A STUDY OF GREASE LUBRICATED BEARINGS

by

FRANCIS JOSEPH SULLIVAN

B. S., Harvard University, 1936

A THESIS

submitted in partial fulfillment of the

requirements for the degree of

MASTER OF SCIENCE

Department of Machine Design

KANSAS STATE COLLEGE

OF AGRICULTURE AND APPLIED SCIENCE

1941

TABLE OF CONTENTS

INTRODUCTION

This study was undertaken with the purpose of determining some of the operating characteristics of grease lubricated bearings. The literature in the field of lubrication contains little information on their performance. The chief interest seems to have been centered on the technical problems of manufacture and the problems of application such as churning and channelling. The analysis of plastic flow and the study of the effect of different kinds of greases on a test bearing would give a more complete understanding of the problem of grease lubrication. With this object in view the problem was begun.

The purpose of any lubricant is to reduce the friction between a moving surface and the bearing surface. There is a wide variety of available lubricants. Kingsbury's first demonstration was with an air lubricated thrust bearing (Hersey, 1936). Some water lubricated bearings are in use. These cases are rare as oils and greases are used nearly universally as lubricants.

GREASE CHARACTERISTICS

A grease is a mixture of a soap and an oil. Generally a mineral oil is used, although vegetable and animal oils are not uncommon. The type of a grease is determined by its soap base. Lime-soap grease, or cup greases, are buttery in texture and are water repellant. Soda-soap greases have a stringy texture and will emulsify in water. The above are the most widely used types. In addition there are various metallic-soap greases that are used for special applications.

No standard commercial classification of greases exists in this country such as there is for oils. Oils are most generally classified by their S. A. E. Number, which is a number arbitrarily given to all oils that fall within a certain range of Saybolt viscosity.

Each manufacturer of greases has his own method specifying the grade of his product. Some greases are number: 0, 1, 2, etc.; others are labelled very soft, soft, medium soft, and so on.

The A.S.T.M. penetration number is the standard test for a measurement of the hardness of a grease. The number is found by measuring the depth to which a weighted point penetrates a sample of the grease. The hardness of a grease varies with the amount of working that it receives. For this reason, the hardness is measured after the grease has been mixed a specified amount.

The dropping point is also a measure of the hardness of a grease. By observing the temperature at which a sample of the grease becomes liquefied and drops from a container, this rating is determined.

REVIEW OF LITERATURE

The modern theory of film lubrication begins with the experiments of Tower in England. Under the sponsorship of the Institute of Mechanical Engineers, Tower experimentally discovered the variation in oil pressure over the surface of a partial bearing (Archbutt and Deeley, 1927).

Reynolds later recognized the hydrodynamic theory which underlay this development, and in 1886 presented the paper which contained the general theory of film lubrication and its application to the particular experiment of Tower's. The best summary of this theory is by Boswall (1928).

This paper of Reynolds' is rather famous for its difficulty. In fact, the greater part of the literature in the field is remarkable for its mathematical abstruseness. The remainder consists largely of rule-of-thumb formulae for bearing design and application. As a result, the amount of information available concerning engineering practice is quite meagre.

Barnard (1938) has reported on the work of Arveson in measuring the apparent viscosity of several greases. This report does not consider the grease as a plastic solid. The viscosity is tabulated as a function of the rate of shear. Since the theory of film lubrication considers the viscosity as a constant in the bearing, these results are probably not directly applicable to bearing design.

THEORY

The theory of film lubrication is based on Newton's Law of Viscous Flow; $du/dy = s/\mu$ *. The velocity gradient across a section of fluid is proportional to the shearing stress. This necessarily assumes streamline flow in the fluid.

One of the more serious difficulties in the mathematical derivation of the theory is the fact that the constant of proportionality, the viscosity μ, is not constant, but varies with both the temperature and the pressure. The change of viscosity with pressure is small. For mineral oils, the per cent increase in viscosity is equal to 5×10^{-6} times the pressure squared (Marks, 1930). Inasmuch as this variation is insignificant when the moderate pressures encountered in bearings are considered, it has frequently been neglected.

