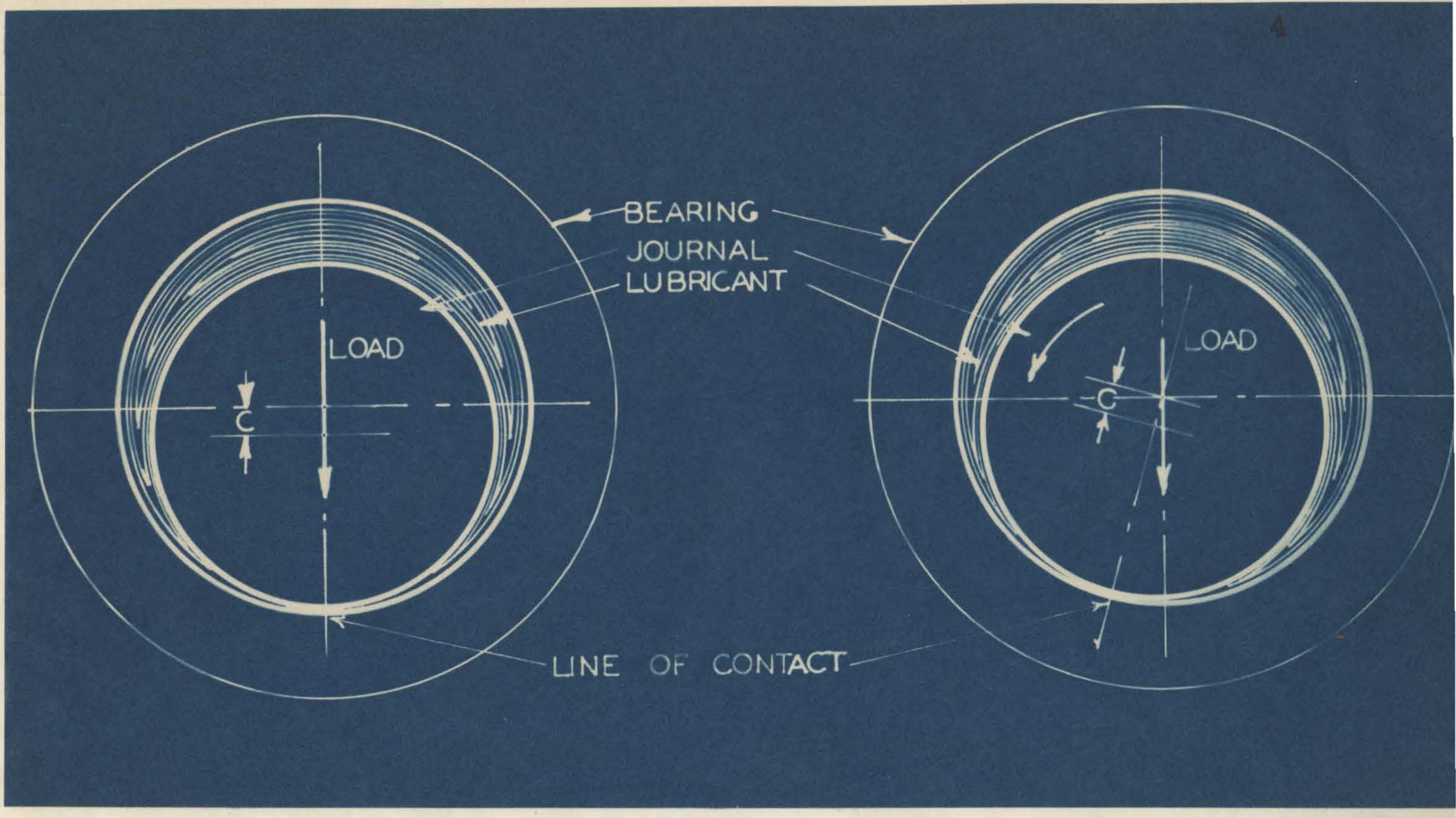


Fig. 2 - Journal at Rest

Fig. 3 - Journal Rotating Slowly, Dry Friction

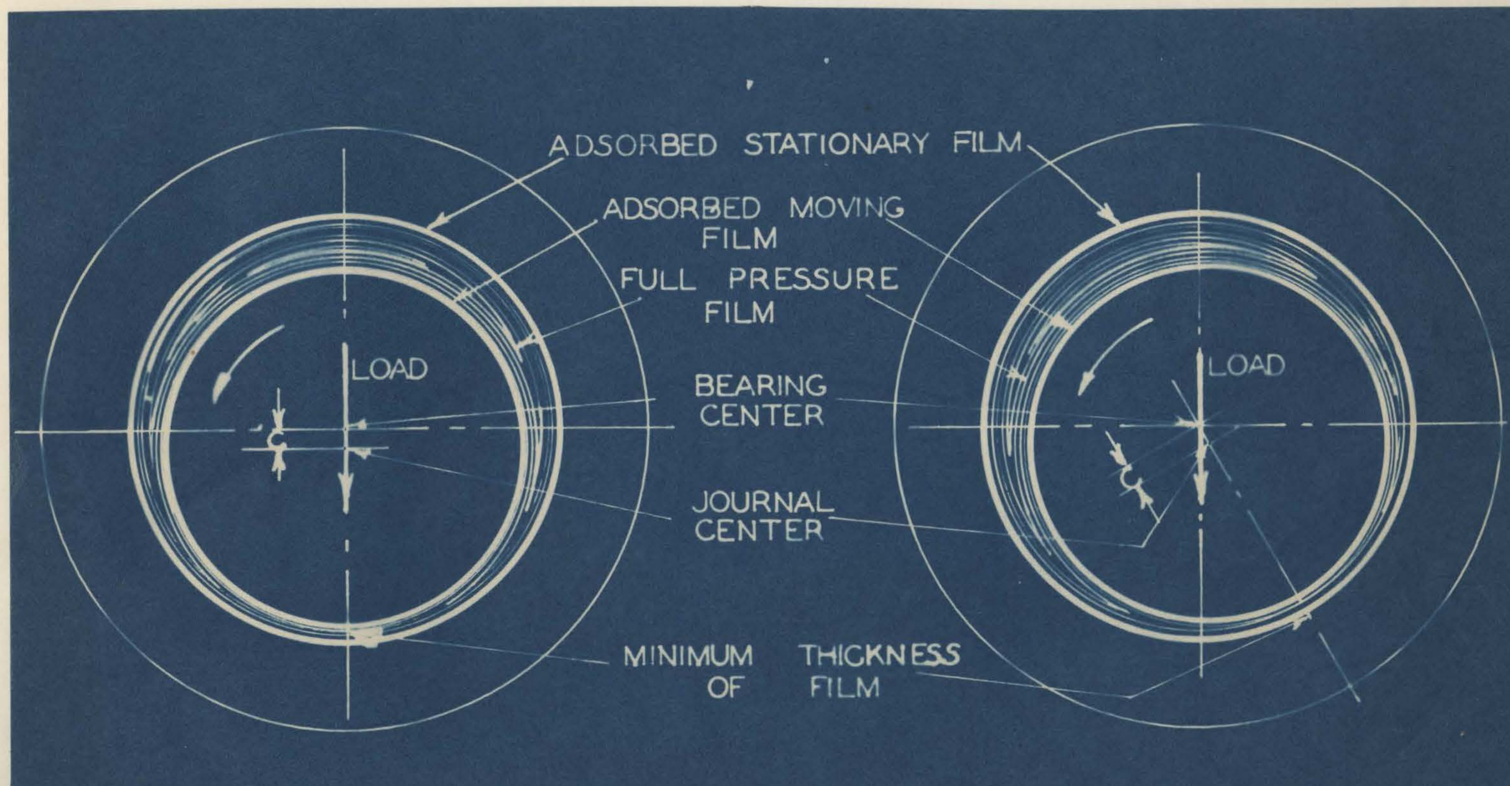


Fig. 4 - Journal Rotating at Moderate Speed, Perfect Lubrication

Fig. 5 - Journal Rotating at High Speed, Perfect Lubrication

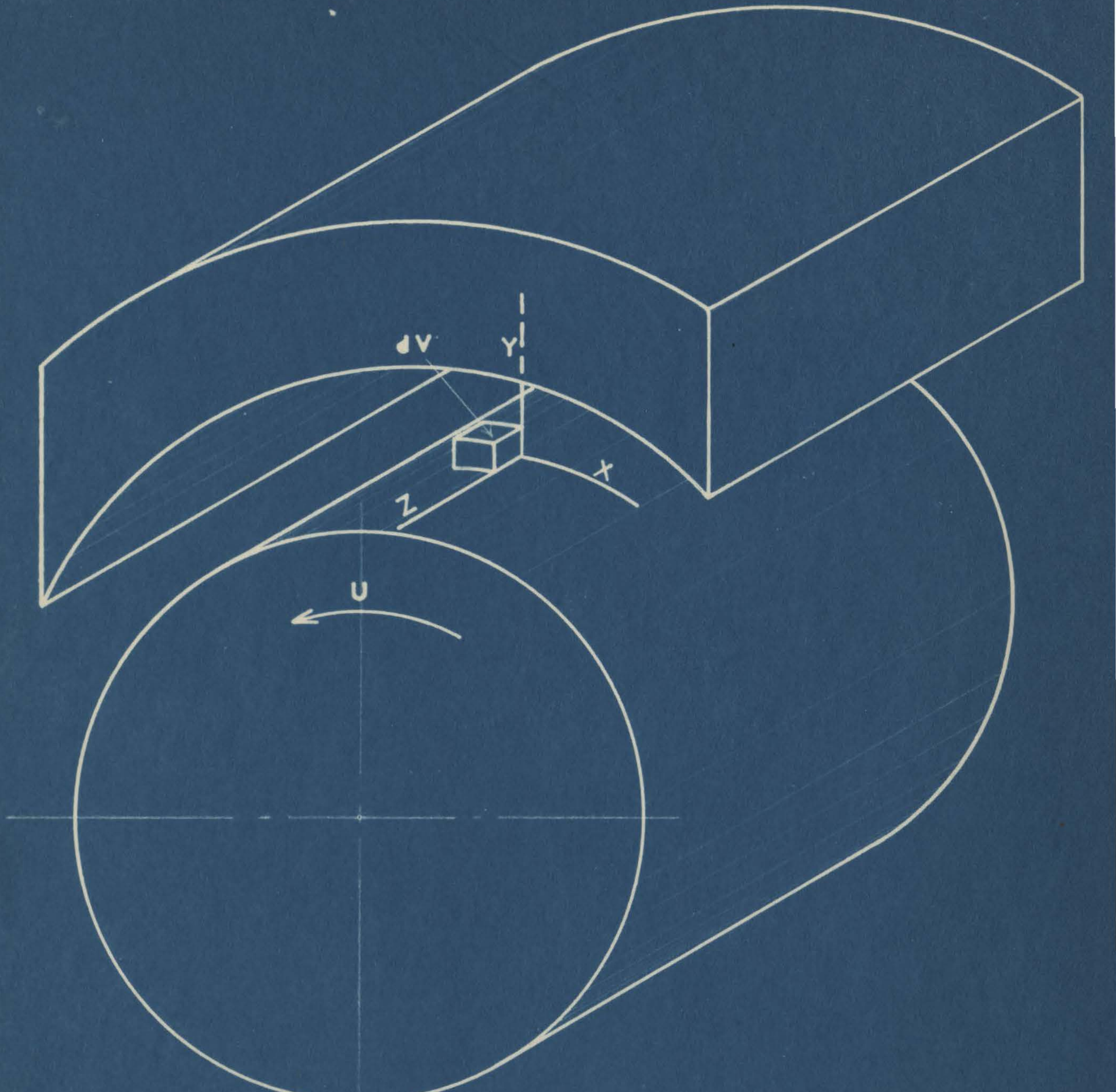


Fig. 6 - Journal Bearing With Clearance Exaggerated to Show Any Small Element of Volume of Oil in the Clearance. (The x axis is taken in the direction of motion, the z axis is perpendicular to the direction of motion and parallel to the surfaces, and the y axis is perpendicular to both the direction of motion and the surfaces.)

perfection in the design of machinery, all bearings should be designed to operate in this region of perfect lubrication. Hence many analyses of this condition have been made, the first having been made in 1883 when Petroff derived an equation for the friction in a concentric, full bearing. (3) The underlying theory of perfect lubrication is understood, but as yet none of the mathematical analyses of the common cylindrical bearing have been completely solved.

Reynolds developed the following differential equation for the pressure distribution in an oil film. (4)

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{U\partial h}{2\partial x}$$

Where: $\frac{\partial}{\partial x}\left[\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right]$ = net volume of oil leaving the small volume <u>dV</u> in the <u>x</u> direction due to the pressure gradient $\frac{\partial p}{\partial x}$, Fig. 6.

$\frac{\partial}{\partial z}\left[\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right]$ = net volume of oil leaving the small volume dV in the z direction, due to the pressure gradient, $\frac{\partial p}{\partial z}$.

$\frac{U\partial h}{2\partial x}$ = net volume of oil entering the small volume <u>dV</u>, due to viscous shear.

h = thickness of oil film measured in the <u>y</u> direction.

U = journal velocity.

Reynolds' equation cannot be solved without making certain simplifying assumptions. But with research, tests and a knowledge of the theory of perfectly lubricated bearings, the designer can in many cases approach an ideal design. He must rely, however, principally on past experience, using as a basis of design the proportions of bearings that have proved satisfactory in service.

In view of these facts, a compilation of modern successful design practices should be of inestimable value to designers. It should also be of benefit to other engineers who operate machinery. It is for this reason that a survey on common cylindrical bearing practices has been made, the results of which are embodied in this thesis.

REVIEW OF LITERATURE

The bearing is not a new development. The records show that a form of a journal bearing in a wheel was existent as far back as 3500 B. C. (5) This bearing was very crude, made of wood, and probably being nothing more than a slice of a tree trunk with a hole in the center. Even earlier, the ancients were well aware of some of the fundamentals upon which our present refined bearings are based. They knew that the rounder a log the easier it was to roll; that it was easier to roll the log than to slide it; that it was easy to keep the log rolling once it was started; that friction creates heat; and that to pull a floating log was child's play.

The bearing in ancient days was mainly used for war chariots and carts. As early as 2000 B. C. chariot wheels consisted of rims, spokes, and hubs. Not much is known of the early development of the bearing, but probably wooden shafts and bearings were used. Later, iron was used for either the bearing or the shaft, or both. Down to the awakening of mechanics, which occurred with the introduction of the steam engine shortly before A. D. 1800, all heavy equipment was made principally of wood. The bearings were generally of the open type and lined with bent metal, which supported a metal bushing on a wooden shaft. Medium weight machinery was made of wood including the bearings. Such intricate and light tools and appliances--lathes, locks, and clocks--were made of iron, brass and bronze.

The earliest record of the use of lubricant is recorded on the old Egyptian tomb of Tehuti-Hetep. (7) The carving on the tomb was made about 1650 B. C. and shows lubricant being poured between the runners of a sledge and the planks upon which it is being slid. A chariot axle of about 1400 B. C. was found to have on it solid animal fat, some quartz, sand, iron, and compounds of aluminum and lime. This analysis indicates the presence of a manufactured grease made with lime soap, and possibly tallow. The impurities were due to the lack of any dust seals. Our wagon bearings today are constructed and lubricated with a remarkable similarity to those of 1400 B. C.

The Egyptians, Greeks, and Romans had various lubricants and used different methods of application for each. For about 1900 years these finer developments were lost. Than as machinery progressed, the development of better lubricants and methods of lubrication took place. Sperm oil was the supreme lubricant for all light machinery during the period just before the awakening of mechanics. In many cases the lubricant was largely determined by the location of the machinery, because of the inability of obtaining a different and better lubricant from a remote source. Olive oil was also used for light weight machinery. Coconut oil was used for medium weight machinery, and beef and mutton tallow and vegetable oils were used for the heavier equipment. These lubricants were sufficient for the bearings existent in that day. They were not only sufficient, but due to their nature of readily oxidizing, would form free fatty acids which would build up almost solid waxes in the clearances. Such performance of lubricants actually aided the operation of the bearings on which they were used, for machinery was then rough; the clearances were large and the bearings generally out of line.

In 1839, Babbitt made known his famous bearing material consisting of 24 parts of tin, eight parts of antimony, and four parts of copper. (5)(8) During this transition period, the ring, circulating, and bath oiling systems were introduced, and the formation of waxes that made the animal and vegetable oils so useful were now detrimental. The introduction of mineral oils was a formidable task. All the leading authorities advocated the use of animal and vegetable oils. Prof. Robert H. Thurston wrote: "Sperm oil is the best known, and for general purposes the most excellent of all lubricants" and "lard oil is the most extensively used of all the animal oils, and is an excellent lubricant, although inferior to sperm oil." (7) Mineral oils were virtually unknown at that time, as the first oil well was not dug until 1859.

Difficulties were encountered when the animal and vegetable oils were replaced by the mineral oils. Within about two to six days after the replacement of the animal or vegetable oil with the mineral oil, deposits that were left by the original oil precipitated in the form of a soft varnish which rolled up on the surfaces, clogging the lubrication system and causing a hot bearing on account of inadequate lubrication. This problem was overcome by the proper compounding of the mineral oils and by the use of the proper viscosity oil. It took persistent work to introduce this radically different lubricant. But the task was accomplished by what is now known as the lubrication engineer, who since has emerged as a trained lubricating specialist. The lubrication engineer cannot be overlooked in a study of bearing practice, for it was he that yesterday tested machines for the purpose of determining the best lubricant to use for a particular bearing. He still performs this same function today, but is much more specialized than formerly. The following significant facts present some indication of the services that the lubrication engineers were capable of prior to even 1906: (7) "Up to 1906, the totals of the tests conducted by these departments on textile mills in England using this power-testing system made this interesting summary:

	Total Horsepower
With mill oils	74,223.44
With new oils	64,163.30
Saving in horsepower	10,060.24 or 13.55%"

Since the inception of the lubrication engineer's task, more and more study has been given to the scientific aspects of the subject. Prior to 1875, Morin had conducted some experiments on friction from the result of which he drew the following conclusions:

"1. The friction between two bodies is directly proportional to the pressure.

"2. The amount of friction (pressure being the same) is independent of the areas in contact.

"3. The friction is independent of the velocity, although static friction is greater than the friction of motion.

"About 1875, however, these laws or theories were found to be almost entirely erroneous and it was proved that the friction between two bodies is not directly proportional to the pressure; that the amount of friction is dependent upon the area in contact, and that the friction is also dependent upon the velocity, but that in neither case are they in direct proportion." (9)

Development of the Theory

The first important results on the friction in bearings were obtained by Beauchamp Tower in 1883 and 1885 while he was conducting research work for the Institution of Mechanical Engineers. This work of Tower inspired Osborne Reynolds to develop the classical mathematical theory of fluid film lubrication. (4) Reynolds' equation, as given on page 6, is still the basis for our modern concept of fluid film lubrication. The results of the work by both of these men were published in England, Reynolds' work being published in 1886. Petroff about the same time was conducting investigations of importance, he in 1883 having deduced the first mathematical formula for the friction of a journal bearing. (10) Petroff assumed in his deductions, however, that the film of oil was of uniform thickness, which is never the case in actual bearings. Kingsbury started work on journal bearings in 1888, and soon discovered that perfect fluid film lubrication was possible due to the wedging action of the film. (11) At this time, Kingsbury knew nothing of Reynolds' mathematical analysis of Tower's experimental results. Kingsbury continued experimenting with lubrication problems and published "Experiments with an Air Lubricated Journal" in the _Journal of the American Society of Naval Engineers_ in 1897. These experiments were in close agreement with Reynolds' theory.

In 1902, Lasche published results of some experiments that he had conducted on oil groove bearings with different clearances. (12) Stribeck at the same time published results of his experiments. (13) Later, Dr. A. Sommerfeld of Munich and W. J. Harrison extended Reynolds' theory. (14) (15)

Since this earlier work, many different experimenters have conducted various experiments, with the main purpose of determining to what extent the theory is capable of predicting the performance of a bearing and at the same time furnish data useful to the designer. Much has been done along this line, so much as to be beyond the scope of this thesis, and there is still much that is unknown. The effect of fluctuating loads and the effects of oiliness are still obscure. Effects of various other factors are not definitely known. Enough information has been assembled, however, to give a rational procedure for the design of certain types of bearings, the results of which are in reasonable accord with the best practice.

Today, experimental work is being directed toward the special problems of the designer which are not capable of being solved by theory: the determination of the conditions of the breakdown of the film, the study of the effects of cyclic fluctuations of load, and the development of new bearing materials and new lubricants.

Fundamentals of Bearing Lubrication

One should possess an accurate concept of viscosity in order to understand properly the mechanism of the lubrication of a journal bearing.

Newton's law of viscous fluids states that at any point in the fluid, the shearing stress is directly proportional to the rate of shear. Therefore, if S denotes the shearing stress (tangential force per unit area), and if R denotes the rate of shear:

$$S \propto R$$

$$S = \text{constant } xR$$

$$S = \mu \; xR$$

where μ is the coefficient of viscosity. Experiments show that Newton's law is reliable for the ordinary purposes of bearing design.

Hagen developed in 1839 the Hagen-Poiseuille equation for the flow of fluid through a capillary. (3)

$$Q = \frac{\pi}{8} \frac{P}{L} \frac{r^4}{\mu}$$

Where: P = pressure at the inlet end of the tube.

L = total length of the tube.

r = radius of tube.

Q = quantity of fluid flowing from the tube.

μ = coefficient of viscosity.

letting:

$$C = \frac{\pi}{8}\frac{r^4}{L}$$

$$Q = C\frac{P}{\mu}$$

A simple method of determining the viscosity of a fluid within two per cent of accuracy is to use a glass or metal capillary, whose temperature can be controlled in a temperature control bath, and whose ratio of length to internal diameter is not less than fifty. The inlet pressure should be kept reasonably constant during the period of efflux. The constant C can be readily obtained by calibrating the capillary with a liquid of known viscosity.

The viscosity of fluids may be determined by the application of any mechanical phenomenon that is sufficiently affected by viscosity. The standard commercial viscometer consists essentially of a standard tube fitted at the end with a standard orifice. This tube is mounted vertically in a temperature control bath. The viscosity of a fluid is equal to the number of seconds required for 60 milliliters of the fluid to pass through this orifice under standard temperature conditions. For example, an oil of 150 seconds Saybolt at 130° F, is an oil such that it will take 150 seconds for 60 milliliters of the oil to pass through the Saybolt Universal Viscometer orifice when the viscometer bath temperature is maintained at a constant temperature of 130° F.

Hersey says of the commercial viscometer: "The commercial viscometers so widely used in the petroleum industry are fitted with outlet tubes that are too short to be described as capillaries. For very high viscosities the efflux times are inconveniently long; for low viscosities they are complicated by kinetic energy and turbulent motion. In either case the flow takes place under a falling head; and the effective temperature of the sample is uncertain. The standardization of such instruments has brought a reasonable degree of uniformity, but at the risk of perpetuating the complications. The situation was reviewed at the World Petroleum Congress, London, 1933, where it was voted to adopt c.g.s. units. Conversion from Saybolt readings into c.g.s. units is accomplished with the aid of two formulas, [used in conjunction with tables] applicable respectively above and below 100 seconds." (3)

The commercial viscometers are very insensitive to differences in viscosity encountered in the low range of viscosity and give at best the kinematic viscosity, which requires a knowledge of the specific gravity of the fluid before the viscosity can be computed.

The c.g.s. or absolute unit of viscosity is the force in dynes per square centimeter required to maintain a rate of shear of one centimeter per second in a fluid film of one centimeter thickness. This unit of viscosity (dyne-seconds per square centimeter) is known as the poise. The centipoise is one one-hundredth of the poise and is used to eliminate fractional viscosity numbers.

Commercially, viscosity determinations take into account the effects of temperature, but not of pressure. The pressures encountered in commercial practice are generally not great enough to warrant a determination of the effects of pressure upon the viscosity of an oil. Nevertheless, this subject of the effects of pressure upon the viscosity of a lubricant is important and should be studied. Experiments indicate that the effects of temperature and pressure are greater on mineral oils than on fatty oils. The viscosity of an oil decreases rapidly as the temperature increases. The viscosity of an oil increases at first slowly and then rapidly as the pressure increases.

An analysis of Newton's law when the viscosity distribution is linear is as follows: (3)

In Fig. 7 is shown a small volume of fluid, the full lines being the original cross sectional shape of the volume of fluid, and the dotted lines being its shape after a short interval of time.

Let: $A =$ area of shear

$F =$ tangential force acting on $\underline{A}$

$h =$ height of element

$V =$ velocity of upper surface relative to lower surface

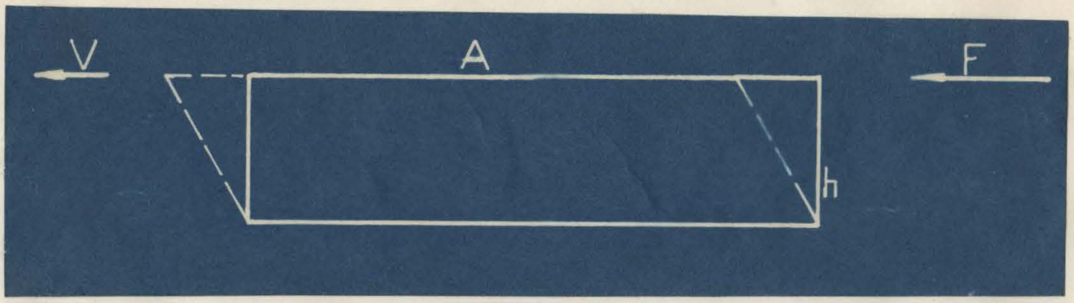


Fig. 7 - Small Volume of Fluid with Linear Velocity Distribution.

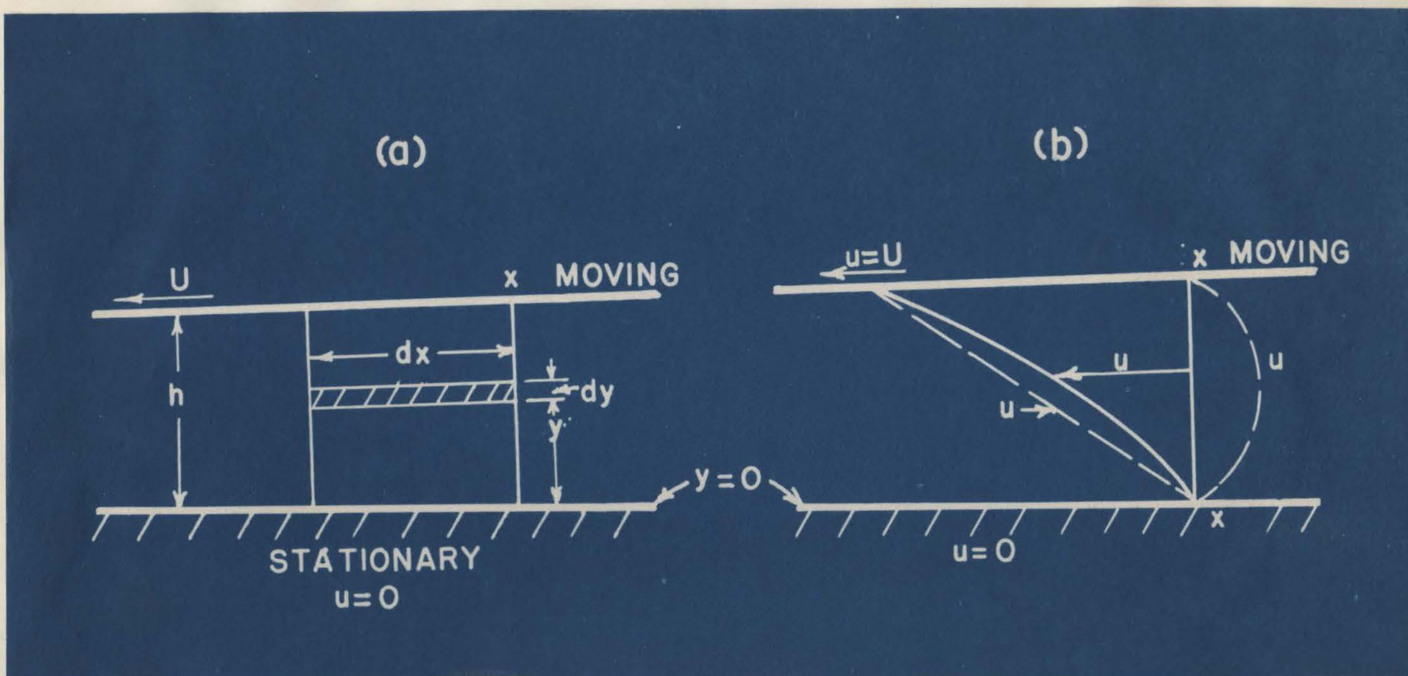


Fig. 8 - Film Pressure and Velocity Distribution. (a) Location of an Element $\underline{dx}$ $\underline{dydz}$ (length $\underline{dz}$ perpendicular to diagram). (b) Velocity Distribution Over a Cross Section of the Film at $\underline{x}$.

R= rate of shear

Therefore:

$$S = \frac{F}{A}$$

$$R = \frac{V}{h}$$

And:

$$\frac{F}{A} = \left(\frac{V}{h}\right)\mu$$

When the velocity distribution is not linear, the gradient may be expressed by the corresponding differential notation.

An example of the application of Newton's law when the velocity distribution is not linear is the following analysis as developed by Reynolds: (3)

Figure 8 shows two surfaces, one moving with a velocity, U, and the other stationary. u denotes the velocity of the fluid in the x direction at any height, y, above the stationary surface; h is the film thickness as identified by the coordinate, x; dp/dx is the pressure gradient in the direction of motion, and p is the hydrostatic pressure at x. S is the shearing stress at any point in the fluid, and S' is the shearing stress at the moving surface. S_0 is the stress at the stationary surface. The shearing stress is the force per unit area. μ is the viscosity of the liquid.

An analysis of the forces acting on the element of fluid shaded in Fig. 8a is as follows:

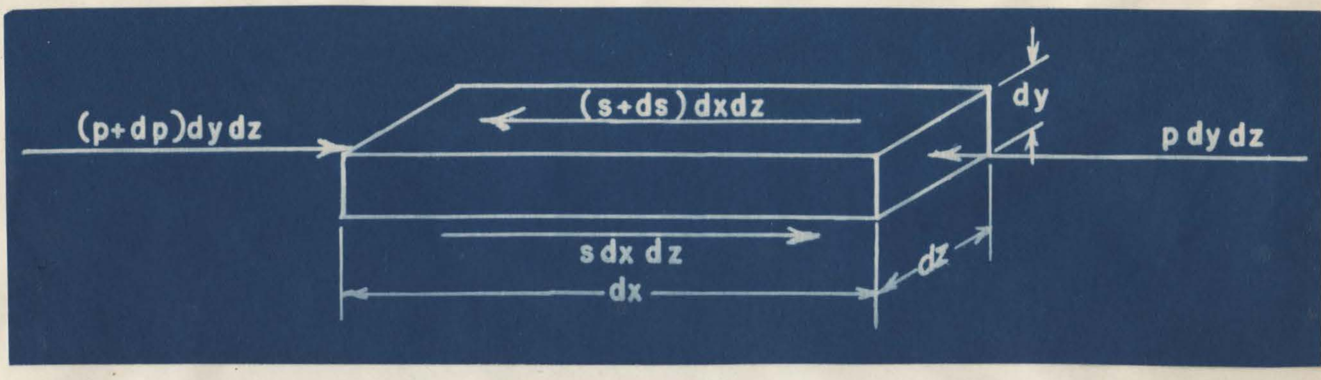


Fig. 9 Shaded Element from Fig. 8, Enlarged for the Purpose of Showing the Forces Acting upon This Element in the Direction of Motion.

