RELATIONSHIP OF VISCOSITY, SURFACE TENSIONS, AND COEFFICIENT OF FRICTION OF LUBRICATING OILS.

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By

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P R E F A C E

The purpose of this treatise is a study of some of the physical properties of lubricants, but especially of the properties which bear directly upon the problem of lubrication. A relationship is shown to exist between the friction coefficient and the viscosity constant, also a relationship is set up between the viscosity constant and the surface tension factor, but no relation has been established between the friction coefficient and the surface tension factor. This is what is needed most at the present time, but before such a relation can be established, there are some other problems to be solved, which have been pointed out in this treatise.

The author has not attempted to make any experimental research on the coefficient of friction, but has quoted freely from Archbutt and Deeley, Beauchamp Tower, and others who have spent several years in the study of the coefficient of friction. The research work which the author has carried on most vigorously has been in the study of the properties, viscosity, and surface tension, and it is hoped that enough has been accomplished to show what the real problems are at the present time along such lines.

The author is particularly indebted to Prof.

W.M. Sawdon, of Cornell University, Prof. F.R. Watson, of the University of Illinois, Prof. A.E. Flowers, of the Ohio State University, Dean P.F. Walker, of the Engineering School of Kansas University, and Prof. A.H. Sluss, of Kansas University, for the many valuable suggestions received from them.

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I N T R O D U C T I O N

The problem of proper lubrication has been and still is one of the most perplexing problems that the engineer has had to deal with in his line. This is not from the fact that experimental investigation has been neglected; as a matter of fact for the last 226 years, a vast amount of experimental data has been collected, but the results not only failed to agree with each other, but they also failed to agree with the general experience of engineers concerning the frictional resistance of machinery.

The full benefit derived from the use of a lubricant was, until a couple of decades ago, seldom taken practical advantage of, as certain adverse conditions almost invariably accompany perfect lubrication. Prof. J. Goodman in 1890 in a paper read before the Massachusetts Association of Engineers stated that, out of every ton of fuel consumed for engine purposes, some 400 to 800 lbs. are wasted in overcoming friction of the working parts of engine, or generator, and a motor also wastes a large percentage of power by its own friction. One would not be far short in saying that from forty to eighty per cent of the

fuel is consumed in overcoming friction. This extremely wasteful state of affairs is greatly improved by a due observance of the laws of friction and lubrication.

CHAPTER I

KINDS OF FRICTION

Nature of friction:-- Friction is not merely a resistance to the relative motions of solid surfaces. The changes of a shape undergone by solids when under stress, as well as the movements of liquids, are opposed by internal friction, which, in such cases, is friction of quite a different nature and obeys laws quite different from that of friction between opposed surfaces.

Friction is divided into two main classes as follows: The friction of quiescence, and the friction of motion. The latter can be further sub-divided into rolling friction, sliding or gliding friction and revolving friction.

Rolling friction is that due to the resistance offered by the circumference of a wheel to the power by which it is propelled, and is met with in roller bearings.

Sliding or gliding friction, is that of ordinary journals or bearings, and is caused by one body rubbing upon another, and it causes the loss of force or motion in amounts varying in accordance with the different natures of the surfaces in contact, and

with the pressure of the one body upon the other.

The ordinary accepted theories of solid and of liquid are as follows: In the first, friction increases in direct proportion to the load, is independent of the extent of surfaces, and diminishes with an increase of velocity beyond a certain limit. In the second, the friction is independent of the pressure per unit of surface, and increases as the square of the velocity.

To give some idea of the vast amount of time that men of science have spent in investigating the subject of friction, the following brief summary of the results of the experiments carried out by the principal investigators will be given as follows:

Amontous, 1669, concluded that friction was not augmented by an increase of surface, but only by an addition of pressure upon the surfaces in contact, that it was the same for woods and metals when lubricants were employed, and that it increased or diminished with velocity.

De Vince, 1784, was of the opinion that with varying surfaces the smallest gave the least friction, the latter being greatly increased by cohesion; that the friction of hard bodies in motion

was an uniformily retarding force, whilst that of soft bodies was productive of an increase of retardation with an increase of velocity; and that friction was about one-fourth the pressure.

George Reunie, 1829, considered the friction of metals to vary in proportion to their hardness, the harder developing less than those of a softer nature.