The change of viscosity with temperature is considerable and should be considered. There is no general rule applying to all fluids, although it is usually a logarithmic relationship. In the case of lubricating oils, an approximate formula is $\log \mu = \log a - b \log (t - c)$; where a and b are experimental constants, c is a reference temperature, and t is the temperature of the oil. The American Society for Testing Materials has prepared a co-ordinate chart where the viscosity-temperature function of an oil plots as a straight line. This differs slightly from regular log-log co-ordinates.

---

* See Appendix for symbols.

It is impossible to determine exactly just what the temperature will be inside a bearing. The temperature depends upon the friction, the specific heat of the oil, and the heat conductivity of the bearing. Boswall (1928) treated the case where the temperature varies uniformly in the direction of motion. Most of the time, however, the temperature change is neglected, and the viscosity through the bearing is assumed some arbitrary value.

<u>The flow of a plastic</u>. A lubricant operating under conditions of purely plastic flow is defined by Houwink (1937) as possessing the properties of an elastic solid up to a shearing stress equal to its yield point, and with greater stress flowing with viscous flow. In other words, the velocity gradient across any section is zero up to s_o, from then on it is proportional to $s - s_o$.

Consider a plastic solid flowing through a pipe of radius R. Acting on a cylindrical element of the plastic of radius r, and length l, considered as a free body, will be a pressure differential and a shearing stress (Fig. 1).

For equilibrium,

$$\Delta p \times \pi r^2 = 2 \pi r \, l \, s$$

$$s = \Delta p \, r/2 \, l \qquad (1)$$

The flow follows Newton's law which takes the form,

$$\frac{d\,u}{d\,r} = \frac{s - s_o}{\eta} \qquad (2)$$

From (1) it will be observed that the shearing stress decreases towards the center of the pipe. When the stress reaches

EXPLANATION OF PLATE I

Fig. 1. Free body diagram of plastic flowing in pipe.

Fig. 2. Tubular section of plastic flowing in pipe.

Fig. 3. Arrangement of apparatus. T_1 and T_2 are belt tensions and R_1 and R_2 are reactions from scales.

PLATE I

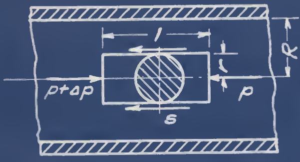


Fig. 1

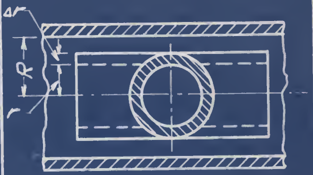


Fig. 2

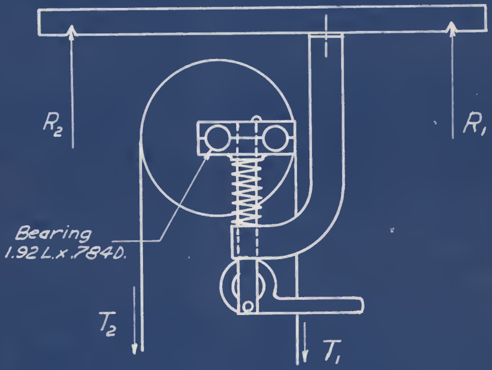


Fig. 3

s_o, the yield point of the plastic, the velocity gradient (2), du/dr, will become zero. This indicates that at the center of the pipe there is flowing with uniform velocity a tubular section of the plastic, whose radius r_o equals $2ls_o/p$. By integrating the above expression (2), the velocity of this section, u_o, will be found to be equal to

$$\frac{\Delta p}{2\eta l}(R - r_o)^2 \qquad (3)$$

These expressions can be used to determine the yield point of the plastic. The velocity at the wall of the pipe is zero. If the pressure differential is adjusted so that flow through the pipe is just about to begin, the shearing stress at the boundary will be equal to s_o, r_o will equal R, and the yield point, s_o, will be equal to $\Delta pR/2l$.