For equilibrium, the sum of the forces in the direction of motion must be zero. Therefore:

$$(S + dS)dxdz + pdydz - Sdxdz - (p + dp)\, dydz = 0$$

$$dSdxdz - dpdydz = 0$$

$$dSdx - dpdy = 0$$

$$\frac{dS}{dy} = \frac{dp}{dx}$$

Integrating with respect to y, treating dp/dx as a constant at any fixed value of x,

$$S = y\left(\frac{dp}{dx}\right) + C$$

$S = S_o$ when $y = 0$, and $C = S_o$

$$S = S_o + y\left(\frac{dp}{dx}\right) \qquad [1]$$

From Newton's law:

$$S = \mu\left(\frac{du}{dy}\right)$$

Equating the two above equations:

$$\mu \left(\frac{du}{dy}\right) = S_o + y\left(\frac{dp}{dx}\right)$$

But: $u = 0$ when $y = 0$, and $u = U$ when $y = h$

$$\int_{u=0}^{u=U} \mu\, du = \int_{y=0}^{y=h} \left(S_o + y\frac{dp}{dx}\right) dy$$

$$\Big[\mu u\Big]_0^U = \left[S_o y + \frac{y^2}{2}\frac{dp}{dx}\right]_0^h$$

$$\mu U = S_o h + \frac{h^2}{2}\frac{dp}{dx}$$

$$S_o = \frac{\mu U}{h} - \frac{h}{2}\frac{dp}{dx} \qquad [2]$$

Substituting $\underline{S_o}$ from Equation [1] in Equation [2]

$$S = \frac{\mu U}{h} - \frac{h}{2}\frac{dp}{dx} + y\frac{dp}{dx}$$

$$S = \frac{\mu U}{h} + \left(y - \frac{h}{2}\right)\frac{dp}{dx} \qquad [3]$$

This is the general equation of the shearing stress within a fluid.

$y = h$ when $S = S'$

$$S' = \frac{\mu U}{h} + \frac{h}{2}\frac{dp}{dx} \qquad [4]$$

Subtracting Equation [2] from Equation [4] we find:

$$S' - S_o = h\frac{dp}{dx}$$

Thus the shearing stress on the moving surface is greater than that on the stationary surface by the amount $\underline{h(dp/dx)}$. This can be explained by Fig. 8b where the velocity distribution curve for $\underline{u}$ has been shown as a function of $\underline{y}$. The curve $\underline{u_1}$ is the velocity distribution curve, neglecting the pressure gradient.

The curve u_2 is the parabolic distribution that would occur due to the pressure gradient alone. The curve u is the resultant of these two curves, and the slope of the u curve is du/dy. Newton's law states: that the stress is proportional to the rate of shear or du/dy. Hence, the shearing stress at the moving surface is greater than that at the stationary surface.

After developing Equation [4], Reynolds needed a knowledge of the pressure gradient so that a full interpretation could be available. Reynolds developed the following general equation of the hydrodynamic theory. (See page 5.)

$$\frac{\partial}{\partial x}\left(\frac{h^3 \partial p}{12\mu \partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3 \partial p}{12\mu \partial z}\right) = \frac{U \partial h}{2 \partial x} \qquad [5]$$

Where: U = surface speed

h = film thickness at any point

μ = viscosity of lubricant

p = hydrostatic pressure at any point in the film whose co-ordinates are x and z.

x = co-ordinate in direction of motion

z = co-ordinate parallel to the axis of the journal.

He assumed steady running conditions, that is, the surface speed and the film thickness at any point are constant with respect to time. He also assumed that the lubricant used was incompressible.

Reynolds integrated this equation after assuming constant viscosity, and negligible side leakage and obtained the following formula for the pressure gradient. (3)

$$\frac{dp}{dx} = 6\mu U \frac{h - h_1}{h^3} \qquad [6]$$

Where: h_1 is the film thickness at the point of maximum pressure, that is, where dp/dx equals zero.

He integrated this equation further for the cases of flat surfaces of infinite width, and for the partial bearings of Tower's experiments under such conditions where the eccentricity was not greater than one-half the radial clearance.

Dr. A. Sommerfeld of Munich extended Reynolds' theory by integrating Equation [6] for all values of eccentricity, and for the cases of the half bearing and the full bearing. (14)

This theory has been extended and confirmed by numerous men since then. The case of rectangular plane surfaces has been completely solved mathematically. This solution is a three dimensional solution, including side leakage, and was developed by A. G. M. Michell. (16) His analysis was extended by H. M. Martin, R. O. Boswall, T. Kobayashi, and others. (17)(18)(19) Such a rigorous mathematical solution including the effects of side leakage has not been developed for the case of the journal bearing. The approximation by assuming no side leakage by setting the term containing $\frac{\partial p}{\partial x} = 0$ is generally made. Many different approximate methods are available. Kingsbury has a precise solution for the side leakage problem, using an ingenious electrical analogy. (20)

From Sommerfeld's results the capacity of a bearing is dependent upon the following variable: (3)

$$s = \frac{\mu N}{P}\left(\frac{D}{C}\right)^2$$

Where:

μ = absolute viscosity of the lubricant

N = R. P. M. of journal

P = average pressure per unit area

D = diameter of journal

C = diametrical clearance

s = Sommerfeld variable.

"The most evident limitations of the hydrodynamic theory in its present state of development and application may be considered under three heads: (a) the point of view of the analysis; (b) physical limitations; and (c) geometrical limitations.

"1. Some of the most ingenious and successful investigations, such as Kingsbury's study of optimum conditions, have been conducted strictly from the viewpoint of bearing design. The results therefore are not always directly applicable in calculating the probable performance of a bearing of any fixed design, or in selecting the most suitable lubricants for existing machinery.

"2. The treatment is usually limited to a steady state, with constant viscosity; and in most investigations, even when side leakage is taken into the picture, the effects of negative pressure are dealt with in some arbitrary or artificial way.

"3. The bearing surfaces are assumed perfectly smooth and rigid, and of some ideally simplified geometrical form; the applied loads, other than the driving torque, reducing to a single resultant force, conveniently located for purposes of calculation.

"Because of these limitations and in order to verify the laws of lubrication there is a constant demand for direct experimentation. Swivel-chair experimental work is both easy and economical, but before any project can be carried through to completion in the laboratory a host of new difficulties may arise. Questions associated with the accuracy and finish of the bearing surfaces and the methods of temperature measurement will immediately come to mind. We wish, however, to call attention especially to the large number of variable involved in lubrication experiments. The number of observations necessary for exploring the field in connection with any one type of machine element may be very considerable, and the expense correspondingly great, when the separate factors are varied over the full range one at a time.

"Dimensional theory offers a possible escape from both sets of difficulties by means of a combined mathematical and experimental approach." (3)

Hersey developed the laws of lubrication of journal bearings, by dimensional analysis, and published this analysis in the _Transactions of the American Society of Mechanical Engineers_, in 1915. (21) A summary of his results are, after imposing the following restrictions:

"(1) The bearing must be in a steady state." Acceleration and deacceleration, heating up and cooling off, and intermittent load are not considered.

"(2) The lubricant must be homogeneous.

"(3) The bearing must be running below the critical speed at which eddy motion would be set up in the lubricant.

"(4) The effect, on the motion of the lubricant, of any other forces than hydrostatic pressure and shearing stress must be negligible.

"(5) The metal surfaces must always be separated by a film of lubricant which is thick enough to have the same mechanical properties it would have in bulk.

"(6) There must be no resultant couple acting on the bearing in the plane of its axis."

$$\phi \left(f \frac{\mu N}{P} \frac{D^3 N}{Q} s \frac{C}{D} \frac{l}{D} r \right) = 0$$

$$f = \emptyset \left(\frac{\mu N}{P} \frac{D^3 N}{Q} s \frac{C}{D} \frac{l}{D} r \right)$$

Where:

f = coefficient of friction

μ = absolute viscosity of lubricant

N = revolutions per unit time

P = pressure per unit of projected area

D = diameter of journal

Q = quantity of lubricant per unit time

S = volume of lubricant in bearing/volume of clearance space

C = diametrical clearance

l = length of journal

r = sum of all ratios that specify the line of action of the load, the shape of the bearing, the arrangement and shape of the oiling mechanism, and all geometrical irregularities.

Should the bearings be free of cavitation, have no end effects, be centrally and uniformily loaded, and be geometrically similar in cross section, then:

$$f = \phi\left(\mu\frac{N}{P}\ \frac{D^3N}{Q}\ \frac{C}{D}\right)$$

In this case where there is no thrust due to forced lubrication, Q cannot enter into the equations and:

$$f = \phi\left(\mu\frac{N}{P}\ \frac{C}{D}\right)$$

In this case where the bearings are geometrically similar:

$$f = \phi\left(\mu\frac{N}{P}\right)$$

Hersey found the carrying power, p, to be dependent upon the factors as:

$$p = \mu N\cdot\theta\left(\frac{x}{c}\ \frac{D^3N}{Q}\ s\ \frac{C}{D}\ \frac{l}{D}\ r\right)$$

Where:

x = minimum film thickness

c = radial clearance

In geometrically similar bearings:

$$p = \mu N\cdot\theta\left(\frac{x}{c}\ \frac{D^3N}{Q}\right)$$

Hersey concludes that the laws of the lubrication of all geometrically similar bearings can be established experimentally by varying the two quantities $\frac{\mu N}{P}$ and $\frac{D^3N}{P}$ of one bearing. In all geometrically similar bearings that are similarly lubricated the laws can be established by varying the one factor, $\frac{\mu N}{P}$. These bearings are defined by:

$$f = \phi\left(\frac{\mu N}{P}\right)$$

$$P_0 = \mu N \cdot \theta\left(\frac{x}{c}\right)$$

The carrying power of any bearing is, therefore, directly proportional to the product of the viscosity and the revolutions per unit time. The constant of proportionality is the same for all geometrically similar, similarly lubricated, and equally safe bearings.

Practically all results today in the experimental field of lubrication are presented as Hersey recommended. His analysis is in thorough accord with the hydrodynamic theory, each development in the hydrodynamic theory having been found to be some analysis of a special case of Hersey's analysis. Hersey's analysis is also confirmed by experiments. Experiments show the equation of the law of a full journal bearing to be:

$$f = A + B\left(\frac{ZN}{P}\right)$$

Where:

N = R.P.M.

P = pressure, pounds per square inch

A and B are constants

Z = viscosity of lubricant, centipoises

To prove further Hersey's agreement with experimentation, Figs. 10, 11, 12, 13, 14, and 15 have been reproduced from various investigations. Figure 10 shows the results of experiments conducted at the Massachusetts Institute of Technology which were first published in the Journal of the Washington Academy of Science, Vol. 4, 1914. (22) These experiments were conducted on a full journal bearing, lubricated with three different oils and under varying loads and speeds. All methods of plotting the data were confusing until this method was adapted. The results were affected slightly when the oil supply was restricted to too great an extent. This is entirely in accord with theory. These experiments show the practicability of Hersey's analysis as applied to the presentation of data. Figure 11 shows experimental data taken under the more developed conditions at a much later date showing the exactness of the method. (23) Figure 12 shows the effect of the length diameter ratio on the bearing. (24) Figure 13 shows the effect of the clearance diameter ratio on the bearing. (25) All the results may be expressed by one curve by plotting the data as in Fig. 14. (25) Figure 15 shows that the relation $\left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right)$ is independent of the $\frac{L}{D}$ ratio when the $\frac{L}{D}$ ratio is greater than 0.75. (25)

Figure 16 is a typical $\frac{ZN}{P}$ curve, showing the three regions of lubrication. In the region of thick film lubrication, the film has been built up as is illustrated by Figs. 5 and 17. In this region, the viscosity of the oil is great enough to

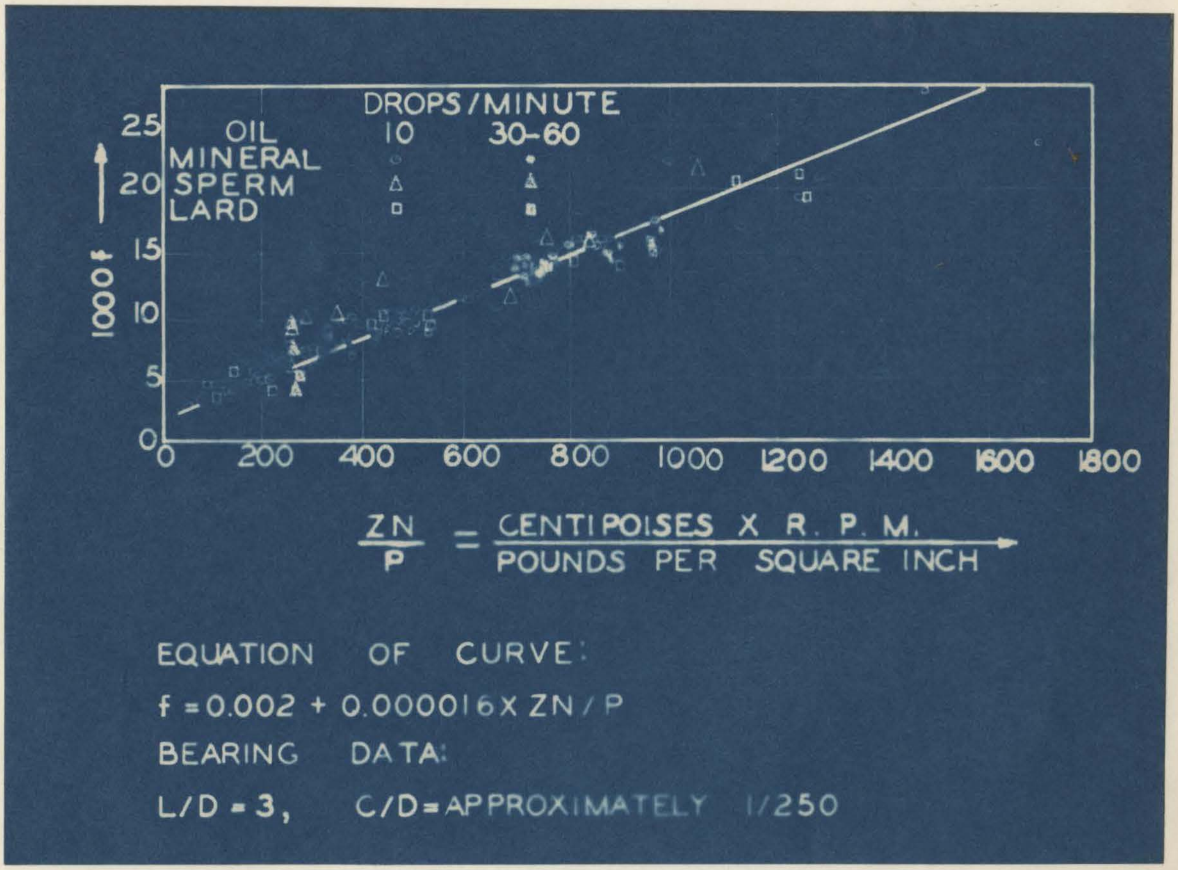


Fig. 10 - Results of Experiments Conducted at M. I. T.

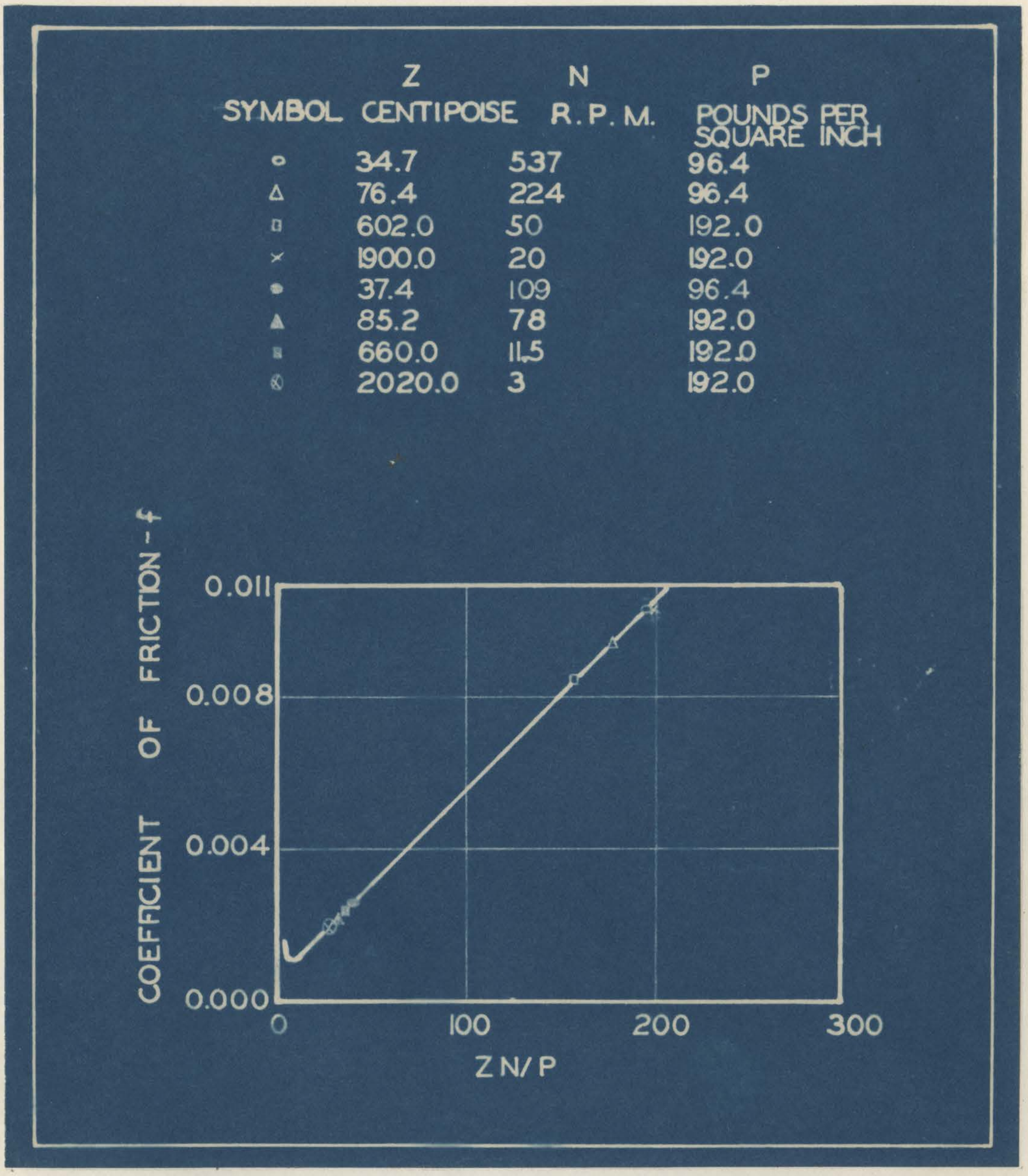


Fig. 11 - Typical f vs $\frac{ZN}{P}$ Curve

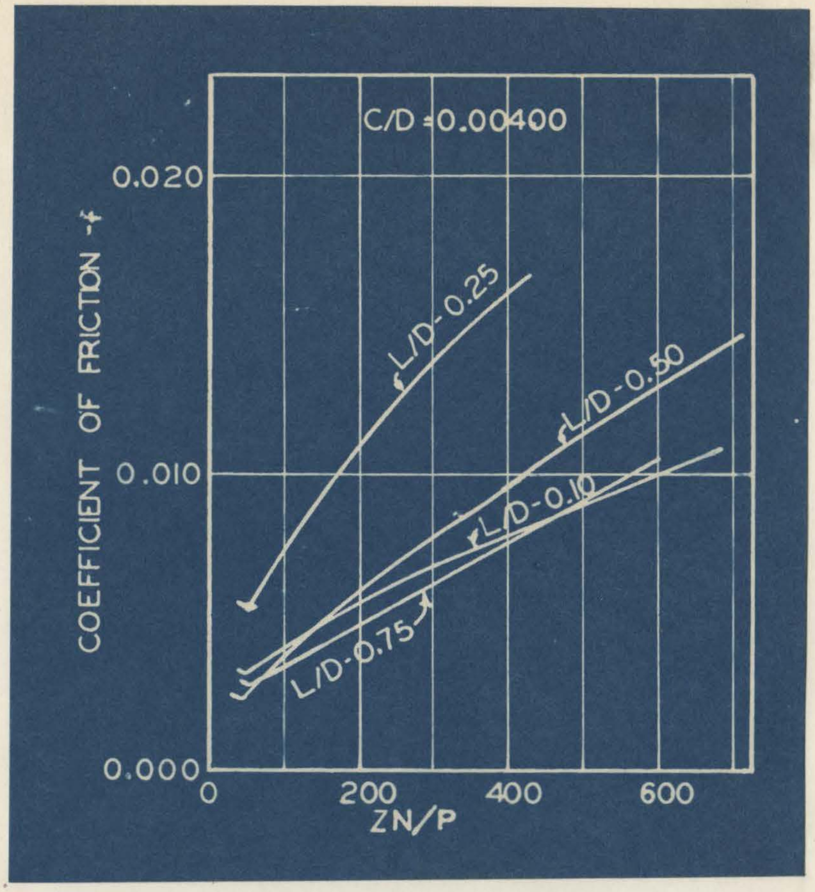


Fig. 12 - Curves Showing the Effect of the Length Diameter Ratio on a Bearing

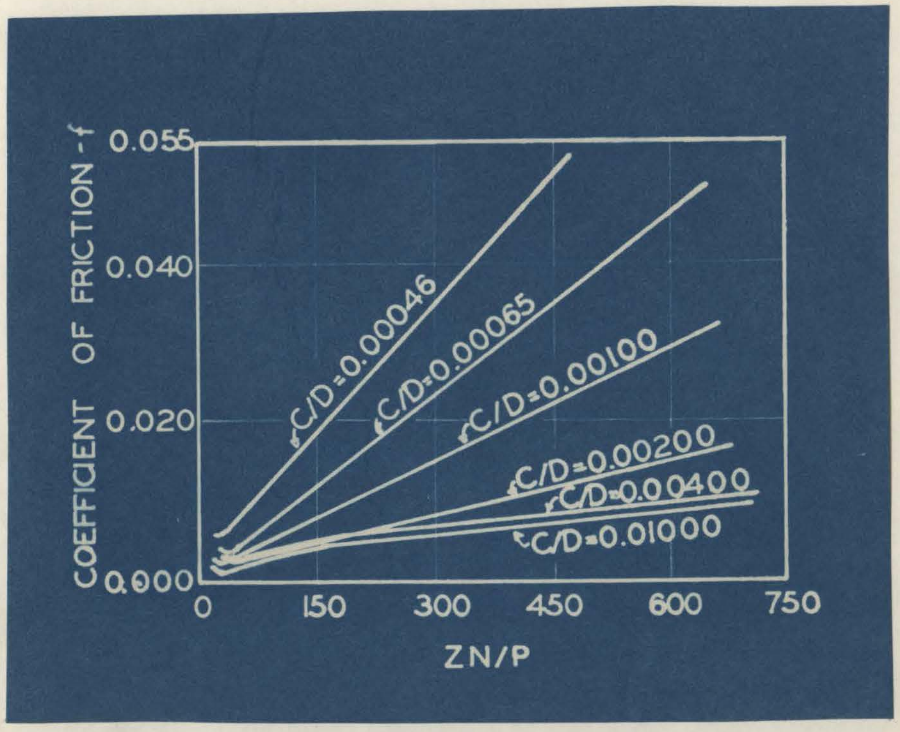


Fig. 13 - Curves Showing the Effect of the Clearance Diameter Ratio on a Bearing

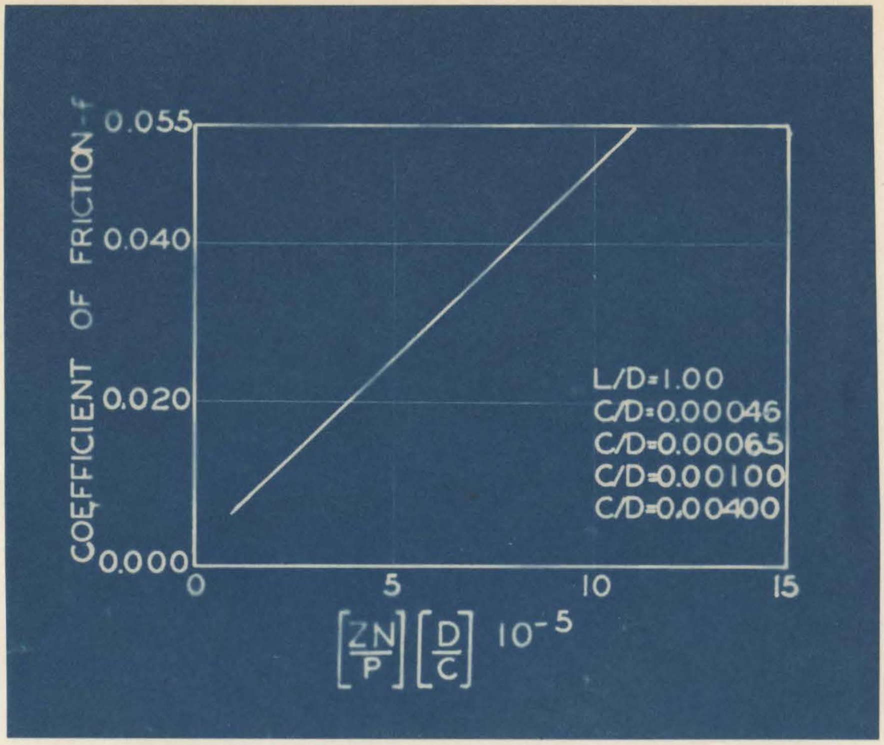


Fig. 14 - For Any Bearing Having a Series of Different Clearance Diameter Ratios, the Results May Be Expressed by One Curve Having as Ordinates the Coefficient of Friction, f, and the Relation, $\frac{ZN}{P}$ $\frac{D}{C}$.

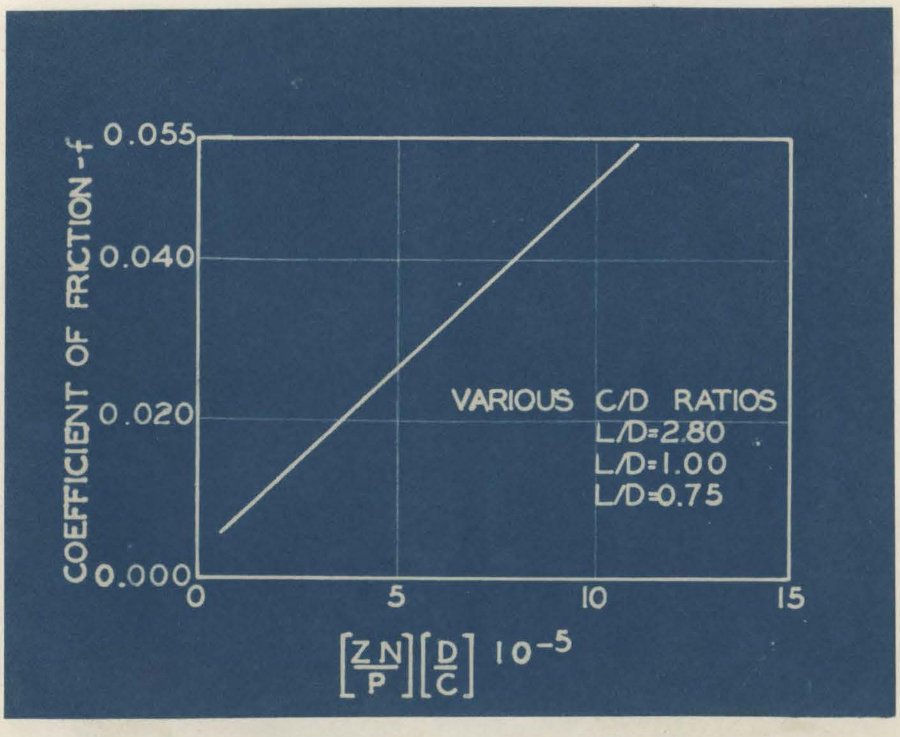


Fig. 15 - The Relation, $\frac{ZN}{P}$ $\frac{D}{C}$, Is Independent of the $\frac{L}{D}$ Ratio when the $\frac{L}{D}$ Ratio Is Greater than 0.75.

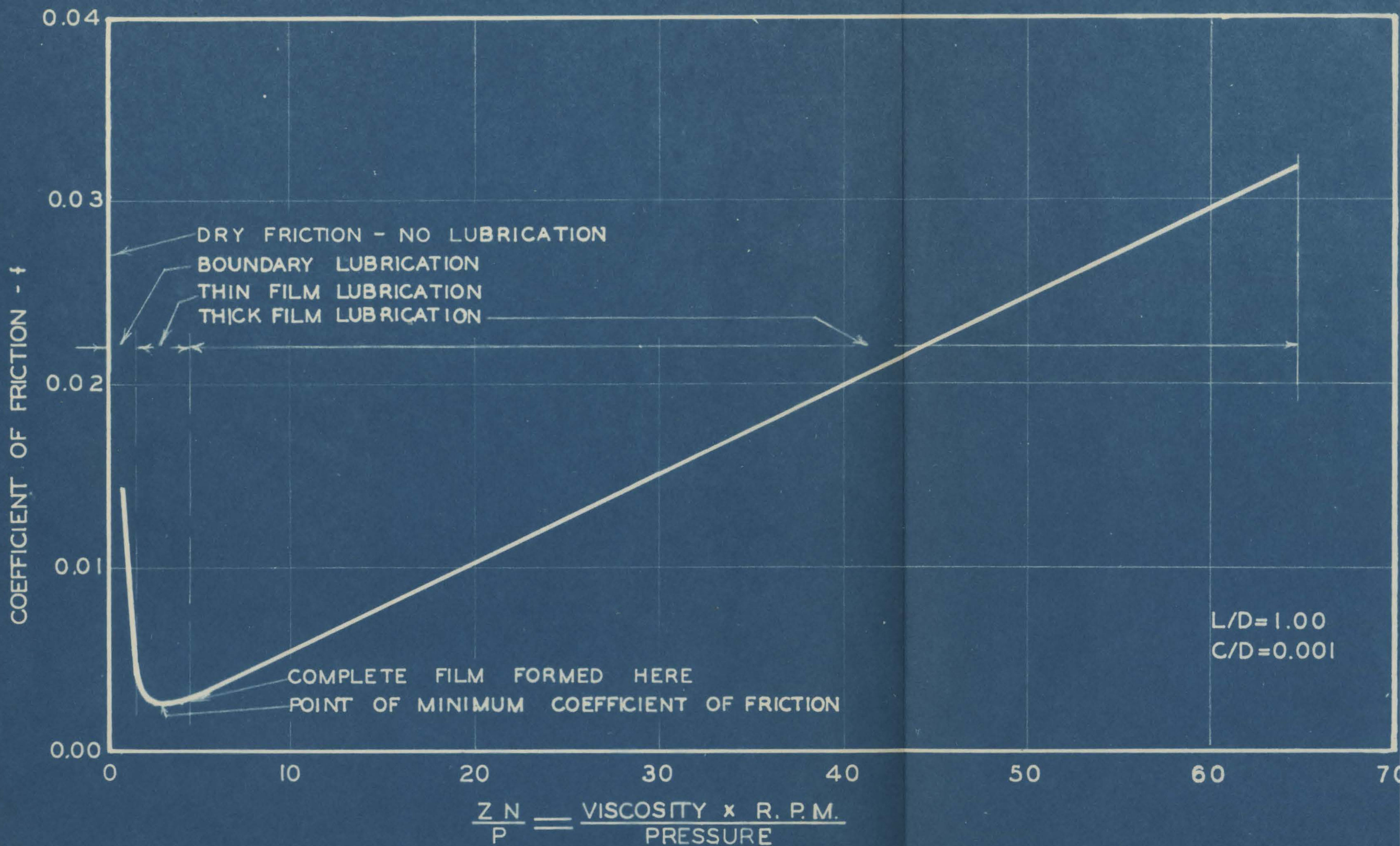


Fig. 16 – Typical f vs $\frac{ZN}{P}$ Curve, Showing the Three Regions of Lubrication.