General Morin, 1838, concluded that the friction of woods and metals rubbing upon each other was practically the same, and that the friction during motion was proportional to the pressure and independent of the area of the surfaces in contact and of the velocity of motion.

Professor Kimball decided that at slow speeds the friction between pieces of pine wood decreased rapidly as the speed increased.

Professor Jenkins, 1877, found that at very slow speeds, with a very marked difference between the static friction or the friction of rest, and the dynamic friction or the friction of motion, the coefficient of friction decreased gradually as the speed increased from .012 to 0.6 feet per minute. When the difference between the coefficient of static friction and dynamic friction was very slight, no difference was observed.

Captain Douglas Galton, 1878-79, ascertained the existence of a temporary decrease in the coefficient of friction with an increase of speed of from 400 to 5,300 feet per minute in the frictional resistances between brake blocks and wheels.

Prof. Thurston, 1878, was led to conclude that the coefficient of friction first decreases and after a certain point increases with the speed, the point of change varying with the pressure and with the temperature.

The Reports of Experiments, conducted by Mr. Tower, 1883-1891, demonstrated that in an oil bath the absolute friction, that is the actual tangential force per square inch of bearing required to resist the tendency was nearly constant under all loads within ordinary working limits, and did not increase in direct proportion to the load, as it should do according to the ordinary theory of solid friction, but approximated more to the laws governing liquid friction, inasmuch as it was nearly independent of the pressure per square inch, and increased with the velocity, though at a rate not nearly sorapid as the square of the velocity. A very considerable diminution in the friction was found to take place as the temperature arose. The friction

seemed to be dependent upon the quantity and uniformity of distribution of the oil, the amount of which might be very small, giving a coefficient of one one hundredth, any diminution and increase of friction much beyond this point, however, producing imminent risk of heating and seizing. The oil bath was found to give the most perfect lubrication, and the limit beyond which friction could not be reduced. With speeds of from 100 to 200 feet per minute the coefficient of friction could be reduced as low as one one thousandth by properly proportioning the bearing surface to the load.

Thus the widely taught law that friction is independent of the extent of surface in contact, but varies only with the pressure, is about ready to be placed among the archives of ancient scientists. The pressure inferred in the relationship is that exerted over the whole surface, and not per square inch, - that is, a surface one square foot in area under a pressure of one pound per square inch would require the same force to move it over a resisting surface as it would if made one square inch in area under a pressure of 144 pounds per square inch.

The extent of surface in contact was sup-

posed to have no effect upon the force or work of friction necessary to move one body upon another, and consequently required no increased effort to produce motion, provided the same total pressure was exerted although the area of the surfaces in contact might be at variance.

Some very recent experiments made with various grades of lubricants, to determine the coefficient of friction of lubricated surfaces under varying conditions have proved conclusively that the amount of surface in contact materially influences the work of friction. If the relationship of the "resistance of friction as independent of the area of surfaces in contact, but dependent upon the pressure" were true, the temptation would be to reduce the work of friction and the abrasion of the materials by increasing the area of the surfaces in contact, which would allow the use of a lighter oil by reducing the pressure per square inch, without increasing the abrasion. Practical demonstration, however, has proven the necessity of avoiding long journals and with the friction of rotation, an increase in the diameter of the journal means a corresponding increase in the work of friction.

It has been established quite conclusively

that when the rubbing surfaces are kept well separated by the lubricant, and the friction is more dependent upon the nature and fluidity of the lubricant than upon the nature of the solids carrying the load.

There seems to be a combined friction consisting of the particles forming the lubricant and of the moving surface in contact with it, with constant pressure and temperature, it is dependent upon the extent of surface in contact and varies directly with it. It is also influenced by unit pressure and varies in the same ratio as has been previously taken for granted.

As the resistance of lubricated surfaces is made up of the resistance of the particles of the lubricant, it is evident that any influence that will change the fluidity or density of the lubricant will also affect the frictional resistance.

Increase of temperature, increasing the fluidity causes a decrease in the coefficient of friction; while increase in unit pressure causes an increase in the density of the fluid, and, necessarily an increase in the friction when motion is produced. The ideal condition of lubrication is attained when the viscosity of the lubricant at the working temperature is sufficient, and no more, to keep the surfaces of the solids apart

under the maximum pressure they may have to sustain.