Further integration of (2) leads to the formula for the velocity at any point in the cross-section. It was found that

$$u = \frac{\Delta p}{4\eta l}(R^2 - r^2) - \frac{S_o}{\eta}(R - r) \qquad (4)$$

Consider a hollow cylindrical element of the plastic, of radius r, and thickness dr (Fig. 2). The cross-sectional area of this element will be $2\pi r\,dr$. Let Q equal the quantity flow, the volume of plastic flowing through the pipe in time t.

The flow through the section will be equal to

$$dQ = u\,2\pi r\,dr$$

Integrating from 0 to r_o with u equal to u_o of (3), and from r_o to R with the velocity of (4), leads to the expression for Q.

$$Q = \frac{\pi\ \Delta P}{2\eta l}\left(\frac{R^4}{4} - \frac{R^3 r_o}{3} - \frac{R^2 r_o^2}{2} - R r_o^3 - \frac{35}{24} r_o^4\right) \qquad (5)$$

Transposing this gives the formula for the viscosity of the plastic,

$$\eta = \frac{\pi\ \Delta P}{48Ql}\left(6R^4 - 8R^3 r_o + 12R^2 r_o^2 - 24R r_o^3 + 35 r_o^4\right) \qquad (6)$$

By using the above formulae in conjunction with experimental data, the two important properties of a plastic lubricant, the viscosity and the yield point, may be found.

METHOD OF TESTS

The general arrangement for the tests is shown in Fig. 3. The drive was from an induction motor rated one-third horse power at 1725 rpm. The load was only a small fraction of this rating, so that the motor speed for all the tests was 1775 rpm. A four-step pulley was on the motor shaft giving the bearing shaft speeds of 1522, 1127, 773, and 394 rpm.

The bearing was supported by its frame on two balance scales. The scales were divided to a quarter of an ounce, and it was possible to read to one-tenth of a division with a fair amount of accuracy. By means of a force analysis of the system, a formula for the effective belt pull was found:

$$T_2 - T_1 = 1.876\ R_2 - 3.09\ R_1 + 3.39.$$

Since the diameters of the pulley and of the shaft were known, the friction force at the shaft was calculated. Dividing this by the total load on the shaft gave the value of the coefficient of friction.

The bearing was unusual in its design. It consisted of two 180° partial bearings, held together by a spring. The load was applied by the two halves of the bearing pressing on the shaft. This differs from actual practice in which the load is applied to the shaft from the bearing or the shaft is supported by the bearing.

The greases tested are shown in Table 1. In addition, a test was run on a sample of 20W motor oil having Saybolt viscosities of 178" at 130° F. and 59" at 210° F.

It will be noted that the greases fell into four groups. One group was made up of lime base greases of different penetrations containing oil of the same viscosity. The second group contained a similar series of soda base greases. The third and fourth groups contained lime and soda base greases respectively; here the penetration was held fixed and the viscosity of the oil was varied. By means of these groupings all but one variable was eliminated.

The object of using these groups of greases was to provide a method of determining the influence on the performance of hardness and of viscosity of oil in the grease. The shaft was run at four speeds so that the variation of friction torque at different speeds might be found. The particular curve of coefficient of friction plotted against revolutions per minute was only of secondary interest.

The properties of a lubricant are very markedly influenced by temperature variations. The Saybolt viscosity of an oil varies inversely with the temperature. The hardness of a grease may be measured by its dropping point, the temperature

Table 1. Showing greases tested.