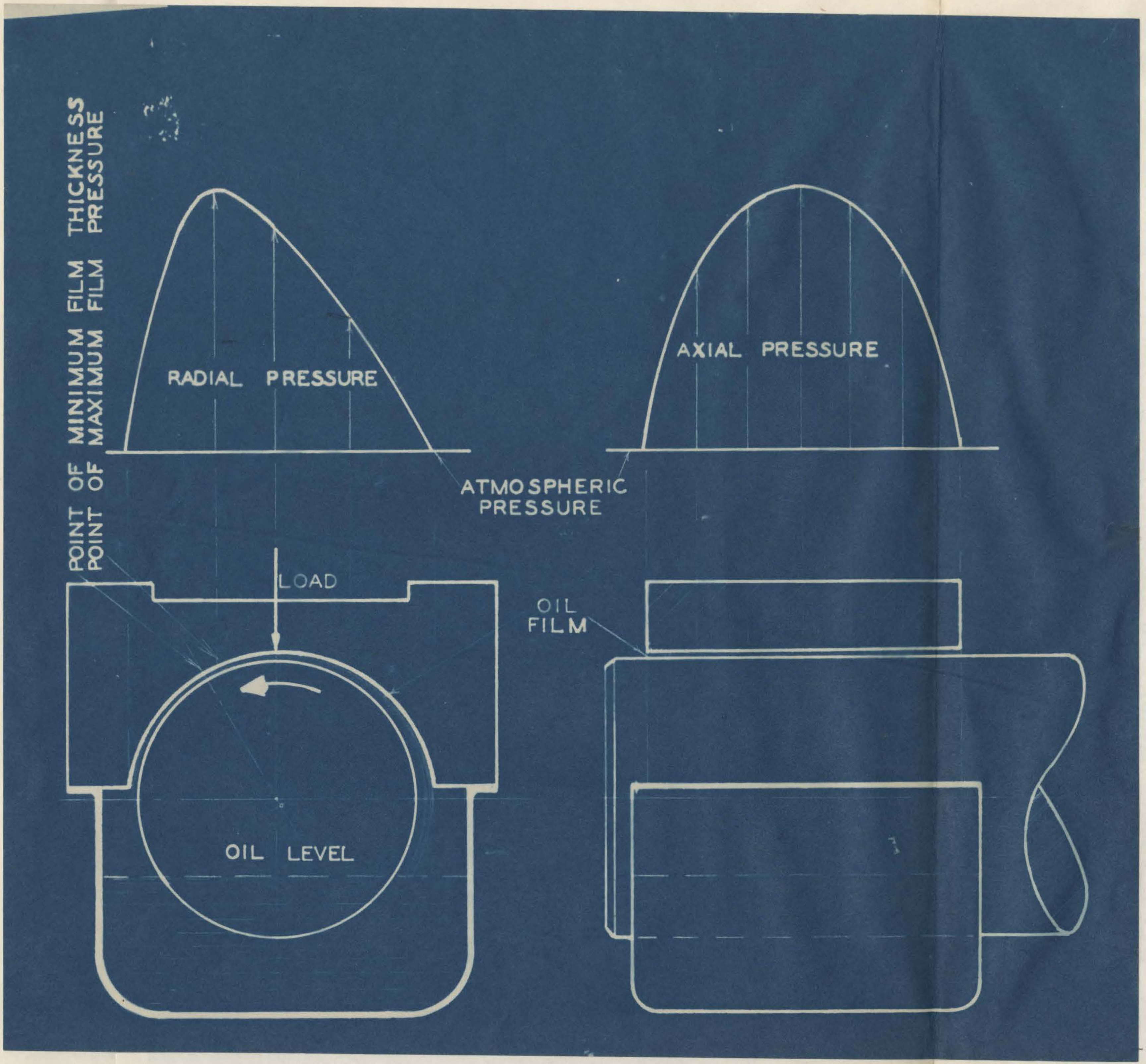


Fig. 17 - Pressure Distribution in the Oil Film of a Partial Bearing.

support the load and to resist squeezing action; the pressure is low enough to allow the film of oil to support the journal; and the rotational speed of the journal with respect to the bearing is great enough to build up a film of oil between the journal and bearing. Pressures are built up within the oil film as shown in Fig. 17. As the speed increases, the frictional force increases, thus increasing the work done in rotating the journal and the temperature of the bearing. This increased temperature decreases the viscosity of the oil, which tends to lower the frictional force and hence heat generation. At the same time the bearing dissipates heat more rapidly, and these effects all tend towards some definite predetermined condition. As the speed of the journal decreases, the heat generated by friction decreases, with a corresponding decrease in operating temperature of the oil with the result that the viscosity of the oil is raised, which tends to raise the carrying capacity of the oil film. These effects tend toward some definite predetermined condition. The heat generated is dependent upon the frictional force, and this force is independent of the pressure. Hence the varying of the pressure does not affect the temperature of the bearing. But it does change the viscosity of the oil, slightly at low pressures and greatly at very high pressures. Thus a raising of the pressure will tend to raise the carrying capacity of the oil film due to a raise in the viscosity of the oil. But the carrying capacity of a bearing as shown by the $\frac{ZN}{P}$ relation is decreased as the pressure is increased. The effects of pressure, therefore, tend toward some definite predetermined condition. It has been shown that while a bearing is operating in the region of perfect fluid film lubrication a very stable condition exists, so the region is oftimes referred to as the region of stable lubrication. The theory of perfectly lubricated bearings is applicable to bearings operating in this region.

In contrast with this region is the region of unstable lubrication, denoted in Fig. 16 as the region of boundary lubrication. In this region, solid friction between the metal surfaces of the bearing and journal is approached. When this condition is approached, a decrease in speed favors the closer contact of the two metal surfaces. This in turn increases frictional resistance, which increases the temperature of the bearing. With this increase in temperature, the viscosity of the oil is decreased which reduces the carrying capacity of the oil and in turn increases the frictional resistance due to the closer contact of the metal surfaces. An increase of pressure provokes this condition. A journal bearing operating in this condition is thus very unstable and likely to fail. The only stabilizing influence is that of the more rapid heat dissipation of the bearing with higher operating temperatures.

The region of thin film lubrication is the condition existing between thick film and boundary lubrication. In this region, the surfaces may be completely separated, yet be so close together that the hydrodynamical theory does not apply. Rather a new phenomenon of the polorization of the oil molecules is applicable. The strength of the absorbed film is increased due to this polorization. Under some conditions, a film of oil may be between the two surfaces, yet so thin that the uppermost ridges of the surfaces of the journal and bearing drag over each other as in Fig. 18. The exact nature of the lubrication acting within this region is not known; however, some theories of the phenomenon in this region have been advanced.

The term oiliness is used to explain some of the effects within this region (see Fig. 19). (26) Oiliness has been defined as "That property which causes two lubricants to possess different coefficients of friction, although their viscosities at the film temperature may be the same." (27) This definition does not account for the effect of pressure upon viscosity. High pressure concentrations may occur in this region due to the irregularities of the surfaces. This pressure will tend to raise the carrying capacity of the oil because of the increased viscosity due to pressure. (28)

Some few definitions of oiliness have been suggested that take into account the effects of pressure on the viscosity. Thus oiliness may be defined as that property which causes two lubricants to possess different coefficients of friction, although their viscosity at film conditions be the same. W. H. Hershell has conducted research on oiliness and has proved the following to be true:

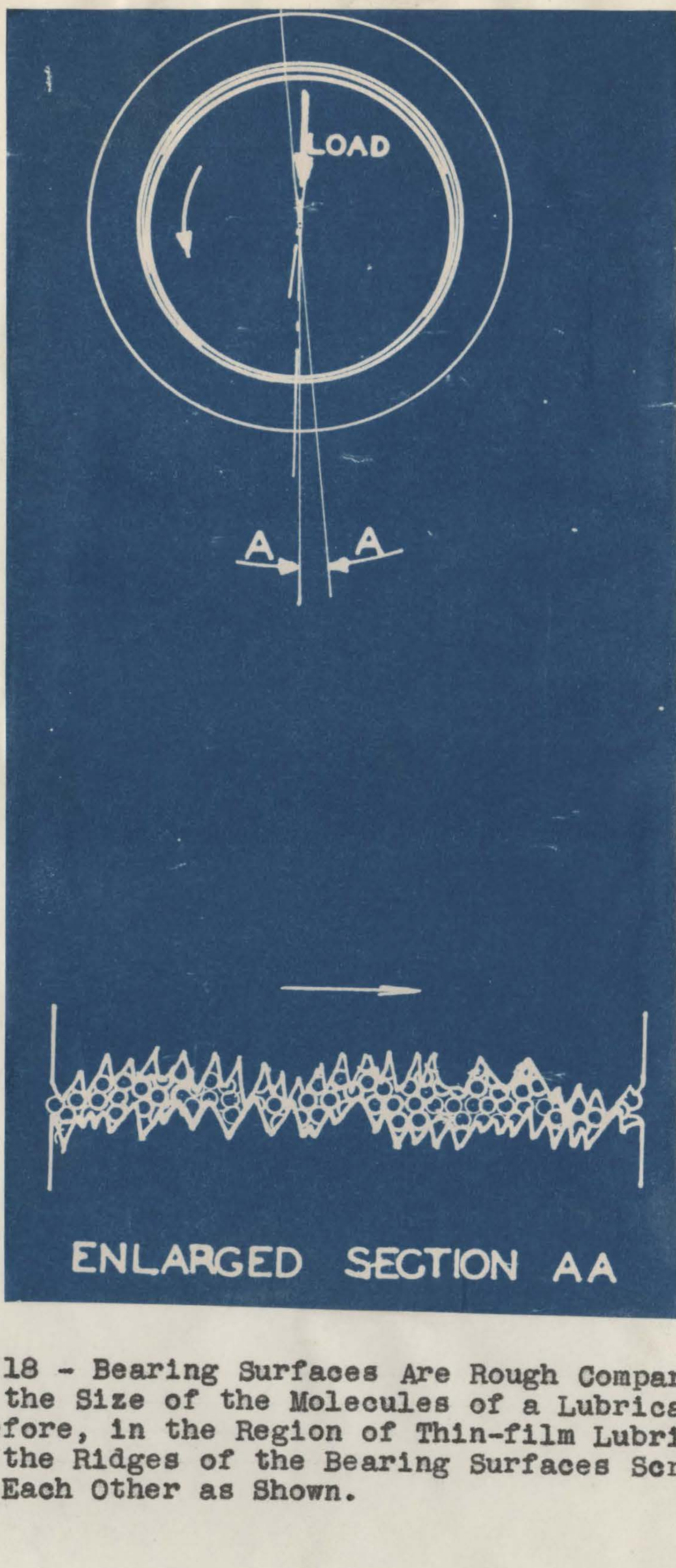


Fig. 18 - Bearing Surfaces Are Rough Compared with the Size of the Molecules of a Lubricant. Therefore, in the Region of Thin-film Lubrication the Ridges of the Bearing Surfaces Scrape over Each Other as Shown.

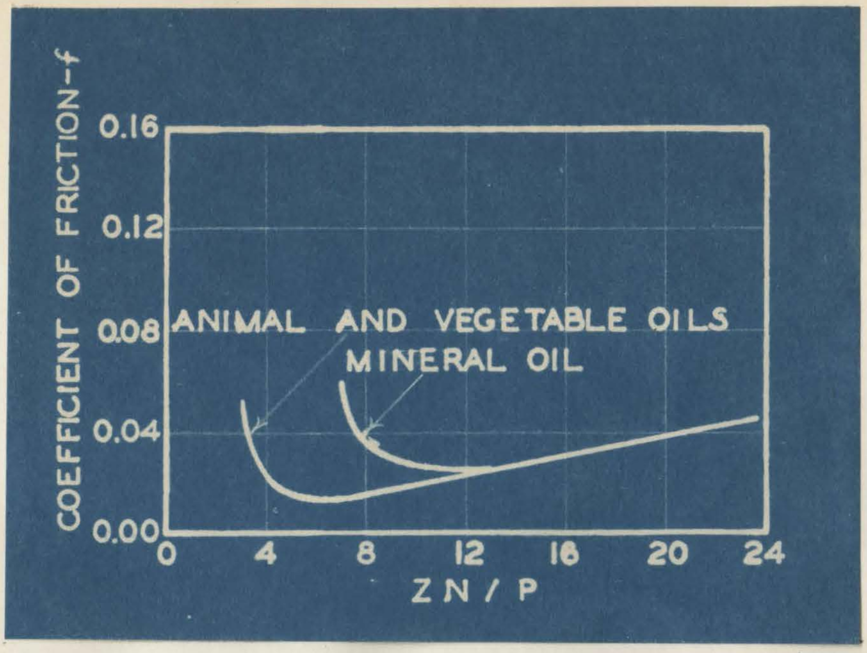


Fig. 19 - Curves (26) Comparing the Oiliness of a Mineral Oil and Animal and Vegetable Oils, the Latter Having Greater Oiliness than the Former. (The effect of oiliness is to lower the minimum value of ZN/P, thus giving the bearing a lower possible coefficient of friction and raising the factor of safety of the bearing.)

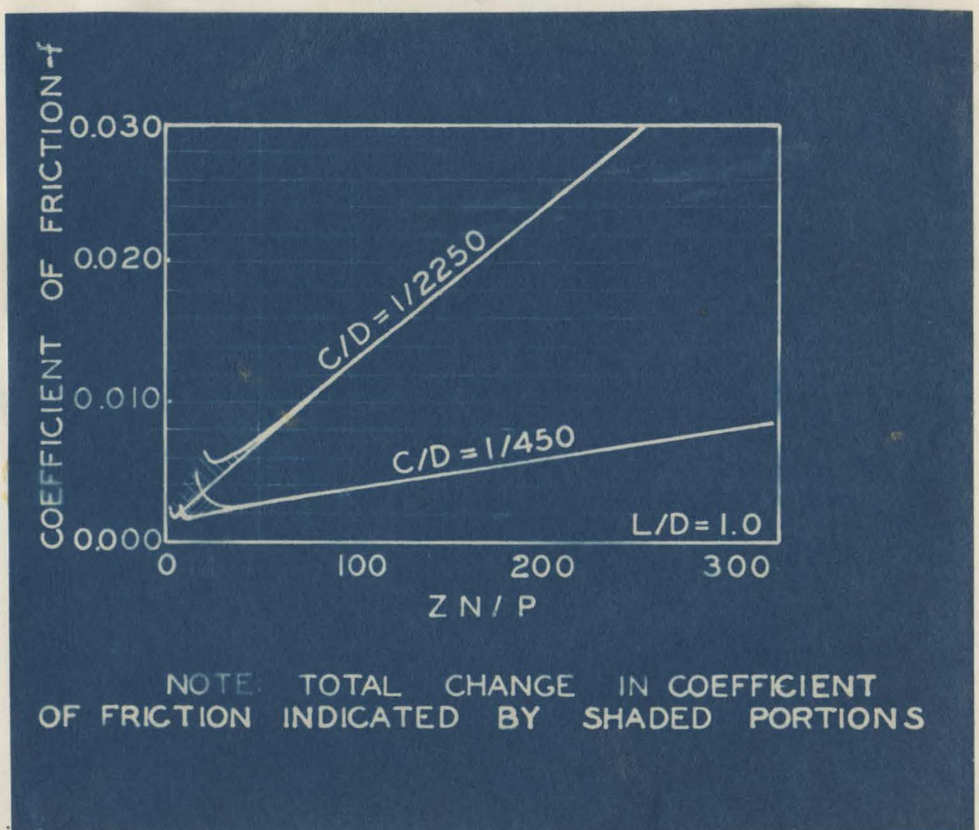


Fig. 20 - The Effects of Running-in on the Performance of Journal Bearings. (The effects have been shown for two different bearings (30), they being to lower the possible coefficient of friction and the minimum value of ZN/P; hence increasing the factor of safety of the bearing.)

"1. Petroleum oils of high viscosity have better oiliness than those of low viscosity.

"2. Lubricants of the type containing lead, soap or oleic acid have remarkable superiority from the standpoint of oiliness.

"3. The oiliness of a given lubricant is not the same when tested in conjunction with different bearing materials." (27)

In connection with the effect of bearing materials, the lead content increases the oiliness. S. A. McKee has shown that the addition of kerosene decreases the oiliness effects and decreases the film strength under severe conditions. (29)

That the effect of running-in on a bearing is much the same as that of oiliness in a lubricant is readily seen by comparing Fig. 20 with Fig. 19. (30) Fig. 20 shows that running-in reduces the frictional losses in the region of thin film lubrication or losses arising from severe operating conditions. For a well designed high-speed bearing, the frictional losses may be reduced greatly by running-in the bearing and then operating at a lower value of $\frac{ZN}{P}$. This reduction may possibly be due to one of two reasons or both. The minute ridges and projections on the surfaces of the journal and bearing may be so smoothed out by running-in that a lower value of $\frac{ZN}{P}$ is reached before these imperfections drag over each other thus increasing the frictional resistance. Or the contour of the bearing may be changed such that this affects the shape of the $\frac{ZN}{P}$ curve. Although these changes do not alter the $\frac{ZN}{P}$ curve at the high values, it is quite possible that they have so great an effect at the low values that the changes will greatly affect such a thin film as is existent under these conditions. (30)

S. A. McKee and T. R. McKee come to the following conclusions regarding the minimum value of $\frac{ZN}{P}$ in their paper, "Journal Bearing Friction in the Region of Thin Film Lubrication:" (23) "Thus, within the scope of these tests, there is a general indication that, for a given bearing and lubricant, an increase in load tends to decrease the value of $\frac{ZN}{P}$ at the point of minimum $\underline{f}$."

"For a given bearing and lubricant an increase in speed tends to decrease the value of $\frac{ZN}{P}$ at the point of minimum $\underline{f}$, and for a given value of $\frac{ZN}{P}$ in the region of thin-film lubrication an increase in speed tends to decrease the coefficient of friction." The McKee tests also show:

That the influence of viscosity on the minimum value of $\frac{ZN}{P}$ must be small, for the tests did not shown any.

That Babbitt bearing metal was superior to bronze in having a lower possible $\frac{ZN}{P}$ value.

It is easily possible, then, that the applied pressure altered the shape of the $\frac{ZN}{P}$ curve due to its effect upon the viscosity of the oil.

The temperature at which a bearing operates has been shown to be one of the chief factors determining the carrying capacity of a bearing. This operating temperature should be controlled. Many analyses and tests show that there is no appreciable temperature rise between the center of the oil film and the bearing itself, thus leading to the advancement of the idea that the heat should be dissipated from the bearing itself and this provision should be made in the design of bearings. (25) (31) But the fact that so many bearings are cooled today by the excess flow of oil through them supports the fact that the rate of oil flow through a bearing has a great influence on the operating temperature of the bearing.

Grooving greatly affects the formation of an oil film. The oil flow out the ends of the journal bearing is dependent largely upon the clearance and indirectly on grooving, for the more perfect the oil film the greater the end leakage.

Howarth's theoretical analysis shows that small clearances give high frictional resistance accompanied by close approach, that moderate clearances give low frictional resistance accompanied by a thick film, and that large clearances give high frictional resistance accompanied by close approach. (32)

An investigation carried out by the National Physical Laboratory of England shows the importance of keeping water from the oil lubricated journal bearing. This investigation shows that 0.05 per cent of water in motor-car oil reduces its seizing temperature by 60^{0} C., and concurrently the minimum friction is increased by as much as 40 per cent. (33)

Experimentation has also shown that the extent of the film of lubricant is dependent upon mechanical stability rather than upon the available bearing arc, and that an excessive bearing length does not increase the carrying capacity of a bearing but only increases its friction. This is due to the fact that the deflection of the shaft, and the impossibility of perfectly machining the journal and bearing causes localized areas to be overloaded in the long bearing. Optimum frictional conditions are obtained with a bearing contact area of approximately square proportions. The offset loading as compared with the central loading of a bearing improves the capacity of the bearing and reduces its friction. (34)

The outline of developments in the theoretical and experimental field of journal bearings just given is the basis for many of the present day methods of design.

One method that has been much employed in the design of journal bearings, and is still used today is the use of the factor <u>PV</u>. All designers of bearings should be familiar with this factor, its importance being indicated by:

Work of friction $=$ F x V

But: $F = W \times f$

$W = P \times A$

Therefore: $F = P \times A \times f$

Work of friction $= P \times A \times f \times V$

$$P \times V = \frac{\text{Work of friction}}{A \times f}$$

Where: F $=$ Force of friction

V $=$ Rubbing velocity

W $=$ Load on bearing

f $=$ Coefficient of friction

A = Area load is distributed over

P = Pressure on bearing

C = Constant

The law upon which this formula is based is that the work of friction is dependent upon the force of friction and the rubbing velocity. This law is applicable to imperfectly lubricated bearings only, for in perfectly lubricated bearings the work of friction is dependent upon the viscosity of the lubricant and the velocity and is independent of the pressure. Hence, the PV factor is not applicable to the perfectly lubricated bearing. It is applicable to the imperfectly lubricated bearing under the following conditions: when the coefficient of friction is constant, and when the bearings compared have the same heat radiating capacity. Under these conditions, this equation is ideal as a basis of bearing design, for bearings fail due to overheating, which is dependent upon the amount of frictional work developed.

Commercial Development of the Bearing after 1900

At the beginning of the nineteenth century, the underlying theory of the bearing was not widely known by engineers. They used as bases for bearing design the empirical rules of thumb. In consequence, some bearings operated satisfactorily, whereas others did not. In many cases the bearings operated under conditions of perfect lubrication, but their designers were unaware of this fact. The journal bearing was generally designed as a structure to carry the imposed load. Lubrication was left to some operator, who generally was not as well qualified to select the correct lubricant as was the designer and maker. With the evolution of large scale production of certain machinery, the responsibility of specifying lubricants and providing for their proper application has been assumed by the manufacturer. The development of this practice was largely dependent upon the standardization of lubricants and lubricant tests--a practice still largely in the developmental stage.

The early practice in bearing manufacture was to hand scrape the bearing, so that the journal fitted rather snugly and to let the journal and bearing wear to a good fit. This practice is still used, but the prevalent method is to machine the bearing carefully to the proper clearance and finish. The new methods of lapping shafts and burnishing bearings give highly finished surfaces, and this practice not only gives rapid and interchangeable production so necessary today but provides highly finished surfaces which are very desirable.

For a long time Babbitt was applied to the bearing shell by pouring Babbitt into the shell from a ladle. A mandrel, positioned in the middle of the bearing, was used as a core. Tinning and peening were found to aid. Small steel anchors or large dovetail anchorage spaces were later provided to aid in holding the Babbitt to the bearing shell. The latest development, that of centrifugally casting the Babbitt to the shell, has overcome many troubles encountered in the older practice.

The development of the journal bearing has been influenced mainly by the demands of manufacturers for increased dependability and lower initial and maintenance costs. The machine makers must design a dependable journal bearing unit, including a dependable method of lubrication. This unit must be of minimum size for low initial cost, and must be efficient, having minimum power losses. This demand has encouraged research in the journal-bearing field. The developments in some fields have been tremendous whereas in other fields, developments have lagged, as the need for better designs has not been as urgent.

Many machines running today are examples of bearing practices of yesterday. The machines are still good and are very useful, but they are usually larger and are not as efficient as are present day machines. The old railroad journal is still very common, and just now the old design is being recognized as being very inefficient. Older practice depended entirely upon oil being fed upward through waste. More positive lubrication is now being provided. Improper grooving of the older bearings was prevalent, the groove often being placed in the high pressure area.

In 1905, mineral oils devoid of vegetable and animal oils were recommended for bearing lubrication for the first time. (35) Solid lubricants were recommended for the most severe cases. Since then, lubricants have been highly developed and can withstand the more severe conditions imposed upon them. These refinements have been brought about by new refining methods and new additive agents. The number of lubricants to choose from today are numerous.

Table I.[2] gives an idea of the practices prevailing in the early part of the nineteenth century.

Table II. gives pressures which were published in 1911 and recommended for use in gas engine design. (36) Table III. shows limits on soft Babbitted bearings specified in literature in 1916. (37), (38), (39)

Due to the constant raising of the specific output of the sutomobile engine, its bearings have had progressively more severe conditions imposed upon them. At first splash lubrication was used, but as conditions within the engine became more severe, forced lubrication was introduced.

---

[2]From "The Mechanism of Lubrication" by Robert E. Wilson and Daniel P. Barnard. (26) Results primarily based on Alford's "Bearings and Their Lubrication."

TABLE 1--VALUES OF zN/p FOR VARIOUS RECOMMENDED CONDITIONS OF OPERATION

Location of Bearing	Lubricant	p	N	z	z N/p	c/D
Automobile Crankshaft	Medium Machine Oil	300 to 700	900 to 1,400	7 to 8	15 to 25	0.0010
Aeronautic Engine Crankshaft	Heavy Engine Oil	300 to 1,800	1,800 to 2,000	7 to 8	15 to 25	0.0010
Stationary Gas-Engine Main	Medium Machine Oil	500 to 700	250 to 800	30	25	0.0010
Stationary Gas-Engine Crank-pin	Medium Machine Oil	1,500 to 1,800	250 to 800	50	15	0.0010
Stationary Gas-Engine Cross-head	Medium Machine Oil	1,500 to 2,000	250 to 800	40	10	0.0010
Diesel Engine Main	Heavy Engine Oil	250 to 600	60 to 160	30	15	0.0010
Diesel Engine Crankpins	Heavy Engine Oil	1,500 to 4,000	60 to 160	40	2 to 5	0.0010
Marine Steam Engine Main	Machine Oil	275 to 500	180	30 to 40	20 to 30	0.0010
Marine Main Crankpin	Machine Oil	400 to 500	180	30 to 40	20	0.0010
Stationary Slow-Speed Main	Heavy Machine Oil	80 to 400	40 to 80	70	20	0.0010
Stationary Slow-Speed Crank-pin	Heavy Machine Oil		40 to 80	80	6 to 8	0.0010
Stationary Slow-Speed Cross-head	Heavy Machine Oil	1,000 to 1,500	40 to 80	70	5	0.0010
Stationary High-Speed Main	Engine Oil	60 to 250	360	15	25	0.0010
Stationary High-Speed Crank-pin	Machine Oil	400 to 1,500	360	30	6 to 15	0.0010
Stationary High-Speed Cross-head	Machine Oil	1,500 to 1,800	360	25	5	0.0010
Locomotive Drive-Wheel	Heavy Machine Oil	550	250	100	30 to 50	0.0010
Locomotive Crankpin	Heavy Machine Oil	1,500 to 2,000	250	100	5 to 8	0.0010
Locomotive Cross-Head	Heavy Machine Oil	3,000 to 4,000	250	130	6 to 8	0.0010
Marine Steam Turbine	Light Machine Oil	85	2,000	10	250	0.0010
Stationary Steam Turbine	Machine Oil	400 to 950	2,000	20	100 to 200	0.0010
De Laval 7-Hp. Steam Turbine	Light Machine Oil	7 to15	30,000	1	1,500 to 3,000	0.0020
De Laval 300-Hp. Steam Turbine	Light Machine Oil	20 to 25	10,500	2	1,000	0.0020
Railway Car Axle	Heavy Machine Oil	300 to 450	300	100	50 to 100	
Generator and Motor	Engine Oil	30 to 80	150 to 500	25	200	0.0010
Rolling Mill Main	Hot Neck Grease	1,800 to 2,500	60	--	1	--
Cotton Mill Spindle	Spindle Oil	1	8,000 to 12,000	2	10,000	0.0050
Gyroscope		750 to 850	800 to 1,500	60 to 30	55	0.0013

The method of splash lubrication required large grooves. Then, too, the length-diameter ratio was relatively large, requiring this type of grooving. The grooving was of helical form, and the grooves crossed at right angles at the oil hole.

As the loads progressively increased with the development of the engine, the demands on the bearing metals increased. At first brass or bronze bearings were used but shafts were unhardened, medium steel that wore rapidly; therefore, the soft bearing metal Babbitt, was introduced. When the load-limit on Babbitt was reached, the bronze-back, Babbitt-lined bearing was developed and this combination was satisfactory for a time. During the World War, however, progress in engine design demanded better bearings. The centrifugal method of casting was soon developed, and this was used in casting Babbitt in the bronze-back. Because of higher temperatures encountered, first in the bearings of airplanes and then in those of automobiles, further developments were necessary. Thin Babbitt-lining in a steel-shell was found to be better than lining in a bronze back. Copper-back bearing alloys were developed. The production cost of the making and the bonding of the copper-lead alloy to the steel-back was very high but necessary in some cases on account of the heavy loads and high temperatures. Satco metal came into use in the United States in 1933. Research has made available the advanced bearing metals we have today, these being the nickel-bronze, the beryllium-copper and the cadmium base alloys. The

Table II. Average Maximum Bearing Pressures for Gas Engines as Calculated in 1911.[3]

For Horizontal Engines						For Vertical Gas Engines						
D	4	8	12	16	20	D	4	8	12	16	20	Assu.*
D_{cp}	1 1/2	3 1/8	4 3/4	6 3/8	8 1/8	D_{cp}	1 5/8	3 1/4	4 7/8	6 1/2	8 3/16	Cal. **
L_{cp}	1 5/8	3 1/4	4 7/8	6 9/16	8 3/8	L_{cp}	1 5/8	3 5/8	5 5/8	7 5/8	9 5/8	Cal.
A_{cp}	2.44	10.15	23.2	41.75	68	A_{cp}	2.64	11.8	27.8	49.75	78.75	DxL
P_m	250	250	250	250	250	P_m	250	250	250	250	250	
K_{cp}	1290	1240	1220	1210	1150	K_{cp}	1190	1065	1035	1015	995	Cal.
P_m	300	300	300	300	300	P_m	300	300	300	300	300	
K_{cp}	1550	1485	1450	1450	1390	K_{cp}	1430	1280	1240	1215	1200	Cal.
P_m	350	350	350	350	350	P_m	350	350	350	350	350	
K_{cp}	1800	1730	1710	1690	1620	K_{cp}	1660	1490	1440	1420	1400	Cal.
P_m	400	400	400	400	400	P_m	400	400	400	400	400	
K_{cp}	2060	1980	1950	1930	1850	K_{cp}	1920	1720	1660	1620	1600	Cal.

D = Cylinder diameter in inches

P_m = Maximum explosion pressure in lb./sq. in. of piston face

D_{cp}= Bearing diameter of crankpin in inches

L_{cp}= Bearing length of crankpin in inches

K_{cp}= Maximum unit, bearing pressure--lb./sq. in.

A_{cp}= $D_{cp} \times L_{cp}$, in sq. ins.

Maximum $K_{cp} = \frac{\pi D^2}{4} . P_m \div (D_{cp} \times L_{cp})$

*Assu.---assumed

** Cal.---calculated empirically

[3]These tables are taken from American Machinist--Nov. 9, 1911.

Table III. Limits on Soft Babbitted Bearings as Specified in Literature in 1916.

Conditions	Allowable Pressure lb./sq. in.
High speed shafts in Babbitted bearings	10-30
Large slow speed shafts in Babbitted bearings	100
Dead load on soft Babbitted bearings	300-500

cadmium-silver alloy was not available commercially until 1935.

The Diesel engine commercially is a relatively new type of engine. The first one produced in America was built in 1898 by Adolphus Busch. Commercial production of the Diesel was started in 1903. In the early days, one type of Diesel engine was lubricated with a mixture of oil and water by the splash method. Another type was lubricated by sight-feed cups, and banjo oilers. Later, the methods used in even one engine were numerous: one type having a forced-feed mechanical lubricator to feed oil to the cylinder, a scoop to feed oil to the piston pin, ring oilers for the main bearings, and banjo oilers for the crankpin bearing. Still later, the mechanical forced-feed lubricator was used to supply a larger portion of the lubrication, and ring and chain oilers came into greater prominence.

The development of the Diesel engine has been similar to that of the automobile. Higher specific outputs have been the tendency. This has been accomplished with correspondingly higher speeds; more compact engines, requiring shorter bearings; lower length-diameter ratios, as the bearings are shorter and the shaft diameters larger to avoid the critical speeds encountered at high speeds; and higher bearning loads due to increased speeds and greater mean effective pressures. Thus, the developments in the bearings for automobiles, airplanes, and Diesel engines have paralleled each other, the development of each engine type augmenting and influencing developments in the others.

Rolling mill problems have been entirely different from those of internal combustion engines. The trend has been to increase the speeds of the mills and at the same time to increase their dependability. In addition, there has been a constant demand for closer tolerances. At first bronze bearings were used so that high loadings could be withstood. Lubrication was effected by packing the necks with a fairly hard grease, and the bearing temperature was kept low by playing a steady stream of water on the roll neck. Often the neck was lubricated only by the water because the grease either ran out or lost contact with the roll neck. Frictional losses and material costs were reduced in some cases by using Babbitted bearings with a bronze base. Not long ago the advantages of supplying a constant supply of grease to the bearings were realized. Mills were often equipped with new automatic systems employing soft grease at this time. These centralized systems lowered lubricating costs and power losses as well as greatly lengthing the bearing life.

Typical of old practice can best be described by a test that was taken on one mill. (40) The mill tested was a strip mill that had ordinary plain open type, short bearings. The wear on these bearings was so great that they had to be taken up after each bar was rolled. Water was used copiously on this bearing to cool it. To show what experimentation on bearings and a knowledge of the theory will accomplish, the bearings were replaced by well designed oil lubricated bearings. The mill was then capable of 50 per cent higher reductions, even with a reduced power consumption of 40-50 per cent. The newly designed bearings did not wear to any noticeable extent.