Group	Code number	Type of soap	A.S.T.M. penetration at 77°F. (worked)	Viscosity of oil
1	9060	Lime	350	105" at 100°F.
1	9061	"	300	"
1	9062	"	250	"
1	9063	"	200	"
2	9130	Soda	370 Max.	200" at 100°F.
2	9131	"	320 Max.	"
2	9132	"	270 Max.	"
2	9133	"	220 Max.	"
3	9061	Lime	300	105" at 100°F.
3	9022	"	300	850" at 100°F.
3	9297	"	290-315	300" at 100°F.
4	9131	Soda	320	200" at 100°F.
4	9222	"	315-345	300" at 100°F.
4	9206	"	270-300	105" at 210°F.

at which the grease liquefies. This temperature falls within a moderate range, since nearly all greases become liquid at less than four hundred degrees F.

An increase in temperature will therefore cause the grease to become more fluid. It would then appear that variation in temperature would be followed by changes in the friction as measured in the test.

For comparable results, the tests should be run at controlled temperatures. The technique used in the experiment endeavored to simulate operating conditions in the bearing. The bearing was first supplied with a more than sufficient quantity of the particular grease to be tested and run for about twenty minutes at the maximum speed of 1522 rpm. At this point the friction torque as indicated by the reactions on the balances would be stable, and the bearing would have heated to its highest temperature. After this, the readings were taken. Before reading at the next lower speed, the bearing was relubricated and run long enough to assure equilibrium.

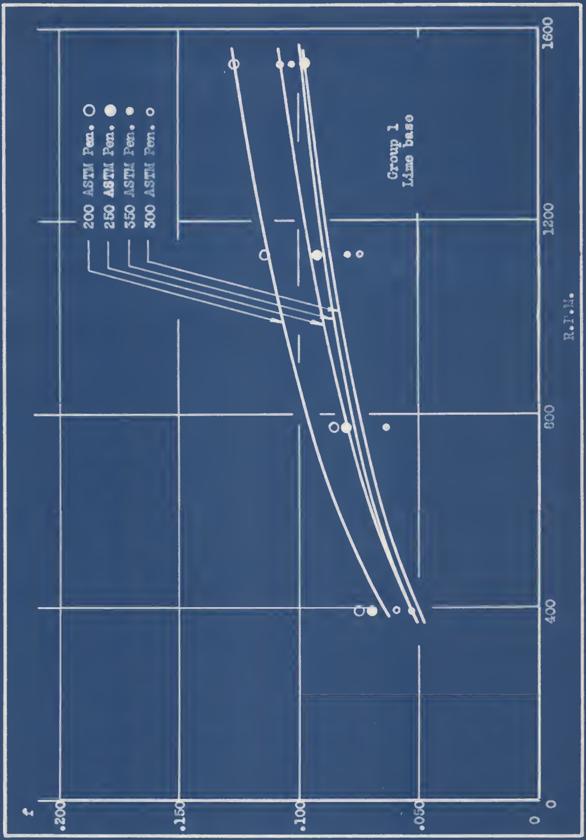
Since all the greases were different in composition and texture, conditions in the bearing would necessarily vary. The harder soda base greases, for instance, have greater internal friction losses than the soft lime base greases, and will therefore cause the bearing temperature to rise higher due to the greater amount of work lost in the film.

This apparent lack of standard for the various tests was not a serious problem. The purpose of the experiment was to find the operating characteristics of these greases, and to study the effect of penetration and viscosity on their performance. The bearing was operating under a fixed load and at room temperature, which was fairly stationary. Any differences in the bearing temperature were therefore a function of the grease sample under test. This, by the way, more closely approached actual conditions than would a laboratory test where the variables are closely restricted.

EXPLANATION OF PLATE II

Curves of coefficient of friction plotted against R.P.M. for lime base greases containing oil of 105 sec. viscosity at 100° F.

PLATE II

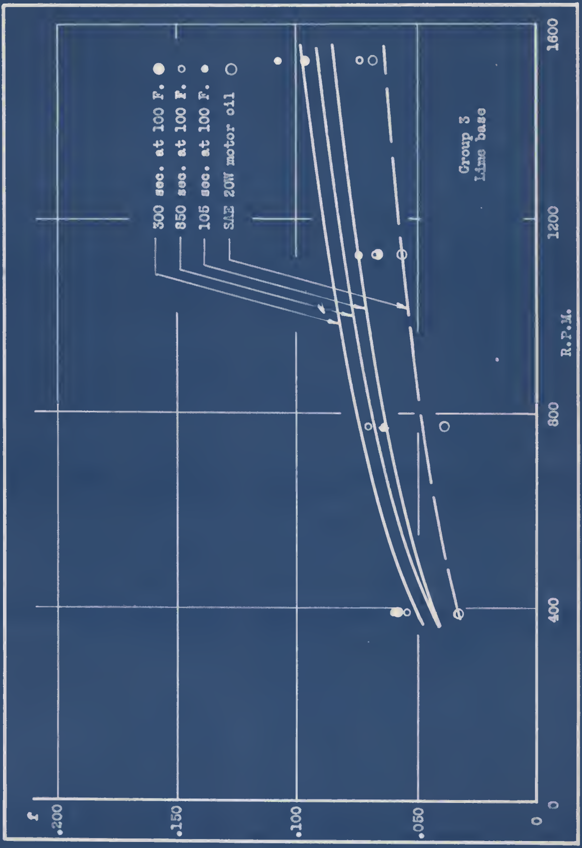


200 ASTM Pen.
250 ASTM Pen.
350 ASTM Pen.
300 ASTM Pen.
Group 1
Lime base
f
.200
.150
.100
.050
0
0
400
800
1200
1600
R.P.M.

EXPLANATION OF PLATE III

Curves of coefficient of friction plotted against R.P.M. for lime base greases of 300 A.S.T.M. penetration.

PLATE III

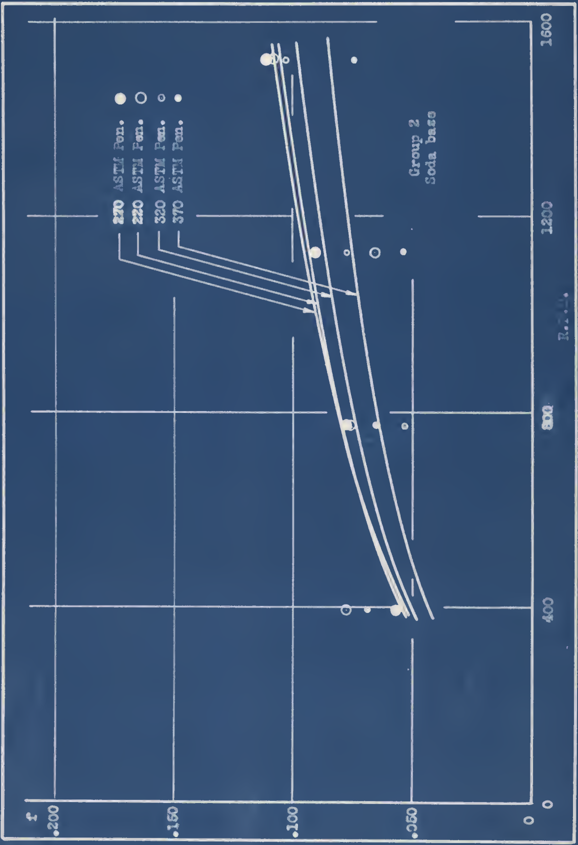


300 sec. at 100 F.
850 sec. at 100 F.
105 sec. at 100 F.
SAE 20W motor oil
Group 3
Lime base
f
.200
.150
.100
.050
0
0
400
800
1200
1600
R.P.M.

EXPLANATION OF PLATE IV

Curves of coefficient of friction plotted against R.P.M. for soda base greases containing oil of 200 sec. viscosity at 100° F.