Table IV. indicates the advantages of the new non-metallic bearing materials. (41)

Rolling mills have been dependent upon the old type of bearings for some time, but the development of the new oil lubricated bearings and the introduction of non-metallic bearing materials have brought about appreciable improvements.

The sleeve bearing industry today is very important. It had its beginning in 1914 when one large foundry set up a standard line of sleeve bearings, using as a basis for its standard sizes the averages derived from a large number of orders. (42) This industry grew rapidly; research has provided better bearing materials and a thorough understanding of the limitations of bearing materials.

During this period of development all the modern methods of lubrication were known but were in a somewhat undeveloped state. The finer developments in the methods of lubricating bearings have come about with the developments in machinery in general. Each is dependent upon the other. A well-built machine is virtually useless if it is not properly lubricated, and likewise an expensive lubricating system on a cheap machine is not economical. Hence lubricating systems and machines have

Table IV. Power Tests on 12-inch Skelp Mill.[4]

Types of Bearing	Bar size, in.	Percentage of Reduction	Rpm	Hp. input	Friction Load, hp.
Lignum Vitae	7 11/16 x 0.225	20	215	362	41.8
Babbitt	7 11/16 x 0.225	20	215	467	73.0
Babbitt	7 11/16 x 0.18	13.9	258	285	28.0
Babbitt	7 11/16 x 0.18	13.9	258	628	131.0
Babbitt	7 11/16 x 0.18	13.9	366	228	50.0
Babbitt	7 11/16 x 0.18	13.9	365	965	104.0

[4] From "Steel-plant Lubrication" by J. F. Pelly, Mechanical Engineering, January, 1935. p. 28.

advanced together. However, the machine owes much of its development to lubrication development. At first, difficulty was encountered in properly lubricating engines and turbines. Wick feeding of the oil to each individual bearing was common in early practice. From this method evolved the mechanical lubricator, which employed a reservoir and a pump that forced the oil through individual pipes to each bearing. Next, the gravity circulation system came into use, the oil flowing down into the bearings from an overhead reservoir, and thence to a sump where it was pumped back to the reservoir. The pressure circulation system was another advancement, the oil being circulated by means of a pump through all the bearings. The oil then drained from the bearings to a sump from which it was again forced through the bearings. Hand oiling, drop feed oiling, waste or pad oiling, ring oiling, chain oiling, and bath oiling were all employed and are still in common use.

PRESENT PRACTICES

Designers today are still after the same results they were a few years ago; viz., that of designing dependable, durable, and efficient machines at low cost. Hence many designers are applying the latest developments possible in bearing practice to their new designs. This effort makes for progressively better design practices. Where the machines are complicated and are depended upon for steady production, an intricate and wholly automatic lubrication system is warranted and incorporated in the machine. Careful consideration is being given to keeping contaminants from the journal bearing units. Testing of designed journal bearings to find their capabilities in service is now common practice in industries that can afford the expense. The design engineer has to rely solely on these tests and past experience in many cases, for he cannot adequately foretell such effects as distortion due to temperature effects, the effectiveness of cooling, or the effects of misalignment and of different surface conditions of the journal and bearing. Also the bearing metal used and its method of application have a great effect on the probable life of the bearing.

In the following discussion of "Present Practices" some of the material published today on bearings will be assembled in the hope that it will be of some aid in the designing and in the future development of bearings.

Lubricating Systems

In order for a bearing to operate efficiently, it must have a sufficient amount of oil in it at all times. A well-designed bearing will soon wear and be ineffective unless it is continuously and correctly lubricated. The proper method of lubrication should be incorporated in the journal bearing when it is designed and built. By proper lubrication of the journal bearing is meant: supplying the correct oil at the right place in the right quantity.

a. The correct oil is one that under all operating conditions will provide perfect fluid film lubrication.

b. By supplying the oil at the right place, it will be distributed uniformily over the bearing, with little loss in end leakage. Grooving is the most important factor in the distribution of the oil to the right place.

c. When the oil is supplied in the right way, it is supplied regularly in the exact quantities that are demanded by the journal bearing. (43) The amount required by the bearing is surprisingly small in those cases where it is not used as a coolant. Only enough is needed to make up the small losses due to evaporation and end leakage.

The engineer today has many lubricating appliances at his command, a list of which is given in Table V.

To meet the third requirement of properly lubricating a bearing, that is, to supply the oil in the exact quantities

Table V. Classification of Oiling Appliances.[5]

I. Hand squirt cans
  1. a. push bottom type
     b. enclosed pump type

II. Drop-feed cups
  1. a. single feed
     b. multiple feed
  2. a. variable level
     b. constant level
  3. a. gravity feed
     b. pressure feed

III. Wick-feed cups and boxes
  1. a. single feed
     b. multiple feed
  2. a. variable level
     b. constant level

IV. Bottle oilers
  1. a. globe body
     b. flat sided body

V. Ring, chain, and collar oilers

VI. Manually or hand operated oilers

VII. Hydrostatic lubricators

VIII. Mechanical operated lubricators
  1. a. ratchet drive
     b. rotary drive
     c. clutch drive
  2. a. single feed
     b. multiple feed

IX. Oil Pumps
  1. a. gear type
     b. rotary type
     c. centrifugal type

X. Centralized appliances

[5] From "Oiling Appliances and Systems" by J. I. Clower. (44)

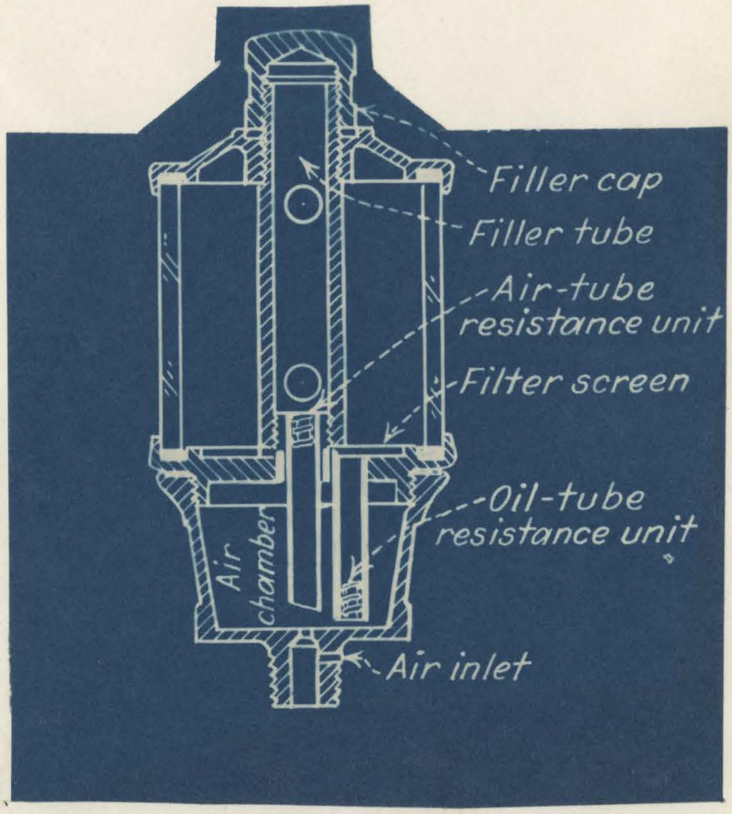


Fig. 21 - Temperature Regulated Oil Cup.

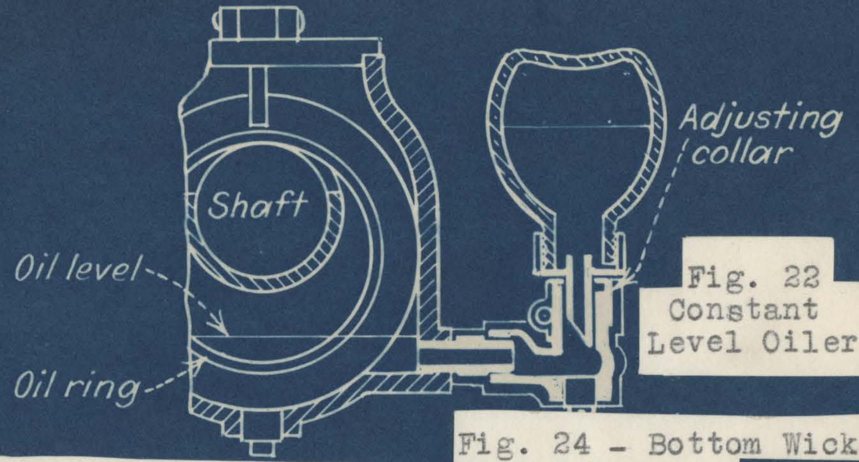


Fig. 22 Constant Level Oiler.

Fig. 23 - Wick-Feed Oil Cup.

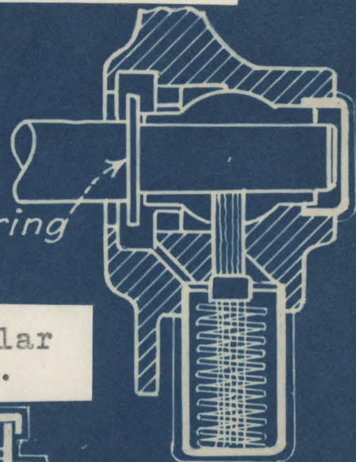


Fig. 24 - Bottom Wick-Feed Oiler.

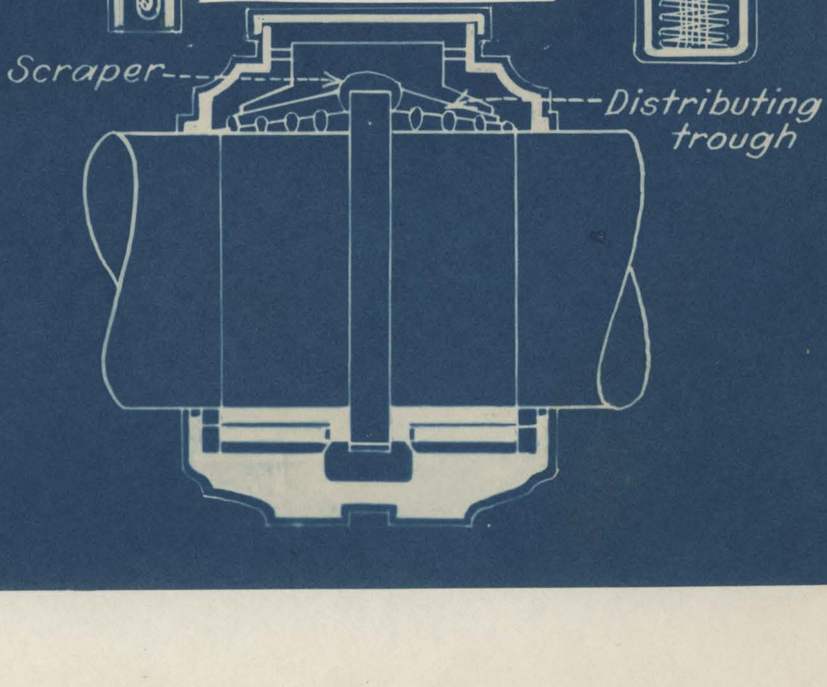


Fig. 25 - Collar Oiled Bearing.

which are demanded by the bearing, the lubricating system must meet the following requirements:

(a) It must always function, for even a momentary failure may result in the bearing's not getting its supply of oil. It must be reliable.

(b) It must always supply oil in the exact quantities demanded by the bearing. A journal bearing needs no lubrication unless it is in operation, so to meet this condition the lubricating system must stop when the journal stops, and it must start when the journal starts. Too much dependency should not be placed upon operators.

(c) To supply the oil in the exact quantities, it is evident that the system used must be susceptible to close-feed regulation, since the quantities demanded in most cases are very small.

(d) In addition to the close regulation required, the system should be capable of supplying variable amounts of oil depending upon the quantity demanded. In general, the amount of lubricant required increases with the speed. When the oil is used as a coolant, more oil is required to cool the bearing at high than at low speeds. Also, end leakage is greater at high than at low speeds.

To reduce maintenance cost, the system should be easily accessible for cleaning and repairing. Inaccessible systems are too often overlooked when a checkup should be made. Expense is incurred in keeping the oiling systems supplied with oil, and for this reason they should be easy to refill with oil.

Table VI. lists most of the oiling systems in use today, the type of service for which they are fitted, and their characteristics.

"In this table by the term non-automatic is meant that the system starts and stops functioning when the machine starts and stops. Only those systems in which the oil is used over and over are considered efficient. Certainly, a system in which oil passes through the bearing once and then runs to waste cannot be classified as efficient. Systems in which the rate of feed is affected by oil level change and temperature variations are not considered to give a regular supply. Hand and drop-feed oiling are not considered reliable because of the human element in the former and the susceptibility of the needle valve in the latter to become clogged with dirt and deposits." (44)

Clower gives a complete discussion on the drop-feed oiler. (44) He also gives pertinent information relative to the design of wick-feed oilers. A booklet on the design and application of the wick-feed oiler has been prepared by the Socony-Vacuum Oil Co., Inc. (45) This booklet also discusses the bottom-feed wick oiler. Karelitz (46) has made a complete investigation of the bottom-feed wick oiler.

The bottle oiler accomplishes proper lubrication at a very low cost. It is a relatively new development and is an important one. The principles of its operation, its construction, application, and field of service are completely described in the booklet, "Bottle Oilers," published by the Socony-Vacuum Oil Co., Inc. (43)

The ring oiled bearing has wide application. It is used widely because it is so efficient, reliable, and economical. It will supply oil in uniform quantity if the oil level is constant and if the ring speed is constant. When the journal increases in speed, the oil supply to the bearing increases within limits. Little attention is needed in a properly designed unit. "It has been found that properly housed motor-sleeve bearings in eight-hour daily use have required oiling not oftener than once every two years and, in some cases, similar bearings have run nearly four years with a single oiling." (47) The housing should be large enough that the impurities getting into the lubricant will settle out and that the heat will be more readily dissipated.

The ring oiling method may not be so satisfactory for a shaft of less than two inches diameter. It is not applicable to cases in which excessive vibration is present. In high speed application, where there is demand for a more copious supply of oil, ring oiling is deficient. The increase in speed causes slippage of the ring, churning, frothing, and centrifugal force throws oil from the ring. Forced-feed lubrication is now replacing the ring-oiled, water-cooled bearing in high-speed applications. The tendency is also to increase the

Table VI. Oiling Systems and Their Characteristics.[6]

Name of System	Type of Service	Characteristics
Hand oiling	Low speed bearings, open gears, chains, wire rope, etc., of relatively cheap and rough machinery.	Non-automatic, unreliable, very inefficient and irregular. First cost low but maintenance usually high.
Drop feed oiling	Plain and anti-friction bearings, chains, gears, etc. Commonly used on steam engines, air and ammonia compressors, internal combustion engines, various machine tools, etc.	Non-automatic, unreliable, irregular, inefficient, adjustable, relatively cheap, adjustable to a certain degree.
Wick-feed oiling	Paper mill, rubber mill, cement mill, woodworking, ceramic plant machinery; railway truck axles, small electric motors, general bearings, etc.	Non-automatic, irregular, reliable, adjustable, moderately efficient, and relatively cheap.
Bottle oiling	Plain horizontal bearings of small and medium sizes of machine tools; shafting in cotton, silk, rayon, and woolen mills; conveyor shaft bearings, blowers, etc.	Automatic, adjustable, reliable, regular, moderately efficient and cheap. Cannot be used where glass bottles are likely to be broken.
Ring, chain, and collar oiling	Electric motors, fans, blowers, centrifugal pumps, steam engines, line-shaft bearings, etc.	Automatic, efficient, reliable, regular, clean, minimum attention, limited to horizontal bearings.
Bath oiling	Enclosed gears, chains, wire cables, plain and anti-friction bearings, thrust bearings, etc.	Automatic, efficient, reliable, regular, minimum attention, requires oil-tight housing, high first cost, limited application.
Splash oiling	Bearings grouped in oil-tight housings, enclosed gears of all types, cross-head pins and guides of steam engines, compressors, internal combustion engines, etc.	Automatic, efficient, reliable, regular, limited application. Oil must be maintained at correct level.
Circulation oiling	High-speed, heavy loaded bearings, high-grade gearing of all types, machine tools, steel mill machinery, steam turbines, fans, blowers, etc.	Usually automatic, reliable, positive, regular, adjustable, efficient, wide range of application, first cost high.

[6]From "Oiling Appliances and Systems" by J. I. Clower. (44)

capacity of the ring oiler that it may replace the more complicated and more expensive forced circulation system. Tichvinsky and Baudry (48) have shown that rings with internal circumferential grooves have greater capacity and are more suitable for high speed applications than the older type of oil ring.

Figure 26 gives various cross-sectional shapes of rings commonly used.

Table VII. indicates what general practice has determined with regard to ring oiling design.

An important development in the lubrication of bearings is the use of very high pressures in the oiling system. Pressures are used in some cases that are capable of supporting the whole load on the bearing. Thus no metal to metal contact can exist at any time and wear is virtually impossible. In other cases high pressures, sufficient to lift the shaft from the bearing, are used only in the starting up period to reduce starting torque and wear.

Refinements are being made in the circulation systems by adding such devices as heaters, coolers, rectifiers, centrifuges, and filters. These devices, used properly, keep down contaminants in the oil. Hermetically sealed units are now in use, these units being free from contamination and oxidation.

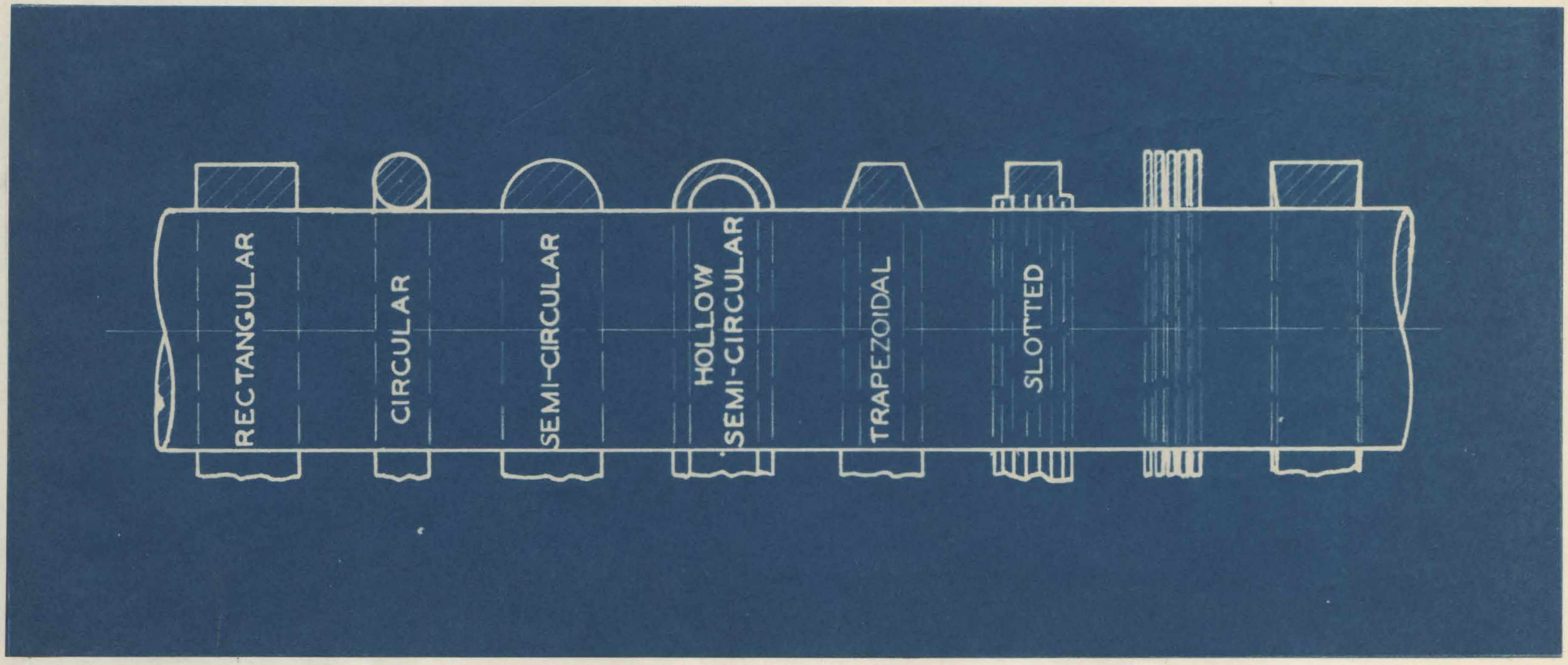


Fig. 26 - Cross Sectional Shapes of Oil Rings Commonly Used.

Table VII. Factors to Observe in the Design and Use of Oil Rings.

Diameter of ring = 2 times diameter of shaft.
One piece ring usually preferred.
Ring should be a perfect circle.
Rings are made heavy enough to stand machinery and handling.
Oil rings are capable of supplying oil to 4 inches each side of ring.
Oil level should be maintained constant at about half the ahaft diameter below the bottom of the shaft.
Oil reservoirs should be large enough to aid cooling and settlement of impurities to bottom of oil well.
For slow speed application, chain oiling is preferable.
For high speed application, the ring may slip and not be able to supply a sufficient amount of oil to the bearing, thus necessitating the use of another method of oiling.

Grooving

The importance of grooving in order to obtain the correct distribution of oil in bearings cannot be overemphasized. In some cases grooving is as unnecessary and objectionable as it is essential in other cases.

The primary purposes of grooving is to promote the formation of a film of oil between the journal and the bearing, and to promote the continuity of this film. Therefore, grooving must be so located that it will conduct oil from the low pressure area to the high pressure area. Grooving in the high pressure area is detrimental, for the pumping action of the journal actually forces oil from the bearing, and results in a deficiency of lubricant. Moreover, grooving in this area actually scrapes oil from the journal and breaks the supporting film. In addition, the load carrying area of the journal is greatly reduced. Figure 27 shows the effects of the incorrect oil groove on hydrodynamic pressures.

Tests on a glass bearing show that with the oil entering through a single hole, a band of oil was formed one-half an inch in width. (49), (50) This result was obtained in the case of no force feed. When forced feed was employed, large quantities of oil passed through the bearing and no cavitation took place. Such results indicate that in normal application, where forced-feed lubrication is not employed, some type of grooving should be used. This is evidently due to the fact that a bear-

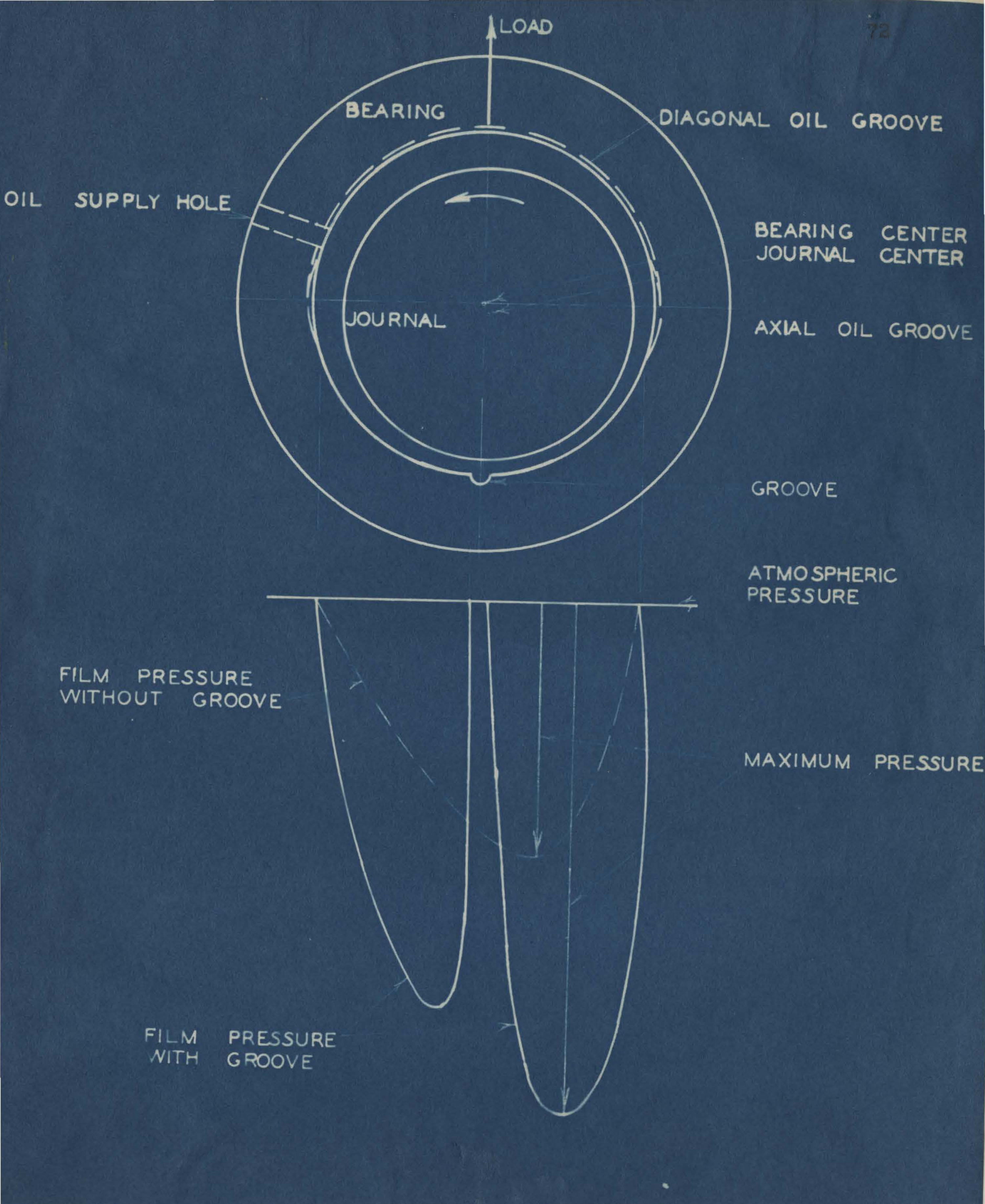


Fig. 27 - Effects of Oil Groove in the High Pressure Area of a Bearing. (The continuity of the oil film is broken, causing excessive film pressures within the bearing. These high pressures are likely to rupture the film with corresponding destructive results.)

ing only partially lubricated, as it would be without grooving, would wear rapidly and be subject to excessive power losses.

These tests also showed that for a light oil, the oil was distributed one-half an inch on each side of a groove, whereas a heavy oil was distributed 1/8 to 3/16 inches each side of the groove. The perfection of the oil film was found to be dependent upon the grooving. In general, a bearing of over one inch in length needs a groove that extends to within one-half inch of each side of the bearing. The groove should not extend further, for then excessive end leakage will result.

Figure 28 shows the correct grooving for a journal bearing. The groove has been placed in the region of lowest pressure, and the oil easily enters the bearing and is pumped by the journal into the high pressure region. The fundamental consideration in the grooving of bearings is to determine the direction of loading, and then the region of low pressure of the bearing, placing the groove in this region.

When the bearing is of the split type, chamfering of the edges is all that is required in the majority of the cases. In any grooving application the groove should be smooth in order to avoid the cracking of the bearing material, and the edges should be well rounded to prevent the scraping of the lubricant from the journal as in Fig. 29. For bearings of long length, more than one source of oil supply should be used to insure the proper distribution of the lubricant. These supply sources should be connected with a groove. Generally, one supply hole

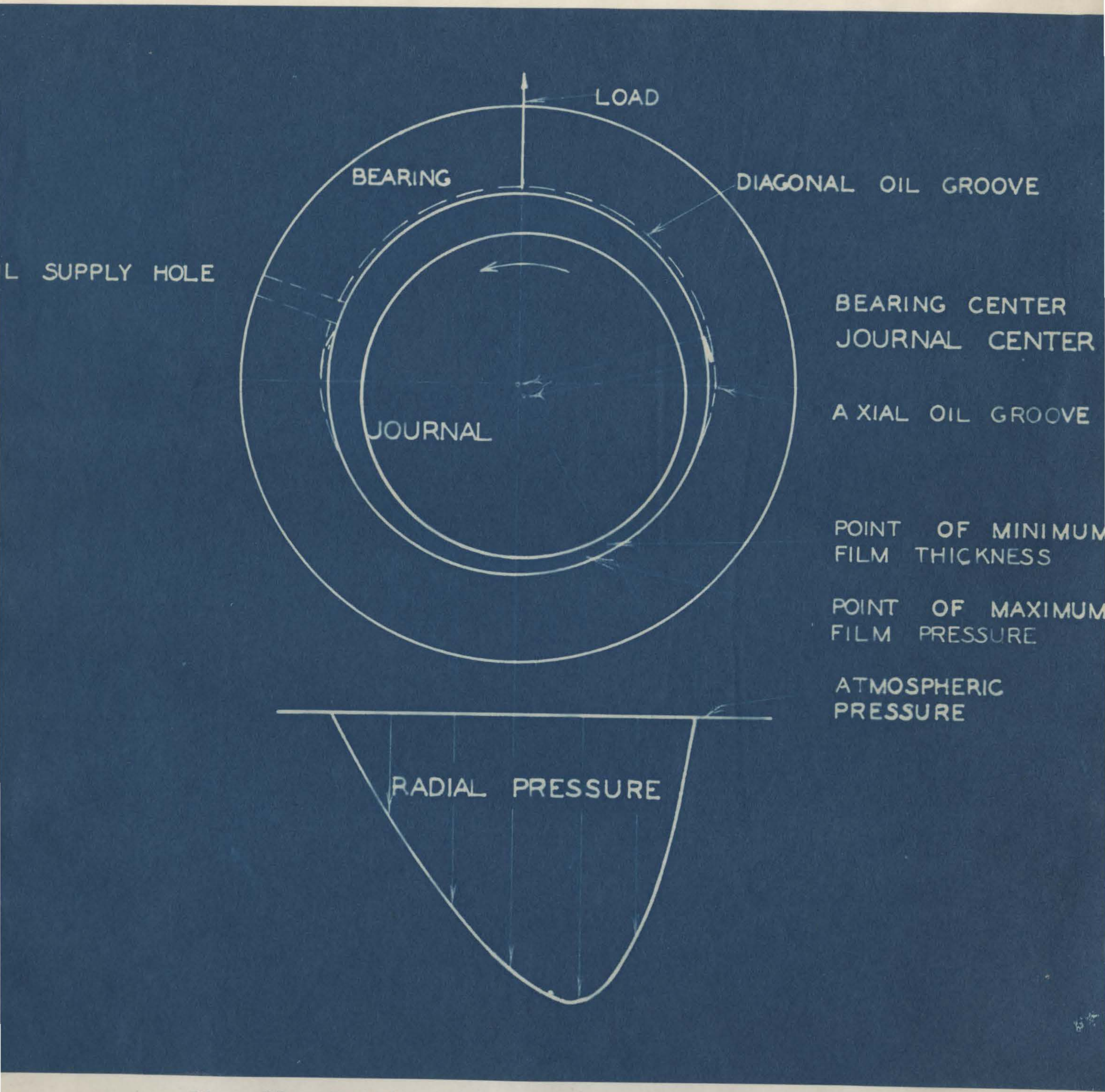


Fig. 28 - Bearing with Grooving in the Proper Region so That the Formation of an Oil Film Is Aided.

will adequately lubricate 8 inches of bearing length, provided the proper axial groove is employed. Should a heavy-bodied oil be used, an additional groove to the chamfer should be used to insure proper distribution of the lubricant.

When the bearing is of the slow speed type and heavily leaded, an auxiliary groove as shown in Fig. 31 should be provided. This groove should be placed ahead of the high pressure region and should be located in the region where the pressure starts building up. It should be shallow, with well rounded edges, and not very wide. Correct grooving for an oil ring bearing is illustrated in Fig. 33.

Accepted methods of grooving are adequately described in Grooving Oil-Lubricated Cylindrical Bearings by the Socony-Vacuum Oil Co., Inc., (51) and "Grooving Bearings in Machines" by G. B. Karelitz. (52)

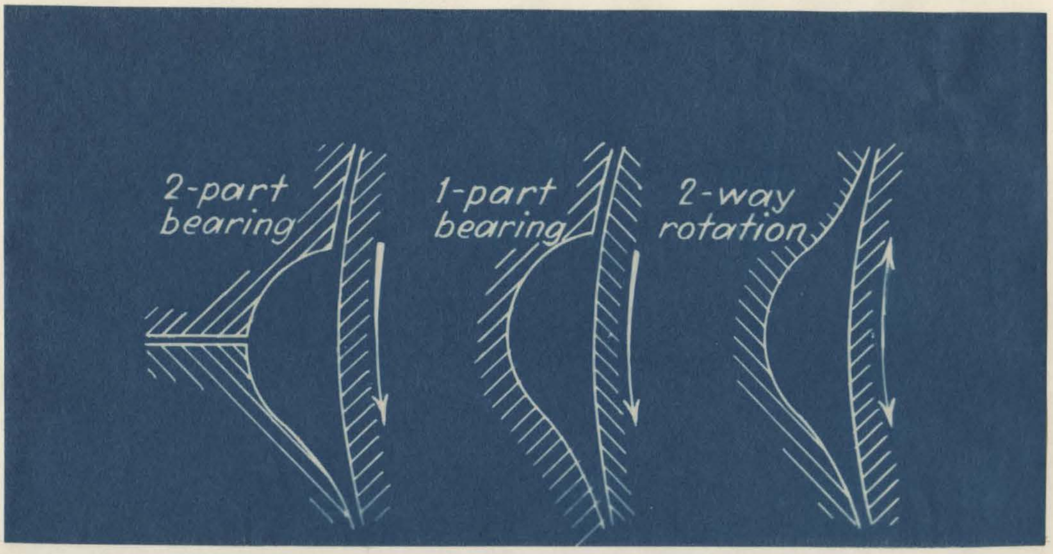


Fig. 29 - How to Cut Oil Grooves.

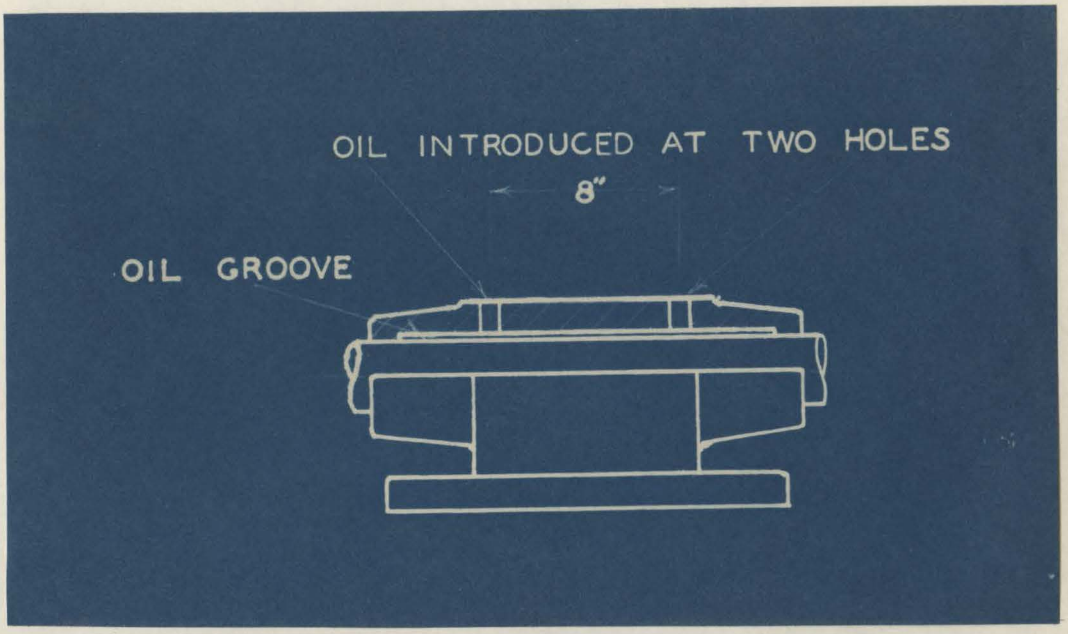


Fig. 30 - Longitudinal Groove Cut in a Long Bearing to Connect Two Supply Sources thus Aiding in the Distribution of the Lubricant.

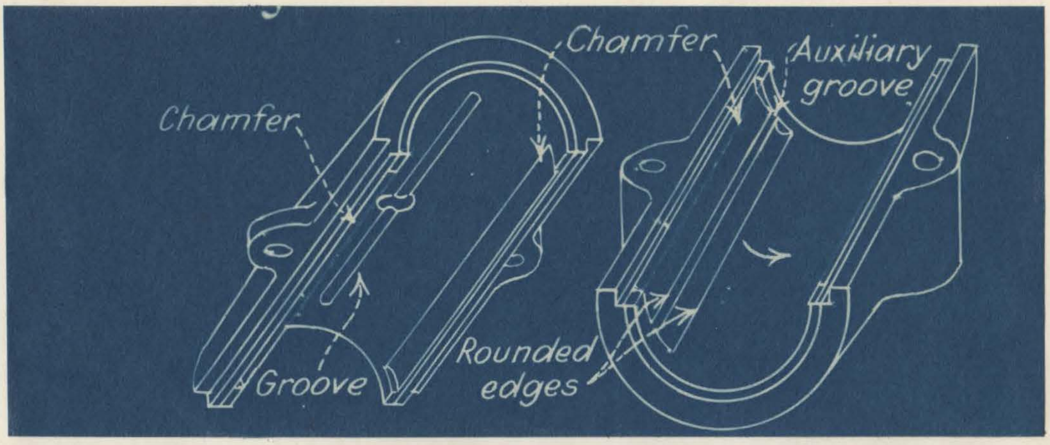


Fig. 31 - Grooving in a Heavily Loaded Bearing, Showing the Auxiliary Groove Just Ahead of the High Pressure Region.

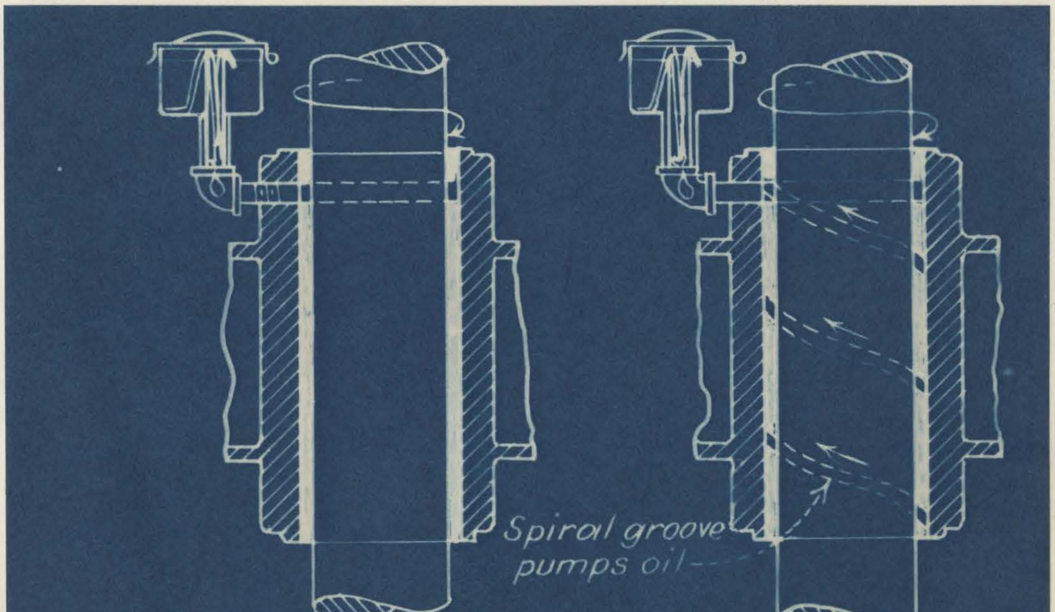


Fig. 32 - Vertical Bearing Grooving.

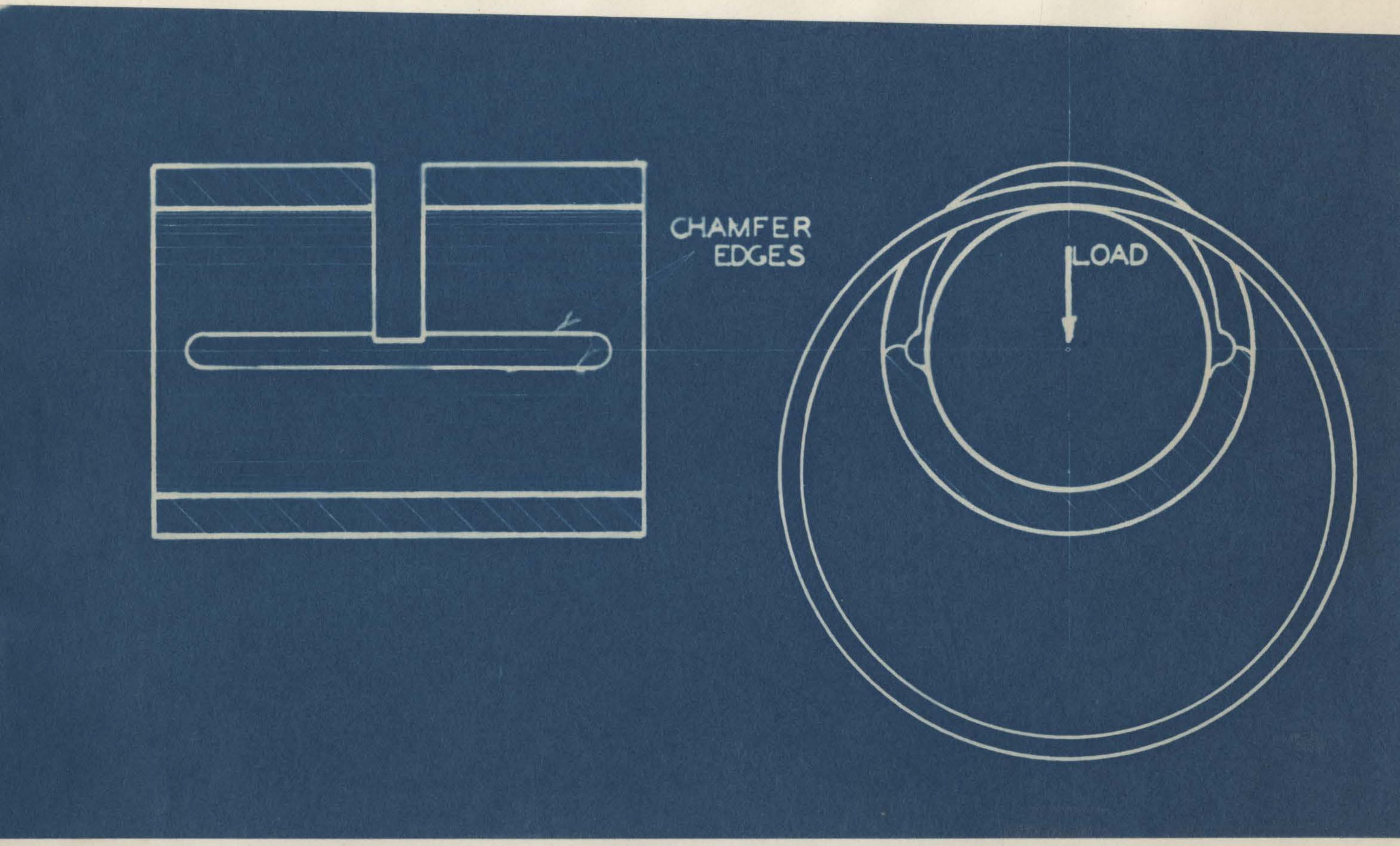


Fig. 33 - Correct Grooving of an Oil Ring Journal Bearing.

Bearing Metals

The general belief that a good bearing metal should have hard crystals to support the load and soft crystals to conform to the shaft may now be disputed, because of the good bearing properties displayed by silver. (53)

A good bearing metal must have sufficient plasticity to conform easily to the shaft, and to be readily run-in. It must be of such nature that small wearing particles will easily imbed themselves and not grind away the metal. It should also have a low coefficient of friction so it will not wear excessively during the starting period, when boundary lubricating conditions prevail. So that the bearing may support the shaft without deforming or failing, the bearing metal should have relatively high compressive strength, resistance to pounding, and fatigue strength. The bearing metal must resist any corrosive action present, and it should have properties such that it may be easily bonded tightly to the bearing shell.

As no bearing metal today has excellence in all of these properties, a bearing metal should be selected after the designer has considered the conditions under which it is to operate. In many cases, by proper analysis, a cheap yet satisfactory material may be used.

In general, tin in a bearing alloy increases its toughness and ductility, and copper hardens it. Copper and tin together make a good combination. To cheapen Babbitt, lead is sometimes employed instead of tin. The addition of antimony to a bearing alloy hardens it, but makes it more brittle.

A summary of the principal features of some commonly used bearing metals is given in Table VIII.

Bronze is used a great deal in heavily loaded, low speed applications, because it withstands heavy pressures. But its conformability is low; so its use is desirable only in those applications where lubrication and alignment are good.

Plastic bronze, which contains about twenty-five per cent of lead, is suitable for higher speeds and lighter loads. Other bearing materials are rubber and synthetic resins, excellent for water lubricated cases.

Table VIII. Principal Features of Commonly Used Bearing Metals.

Tin-base Babbitt:
- High anti-score properties
- Conformability excellent
- Good embeddibility properties
- Corrosion resistance
- Good bonding properties
- Fatigue and mechanical strength low

Lead-base Babbitt:
- High anti-score properties
- Softer than tin-base Babbitt
- Conformability better than tin-base Babbitt
- Embeddibility excellent
- Corrosive resistent
- Fatigue and mechanical strength low
- Cheaper than tin-base Babbitt

Hardened lead (Lead-base Babbitt hardened with 1-2% tin and small percentages of other metals):
- Good wear properties
- Not corrosive resistance
- Mechanical properties slightly better than standard Babbitts

---

Cadmium base alloys:
- Plasticity good
- Harder than above alloys, particularly at high temperatures
- Longer fatigue life and compressive strength than above alloys
- Very susceptible to corrosion at temperatures of greater than 250° F.
- Addition of iridium prevents corrosion, but raises cost far above that of tin-base Babbitt

---

Copper-lead (Lead 30-45% and remainder copper):
- Anti-score properties not so good
- Conformability not so good
- Embeddibility poor
- Very high fatigue and compressive strength
- Susceptible to corrosion
- Corrosion arrested by iridium treatment until the iridium wears off
- Good for low and medium speed internal combustion engines

Satco ($CaPb_3$ dispersed throughout a ground mass of lead containing tin and calcium in solid solution:
- Conformability satisfactory
- Good frictional qualities
- Very high fatigue and compressive strength even at high temperatures
- Good bonding qualities
- Good corrosion resistance

Specification of Lubricants

The ideal basis for the design of a journal bearing would be that of the lubricant. But this is an impossibility, for the same design encountering different conditions of operation will require different lubricants. Therefore, lubricants should oftentimes be specified after the installation of the unit. In many cases, however, the lubricant should be specified by the manufacturer.

The late W. F. Parish, in speaking of this lubricating problem said: "This lubrication problem cannot be solved by the 'average' method; that has been tried and has not been successful, as can be seen when an oil company will state in its charts that only one of its oils shall lubricate all of the Ford cars, and yet it will have to sell at least four different grades to satisfy the owners, who have found something wanted in the oil recommended, which was supposed to satisfy the 'average' condition!" (54)

The need for individual attention in the selection of lubricating oils is being met by the lubricating engineer. He can draw on vast and varied experience in recommending lubricants, conducting power tests when necessary. His aim is to provide safe lubrication with a minimum of power loss, with a maximum of production output, and with highest economy. By making an individual survey of a plant, he can reduce to a minimum number the different lubricants required, thus saving on inventories and storage.

The lubrication engineer also lends his valuable practical training to the designer.

Automobile Engines

The trend in automobile engine design is toward higher specific output. This is accompanied with an increase in speeds and pressures. Vibration and deflection at the higher speeds necessitate stiffening the crankshaft by shortening the bearings, and increasing the journal diameters. At the same time there is a demand for more reliable and durable engines.

The designer of the automobile bearing has not relied on the hydrodynamic theory as has the designer of the steadily loaded bearings of the turbine or electric motor. The effects of fluctuating loads are not generally understood. Past experiences indicate that there is a definite increase in load capacity due to fluctuating loads at low rubbing speeds, but at high speeds the effects are unknown. More information is needed on the relation between the heat dissipation of the bearings and their temperature rise. Flexibility limits the application of any theoretical analysis to the problem.

The necessity of the application of design technic and the reliance upon past experience in the design of automobile bearings is apparent. The details of present design practices were not obtained, but evidences are that the PV factor is used, P being derived from the inertia forces.

Professor C. F. Taylor has called attention to a "factor that may contribute to the troubles of heavily loaded bearings; namely, flexure of the parts. Considering a big-end at firing top centre, it is clear that the loaded face of the crankpin will tend to become concave under load, and the big-end convex. If the two have the same curvature, the load will be evenly distributed, otherwise it will be concentrated either in the center or at the ends. This may account for the erratic nature of white metal big-end troubles, some designs being free from them, whereas others with apparently similar bearing conditions fail. It may be that the successful designs are those in which, by good design or good luck, the two curvatures are equal." (55) The main bearing caps in the automotive engine are very stiff, and the connecting rod bearings follow the same practice as far as is possible. The rod has to be as light as possible to give minimum inertia forces and best operation. At the same time it is beneficial to make it and the bearing rigid. The bearing is made as rigid as possible by leaving the bosses at the connecting-rod bolt intact and sometimes by providing the shell with circumferential ribs or flanges. The journals are stiffened by increasing their diameter and by adding counterweights to the crank shaft.

Dicksee has emphasized that the ordinary forced circulation system is disadvantageous, as it requires oil grooves that reduce the available bearing surface, and as it circulates an unnecessary amount of dirt. (34) At present, however, 40 of a total of 44 automobile engine types have pressure lubrication to the main and connecting rod bearings. (56) Attempts are being made to limit the amount of sludge formation by keeping the temperatures lower. The recent application of oil filters to many engines also helps to reduce the circulated dirt and sludge.

A momentary stoppage of the oil to any one of the high-speed bearings in the engine would likely result in its failure; therefore, provisions must be made for an unfailing supply of lubricant to all the bearings. The usual method of circulating the oil is by a gear pump. Should the pump lose its prime, the flow of oil would cease; so the pump is generally immersed in the oil. Drilled oil passages in the block and adequate protection of the oil lines reduce danger of stoppage of the oil flow should an oil line break. Proper design of oil lines avoids crucial places where the line may have a fatigue failure.

Not only are excessive temperatures an aid to sludge formation, but they also may be the cause of bearing failure. A high temperature may cause breakdown of the film due to the lowering of the viscosity of the oil. The viscosity of the oil may become so low that excessive leakage in one bearing will rob another of its supply. As the viscosity of the oil decreases, the pressure of flow decreases in the majority of the present engines. This pressure must work against the centrifugal force of the oil in the supply line from the main journal to the connecting rod journal. This centrifugal force is not affected appreciably by the change in temperature, so less pressure is available to do the same work whereas there should be more.

This condition is likely to result in the inadequate lubrication of the big-end connecting-rod bearing.

The oil temperature in the crankcase is lowered by correctly designing the crankcase and by applying crankcase ventilation, a practice in common usage today. The inlet for the ventilating air is the oil inlet, which is equipped with an air filter. An outlet is provided at a point suitable for the proper circulation of air through the crankcase; the circulation being caused by the whirling of the crankshaft. The crankcase should provide sufficient cooling area which is exposed to the air stream when the automobile is in operation. That the control of crankcase temperatures is the crux of the problem in automobile engines is borne out by the following statement: "According to some experimental evidence at hand, the minimum value of $\frac{ZN}{P}$ for a well-run-in bearing is 30, and an increase in the speed of the crankcase reduced the value of $\frac{ZN}{P}$ from 510 to 60, and the factor of safety therefore from 17 to 2. By decreasing the temperature of the air in the crankcase from 149 to 104 degrees Fahrenheit the factor of safety at 3000 r.p.m. was increased from 2 to 8." (57) It is interesting to note that the use of the overdrive in the Studebaker reduces its crankcase temperature about fifty degrees Fahrenheit.

The crankcase ventilation system performs a valuable function, in addition to cooling, by sweeping out the water vapors before they can unite with the sulphur compounds in the crankcase oil. The sulphur compounds present in the crankcase oil are derived principally from the products of combustion blowing by the rings; and the uniting of these sulphur compounds with water results in sulphurous acid, which is very corrosive.

The oils used today in the automobile normally are those recommended by the manufacturers. That their recommendations are beneficial is indicated by the large number of engines that are operating satisfactory today using the recommended oils. S. W. Sparrow shows that the viscosity of the oil used in winter in the automobile should be as high as that used in the summer. (58) Recommendations of manufacturers, however, specify lower viscosity oils for the winter than for the summer.

There is much discussion on the methods of introducing the oil to the journals in the automobile engine because of the variances of the loads and the lubrication required. In all cases the theory is observed, and the oil is introduced on the unloaded side of the bearing as nearly as possible. The crank angles at which the oil should be discharged, the angles of incidence of the outlet and inlet holes, the number of oil holes required, and the grooving of the bearings are all controversial problems and complicate the design of the lubricating system.

Generally, the oil is fed through a distributing manifold or through the block to the main bearings and from thence to the connecting-rod bearing through holes drilled in the web of the crankshaft. When oil is fed through the mains to the more heavily loaded crankpins, the oil must flow readily. This is accomplished by having a circular groove in the main bearing, generally giving continuous connection between the crankpin and the main supply line. In some cases, this inlet to the crankpin is only connected for the greater part of the time. The continuous groove assures better lubrication than the non-continuous groove and does not cause oil knock, but it wears a ridge in the journal. The non-continuous groove causes oil knock, does not lubricate as well, but wears more evenly. Usually, when lubrication of the piston pin and cylinder walls is accomplished through a drilled hole in the rod, the connecting-rod big-end bearing has circumferential grooving in it. In such cases the groove is no longer than $\frac{1}{2}$ to $\frac{3}{4}$", and provision is always made that the inlet to the connecting rod be supplied indirectly from the main distributing header while this groove is furnishing oil to the piston pin and cylinder walls. The Chevrolet is a very special case, employing pressure stream lubrication and has a circumferential groove extending entirely around the big-end bearing.

The majority of main and connecting-rod bearings today are the thin-walled replaceable type. These bearings are provided with a small amount of relief at the parting to prevent binding with the journal when the bolts are drawn tight. Sometimes the relief is large to provide better distribution of the oil and to provide a reservoir for impurities. When the relief is large, however, it does not extend the whole distance of the bearing so oil will not leak out before it has reached all parts of the system.

The bearing liners of these thin-walled bearings are about 0.010 to 0.025 inches in thickness and are well bonded to the shell. This thinness is actually advantageous as it raises the mechanical strength of the unit, especially at the high temperatures encountered. The bearings are machined to close tolerances and are provided with excellent finishes on their surfaces. The overall thickness is often less than 1/16 of an inch. Such construction costs little, and permits a light assembly.

Most of these bearings are of the steel back, Babbitt lined construction. Some are Babbitt, with no backing, and some are Babbitt linings in the bronze back. Others are cadmium silver lining in the steel back, copper lead lining in the steel back, white bearing metal alloy lining in the steel back, and copper lead alloy.

Application of the Thin Wall Automotive Bearing to Industry

Because the thin wall type bearing is so economical to manufacture, so easy to install and replace, and has such a high load carrying capacity, it is used in many industrial applications.

Diesel Engines

The trend of the development in the present high-speed Diesel engine is so similar to that of the automobile engine that the problems and solutions of one parallels the problems and solutions of the other. The tendency is to raise the specific output, with a corresponding increase of speeds and shortening of engine and bearings. The main difference in the two is that the high-speed Diesel encounters slightly more difficulty in its bearings and its lubrication, for operating temperatures are higher. Its parts are heavier and combustion pressures are higher, thus sometimes giving heavier loadings. However, the speeds encountered in the Diesel are not nearly as high as those of the automobile engine, although they have risen considerably in the past years.

The Diesel engine manufacturers utilize the latest improvements in bearing construction; namely, the thin-shelled replaceable bearing. The temperatures are rising correspondingly with the specific output. For this reason, the oil pump capacities existent in the engines today are higher than those of yesterday; and the oils in use today are lighter than those of yesterday, so that the effectiveness of cooling by oil may be increased. Also, the clearances in the Diesel have been reduced to give smooth and quieter operation, thus requiring a lighter oil.

The low-speed Diesel has the same tendency as the other engines, but the promotion of higher specific outputs has not been so urgent as in the high-speed units.

The bearings in the high-speed Diesel are designed on the basis of the inertia forces, whereas the bearings in the low-speed engines are designed on the basis of maximum unbalanced forces acting on the piston. The only authoritative statement found on this point is: "It has been the custom to judge bearing sizes on the basis of maximum pressure, and the product of the mean pressure and the velocity." (59) The maximum pressure refers to the maximum combustion pressure.

The temperature problem is the most acute problem the Diesel manufacturer faces. As a consequence, it is common practice to provide oil coolers for the Diesel engine. In large Diesel engines, oil is curculated through the various parts to reduce the temperatures in the engine. These temperatures, along with the pressures encountered, require that the bearings be constructed of the best material. At the high temperature reached, Babbitts rapidly lose their strength and are incapable of withstanding the loads imposed upon them. So the higher strength materials such as copper lead, cadmium silver, and nickel cadmium, are now employed in many cases. But these bearing materials are susceptible to corrosion by some organic acids already present or developed during service in certain oils.

Diesel engines are also susceptible to ring sticking, scratching, scuffing, and wearing of the cylinders. Therefore, the Diesel engine oil must have high film strength, a high degree of oiliness, low carbon forming tendency, stability against oxidation and engine temperature, and detergency properties. So, some Diesel oils contain certain additive agents to give the oil these needed properties, and experience shows some of these additive agents to be very beneficial. The additive agents usually used are: aluminum naphthenate, calcium naphthenate, calcium oleate, calcium phenyl stearate, and calcium dichlorostearate. Each of these additive agents is corrosive to cadmium-silver, copper lead, and nickel cadmium alloys at the temperatures encountered in the Diesel engine. Therefore, the Diesel manufacturer has had to use a lubricant that is not entirely satisfactory when the better bearing materials were used.

However, recently there was announced a new R.P.M. Delo oil produced by the Standard Oil Company of California in ranges of viscosity to meet all Diesel needs. This oil is a new development and is not in general use now but promises to aid the development of the Diesel engine a great deal. Claims of the proficiency of this oil are: "Prevents ring-sticking under the most severe engine operating conditions. Superior in the maintenance of piston cleanliness and the prevention of carbon and sludge deposits in oil ring slots. Non-corrosive to alloy bearings and is, therefore, suitable for use in all engines, regardless of bearing material, such as copper-lead, cadmium-silver, Satco and similar types." (60) The November, 1939, issue of the S.A.E. Journal gives a complete account of the development of this oil and of its testing upon which testing the above mentioned claims are based. (61)

Today, the materials generally used in the Diesel engines for main and connecting-rod bearings are: Babbitt, copper-lead, cadmium-silver, nickel-cadmium, and Satco. These are generally lined in a steel or bronze back. Satco was first tested as connecting-rod bearing material in 1933 on gas electric cars of a western railroad. (62) Since then it has demonstrated its usefulness by proving more satisfactory than other materials under conditions of the severest of bearing applications, that of railroad Diesel engines. It is used in the Diesel engines of the streamlined engines of the Union Pacific, Burlington, Boston & Main, and New Haven railroads. It is also used in many industrial Diesels.

The common practice is to provide the main and connecting-rod bearings of all Diesel engines with oil under pressure employing the circulation system. Some of the smaller engines use the mechanical-force-feed system or the splash system. Ring oiling for the main bearings is used to a limited extent. Generally, in the larger and slower speed engines two oils are used in lubricating the engine, a heavy oil supplied by force-feed lubricators to the cylinders, and a light oil supplied to the bearings by a circulation system. Frequently, the dry sump is employed, which permits a better control of oil temperatures.

Diesel units are usually equipped with filters. The absorbent type, which is usually made of Fuller's earth, charcoal Bauxite, or other absorbent materials, is capable of removing the additive agents in the commonly used oils that are so beneficial to Diesel life, and so the use of such filters is in this respect detrimental. The absorbent filter, which is usually made of cotton filler, yarn, or other materials of this nature, are beneficial in removing most of the insoluble impurities in the oil. (63)

Large type units are equipped with elaborate oil purification systems. Diesel consumers now use two hundred times the amount of lubricating oil that the steam plant field uses. Thus, the use of the purification systems in Diesel power plants is necessary to cut down lubrication costs.

The three systems used in the purification of Diesel engine oils are:

(a) Continuous by-pass system

(b) Continuous total purification system

(c) Batch purification

The continuous by-pass system is the most commonly used system today. This system is usually equipped with heaters to heat the oil to be purified, and the purification is usually accomplished with the aid of a centrifuge. The older plants using the batch purification system employ the centrifuge, whereas the new ones generally use chemical coagulants. Activated clay is used in conjunction with both the continuous by-pass and the batch system in some quarters. The continuous total purification system is limited in its use to Diesels having slow rates of oil circulation.

Airplane Engines

There are two types of airplane engines today, the in-line type and the radial type. The in-line type of airplane engine is very similar to the automobile engine. The airplane engine has to be more reliable, and this factor affects the design to some extent. The lubrication is similar to automotive practices except that the dry sump system is employed, more effective cooling of the oil being obtained with the use of an outside storage tank and oil cooler. This system is so constructed that stoppage of one passage in the crankshaft does not cause failure. In addition to this, the oil used at the crankpins is centrifuged by the crankshaft before it enters the bearing. This centrifuging is accomplished by taking the oil from the center of the hollow pin by means of a short oil line, as in Fig. 34. The oil is fed to the bearings under higher pressures than in the automobile engine, and the clearances are greater. Little grooving is employed that better utilization of the bearing areas be made. All of these practices tend to make the airplane motor more reliable.

The radial type of engine is lubricated on the same principle as the in-line engine, that is, the system must be reliable. Therefore, the dry sump type is commonly used, the oil going to an outside reservoir which enables better control of the oil temperature and flow back to the engine. Oil coolers are commonly used with temperature controls attached. The main bearings are

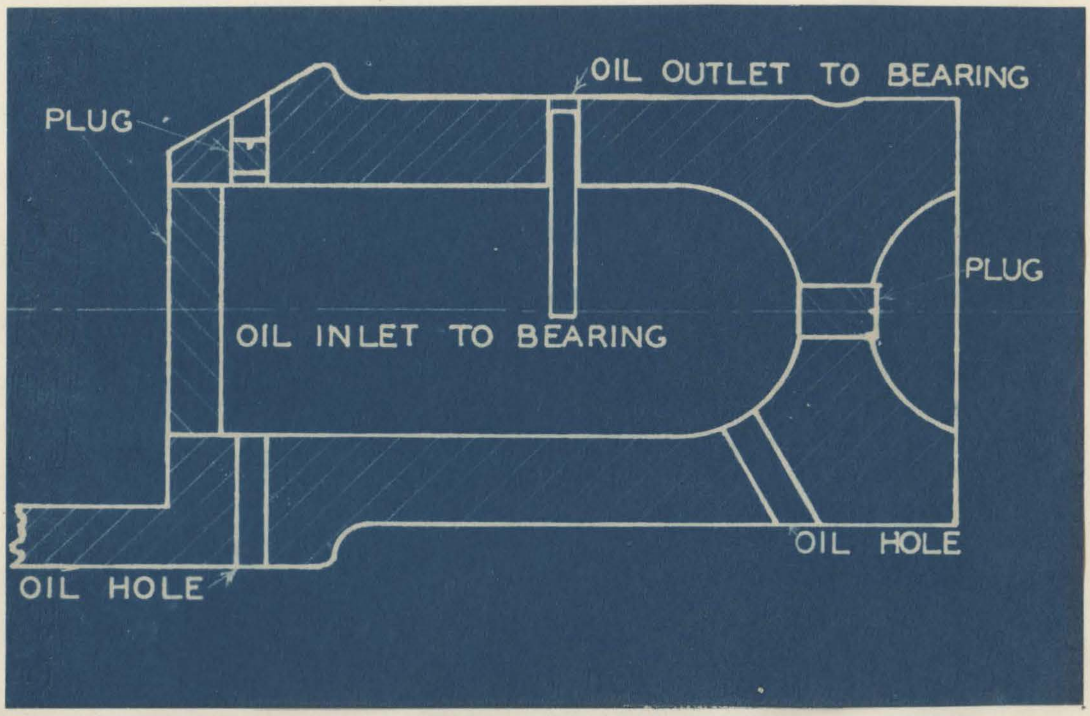


Fig. 34 - Cross Section of a Crankpin of a Radial Engine, Showing the Method of Centrifuging the Oil. (The same principle is employed on the in-line engines.)

usually ball or roller, but the master rod, and articulate rod bearings are of the cylindrical type. The oil is fed by way of the shaft to the crankpin through a centrifuging device as described previously. All lubricant for the crankpin and articulate rod bearings passes through the crankpin, the oil being conducted to the articulate rod bearings by means of passages and drilled holes. Small supplementary oil grooves are sometimes placed in the master rod to act as reservoirs, so that more oil may be available at points of heavy loading.