PLATE IV

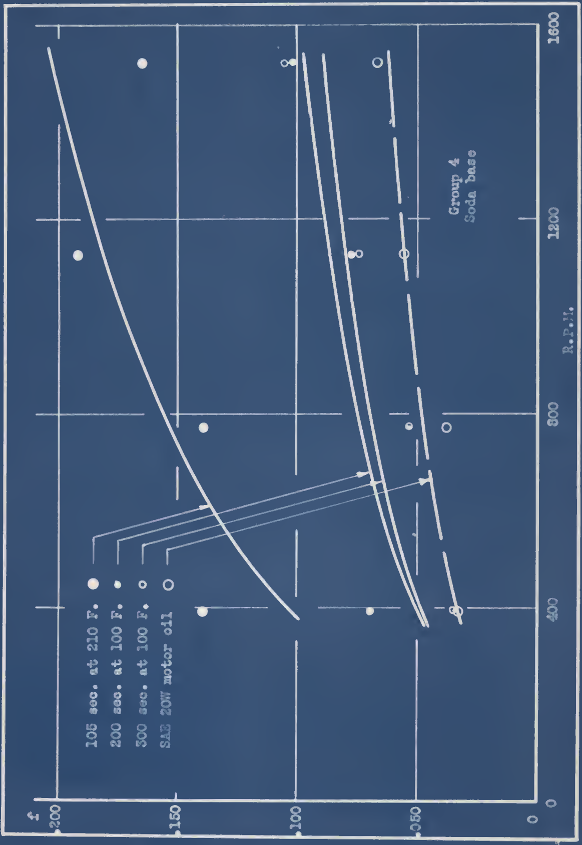


270 ASTM Pen.
220 ASTM Pen.
320 ASTM Pen.
370 ASTM Pen.
Group 2
Soda base
f
.200
.150
.100
.050
0
0
400
800
1200
1600
R.P.M.

EXPLANATION OF PLATE V

Curves of coefficient of friction plotted against R.P.M. for soda base greases of 320 A.S.T.M. penetration (approx.).

PLATE V


105 sec. at 210 F.
200 sec. at 100 F.
300 sec. at 100 F.
SAE 20W motor oil
Group 4
Soda base
f
.200
.150
.100
.050
0
0
400
800
1200
1600
R.P.M.

RESULTS

The results of the tests are shown on Plates II, III, IV, and V. There are four sets of curves corresponding to the four groups of greases that were tested. The curves were drawn by using the method of least squares on semi-logarithmic co-ordinates.

It will be noted that the values of the coefficient of friction as determined from the experiments did not fall upon a smooth curve. For that reason, the method of least squares was used to find the best expression for the relationship between the coefficient of friction and the speed of the journal.

The first attempt at curve fitting was done by using log-log co-ordinates. The expressions that were calculated were found to diverge considerably from the given values of the coefficient of friction at the maximum speed of 1522 rpm, the values from the curve being greater than those observed. Using a semi-logarithmic plot the curves were found to fit the data fairly closely. The relationship between the speed and the coefficient of friction took the form $N = a^{bf}$, where a and b are constants that differ with the different greases.

When a bearing is lubricated by an oil of constant viscosity at various speeds, or when the coefficient of friction is plotted against the value of ZN/P, the plot of the coefficient of friction is a straight line and its value is proportional to the abscissa except near the origin where boundary friction occurs. If it had been possible to give the bearing a continuous and constant supply of lubricant, the plot of the coefficient of friction would have been closer to a straight line. In the test, the bearing was supplied with aninitial quantity of lubricant and run until a steady state was reached. At the higher speeds, the friction loss was higher, the temperature in the bearing was higher, and the viscosity of the grease was therefore less. Since the friction force is a function of the viscosity, the coefficient of friction at the higher speeds was reduced from a straight line value.

The curves of Plates II and IV show the effect of the change of the hardness of the grease on the coefficient of friction. The coefficient of friction increased slightly with the harder greases, but not as much as might be expected. It was noticed during the tests that the bearing tended to run hotter with the harder greases, thus reducing the viscosity and lowering the coefficient of friction.

The two softest lime base greases have curves that lie close together. These greases were both quite soft and oily. Apparently, in this case the hardness of the grease did not have appreciable effect on the coefficient of friction.