Outboard Gasoline Motors for Motorboats

Four large manufacturers of outboard gasoline motors for motorboats were written since no information concerning these motors could be obtained from the library. One reply was obtained, which is probably typical of all outboard motor boat engines. The author of this reply believed that a tabulation of specific data on such motors would be of little use because of the method of lubrication.

All the outboard motors made by this company are lubricated by mixing lubricating oil with the gasoline, and the temperatures at which the engines operate depend to a great extent upon the temperature of the water in which the motors are operated. Therefore, the most important factor in bearings, the operating viscosity of the oil, is indeterminate, and the outboard motor manufacturers naturally lay less stress on this factor than do those other manufacturers whose engine has an oil sump.

The bearing loads have been found to be affected to a much greater degree by vibration than by the actual loads from the natural working forces.

The design of bearings of outboard motors is probably based largely on past experience, and the good designs will be those produced by designers who understand the conditions under which this engine operates.

Steam Engines

Steam engine builders have devoted the larger part of their development work to better valve design and operation so that efficiencies would be raised. Builders have demanded only that the bearings withstand the conditions imposed upon them, and if these bearings proved satisfactory, they were considered adequate, and other bearings were designed similarly. No radical changes have taken place in bearing design in steam engines for many years. However, the steam engine builders have not overlooked new developments that have taken place in bearing practices and have adopted some, the main one being modern systems of lubrication.

The bearings are designed on the basis of experience gained in the construction of steam engines for many years. Generally the basis for comparing the new design with the old successful designs is maximum force occurring on the bearing due to the unbalanced steam pressure alone. This is an approximate method of calculating the force as inertia forces of the reciprocating parts will reduce this force from thirty to fifty per cent at either dead center. The average force on the bearing may be only half as much as this approximate value, but by proper comparison with older designs, this method works. Then, too, the calculation of the actual forces occurring on a bearing in the steam engine involves tedious work. The indicator card of the engine must be on hand before accurate calculations can be made, and these vary widely, depending upon the load and speed. A few companies make corrections for the inertia forces. In many cases the limiting factors of the bearing are determined by the design of the cylinder, piston and related parts. The design of these is the primary consideration of the design of the engine as has been indicated before; the design of the bearing has usually been a secondary consideration. In large engines, rotational speed of the crankshaft is determined by the limiting piston speed. The crosshead pin bearing is made so its length-diameter ratio is greater than one in practically all cases so side slap of the connecting rod may be reduced as much as possible.

The bearings are constructed so that they are easy to reach and easy to adjust. Several different methods are used to promote easy adjustment of these, the chief of which are: easily replaceable bearings, adjustment by means of set screws tightening up directly on the bearing or indirectly by means of wedges, adjustments by removing laminations from shims, and take up of wear on the crosshead pin by rotating it through ninety degrees and relocking it by a simple device.

Many bearings used today employ the new principles of grooving thus allowing the lubricant to reach all parts of the bearing surface and yet not interrupt the formation of any oil film. There are many designs, however, employing numerous grooves in the high pressure area. Lubricant is distributed throughout the bearing surface, but adequate provision is made for taking up wear, which indicates that many of these designs do not operate with perfect lubrication and are therefore susceptible to wear.

The more advanced designs today employ a simple but very adequate supply of lubricant to all bearings. Some models have a combination splash and forced feed lubricating system. The majority of engines built today have the following features: gravity circulation of the oil to all bearings, adjustable sight feed that the proper amount may be fed at all times, drainage of the oil to an oil sump, a method for the separation of water and dirt from the oil, a large capacity pump to force the purified oil back up to the overhead reservoir, and an overflow to the reservoir, the excess oil being recirculated through the pump. Provision is often made for the flooding of the bearings with oil prior to starting. Often, supplementary oiling to the outboard and main bearings is employed by using ring or chain oiling. The vertical type engine, which is very similar to the internal combustion engine, uses a circulation system very similar to that of the automobile engine, the oil being forced to all bearings under high pressure through drilled holes.

Often, very ingenious devices are utilized to keep the water from the oil. The system is frequently equipped with an oil cooler. Devices to indicate that the oiling system is operating properly are very common. The gravity system is generally provided with sight feed glasses and sight glasses on oil reservoirs. The pressure system is generally provided with pressure gages and automatic signaling devices that notify the operator of failure in oil pressure.

The crankpin bearings are generally of bronze, Babbitt, or bronze backed Babbitt lined construction. The main bearings are generally genuine Babbitt; some, however, are of the roller type. For smaller engines, where velocities at the bearing surface of the eccentric are low, cast iron straps on cast iron eccentrics are found. For larger engines, periferal velocities are usually so high that a Babbitt lines eccentric strap is necessary. The crosshead pin is constructed of steel and in many cases hardened. This pin is usually in a Babbitt, a bronze, or a Babbitt lined bronze backed bearing. Generally, lightly loaded crosshead shoes have only cast-iron bearing surfaces; the more heavily loaded shoes are lined with Babbitt; and the most severely loaded shoes are lined with bronze. Thus, for some crosshead shoes on horizontal engines, the lower bearing surfaces are lined with bronze, and the upper bearing surfaces are lined with Babbitt. In practically all engines adequate provision is made for taking up wear on these shoes.

Turbines, Generators, and Motors

Since the bearings of the turbine, the generator, and the motor are similar in all respects and have the same conditions imposed upon them, they will be considered together.

Bearing design and lubrication of these bearings presents an entirely different picture from that of the internal combustion engine. The loads encountered are of the steady type. The length diameter ratio generally ranges around one and one quarter or a little more. The length of the bearing in this case is long enough so that end leakage is easily taken care of by end leakage factors applicable to theoretical analyses, and yet the length is not great enough to influence the results due to concentration of stresses within the bearing caused by excessive deflections of the shaft of a different order than that of the bearing. In shafts exceeding three and one half inches in diameter, spherical seating is commonly employed insuring good alignment of the shaft and bearing. These bearings are easily tested, and the bearing can be well designed.

Power losses in the larger bearings are considerable due to the high rates of shear encountered. Thus, everything possible is done to bring these losses to a minimum and still have a safe bearing. The diameter of the shafts are made as small as is feasible so that the friction torque be low and yet the design be safe. The bearing contact arcs are approximately one hundred and twenty degrees, which is good practice. A substantial relief is usually given the upper half as this half supports no load, yet is responsible for frictional losses of a large magnitude. A large relief in the upper part of the bearing reduces losses in this area considerably.

Small units that are built on a production basis usually have sleeve or antifriction bearings. The sleeve bearings are easily and economically made and installed. They are well designed and entirely satisfactory. These units are generally ring oiled, which method insures adequate lubrication at all times with a minimum of attention. The lubrication of some small units depends upon the human element, while lubrication of others requires no attention, although the latter do not operate in the region of perfect lubrication.

Continuity of the oil film in the large units is imperative. The high rotary speeds at which these units operate along with the weight of the rotating mass would in all probability result in instant wreakage should the oil film fail. This is the reason for the careful designing of such bearings. A copious supply of oil must be provided, not only to maintain the film of oil, but also to aid in cooling the bearing. The use of circulating oil to cool these bearings is now common practice. Therefore, the imperative need of an unfailing lubricating system is apparent. The circulating system is usually provided with two pumps, a main pump and an auxiliary pump. One operates with the machine, and the other operates when the machine is started or when the oil pressure provided by the other pump decreases to some extent. Often loud signaling devices will summon the operator in case of failure of the pump to perform satisfactorily. Constant check is kept on various factors concerning bearing operation that continuity of operation be assured. The oiling system is generally equipped with an oil cooler and frequently with a purification system.

In England a great deal of information has been compiled on the practices in that country prevailing in such bearings by Mr. H. L. Guy, F. R. S., and Dr. D. M. Smith. Similar practice prevails in this country as has been shown by Mr. Soderberg. (34)

Rolling Mill

The demands in a steel mill are: higher speeds, continuous operation, and a better product. The mechanization of the steel industry has required that no one machine fail and thus hold up production. And the demand today is to incorporate machine type accuracy to the rolling mill; so, tolerances are being brought closer and closer. These requirements are being met by utilizing every bit of knowledge in the solution of these problems.

The frictional power losses in the roll-neck bearings of many of the mills range from thirty to fifty per cent of the total power imput. In view of this fact, the importance of attacking this roll-neck bearing problem can be realized. Two factors must be kept in mind concerning the solution of this problem: The first demands that no oil leak from the bearing and contact the metal being rolled. The second requires that the roll-neck bearings operate in very dusty and dirty atmospheres, so adequate provision should be made to exclude detrimental effects.

The trend is use the ball and roller bearings for the lighter and high-speed mills. The perfectly lubricated bearing is finding wide application as are the non-metallic bearings, Bakalite and lignum vitae.

Definite advantages are to be realized from the use of certain well-designed perfectly-lubricated roll-neck bearing units. The effects of pressure on film viscosity have been studied to aid in the design of heavily loaded units, because their advantages were foreseen some time ago by Dr. Kingsbury. (64) One design which is manufactured and sold as the Morgoil Roll-Neck Bearing (40) has demonstrated these advantages of the perfectly lubricated bearing. This was the bearing that was shown to be so superior to the old roll-neck bearings in the discussion of roll-neck bearing development. Indications are that forty per cent of the power used can be saved by the introduction of these bearings to the rolling mill, or ninety-five per cent of the power lost in roll-neck bearings can be saved. This bearing is quite capable of withstanding the loads imposed upon it, even being able to withstand such loads which would break off the neck of the roll. The sealing of the bearing unit is very effective, yet the whole bearing unit is easily removed. Very little wear occurs as would be expected.

The Bakalite and the lignum-vitae bearing have shown great savings in power in many successful applications. These bearings are now operating under pressures of 4500-6500 pounds per square inch. Some trouble has been experienced with wearing of the thrust collars, but wear is confined mainly to these and not to the bearing proper. They are not suitable to mills having roll-neck speeds under one hundred feet per minute or to those cooled with salt water. These bearings are usually lubricated with water, although some Bakalite bearings are lubricated with grease.

Railway Journal Bearing

"From a broad point of view, there has been no fundamental change in journal-bearing or box design of the A.A.R. type since 1875, and no change in the method of lubrication since 1850, which is truly a remarkable record." (65) The Association of American Railroads standard bearing is used universally today, and the methods of lubrication vary but little on different roads. This practice exists because it has been necessary to standardize the practices, for cars must be interchanged between various roads. The old practices have become deeply rooted, and the operators are realizing as the limits on some bearings are being reached, that the methods used are outmoded.

The recent trend in railway transportation has resulted in faster schedules and more reliable service. This has resulted in higher journal speeds, higher loadings on the journals, and a demand for a journal bearing that will not delay service due to hotboxes or other types of failures. As a consequence, more reliable and better journal bearing practice has prevailed during the last decade.

The journal bearing in railway service has to start with the full load upon it, which is not conducive to easy film formation. In addition to this, it is difficult to provide practically a continuous supply of oil to the railway bearing. But in spite of these impediments, the perfectly lubricated railway bearing is a reality.

The application of roller bearings to railroad service has proved to be very advantageous, especially in diminishing the torque necessary to start the train. However, the cost of these bearings is high and their reliability, although good, is not as much as could be hoped for.

The standard A. A. R. box consists essentially of a box, in which rests a partial bearing on a wedge. Oil is fed from the bottom of the box up through waste packing to the journal. The box is equipped with dust guards and a lid. Serious trouble has been encountered in some cases because of the unstable feeding of the oil to the journal. Dirt easily enters these boxes and gets in the packing retarding the flow of lubricant to the journal. Water getting in the box weakens the oil film, causing a hot box. Other factors contributing to failures are improperly fitted brasses, tapered journals, improper packing, and waste grab.

To minimize this waste grab, B. F. Hunter (66) recommends that a high viscosity index oil be used, one of over 100, and that this oil be of a paraffin base. Today, the lubricants used are diverse, each road purchasing under different specifications. Hunter advocates the standardization of specifications for the railway lubricants, and presents evidence that an oil can be specified which will perform better than those generally used today and at the same time be suitable for year around use instead of the present custom of alternately changing the kind of oil used in summer and in winter.

Many driving journals today are lubricated with hard grease that melts at about 325° F. This grease contains from forty to fifty per cent soap, which has no lubricating value. Friction must generate enough heat to melt the grease. Locomotives are operating on this, however, and often this hard grease cake lubrication is supplemented with forced-feed lubrication. The specifications for this grease are unlike for different roads although the greases are very similar. Hunter advocates the standardization of this grease as its properties can be improved slightly and at the same time its cost be considerably reduced.

The perfectly lubricated journal bearing has shown decided advantages over the older types. The Railway Service and Supply Corporation set up a laboratory to develop a new railway journal unit. They advanced a journal bearing unit, using the present structure with certain modifications. This new type bearing is a vacuum type so called because vacuum in certain grooves in the bearing retain oil to assure adequate lubrication. Positive dust sealing is provided. New methods, recommended by developers, to be employed in constructing this bearing are: forging of the brass back instead of casting, as was often done in the past; contrifugal lining of the bearing instead of mandrel lining; and use of the improved bearing linings. This bearing, as tested in the laboratory, showed a very low coefficient of friction, being less than that of roller bearings for speeds over six miles per hour. Advantages of this bearing, as summed up by its makers, are:

a. There is sufficient oil stored in the bearing to operate for a protracted period without dependence on the packing.

b. Lubricant is adequately distributed over the journal surface.

c. Oil serves both as a lubricant and cooling medium for the bearing.

d. The circulation of oil through the bearing promotes feeding from the waste.

e. Foreign matter is prevented from getting under the bearing and from shutting off the oil supply at the rising side of the journal.

f. End leakage because of oil-wedge pressure is practically eliminated, for instead of pressure, there is a vacuum. (67)

The suitability of this bearing for railroad service has been demonstrated by its performance in many installations.

Another development in locomotive bearings has been made by the Motive-Power department of the Southern Pacific Railroad. A serrated recess is cut in the brass crown bearings, which is filled with Babbitt. A more important development is the manner of the lubricating these bearings. Lubricant is supplied by spring supported pads which are fed oil from the oil cellar through many wicks. The old grease boxes were modified slightly to accommodate the oil. Auxiliary oiling of these bearings is effected by feeding oil continuously from the mechanical lubricator of the engine. A large reduction in maintenance costs and a reduction of frictional losses is indicated by the fact that these new bearings operate at temperatures more than 100^{o} F. lower than the older bearings. (68)

The Research Laboratories of the National Bearing Metals Corporation has developed the "Disc-Flo" Journal Bearing. (69) The bearing unit looks very much like the A. A. R. unit.

The important functional design features are:

a. To lubricate the journal bearing with free oil in a circulating system, using a rotating disk so shaped as to keep the circulating oil in flowing channels without aeration.

b. To provide self-alignment for the journal bearing to compensate for all possible misalignment of the journal box in reference to the journal.

c. To employ a flat-back bearing for the purpose of eliminating bearing tilt due to shock impacts.

d. To extend the sides of the journal bearing to approximately 180 degrees of journal coverage to prevent excessive side movement of the bearing in the box, caused by accelerating and decelerating forces of the axle against the box, resulting from power being taken from or applied to axles, as well as by forces encountered in starting and stopping.

e. To provide for all lateral thrust of the axle to be applied against the fillet and of the journal bearing by the journal fillet, except in some locomotive applications, where a separate thrust bearing should be provided.

f. To vent the box to equalize pressures inside and out, and prevent moisture accumulation by condensation.

g. To employ renewable hardened box liners for pedestal-jaw fits to minimize wear at these points, and provide for control of that wear.

h. To incorporate in the journal box a combination fitting for oil gaging and filling, arranged to permit quick checking of the oil level without the use of tools or gages, and the addition of make-up oil with unrestricted flow to minimize inspection time. (65)

The advantages of this unit, signified by laboratory tests, have been summed up as follows: (65)

a. Improved reliability of operation is secured by stability of the lubricating system under all weather conditions.

b. Speeds can be safely increased due to the low-temperature high-speed characteristic.

c. Power imput is reduced by lower journal friction.

d. The life of the box, bearing, and axle will be prolonged by the self-alignment, the increased thrust capacity, and the non-tilting features of the modified journal bearing.

e. Maintenance costs will be decreased by the simplified lubrication inspection.

At the present time the developments in railway journal bearings are very rapid, and it remains to be seen just how quickly the railroads will take advantage of these new developments.

Heavy Duty Bearings

Heavy duty bearing practice has advanced to a much greater degree than the railroad bearing practice. However, the problems encountered in many applications were not nearly so hard to solve. The old practice was to use hardened grease cakes, as is often the case in the railway practice. The application today of these grease cakes to heavy duty bearings is practically nil. The present practice is to use a circulation system for oiling. This has been found to be satisfactory and efficient. In some cases the lubricant is supplied at high pressures, from one hundred to two hundred atmospheres, to eliminate the possibility of any metal to metal contact. Such high pressure lubrication has been found to lower frictional losses considerably in these bearings. It is customary to provide the bearing surfaces with superior finishes, sometimes a mirrow-like finish being given the bearing. High strength bearing alloys like cadmium silver are employed. Fabric and synthetic resin bonded bearings are used to some extent, being lubricated with water, and they give low power losses and long service.

Line-Shafting Bearings

Line shafting practices are much the same as they were years ago. Generally, these bearings consist of a cast-iron housing with Babbitt linings, and are constructed in two parts so the cap can easily be removed from the shaft. These bearings are usually very long, their length-diameter ratio is about three to one. Lubrication is accomplished by grease or oil fed intermittently; therefore, the bearings operate nearly always in the region of boundary lubrication. The shaft speeds of a majority of these bearings are limited to two hundred and fifty revolutions per minute. Some line shaft bearings are ring oiled and thus receive excellent lubrication, so that the bearings will operate in the region of perfect lubrication with a corresponding decrease in frictional power losses and wear. Other types of lubrication, such as wick and bottle oiling, are also used for these bearings.

Gyroscope Bearings

This discussion will be confined to the bearings of the large gyroscopes used to stabilize ships. The bearing practices employed in the design and the making of a bearing for a gyroscope are representative of the refinements possible in the journal bearing today. Probably no other bearings are designed and constructed with such care. Mr. A. E. Schein says of these bearings: "There are probably no bearings, operating under conditions such as imposed upon the bearings of gyroscope rotors, carrying equally high pressures per unit of projected area." (70) The maximum loading of one of the bearings in the ship, Conte di Savoia, is 200 tons. Surface speeds of these bearings run as high as 4300 feet per minute.

These bearings must operate continuously with minimum frictional losses and minimum maintenance. That the requirements of continuous operation and serviceability are being met is borne out by the fact that bearings for gyroscopes have operated as long as eight years without any adjustments or repairs.

Typical design of these bearings as was discussed by Mr. A. E. Schein in "Overcoming Bearing Deficiencies" (70) will be summarized, indicating the fundamental principles that are employed in such good bearing practice.

The loading on the gyroscope is in one of four directions, consisting of a main load and a secondary load at right angles to the main load. This unusual condition of loading enabled two fundamental principles to be utilized. The first is that described by Bradford: "There are indications that clearances as large as 0.002 inches to 0.003 inches per inch of journal diameter are necessary to produce minimum values of friction when complete film lubrication is secured." (71) The second is that under the same film conditions a larger load can be supported by a greater area. In many applications, large clearances cannot be used without sacrifice of bearing area. This is illustrated by Fig. 35, in which the clearance is 0.002 inches per inch of journal diameter and the effective bearing area only approximately forty degrees of arc. The effective bearing area is about one hundred and ten degrees of arc; yet the clearance is still 0.002 inches per inch of diameter by designing the gyroscope bearing as shown in Fig. 36. The journal diameter is 18 inches, so the total clearance is 0.018 inches. The grooving is designed to promote rapid and perfect film formation. The grooving extends 80 per cent over the bearing length and is closed off at the ends to prevent excessive leakage of the oil. Experiments show that fitted bearings have more load carrying capacity and lower friction coefficient than the other types provided that the bearing arc is very small. Experiments show that for perfect film lubrication, the friction coefficient is lower and the load capacity greater if the bearing radius is greater than the journal radius provided the bearing arc is any appreciable amount. The radius of the bearing in the gyroscope

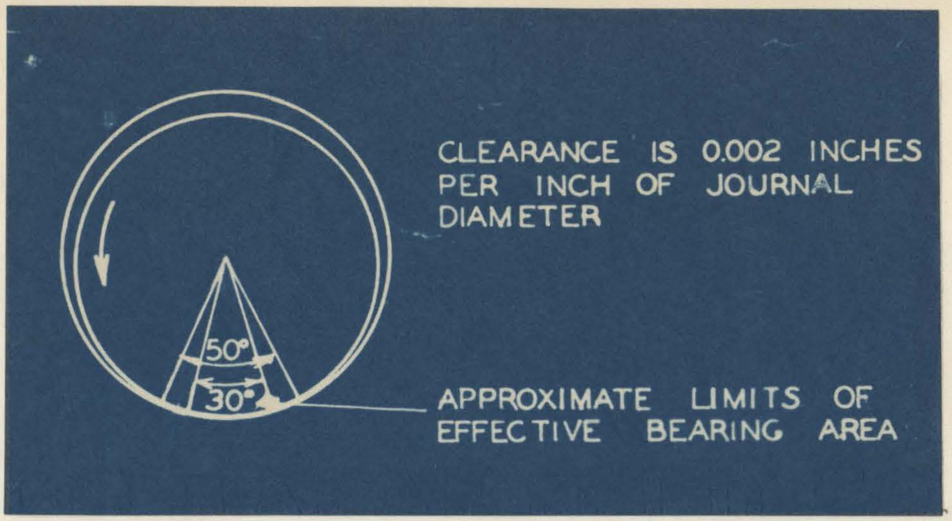


Fig. 35 - Approximate Limits of Effective Bearing Area.

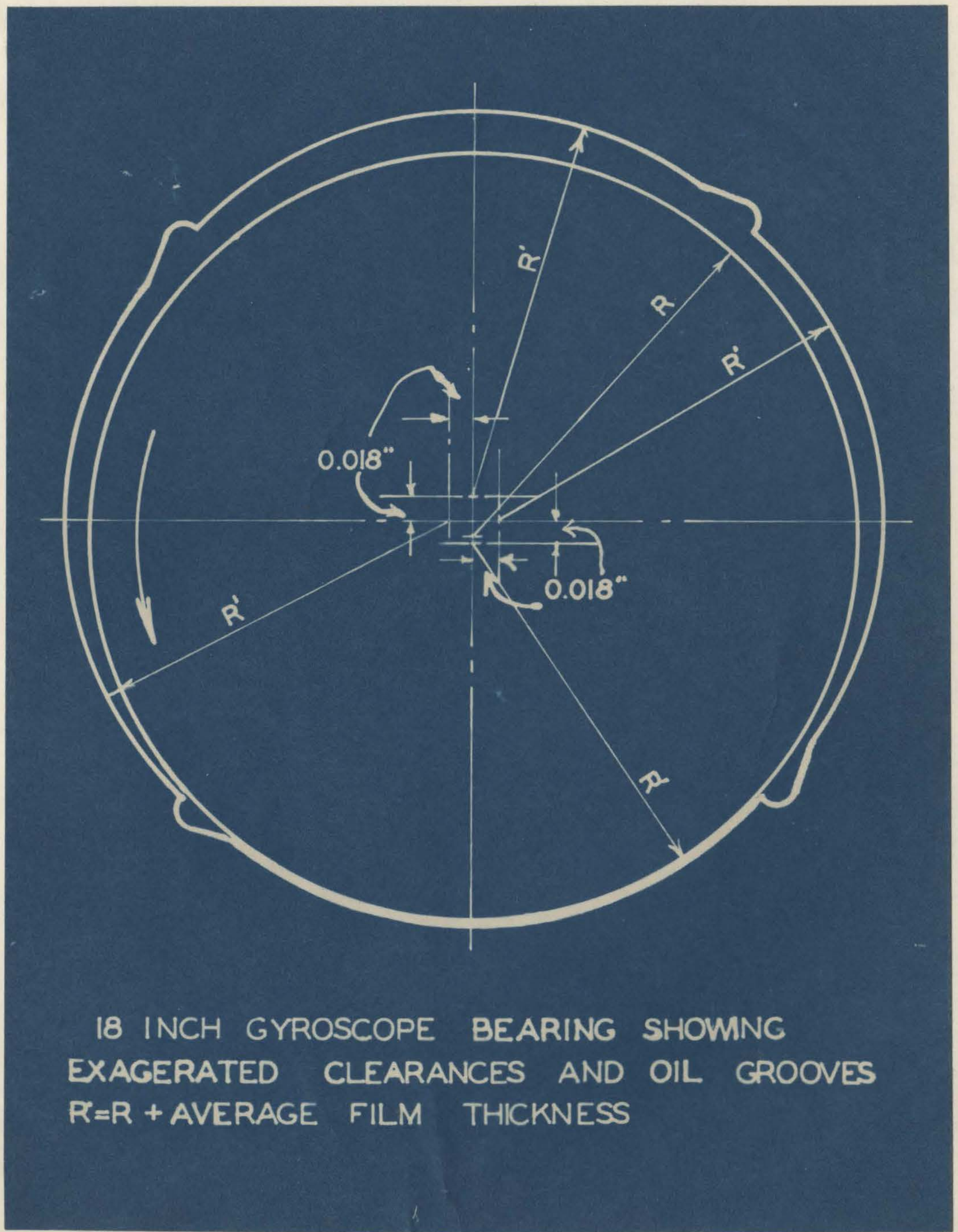


Fig. 36 - Gyroscope Bearing.

bearing is slightly larger than that of the journal, it being greater by the thickness of the film. The radius of the journal and the radius of the bearing is held to a very close limit, the tolerance being 0.0001 inch. The journal is lapped and the bearing scraped to an almost perfect surface. Such consummate fitting enables large load carrying capacity at low frictional coefficients. Running-in experiments show this to be a fact. (30) The bearings are self aligning in order that the load be evenly distributed. In addition to this, the bearings are so designed that their deflection equals the deflection of the shaft. The Babbitt, after being cast in the bearing shell, is subjected to pressure until all stresses are relieved. The pressures to which this Babbitt is subjected are high enough to cause plastic flow; therefore, no trouble is encountered due to the Babbitt breaking away from the shell or cracking. Oil coolers are an integral part of a circulation system that supplies the bearings with ample oil for lubrication and for cooling. A strainer is provided in the system, and grit does not circulate through the bearing.

Standard Bearing Units

The ball and roller bearing industry has a line of bearings that are standardized to sizes. Cylindrical bearings are also available from the Fast Bearing Co., Baltimore, Md., in these same standard sizes for bores up to 5.9 inches. The Fast Bearing Co. manufactures the Fast's Precision Oil Film Bearing. (72) They may be obtained in a regular cylindrical bearing type or in a spherical type that is self aligning and capable of carrying thrust as well as radial loads. The precision with which these bearings are made enables them to withstand very high loading, for the largest of these bearings is capable of supporting a load of 21,000 pounds. Each bearing acts as a self-contained centrifugal oil pump, which acts as an individual lubricating system. Almost unlimited bearing life is predicted. The development of such cylindrical bearings is comparatively new, and the possibilities are very great. These bearings have low frictional characteristics, high load carrying capacities, good performance characteristics, excellent wearing qualities, and they require very little attention.

Rubber Bearings

Rubber bearings offer certain advantages over other types and therefore find useful application in many fields. They wear unusually well, in some cases lasting three or four times as long as the harder surfaced bearings. At very low speeds, the friction torque for rubber bearings is greater than that of metal bearings; but at the higher speeds, the friction torque for rubber bearings is less than that of metal bearings. Less scoring occurs in a rubber bearing, and the rubber fits the shaft well. The lubricating medium used is water, which forms a good film in the well-designed bearing.

The use of water as a lubricating medium makes this bearing particularly applicable to certain foodstuffs machinery. The foods are not then exposed to the possibility of becoming contaminated with oil. The rubber bearing is replacing the lignum-vitae bearings so often used in the past on the propeller shafts of ships. The rubber bearing finds application to many uses as: around vats; chemical agitators; and on hydraulic devices such as pumps, hydraulic turbines, and deep well pumps, dredges, sand and gravel equipment, paper mill equipment, mining machinery, and washing machines.

Brazier and Bowyer (73) suggest the following criteria as requirements for the successful operation of rubber bearings:

a. The size of the bore must be accurate and give sufficient clearance for an easy running fit.

b. Owing to the low heat conductivity of rubber, any frictional heat must be carried away by the lubricating water. A continual flow of cooling water is therefore imperative. The bearing must never run dry, even at starting.

c. The shaft and the bearing surfaces must be smooth and unscored.

d. The lubricating grooves must be placed sufficiently close together, so that the temperature of the water film is not raised excessively.

e. Oil and grease, especially mineral oil, even in small quantities, can often be detrimental to rubber bearings and cause "tackiness." It is possible by suitable compounding, to give rubber considerable resistance to oil.

The grooving commonly employed is straight longitudinal flutes. These are rather close together and well rounded. The number and size of these flutes aid a great deal in promoting a copious amount of cooling water to flow through the bearing and in promoting the discharge of impurities from the bearing. Occasionally, spiral grooving is employed but only when the bearing is in a vertical position.

The pressures that these bearings can stand are limited and are determined largely by the speed of the shaft. Where the shaft is rotating as high as approximately eighteen to twenty revolutions per minute, Haushalten and Moffitt (74) recommend that the pressures not exceed twenty-five pounds per square inch. They recommend that when the shaft is rotating as high as one hundred revolutions per minute, the pressure not exceeding fifty pounds per square inch. Marine propeller shaft bearings usually are loaded to about thirty to thirty-two pounds per square inch.

FUTURE DEVELOPMENTS

The trend of the internal combustion engine will probably follow that of today; namely, higher specific outputs. Contributions to this development will without doubt be better alloys and higher octane fuels that will permit higher compression ratios. The loads imposed on the Diesel bearings are increasing. A factor that must be solved in this development is the production of a better bearing material that is available commercially. The bearing must have a greater fatigue life to withstand the conditions that are going to be imposed upon it. At the same time, the bearing metal of the future must still have its anti-score properties and be even less susceptible to dirt and deflections. It must be non-corrosive. Such a bearing has already been produced experimentally but is not yet commercially available. This experimental bearing has as high a fatigue strength as the copper-lead alloys, as good an anti-friction quality as tin base Babbitt, extra good bonding strength, unusually high mechanical strength corresponding to that of silver, and has the embedability, the conformability, and corrosion resistance of Babbitt. This bearing material combines all the best properties of presently used bearing materials into one bearing metal. An engine equipped with these bearings ran at the speed of 80 miles per hour for the equivalent of 30,000 miles. (53) This is unequalled performance for a bearing metal. Some method may be used in the future internal combustion engine to regulate temperatures of the bearings. Another method that may be used to raise the capacity of the bearings is the elimination of oil grooving.

Other industrial applications of bearings will benefit by the developments taking place in the internal combustion engine bearings. Bearings on turbines, motors and the like are being constantly improved and are being put on sounder scientific bases.

Experimentation and development of the theory will continue and commercial bearings will profit, some taking a long time to make use of these developments, whereas others will make immediate use of them. There is need for such developments as: standards for finishes, better standards for measuring viscosity, the determination of the nature of roughness, the determination of temperature flow from the bearing and shaft, the more accurate determination of temperatures, the more effective sealing of bearings, the determination of the effects of variable loads, and the solving of a host of other problems. At the present, so much needs to be known about the subject that there is disagreement on many points. Such a condition makes a simple presentation of the problem of bearings impossible; consequently, commercial development is hampered. When certain problems are solved, this will not be the case; therefore, the changes in existing practices may be rapid. It is impossible to state definitely what the future has in store for journal bearings.

PLAN OF INVESTIGATION

At first a study of the fundamental aspects of bearings was attempted so that a general understanding of bearing principles would be familiar. All available literature was reviewed so that certain questions on bearing action could be cleared up. In some cases, the questions remained unanswered, for no solution has been found by any one.

All literature in the library was scrutinized for information on bearing practices. For many types of bearings only obsolete data were available. In many cases, a limited amount of up-to-date information was available, but was not as complete as was desired. Letters were therefore sent out to many different companies asking for information needed to complete this survey of modern bearing practices. Scant information was available on the practices in the internal combustion engine. Every manufacturer of automobile engines, Diesel engines, gasoline engines, airplane engines, and motor boat engines listed in the latest statistical issue of the "Automotive Industries" (56) was written. This list of manufacturers was supplemented by others from various other sources. (75)(76)(77) The larger steam engine manufacturers were written and two principle manufacturers of line shafting equipment were written.

From all data accumulated from these sources, a good perspective of modern bearing practices could be acquired.

DISCUSSION OF RESULTS

The results of the library research undertaken in the preparation of this thesis have been given in the review of literature. The latest information on journal bearings published in periodicals has been collected, correlated, and presented in a concise form. The scope of this information was augmented by information supplied by various manufacturers. Space did not permit a complete analysis of all factors, but references referred to may be sought out by those interested in further details of the subject.

Actual data on bearings as given by manufacturers in answer to questionnaires appear in Tables IX, X, and XI. Table IX. lists all data obtained on automobile engine bearings. Table X. lists data on many Diesel engine bearings, and Table XI. lists data on many commercial and industrial gas and gasoline engine bearings. For obvious reasons, the make of the engines has not been given.

Every questionnaire asked for the maximum force on the bearing. Many questionnaires also asked for the average force on the bearing. Some companies gave only the maximum force; other companies gave both the maximum and average force; whereas others gave only the average fore. In general, data requested were sent but ofttimes only one force was calculated by the company, so they sent only that one.

Table IX   1939 AUTOMOBILE ENGINES

Make	General: Lubricant Recommended S A E No.		Big End Connecting Rod Bearing											Main Bearing									
	Winter	Summer	Temperature °F	╤ Bearing Metal	Length, ins. L	Dia., ins., D	Ave. Clearance, ins., C	Max. Press., lb./sq. in. P_{max}.	Viscosity, Z centipoises+	R. P. M., N	ZN/P_{max}.	C/D	L/D	╤ Bearing Metal	Temperature °F	Dia., ins., D	Length, ins. L	Ave. Clearance, ins., C	Max. Press., lb./sq. in. P_{max}.	Viscosity, Z centipoises+	ZN/P_{max}.	C/D	L/D
A	20 W	30	200	BSb	1 7/32	2 1/8	0.0015	1058∘	10.25	4000	38.8	0.00071	0.573	BSb	200	2 1/2	1 5/16	0.0015	480▿	10.25	85.5	0.00060	0.525
	20 W	30	225	BSb	1 1/8	2 3/16	0.0020	1110∘	7.18	4000	28.4	0.00092	0.514	BSb	225	2 45/64	1 1/8	0.0015	1385▿	7.18	20.7	0.00056	0.416
	20 W	30	225	BSb	1 1/8	2 3/16	0.0020	1090∘	7.18	4000	26.3	0.00092	0.514	BSb	225	2 45/64	1 1/8	0.0015	1385▿	7.18	20.7	0.00056	0.416
B	20 W	30	325-	Bab	1.212	2.000	0.0013	1480∘	1.95	4200	5.53	0.00065	0.607	BSb	275-	2 5/16	1 17/64	0.0014	725▿	3.23	18.7	0.00061	0.547
														BSb	275-	2 3/8	15/16	0.0014	1090▿	3.23	12.4	0.00059	0.395
														BSb	275-	2 7/16	1 5/8	0.0014	1030▿	3.23	13.2	0.00058	0.668
														BSb	275-	2 1/2	15/16	0.0014	1040▿	3.23	13.0	0.00056	0.375
														BSb	275-	2 9/16	1 25/32	0.0014	441▿	3.23	30.8	0.00055	0.695
B	20 W	20	325-	Bab	1.306	2 1/4	0.0013	1565∘	1.95	4130	5.15	0.00058	0.580	BSb	275-	2 9/16	1 9/32	0.0014	884▿	3.23	15.1	0.00055	0.500
														BSb	275-	2 5/8	31/32	0.0014	1395▿	3.23	9.56	0.00053	0.369
														BSb	275-	2 11/16	1 15/32	0.0014	1480▿	3.23	9.02	0.00052	0.547
														BSb	275-	2 3/4	31/32	0.0014	1330▿	3.23	10.0	0.00051	0.352
														BSb	275-	2 13/16	2 15/32	0.0014	400▿	3.23	33.4	0.00050	0.880
C	20 W	20	325-	BSb	2 1/32	2.46	0.0015	916∘	1.95	4000	8.53	0.00061	0.825	BBSb	275-	2 1/2	1 1/16	0.0015	155▿	3.23	83.4	0.00060	0.425
														BBSb	275-	2 1/2	1 5/32	0.0015	273▿	3.23	47.4	0.00060	0.463
														BBSb	275-	2 1/2	2 31/32	0.0015	72▿	3.23	179	0.00060	0.788
C	20 W	20	325-	BSb	2 1/32	2.46	0.0015	916∘	1.95	4000	8.53	0.00061	0.825	BBSb	275-	2 1/2	1 1/16	0.0015	155▿	3.23	83.4	0.00060	0.425
														BBSb	275-	2 1/2	1 5/32	0.0015	273▿	3.23	47.4	0.00060	0.463
														BBSb	275-	2 1/2	1 31/32	0.0015	72▿	3.23	179	0.00060	0.788
C	20 W	20	323-	BSb	1 3/4	2.00	0.0015	450	1.95	4000	17.3	0.00061	0.875	BBSb	275-	2 1/2	1 5/16	0.0015	1523▿	3.23	8.47	0.00060	0.526
														BBSb	275-	2 1/2	1 1/16	0.0015	1550▿	3.23	8.34	0.00060	0.425
D	20 W	20	295	BSb	1 3/8	2 1/8	0.0015	1810	2.63	4500	6.54	0.00071	0.648	BSb	295	2 35/64	1 1/2	0.0017	1220	2.63	9.70	0.00067	0.584
	20 W	20	295	BSb	1 3/8	2 1/8	0.0015	1515	2.63	4500	7.81	0.00071	0.648	BSb	295	2 35/64	1 3/16	0.0020	1895	2.63	6.25	0.00079	0.466
E	20	30	280	Bab	1 3/8	2 3/8	0.0012	1540	3.89	4000	10.1	0.00051	0.580	BSb	280	2 1/2	1 1/8	0.0015	1278	3.89	12.1	0.00060	0.450
	20	30	290	Cle	1 3/16	1 7/8	0.0017	1810	3.42	4000	7.56	0.00097	0.633	BSb	290	2 11/32	29/32	0.0020	1580	3.42	8.60	0.00085	0.387
F	10 W	20	320-	Bab	1.201	2 1/8	0.0010	2312	2.03	4500	3.95	0.00047	0.567	Bab	300-	2 1/2	1.045	0.0020	1229	2.51	9.18	0.00080	0.418
														Bab	300-	2 17/32	.982	0.0020	1288	2.51	8.76	0.00079	0.388
														Bab	300-	2 17/32	.979	0.0020	1261	2.51	8.96	0.00077	0.377
														Bab	300-	2 5/8	1.295	0.0020	945	2.51	12.0	0.00076	0.494
F	10 W	20	320-	Bab	.9825	2.000	0.0010	2250	2.03	4300	3.88	0.00050	0.492	Bab	300-	2 3/8	1.045	0.0020	950	2.51	11.4	0.00084	0.440
														Bab	300-	2 13/32	.982	0.0020	1330	2.51	8.11	0.00083	0.407
														Bab	300-	2 7/16	1.232	0.0020	1200	2.51	8.99	0.00082	0.505
														Bab	300-	2 15/32	.980	0.0020	1300	2.51	8.30	0.00081	0.397
														Bab	300-	2 1/2	1.3875	0.0020	685	2.51	15.7	0.00080	0.555
F	10 W	20	320-	Bab	1.201	2 1/8	0.0010	1469▪	2.03	4500	6.22▫	0.00047	0.567	Bab	300-	2 1/2	1.045	0.0020	783▪	2.51	14.4▫	0.00080	0.418
														Bab	300-	2 17/32	.982	0.0020	817▪	2.51	13.8▫	0.00079	0.388
														Bab	300-	2 19/32	.979	0.0020	800▪	2.51	14.1▫	0.00077	0.377
														Bab	300-	2 5/8	1.295	0.0020	601▪	2.51	18.8▫	0.00076	0.494
F	10 W	20	320-	Bab	.9825	2.000	0.0010	1355▪	2.03	4300	6.44▫	0.00050	0.492	Bab	300-	2 3/8	1.045	0.0020	540▪	2.51	20.0▫	0.00084	0.440
														Bab	300-	2 13/32	.982	0.0020	670▪	2.51	16.1▫	0.00083	0.407
														Bab	300-	2 7/16	1.232	0.0020	810▪	2.51	13.3▫	0.00082	0.505
														Bab	300-	2 15/32	.980	0.0020	650▪	2.51	16.6▫	0.00081	0.397
														Bab	300-	2 1/2	1.3875	0.0020	390▪	2.51	27.7▫	0.00080	0.555
G	20 W	20	300-	Bab	1 1/2	2 5/16	0.0017	761▪	2.51	4000	13.2▫	0.00074	0.650	BSb	300-	2 11/16	1 3/16	0.0030	562▪	2.51	17.9▫	0.00111	0.441
	20 W	20	300-	Bab	1 1/2	2 5/16	0.0017	1300◸	2.51	4000	7.73	0.00074	0.650	BSb	300-	2 23/32	1 3/16	0.0030	668▪	2.51	15.o▫	0.00110	0.437
														BSb	300-	2 3/4	1 7/16	0.0030	660▪	2.51	15.2▫	0.00109	0.523
														BSb	300-	2 25/32	1 5/8	0.0030	389▪	2.51	25.8▫	0.00108	0.585
														BSb	300-	2 25/32	1 5/8	0.0030	395▪	2.51	25.4▫	0.00108	0.585
±Average					1.3864	2.1738	0.0015	1408	3.57	4142	12.41	0.00068	0.635			2.5248	1.2548	0.0016	979	3.47	31.07	0.00065	0.496
+Average					1.2278	2.1458	0.0012	1195	2.19	4267	8.62	0.00057	0.570			2.5826	1.1334	0.0024	624	2.51	18.2	0.00091	0.468

+ All absolute viscosities are calculated from the oils recommended for summer use.
− Approximated using data from manufacturers as a basis.
∘ No allowance made for grooving or fillets in calculating the pressure.
▿ Two tenths of an inch allowed for grooving and fillets in calculating the pressure.
◸ Derived from the average pressure using the factor 1.71 as the ratio of the maximum pressure to the average pressure.( )
▪ Average pressure, pounds per square inch.
▫ Calculated from the average pressure, thus giving ZN/P_{ave}.

╤ BSb-babbitt, steel back; Bab-babbitt; Cle-Clevite; BBSb-babbitt, bronze or steel back.
± Average of maximum values only.
+ Average of average values only.

Table X 1939 DIESEL ENGINES

Make	General: Lubricant Recommended S A E No., Winter	General: Lubricant Recommended S A E No., Summer	General: Temperature ° Fahrenheit	Big End Connecting Rod Bearing: Length, L inches	Diameter, D inches	Average Clearance, C inches	Average Pressure, P_{ave} pounds per square inch	Maximum Pressure, P_{max} pounds per square inch	Viscosity, Z centipoises ‡	Revolutions per Minute, N	$ZN/P_{ave.}$	$ZN/P_{max.}$	C/D	L/D	Main Bearing: Diameter, D inches	Length, L inches	Average Clearance, C inches	Average Pressure, P_{ave} pounds per square inch	Maximum Pressure, P_{max} pounds per square inch	$ZN/P_{ave.}$	$ZN/P_{max.}$	C/D	L/D
A	30	40	190-	3.625	5.5	0.005	94.8 °ø	642 °ø	17.2	900	163	24.1	0.00091	0.659	5.5	2.875	0.005	° 75.3		205		0.00091	0.522
	30	40	190-	5.625	7.5	0.006	36.5 °ø	591 °ø	17.2	750	353	21.8	0.00080	0.751	7.5	4.125	0.006	° 33.9		381		0.00080	0.551
B		30-	220-	2.30	4.00	0.0025	581	1847	7.62	1225	16.1	5.05	0.00063	0.575	4.50	2.06	0.0028	534	1280	17.5	7.29	0.00062	0.458
		30-	220-	2.69	5.75	0.0034	621	1764	7.62	1000	12.3	4.32	0.00059	0.468	6.75	4.25	0.0038	470	614	16.2	12.4	0.00056	0.630
		30-	220-	2.75	6.25	0.0038	544	2810	7.62	750	10.5	2.03	0.00061	0.440	7.25	3.69	0.0038	647	1195	8.83	4.78	0.00052	0.509
C	30	30	235-	1.78	2.750	0.003		2020	6.30	2000		6.24	0.00109	0.648	3.500	1.125	0.003		1740		7.25	0.00086	0.322
	30	30	235-	1.78	2.750	0.003		2020	6.30	2000		6.24	0.00109	0.648	3.500	1.125	0.003		1630		7.73	0.00086	0.322
	30	30	235-	1.78	2.750	0.003		2020	6.30	2000		6.24	0.00109	0.648	3.500	1.125	0.003		1780		7.08	0.00086	0.322
D	30	40	120	1 7/8	1 7/8	0.002	197 °		88.9	1200	542		0.00106	1.000		Timken	Roller						
	30	40	120	2 3/8	2 1/4	0.003	204 °		88.9	850	371		0.00133	1.055		Timken	Roller						
	30	40	120	2 1/4	2 3/4	0.003	215 °		88.9	1200	496		0.00109	0.818		Timken	Roller						
	30	40	120	3	2 7/8	0.003	225 °		88.9	720	285		0.00104	1.040		Timken	Roller						
E	20	30	220-	3 1/4	2 3/4	0.003	203 °	335 ø	7.62	1050	39.4	23.9	0.00109	1.182	3	3 1/2	0.003	146 °		54.7		0.00100	1.168
	20	30	220-	2 1/2	3 1/4	0.003	215 °	401 ø	7.62	1050	37.2	20.0	0.00092	0.769	3 5/16	3 3/8	0.003	130 °		61.6		0.00096	1.081
	20	30	220-	3 1/4	2 3/4	0.003	184 °	335 ø	7.62	1050	43.4	23.9	0.00109	1.182	3	3 1/2	0.003	131 °		61.0		0.00100	1.168
F		40-	165-	7 1/4	14 3/4	0.006	227	2430	28.2	225	28.0	2.61	0.00041	0.492	16	12	0.010	65	680	97.7	9.34	0.00062	0.750
G	40	40	200	6 3/8	8 1/4	0.0045		1250	14.4	700		8.06	0.00055	0.773	9 1/2	5	0.0065		723		13.9	0.00069	0.527
	40	40	175	7 1/4	10 1/2	0.008		1740	22.8	257		3.37	0.00076	0.691	11 3/4	7 1/4	0.010		720		8.15	0.00085	0.617
	40	40	200	5	6 1/4	0.004		1525	14.4	700		6.61	0.00064	0.800	7 1/8	6 1/2	0.005		900		11.2	0.00070	0.912
	40	40	175	6	9 1/2	0.007		2160	22.8	277		2.91	0.00074	0.632	9 1/2	6	0.007		1080		5.83	0.00074	0.632
	30	30	160	7 3/4	14	0.010		1570	31.4	425		8.50	0.00071	0.554	14	7 3/4	0.010		885		15.1	0.00071	0.552
H	20	30	160	4 3/4	5	0.004	343 °	1090°	20.9	720	43.8	13.8	0.00080	0.950	5 3/4	6	0.0045	330 °	456°	45.6	33.0	0.00078	1.043
I	10-20	30	220-	2.03	3.248	0.0025	218		7.62	1100	38.5		0.00077	0.625	3.248	1.995	0.0025	470	603	17.8	13.9	0.00077	0.617
						0.0035		1600	7.62	800		3.80	0.00107				0.0035					0.00107	
	10-20	30	220-	2.03	3.248	0.0025	307		7.62	1250	31.1		0.00077	0.625	3.248	1.995	0.0025	611	790	15.6	12.1	0.00077	0.617
						0.0035		1810	7.62	800		3.36	0.00107				0.0035					0.00107	
	10-20	30	220-	2.03	3.248	0.0025	307		7.62	1400	34.8		0.00077	0.625	3.498	2.60	0.0025	574	718	18.6	14.8	0.00072	0.743
						0.0035		1810	7.62	800		3.36	0.00107				0.0035					0.00100	
J	30	30	220-	3 13/16	5	0.0055	938 °		7.62	720	5.85		0.00110	0.763	5 1/2	3 9/16	0.007	38 °		14.4		0.00127	0.647
	10-20	10-20	220-	2 9/16	3 5/8	0.0047	204 °		6.01	900	26.5		0.00129	0.708	5 1/2	1 7/8	0.0059	80 °		67.6		0.00107	0.341
	10-20	10-20	220-	1 7/16	2 3/4	0.0028	586 °		6.01	1200	12.3		0.00100	0.523	3 7/16	1 5/8	0.0036	132 °		54.6		0.00105	0.473
	30	30	220-	4 1/2	5	0.004	83 °		7.62	450	41.3		0.00080	0.900	5	5 3/8	0.002	27.5		125		0.00040	1.075
	30	30	220-	5 1/2	6	0.004	40 °		7.62	400	76.3		0.00067	0.917	6	5 3/4	0.003	° 18.3		167		0.00050	0.958
	30	30	220-	6	8	0.005	34 °		7.62	400	79.2		0.00063	0.750	8	6 7/8	0.006	° 9.5		321		0.00075	0.860
	30	30	220-	8 5/8	10	0.004	21 °		7.62	250	90.7		0.00040	0.863	10	9	0.000	° 7.1		272		0.00000	0.900
Average +				3.5841	4.8904	0.0037	276		23.2	865	120		0.00088	0.778	5.7897	4.3014	0.0041	227		101		0.00077	0.756
Average ±				3.8938	5.9045	0.0043		1513	13.2	928		9.53	0.00082	0.702	7.0387	4.4041	0.0051		987		11.5	0.00075	0.598

‡ All absolute viscosities are calculated from the oils recommended for summer use.
\- Approximated using data from manufacturers as a basis.
° No allowance made for grooving or fillets in calculating the pressure from the imposed force.
ø Calculated using P. M. Heldt's method as described by him in ''Rising Engine Speeds Lift Average Big End Bearing Loads to 1200 Pounds per Square Inch'', Automotive Industries, June 8, 1935.
± Average of values corresponding to $P_{max.}$
\+ Average of values corresponding to $P_{ave.}$

Table XI 1939 GASOLINE ENGINES

Make	General: Lubricant Recommended S A E No., Winter	Summer	Temperature °F	Big End Connecting Rod Bearing: Length, ins. L	Dia., ins., D	Ave. Clearance, ins., C	Ave. Press., lb./sq. in. $P_{ave.}$	Max. Press., lb./sq. in. $P_{max.}$	Viscosity, Z centipoises +	R. P. M., N	$ZN/P_{ave.}$	$ZN/P_{max.}$	C/D	L/D	Main Bearing: Dia., ins., D	Length, ins. L	Ave. Clearance, ins., C	Ave. Press. lb./sq. in. $P_{ave.}$	Max. Press. lb./sq. in. $P_{max.}$	$ZN/P_{ave.}$	$ZN/P_{max.}$	C/D	L/D
A	10	30	220 –	1 7/32	2 1/16	0.0015	179 °	241 ▽°	7.62	1400	59.6	44.3	0.00073	0.593	2 1/4	1 3/8	0.002	226 °		47.2		0.00089	0.611
	30	40	220 –	1 15/32	2 3/8	0.002	179 °	246 ▽°	10.5	1300	76.2	55.4	0.00084	0.619	2 1/2	1 15/32	0.0025	177 °		77.0		0.00100	0.775
	30	40	220 –	2 3/32	2 3/8	0.002	111 °	295 ▽°	10.5	1200	114	42.7	0.00084	0.881	2 1/2	2 7/32	0.002	135 °		93.3		0.00080	0.887
	30	40	220 –	3 1/4	2 3/4	0.003	139 °	314 ▽°	10.5	1050	79.3	35.1	0.00109	1.182	3	4 3/4	0.003	68 °		162		0.00100	1.584
B	10-20	30-40	180	1.28	2.18	0.0015	550 °	831 °	14.1	2800	71.7	47.5	0.00069	0.635	2.62	1.90	0.0015	165	231	239	171	0.00057	0.726
															2.62	2.53	0.0015	132	174	299	227	0.00057	0.966
	10-20	30-40	180	1.28	2.18	0.0015	657 °	987 °	14.1	2800	60.1	40.0	0.00069	0.635	2.62	1.90	0.0015	217	296	184	133	0.00057	0.726
															2.62	2.53	0.0015	170	222	232	178	0.00057	0.966
C	10W-20W	30	180	1.50	2.62	0.0023	224 °	682 °	14.1	1500	94.5	31.0	0.00088	0.573		Double Shielded Ball Type							
D		30-50	220	2.00	2.87	0.0025	618 °	1110 °	13.6	2400	52.8	29.4	0.00087	0.697									
E	20	30	230	1.81	2.50	0.002	480 °	706 °	6.70	2350	32.8	22.3	0.00080	0.724									
	20	30	230	1.81	2.50	0.002	500 °	773 °	6.70	2350	31.5	20.4	0.00080	0.724									
	20	30	210	2.09	3.00	0.002	354 °	598 °	8.75	2000	49.4	29.3	0.00067	0.697									
	20	30	210	2.09	3.00	0.002	380 °	742 °	8.75	2000	46.0	23.6	0.00067	0.697									
	20	30	210	2.09	3.00	0.002	379 °	957 °	8.75	1800	41.6	16.5	0.00067	0.697									
F	20-30	40	210	1.25	1	0.002		1423 °	12.25	1200		10.3	0.00200	1.250		Ball Bearings							
	20-30	40	180	1.50	1.50	0.002		790 °	20.7	1200		31.4	0.00133	1.000		Tapered Roller Bearings							
	20-30	40	190	1.50	1.50	0.002		950 °	17.2	1200		21.7	0.00133	1.000		Ball Bearings							
	20-30	40	190	1.50	1.50	0.002		1320 °	17.2	1200		15.6	0.00133	1.000		Ball Bearings							
	20-30	40	165	1.50	1.50	0.002		1320 °	28.2	1200		25.6	0.00133	1.000		Ball Bearings							
	20-30	40	165	1.50	1.50	0.002		950 °	28.2	1200		35.6	0.00133	1.000		Ball Bearings							
G	20	30	235 –	1 5/16	2.312	0.0025		1390 °	6.30	3000		13.6	0.00108	0.568									
H		30	280	1 9/16	2.25	0.002		1325	3.85	3000		8.71	0.00089	0.694	2 5/8	1 13/32	0.002		1080		10.7	0.00076	0.535
			274						4.08	3000					2 5/8	1 1/2	0.002		1240		9.86	0.00076	0.572
			267						4.43	3000					2 5/8	2 5/16	0.002		1360		9.76	0.00076	0.882
			284						3.64	3000					2 5/8	2 5/32	0.002		740		14.8	0.00076	0.821
I		10	120 –	4	4.75			3850 °	24.4	250		1.58		0.842									
		10	120 –	6	8			1940 °	24.4	250		3.14		0.750									
+Average				1.8447	2.570	0.0020	365		10.4	1920	62.3		0.00079	0.720	2.5913	2.3341	0.0019	161		167		0.00075	0.905
±Average				1.9829	2.575	0.0020		1032	13.8	1680		26.3	0.00099	0.803	2.6225	2.0294	0.0018		668		94.3	0.00067	0.774

+ All absolute viscosities are calculated from the oils recommended for summer use.
– Approximated using data from manufacturers as a basis.
° No allowance made for grooving or fillets in calculating the pressure from the imposed force.
▽ Calculated using P. M. Heldt's method as described by him in ''Rising Engine Speeds Lift Average Big End Bearing Loads to 1200 Pounds per Square Inch'', Automotive Industries, June 8, 1935.
± Average of values corresponding to $P_{max.}$
+ Average of values corresponding to $P_{ave.}$

Some companies design bearings on the basis of the average pressure on the bearing, whereas others base the design on the maximum pressure on the bearing. There is a wide difference in opinion whether to use as a basis the maximum or the average pressure. Usually, a manufacturer bases his design on some similarly tested and reliable bearing, using as a basis the pressure that past experiences indicate safe and satisfactory.

The relation between the maximum force and the average force in similar engines probably does not vary beyond certain rather narrow practical limits. If this is true, it makes no difference whether the maximum or the average pressure is used in comparing similar designs. The basis used for the analysis of the bearings in this thesis is the maximum pressure.

The true temperature of the variable oil film in a bearing is very hard to determine. Those listed in the tables are probable maximum operating temperatures of the oil film in the bearings. The operating temperature must be available so that the absolute viscosity of the oil in the bearing may be obtained. Also, the oils used commercially have a variable viscosity, although rated the same. Therefore, one of the most important factors in bearing lubrication is indeterminate to a certain degree. The determination of the operating viscosity of the oil in centipoises is accurate enough, however, to enable reasonable analyses to be made of the different bearings investigated.

The absolute viscosity of the oil was obtained, using the following average values:

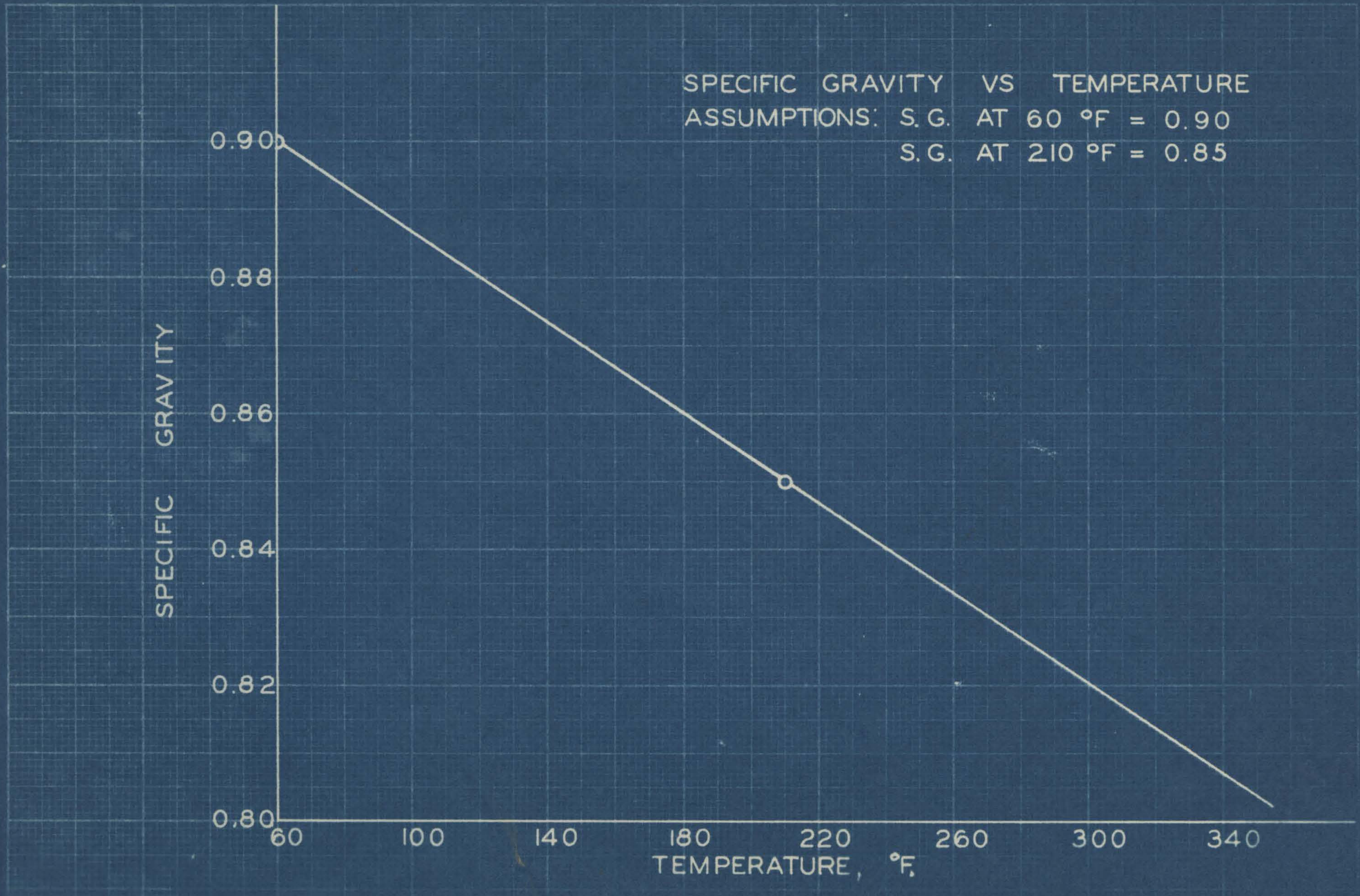
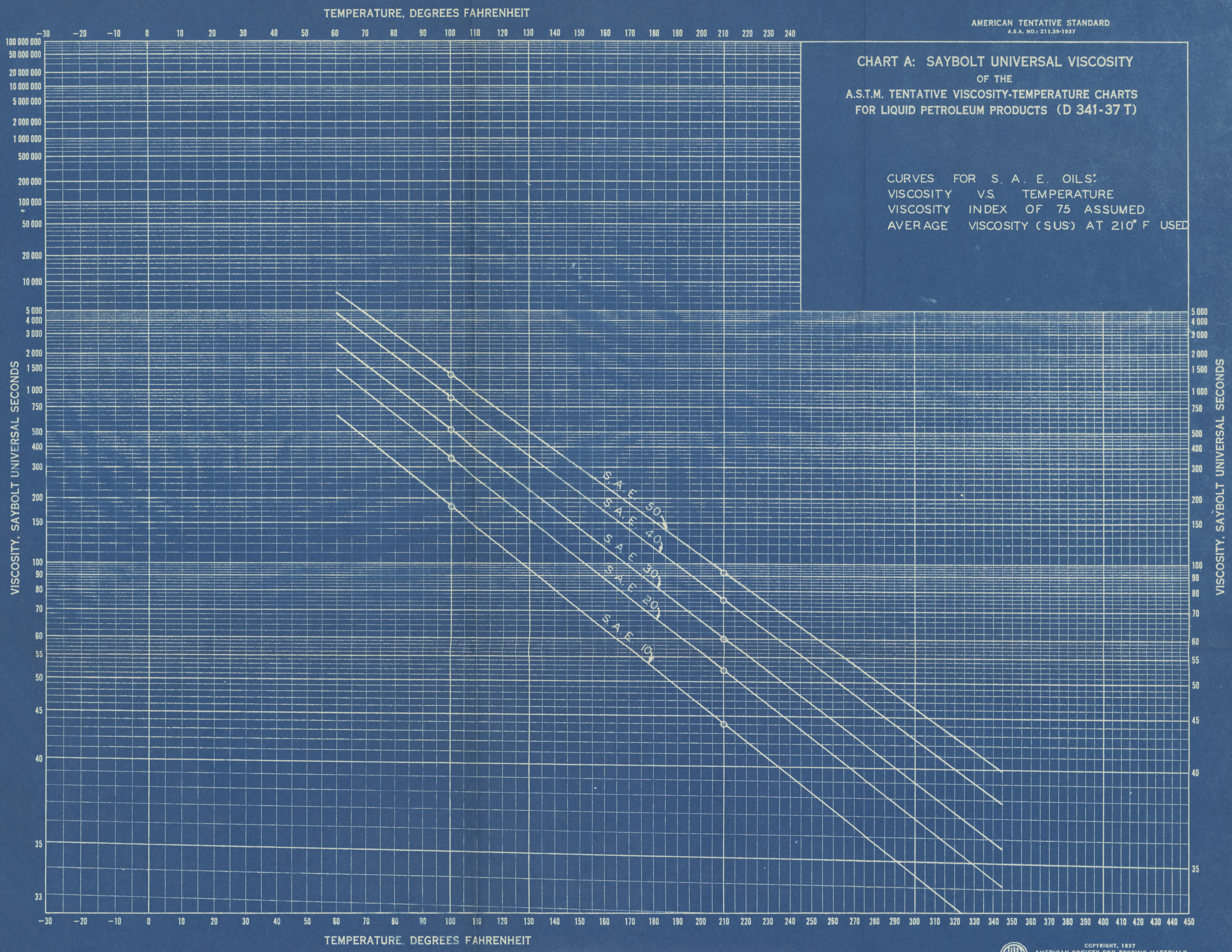


Fig. 37 - Specific Gravity as a Function of the Temperature.