It will also be noted that the two hardest soda base greases have coefficients of friction that are similar. This is doubtless due to the structure of the greases. As noted, soda base greases have a stringy and adhesive texture, and this property becomes more pronounced in the harder greases. For that reason, the internal friction in these greases was greater and caused heating, followed by an increase in their fluidity with a lowering of the friction. The bearing temperature ran higher for the soda base greases and was quite hot for the hard ones.

In studying the effect of varying viscosities on the coefficient of friction, the curves of Plates III and V did not follow expectations. The 300-second lime base grease has the greatest friction in its group, and the 300-second soda base grease has the least friction in Group 4. The viscosity at standard test temperature of the oil used in the grease does not seem to give a satisfactory indication of the friction.

The most viscous soda base grease had coefficients of friction which were about twice as great as the others. This was a dark sticky grease, probably designed for very severe conditions. It caused the bearing to run at a high temperature. The fact that this grease was adhesive would make its shearing strength high and so affect its lubricating properties.

It is quite generally accepted that at high rates of shear, the apparent viscosity of a grease approaches that of the oil which is contained in it. Barnard (1938) showed curves that indicate that this is a correct statement. It would be necessary to conclude that the viscosity of the grease varies widely as the rate of shear is changed.

As mentioned above, there is not any reference in the literature to grease considered as a plastic solid. An analysis of a grease by this method may indicate that its viscosity is different from although related to the oil contained therein. It does not appear reasonable to assume that a number of greases, all made with oil of a given viscosity but of different compositions, proportions, and bases, should have the same viscosity.

CONCLUSIONS

The study of grease lubrication has not been given much attention as regards the mechanics of the behavior of the lubricant. The analysis of the flow of a plastic will enable the measurement of the important properties of the yield point and viscosity.

The general effect of increasing hardness is to cause greater friction in the bearing. The softest lime base greases had equivalent coefficients of friction, which indicated that there is no advantage to applying a very soft lime grease. The fibrous structure of soda base greases caused considerable heating in the bearing for the harder greases and greater fluidity in the grease.

For reducing the friction in a bearing to a minimum oil is superior to grease. The use of a grease lubricant is governed by the application. For instance, in food machinery, the lubricant must not leak into the product; or in complex machinery, a continuous and sufficient supply of oil is impossible to provide. However, the purpose of the grease is still to reduce friction.

It does not appear possible from the above data to generalize as to what is the best grease. The consideration of the factors of penetration and viscosity in a given case will supply a more complete understanding of the problem and a more judicious choice of lubricant.

ACKNOWLEDGMENTS

Indebtedness is acknowledged to Professor C. E. Pearce for his assistance during the experiments and in the preparation of this thesis, and to the Socony-Vacuum Oil Company for furnishing the classified samples of grease used in the tests.

BIBLIOGRAPHY

Archbutt, L. and Deeley, R. M.
Lubrication and bearings. London. Griffin. 650 p. 1927.

Barnard, D. P.
The flow characteristics of petroleum lubricants. In proceedings of the general discussion on lubrication and lubricants, 13th-15th October, 1937. New York. Amer. Soc. Mechan. Eng'rs. Vol 2. 507 p. 1938.

Boswall, R. O.
The theory of film lubrication. London. Longmans Green. 219 p. 1928.

Hersey, M. D.
Theory of lubrication. New York. John Wiley and Sons. 152 p. 1936.

Houwink, R.
Elasticity, plasticity and structure of matter. Cambridge. Cambridge Univ. 376 p. 1937.

Marks, L. S.
Mechanical engineers handbook. New York. McGraw-Hill. 2264 p. 1930.

APPENDIX

List of Symbols

f - Coefficient of friction

l - Length

p - Fluid pressure

P - Bearing pressure

r, R - Radius

s - Shear stress

t - Temperature

u - Fluid velocity

y - Displacement across fluid film

Z - Viscosity, centipoises

μ - Viscosity of oil

η - Viscosity of grease