PRINTED IN U. S. A.

Fig. 38 - Viscosity as a Function of the Temperature.

A Viscosity Index of 75 for S. A. E. Oils.

Average Viscosity (S. U. S.) at 210° F. of 44, 52, 60, 75, and 92 for S. A. E. 10, 20, 30, 40, and 50 oils respectively.

Specific gravity at 60° F. of 0.90.

Specific gravity at 210° F. of 0.85.

K V (centistokes) $= 0.226t - \frac{195}{t}$

Where: $t = 100$ S. U. S. or less

K V (centistokes) $= 0.220t - \frac{135}{t}$

Where: $t =$ more than 100 S. U. S.

Kinematic viscosity in centistokes x specific gravity = absolute viscosity in centipoises.

Figures 37 and 38 were first prepared, and then the viscosity in centipoises was calculated from the given data.

Due to the diversity of engine bearings produced by different manufacturers, the results were averaged. These average bearings should be safe bearings and were used to check Needs' rational method of designing bearings. This method can best be understood by referring to the original presentation, "Effects of Side Leakage in 120-Degree Centrally Supported Journal Bearings." (78)

Sample calculations for an automobile connecting-rod bearing of l/w ratio of 1.00 are given in Table XII. Figure 39 shows the analysis of this automobile bearing. The curves were drawn through points which were obtained as in Table XII. for various l/w ratios.

Table XII. Sample Calculations for Needs' Method of Analysis.

<u>Symbols are as follows:</u>

η = radial clearance
c = eccentricity factor, ratio of eccentricity to the radial clearance
$c\eta$ = eccentricity
h_o = minimum film thickness, or $\eta(1 - c)$ when c = 0.141
U = linear velocity, in./sec., of journal surface, or $\pi DN/60$
l = developed bearing length in the direction of motion, or $\pi D/3$ for the 120-degree bearing.
w = axial width of bearing
P_o = load per unit of projected journal area, or W/D
W = mean load per unit bearing width
D = diameter of journal
a = radius of journal
N = R. P. M.
λ = coefficient of friction
μ = average film viscosity in absolute units
L = load on bearing in pounds
Z = viscosity-centipoises

Bearing: Average bearing for automobile connecting rod bearing.

$a = \frac{D}{2} = 1.0869$ in.

$l = \frac{\pi D}{3} = \frac{2.1738\pi}{3} = 2.275$ in.

N = 4142 rpm.

$U = \frac{\pi DN}{60} = \frac{\pi 2.1738 \times 4142}{60} = 471$ in./sec.

$\mu = 1.46 \times 10^{-7} Z = 1.46 \times 10^{-7} \times 3.57 = 5.21 \times 10^{-7}$ lb.-sec./in.^{2}

$L = 1408 \times 1.3864 \times 2.1738 = 4240$ lbs.

Assume $l/w = 1.00$; then $l = w$

$W = \frac{L}{w} = \frac{4240}{2.275} = 1863$ lb./in. width

From Fig. 15 (78) for $l/w = 1.00$

$\lambda = \frac{5.55\, h_o}{a}$; C = 0.612

From Fig. 14 (78) for $l/w = 1.00$ and C = 0.612

$W = 0.308 \frac{\mu U a^2}{h_o^2}$

$0.308 \frac{\mu U a^2}{h_o^2} = 1863$

$h_o = \sqrt{\frac{0.308 \mu U a^2}{1863}} = \sqrt{\frac{0.308 \times 5.21 \times 10^{-7} \times 471 \times 1.0869^2}{1863}}$

$h_o = 0.000219$ in.

$= \frac{5.55 \times 0.000219}{1.0869} = 0.00112$

Since $h_o = \eta(1 - c)$; $\eta = \frac{0.000219}{(1-0.612)} = 0.000519$ in.

$P_o = \frac{1863}{2.1738} = 858$ lb./sq. in.

FIG. 39 — AUTOMOBILE CONNECTING ROD BEARING

FIG. 40 — AUTOMOBILE MAIN BEARING

FIG. 41 — DIESEL CONNECTING ROD BEARING

FIG. 42 — DIESEL MAIN BEARING

FIG. 43 — GASOLINE ENGINE CONNECTING ROD BEARING

FIG. 44 — GASOLINE ENGINE MAIN BEARING

NEED'S METHOD OF ANALYSIS, SHOWING THE EFFECT OF THE LENGTH-WIDTH RATIO ON THE MINIMUM FILM THICKNESS, FRICTION COEFFICIENT, UNIT LOAD, AND RADIAL CLEARANCE FOR THE AVERAGE BEARINGS INDICATED.

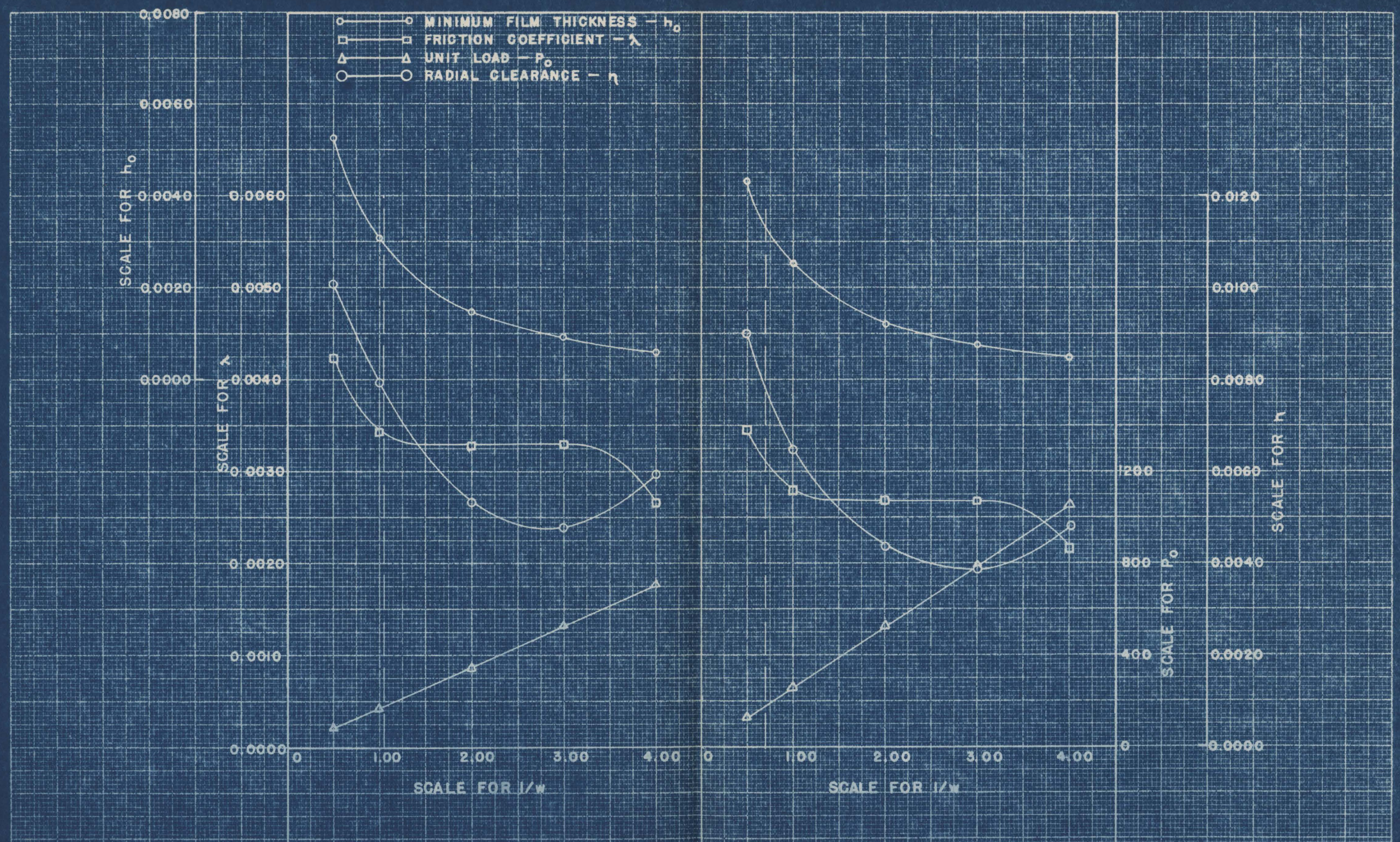


FIG. 45 — TURBINE BEARING A

FIG. 46 — TURBINE BEARING B

NEED'S METHOD OF ANALYSIS, SHOWING THE EFFECT OF THE LENGTH-WIDTH RATIO ON THE MINIMUM FILM THICKNESS, FRICTION COEFFICIENT, UNIT LOAD, AND RADIAL CLEARANCE FOR THE TURBINE BEARINGS INDICATED

"In operation, the most important safety feature is the minimum film thickness. It is desirable therefore to keep h_o as large as possible. Operating economy is governed by the coefficient of friction: hence, λ should be as low as possible. When starting and stopping the machine, there will be a short period of metallic contact in the bearing. Hence, the unit pressure should be reasonably small." (78)

From Fig. 39, there is practically no change in the friction coefficient between $l/w = 1.5$ and $l/w = 3.00$. Since h_o is greater at $l/w = 1.5$, this is a better bearing than $l/w = 3.00$. At $l/w = 1.5$, the friction coefficient is about as low as that at $l/w = 3.00$, but h_o is over 133 per cent greater in the wider bearing. The width $l/w = 1.5$, also, has the advantage of having P_o only about one-half the value at $l/w = 3.00$. This is important in starting and stopping.

Below $l/w = 1.5$, the coefficient of friction increases rapidly. This is also true of h_o but the gain in safety factor is not so great as may be expected, for the bearing is increasing in width, which is undesirable in design and for the fact that the additional heat generated causes greater power loss and lowers the operating viscosity of the oil, which, in turn, tends to decrease the film thickness.

The total frictional drag will be $0.00106 \times 4240 = 4.5$ pounds and the power loss will be $\frac{471}{12} \times \frac{4.5}{550} = 0.321$ horsepower. Total clearance $= 2 \times 0.00043 = 0.00086$.

The l/w ratio for this bearing as determined by actual present practice is 1.64. The bearing having a l/w ratio of 1.64 is practically the same as the bearing having a l/w ratio of 1.5. Hence we have proof that Needs' method of analysis is applicable to this bearing.

To make a more extensive check of Needs' rational method of analyzing bearings, Figs. 40, 41, 42, 43, 44, 45, and 46 have been prepared. These figures give analyses of the other bearings listed in Table XIII.

The dotted lines represent the l/w ratios for the bearings as were determined by actual present practice. By analyzing each of these bearings independently, it is seen that Needs' method of analysis is sufficient for each bearing.

One outstanding feature of these analyses indicates that present practice calls for more safety in the turbine bearings than in the engine bearings. Safety in engine bearings has been sacrificed that they may be made shorter. This shortening of the engine bearings gives a better designed engine, when all factors are considered.

Table XIII. Values for Needs' Analysis of Various Bearings, Used in Plotting Figs. 39, 40, 41, 42, 43, 44, 45, and 46.

	Analysis																Actual	
Kind of Bearing	D	a	l	N	U	μ	L	W	$\lambda\left(\frac{a}{\eta}\right)$	c	$W\frac{h_o^2}{\mu U a^2}$	h_o	λ	P_o	1/w	N	Width	1/w
Automobile Connecting Rod	2.1738	1.0869	2.275	4142	471	5.21×10^{-7}	4240	1863	5.55	0.612	0.308	0.000219	0.00112	858	1.00	0.000519	1.3864	1.64
								3726	11.0	0.72	0.143	0.000105	0.001063	1716	2.00	0.000375		
								5589	18	0.81	0.08	0.0000643	0.001068	2574	3.00	0.000339		
								7452	22.5	0.900	0.045	0.0000418	0.000865	3430	4.00	0.000418		
								932	4	0.52	0.45	0.000355	0.001308	428	0.50	0.000740		
Automobile Main Bearing	2.5248	1.2548	2.645	4181	552	5.07×10^{-7}	3100	587	4	0.52	0.45	0.000582	0.001855	233	0.50	0.001213	1.2548	2.105
								1173	5.55	0.612	0.308	0;000341	0.001505	466	1.00	0.000853		
								2346	11.0	0.72	0.143	0.000164	0.001438	932	2.00	0.000586		
								3519	18	0.81	0.08	0.000100	0.001435	1398	3.00	0.000526		
								4692	22.5	0.900	0.045	0.0000651	0.001168	1864	4.00	0.000651		
Diesel Connecting Rod	5.9045	2.9523	6.18	928	287	19.3×10^{-7}	34,800	2815	4	0.52	0.45	0.000879	0.00119	477	0.50	0.001832	3.8938	1.588
								5630	5.55	0.612	0.308	0.000514	0.000965	954	1.00	0.001320		
								11260	11.0	0.72	0.143	0.000248	0.000923	1908	2.00	0.000836		
								16890	18	0.81	0.08	0.0001513	0.000932	2862	3.00	0.000796		
								22520	22.5	0.900	0.045	0.0000984	0.00075	3816	4.00	0.000984		
Diesel Main Bearing	7.0387	3.5194	7.37	1002	369	20.05×10^{-7}	30,600	2075	4	0.52	0.45	0.00141	0.001604	295	0.50	0.000284	4.4041	1.672
								4150	5.55	0.612	0.308	0.000826	0.001302	590	1.00	0.002115		
								8300	11.0	0.72	0.143	0.000398	0.001245	1180	2.00	0.001422		
								12450	18	0.81	0.08	0.000243	0.001242	1770	3.00	0.00128		
								16600	22.5	0.900	0.045	0.000158	0.00101	2360	4.00	0.00158		
Gasoline Connecting Rod	2.575	1.288	2.7	1680	227	2.015×10^{-6}	5270	976	4	0.52	0.45	0.000592	0.001838	379	0.50	0.001234	1.9829	1.362
								1952	5.55	0.612	0.308	0.000346	0.00149	758	1.00	0.000888		
								3904	11.0	0.72	0.143	0.000167	0.001425	1516	2.00	0.000596		
								5856	18	0.81	0.08	0.000108	0.001425	2274	3.00	0.000537		
								7808	22.5	0.900	0.045	0.0000662	0.001155	3032	4.00	0.000662		
Gasoline Main Bearing	2.6225	1.3113	2.75	2900	399	1.322×10^{-6}	3560	647	4	0.52	0.45	0.000794	0.002424	247	0.50	0.001653	2.0294	1.355
								1294	5.55	0.612	0.308	0.000464	0.001964	494	1.00	0;00119		
								2588	11.0	0.72	0.143	0.0002235	0.001875	988	2.00	0.000798		
								3882	18	0.81	0.08	0.0001365	0.001875	1482	3.00	0.000718		
								5176	22.5	0.900	0.045	0.0000889	0.001525	1976	4.00	0.000389		
*Turbine Bearing (A)	10	5	10.48	3600	1885	11.6×10^{-7}	18,500	882	4	0.52	0.45	0.00527	0.00422	88.2	0.50	0.01098	10	1.048
								1763	5.55	0.612	0.308	0.00309	0.003435	176	1.00	0.00793		
								3526	11.0	0.72	0.143	0.00149	0.00328	353	2.00	0.00532		
								5289	18	0.81	0.08	0.00091	0.00328	529	3.00	0.00479		
								7052	22.5	0.900	0.045	0.000592	0.002665	705	4.00	0.00592		
*Turbine Bearing (B)	10	5	10.48	3600	1885	11.6×10^{-7}	27,750	1323	4	0.52	0.45	0.00432	0.003455	132	0.50	0.00900	15	0.698
								2645	5.55	0.612	0.308	0.002525	0.00289	265	1.00	0.00647		
								5290	11.0	0.72	0.143	0.001218	0.00268	529	2.00	0.00435		
								7935	18	0.81	0.08	0.000743	0.00268	794	3.00	0.00391		
								10580	22.5	0.900	0.045	0.000483	0.002175	1058	4.00	0.00483		

*From: Howarth, H.A.S., "Current Practice in Pressures, Speeds, Clearances, and Lubrication of Oil-Film Bearings." (79)

SUMMARY AND CONCLUSIONS

A complete summary of this thesis would be too lengthy due to the amount of material covered, so only the most important points will be brought out.

(a) The journal bearing is one of the most important elements of machinery.

(b) An outline of the fundamentals of oiling and grooving bearings has been given. The bearing metals and the lubricants in use today and the functions of the lubricating engineer have been discussed.

(c) Present day journal bearing practices have been described for the following types of equipment: automobile engines, aircraft engines, Diesel engines, steam turbines, electric motors, and generators, rolling mills, railroad cars and engines, heavy duty equipment, line shafting, gyroscopes, ships and steam engines.

(d) The basic theory of the perfectly lubricated journal bearing is understood, but exact mathematical solution of the phenomena occurring within the bearing is as yet impossible.

(e) By coordinating mathematical investigation and experimental evidence, a fairly good analysis can be made of journal bearing behavior.

(f) Needs' rational method of designing bearings is a good example of well coordinated mathematical investigation and experimental results. This method of analysis gives the designer a fairly accurate indication of the performance of a journal bearing to be expected.

(g) Because of the limitations of theory a large number of manufacturers do extensive research and experimental work in conjunction with their products, to aid in the development of the journal bearings used.

(h) Those interested in further study on this subject are advised to study references 79 and 80 in addition to those already mentioned. Reference 79 is a survey of bearing practice undertaken in 1934 by H. A. S. Howarth. This survey is a valuable adjunct to this thesis, yet is too long to repeat here. Reference 80 is another article by Howarth summarizing the important formulas and charts that had been offered by bearing analysts during the years prior to 1935.

SUGGESTIONS FOR FURTHER RESEARCH

This survey on bearing practices has indicated that the following research may be beneficially pursued:

(a) A study of the methods of finishing bearing surfaces.

(b) A determination of the relation between the degrees of finish and minimum allowable film thickness.

(c) An analysis of bearing temperatures in internal combustion engines.

(d) The determination of the effects of variable loading.

(e) The continuance of the study of the effects of pressure on the viscosity of oils.

(f) Further correlation on the design of bearings, simplifying methods, formulas, and diagrams for the designer.

(g) The betterment of the present system of measuring viscosity.

Although this survey has been rather extensive, it can be continued. The obtaining of more accurate temperatures would be advantageous. Also, other bearing data can be gathered together by obtaining the sizes and weights of the parts and calculating the loads on the bearings, for in some cases the manufacturers have not calculated their loads, yet have this data. Some work has already been started toward the collection of bearing data on other types of equipment than that given in this thesis. This work is not complete today for two reasons:

inability of some manufacturers to furnish such data, and reluctance of other manufacturers to part with this information. It is believed that many more manufacturers will be able to furnish such information in a few years as rapid advances are being made by many in experimental investigation. Such information will be gladly given by the manufacturers if proved to them that it will not be misconstrued or misrepresented and presented in a clear, accurate and informative manner.

BIBLIOGRAPHY

1. Webster, Noah. New Universal Self-Pronouncing Dictionary of the English Language. Edited by W. J. Pelo, Philadelphia, John C. Winston Co., 1931, p. 105.

2. Ainsworth, Marcus, and Schwartz, Adolph F., "Today's Cars Are Longer Lived," Automotive Industries. February 25, 1939, Vol. 80, No. 80, pp. 201-202 and 275-276.

3. Hersey, M. D. Theory of Lubrication. New York, John Wiley & Sons, Inc., 1938.

4. Reynolds, O. "On the Theory of Lubrication and Its Application to Mr. Beauchamp Tower's Experiments," Philosophical Transactions of the Royal Society of London, Vol. 177, 1886, part 1, pp. 157-234.

5. Hillman, L. A. "Changes That Have Taken Place in Bearings," Industrial Engineer, March, 1925, Vol. 83, No. 3, pp. 127-130.

6. Parish, W. F. "Frictional Resistance to Motion," Mill and Factory, October, 1935, Vol. 17, No. 4, pp. 43-4 and 102.

7. Parish, W. F. "Guide to Modern Industrial Lubrication," Mill and Factory, March 1925, Vol. 16, No. 3, pp. 27-30 and 86-7.

8. Notes on Bearing Babbitts," American Machinery, February 9, 1928, Vol. 68, No. 6, p. 271

9. Duiguid, Jas. "Lubricants and Lubricating," Mechanical Engineering, November, 1925, Vol. 47, No. 11, pp. 887-894.

10. Petroff, N. "Friction in Machines and the Effect of the Lubricant," (a) In Russian: Engineering Journal, St. Petersb. 1883; No. 1, pp. 71-140; No. 2, pp. 228-279; No. 3, pp. 377-438; No. 4, pp. 535-564. (b) German translation by L. Wurzel, Hamburg; L. Voss, 1887.

11. Howarth, H. A. S. "Journal Bearing Lubrication," S. A. E. Journal, November, 1926, Vol. XIX. No. 5, pp. 453-460 and 466.

12. Lasche, O. Die Reibungsverhältnisse in Lagern mit höher Umfangsgeschwindigkeit ("Friction Relations in High-Speed Bearings"), (a) Zeit. V. D. I., Bd. 46, 1902, pp. 1881-1890. (b) English Translation: Traction and Transmission, Vol. 6, 1903, pp. 33-65.

13. Stribeck, R. Diewesentlichen Eiginschaftin der Gleit und Rollenlager ("Characteristics of Plain and Roller Bearings"), (a) Zeit. V. D. I., Bd. 46, 1902, pp. 1341-1348, 1432-1438, and 1463-1470. (b) Forschungsarbeit, 7, 1903.

14. Sommerfeld, A. "Zur hydrodynamischen Theorie der Schmiermittelreibung," Zeit. Math. Phys., Bd. 50, 1904, pp. 97-155.

15. Harrison, W. J. "The Hydrodynamical Theory of Lubrication with Special Reference to Air as a Lubricant," Trans. Camb. Phil. Soc., Vol. 22, 1913, pp. 39-54.

16. Michell, A. G. M. "The Lubrication of Plane Surfaces," _Zeit_. _Math_. _Phys_., Bd. 52, 1905, pp. 123-137.

17. "Theory of Lubrication," _Engineering_, Vol. 100, 1915, pp. 101-103, 154-155, 196-197, and 207-208.

18. Boswall, R. O. _The Theory of Film Lubrication_, London, Longmans, 1928.

19. Kobayashi, T. "A Development of Michell's Theory of Lubrication," Rep. 107, _Aero_. _Res. Inst._, Tokyo Imp. Univ., Vol. 8, No. 11, 1934, pp. 385-414.

20. Kingsbury, A. "On Problems in the Theory of Fluid-Film Lubrication, With an Experimental Method of Solution," _A_. _S_. _M_. _E_. _Transactions_, Vol. 53, 1931, paper APM-53-5, pp. 59-75.

21. Hersey, M. D. "On the Laws of Lubrication of Journal Bearings," _A_. _S_. _M_. _E_. _Transactions_, Vol. 37, 1915, pp. 167-202.

22. Hersey, M. D. "The Laws of Lubrication of Horizontal Journal Bearings," _J_. _Wash_. _Acad_. _Sci_., Vol. 4, 1914, pp. 542-552.

23. McKee, S. A., and McKee, T. R., "Journal-Bearing Friction in the Region of Thin-Film Lubrication," _S_. _A_. _E_. _Journal_, September, 1932, Vol. 31, No. 3, pp. 371-377.

24. McKee, S. A., and McKee, T. R., "Friction of Journal Bearings as Influenced by Clearance and Length," _A_. _S_. _M_. _E_. Transactions, May-August, 1929, Vol. 51, No. 16, paper APM-51-15, pp. 161-171.

25. Dickinson, H. C., and Bridgeman, O. C., "Fundamentals of Automotive Lubrication," S. A. E. Journal, Vol. 31, No. 1. 1932, pp. 278-282 and 304.

26. Wilson, R. E., and Barnard, 4th, D. P., Mechanism of Lubrication. Serial No. 47. Publication of Massachusetts Institute of Technology, 1922.

27. Leis, C. H. "Fundamental Requirements for Bearing Lubrication," Product Engineering, August, 1934, Vol. 5, No. 8, pp. 303-5.

28. Everett, H. A. "High-Pressure Viscosity as Explanation of Apparent Oiliness," S. A. E. Journal, November, 1937, Vol. 41, No. 5, pp. 531-537 and 537-544.

29. McKee, S. A. "The Effect of the Addition of Kerosene on Oiliness of Lubricating Oils," S. A. E. Journal, October, 1926, Vol. 19, No. 4, pp. 356-360.

30. McKee, S. A. "The Effect of Running-in on Journal-Bearing Performance," Mechanical Engineering, December, 1927, Vol. 49, No. 12, pp. 1335-1340.

31. Barnard, 4th, D. P. "A Possible Criterion for Bearing Temperature Stresses," S. A. E. Journal, May, 1932, Vol. 30, No. 5, pp. 192-197.

32. Howarth, H. A. S. A Graphical Analysis of Journal Bearing Lubrication. New York, The American Society of Mechanical Engineers, 1923, 1924, 1925.

33. "Lubrication, Eccentricity of Bearings," Engineering, July 30, 1937, Vol. 144, pp. 130-131.

34. Swift, H. W. "Journal and Thrust Bearings," Engineering, Vol. 144, 1937, pp. 436-437.

35. Moore, S. H. "Friction, Lubricants and Lubrication," Power, December, 1905.

36. Lewis, G. W. "Automobile Engine Bearing Pressures," American Machinery, November 9, 1911.

37. Power. January 4, 1916, Vol. 43, No. 1, pp. 32.

38. Illmer, Louis. "Bearing-Design Constants--I," Power, February 22, 1916, Vol. 43, No. 8, pp. 251-255.

39. Illmer, Louis. "Bearing-Design Constants--II," Power, February 29, 1916, Vol. 43, No. 9, pp. 280-282.

40. Dahlstrom, F. P. "The Morgoil Roll-Neck Bearing," A. S. M. E. Transactions, June 13, 1933, Vol. 55, No. 7, paper IS-55-2, pp. 9-18.

41. Pelly, J. F. "Steel Plant Lubrication," Mechanical Engineering, January, 1935, Vol. 57, No. 1, pp. 25-30.

42. Evans, R. G. N. "Standard Sleeve Bearings," Mill and Factory, March, 1937, Vol. 20, No. 3, pp. 54-58.

43. Bottle Oilers. New York, Socony-Vacuum Oil Co., Inc., 1931.

44. Clower, J. I. "Oiling Appliances and Systems," Mill and Factory, July, 1936, Vol. 19, No. 1, pp. 49-55, 120, and 124.

45. Wick Feed Oilers. New York, Socony-Vacuum Oil Co., Inc., 1936.

46. Karelitz, G. B. "The Lubrication of Waste-Packed Bearings," Mechanical Engineering, June, 1927, Vol. 49, No. 6, pp. 663-670.

47. Brown, J. L. "Selecting the Bearing to Suit the Job," Electrical Manufacturing, May, 1929.

48. Baudry, R., and Tichvinsky, L. M., "Performance of Oil Rings," Mechanical Engineering, February, 1937, Vol. 59, No. 2, pp. 89-92.

49. Gilson, E. G. "Some Bearing Investigations," General Electric Review, May, 1924, Vol. 27, No. 5, pp. 318-327.

50. Barnard, D. P. "Lubrication of Plain Bearings," S. A. E. Journal, May, 1926, Vol. 18, No. 5, pp. 483-485.

51. Grooving Oil-Lubricated Cylindrical Bearings, New York, Socony-Vacuum Oil Co., Inc., 1935.

52. Karlitz, G. B. "Grooving Bearings in Machines," A. S. M. E. Transactions, January-April, 1929, Vol. 51. No. 4, paper MSP-51-5-21, pp. 21-29.

53. Underwood, A. F. "Automotive Bearing Materials and Their Application," S. A. E. Journal, September, 1938, Vol. 43, No. 3, pp. 485-492.

54. Parish, W. F. "Testing Oil on Plant Equipments," Mill and Factory, November, 1935, Vol. 17, No. 5, pp. 48-49 and 107-109.

55. Ricardo, H. R. "Engine Lubrication-Internal Combustion Engines," Engineering, Vol. 144, 1937, pp. 467-468.

56. "1939 American Passenger Cars," Automotive Industries, February 25, 1939, Vol. 80, pp. 214.

57. Heldt, P. M. "PV Factor Is Not Proper Measure of Bearing Capacity," Automotive Industries, June 11, 1932, Vol. 66, No. 24, pp. 842-845.

58. Sparrow, S. W. "Oil for Winter Driving Should Be of No Lower Viscosity than that Used in Summer," Automotive Industries, August 5, 1938, Vol. 79, No. 6, pp. 174-176.

59. "Progress in Diesel Accessories," Diesel Power and Transportation, May, 1938, Vol. 16, No. 5, pp. 456-460.

60. "A Non-Corrosive, Compounded Diesel Lube Oil Is Here," Power Plant Engineering, April, 1939, Vol. XLIII, No. 4, pp. 92.

61. Neely, G. L. "Recent Developments in Diesel Lubricating Oils," S. A. E. Journal, November, 1939, Vol. 45, No. 5, pp. 485-500.

62. Heldt, P. M. "Bearing Materials," Automotive Industries, March 19, 1938, Vol. 78, No. 12, pp. 412-422.

63. McDonald, A. T. "Some Developments Relative to Crankcase-Oil Filtration," S. A. E. Journal, January, 1939, Vol. 44, No. 1, pp. 23-28.

64. Needs, S. J. "Influence of Pressure on Film Viscosity in Heavily Loaded Bearings," A. S. M. E. Transactions, May, 1938, Vol. 60, No. 4, paper RP-60-7, pp. 347-358.

65. Vigne, Albert, and Cox, E. I., "An Oil-Bath-Lubricated Railway Bearing," A. S. M. E. Transactions, August, 1938, Vol. 60, No. 6, paper RR-60-3, pp. 499-506.

66. Hunter, B. F. "Railway Lubricants," Mechanical Engineering, April, 1937, Vol. 59, No. 4, pp. 235-238.

67. Pearce, E. S. "Locomotive and Car Journal Lubrication," A. S. M. E. Transactions, January, 1936, Vol. 58, No. 1, paper RR-58-1, pp. 37-45.

68. "Locomotive Driving Journals Are Oil Lubricated," Railway Age, June 10, 1939, Vol. 106, No. 23, pp. 990-991.

69. "Disc-Flo Journal Bearing for High-Speed Service," Railway Age, January 30, 1937, Vol. 102, No. 5, pp. 229-230.

70. Schein, A. E. "Overcoming Bearing Deficiencies," Product Engineering, January, 1933, Vol. 4, No. 1, pp. 2-4.

71. Bradford, L. J. "Effect of Fitting on Plain Bearing Performance," Product Engineering, April, 1932, Vol. 3, No. 4, pp. 150-152.

72. "Oil Film Bearing," Power Plant Engineering, August, 1937, Vol XLI, No. 8, pp. 504.

73. Brazier, S. A., and Bowyer, W. H., "Rubber as a Material for Bearings," Mechanical Engineering, April, 1938, Vol. 60, No. 4, pp. 336.

74. Haushalter, F. L., and Moffitt, L. Q., "Water Lubricated Rubber Bearings," Product Engineering, March, 1935, Vol. 6, No. 3, pp. 103-105.

75. Rosebloom, Julius. Diesel Reference Guide, Section II, Jersey City, Industrial Institute, Inc., 1931.

76. Thomas' Register of American Manufacturers. New York, Thomas Publishing Co., 1937.

77. Sweet's Catalog File. New York, Division F. W. Dodge Corporation, 1938.

78. Needs, S. J. "Effects of Side Leakage in 120-Degree Centrally Supported Journal Bearings," A. S. M. E. Transactions, Vol. 56, 1934, paper APM-56-16, pp. 721-732.

79. Howarth, H. A. S. "Current Practice in Pressures, Speeds, Clearances, and Lubrication of Oil-Film Bearings," A. S. M. E. Transactions, Vol. 56, 1934, paper MSP-56-2, pp. 891-902.

80. Howarth, H. A. S. "The Loading and Friction of Thrust and Journal Bearings," A. S. M. E. Transactions, Vol. 57, 1935, pp. 169-187.