Coefficient of Friction:-- With the exception of the method of lubrication there is no other element in connection with the subject under consideration that has received more attention than that of the coefficient of friction, and yet there is no other element that is in as crude and indeterminable a state. As the investigators progress, the subject seems surrounded with more and more variables of a complicated nature, which indicated the importance, if not the necessity, of the utmost care when the best results from lubrication are desired.

As long as the metals are prevented by the lubricant from coming in contact, it is found that the friction is dependent upon the fluidity of the lubricant, and varies with changes of the fluid condition, decreasing with a higher temperature and increasing with a less degree of heat.

It was found conclusively by some experiments carried on by Woodbury and Tower, the first under light pressures, and the latter's under heavy pressures, that uniform results could not be looked for unless constant temperature, velocity, pressure, area of surface, in contact, and thickness of the film of the oil between the sur-

faces were maintained. The results indicating clearly that the resistance of friction was dependent upon and changed with a variation in any of the above conditions.

The results obtained by Woodbury are the most accurate of any ever published and probably ever made, there still lacks sufficient uniformity for the derivation of friction with changes in temperature and pressure.

The variation in the coefficient of friction with changes of temperature can readily be carried to an extreme, as it has been found that while the resistance decreases as the temperature is raised, there is a point, depending upon the unit pressure and viscosity of the lubricant where the coefficient starts to increase very rapidly with increase of temperature. The same holds true with a variation in pressure while these laws are true as stated in a general way they depend and a are limited by the viscosity of the lubricant used.

An attempt has been made to prove a positive relationship between the viscosity of an oil and its coefficient of friction, they are no doubt, more or less dependent, there is ~~hardly~~ sufficient data at hand to resolve this to a definite basis. In addition to that of viscosity, lubricants possess a property designated as unctousness which seems to influence the coeffic-

ient of friction as much, if not more than the viscosity. It appears and there seems sufficient information at hand to anticipate it, that a relationship of a positive and determinable nature between the three elements, coefficient of friction, viscosity, and unction is obtainable,

The general conclusions to be arrived at are: That the friction of a shaft or axle depends upon the amount of load upon the bearings, the finish, the truth of allignment, the nature of the lubricants and the efficiency of the lubricant. In ordinary engine bearings the coefficient of friction is about .002, under exceptionally favorable circumstances this may be reduced as low as .001. When cast iron bearings are used the pressure should not exceed 100 lbs. per square inch and the speed 150 feet per minute.

Definition for Coefficient of Friction: The coefficient of friction is that resistance due to friction which is caused by a pressure or load of one pound, hence to find the amount of friction between any two surfaces the load must be multiplied by the coefficient.

CHAPTER TWO

THEORY OF LUBRICATION

A simple definition of lubrication would be the act of lubricating, which means to make smooth or slippery. Lubricants are, with few exceptions, fluid or semi-fluid substances, which have the property of forming and maintaining between friction surfaces films of sufficient thickness to keep the surfaces apart, thus doing away with solid friction and substituting for it much less considerable friction of the liquid itself.

The compounds which have this property in the highest degree are known as fats and fixed oils. Until the latter half of the present century lubricating oils were almost exclusively derived from the animal and vegeatable kingdoms, although a grease had been prepared from petroleum in Galicia from a very early period, while early in the present century the advantages of petroleum as a lubricant free from "gumming" properties were sufficiently well known to lead to its more general use in places where it could be obtained. The rise and development of the modern petroleum and shale oil industries, stimulated by the increased demands for lubricants due to the introduction of railways and the extended use of

machinery has led to the production of immense quantities of lubricating oils, which, besides augmenting the general supply, have, owing to their cheapness and other advantages displaced to a great extent the older lubricants.

The origin of crude petroleum is one of the mysteries of nature and to science about which scarcely two authorities agree. The most recent theory is, in a sense the simplest and is expressed in the word "mud". The mud was formed on the top of mountains, at the bottom of the ocean, and all other levels in the ages long ago, by innumerable agencies in greatly diversified ways. Ages and ages have been allowed for these to do their work. Thus it is seen that crude oils are very complex in their composition and that they must differ greatly one from another.

Under the heading mineral oils, are the hydrocarbons which are derived from the distillation of the products of the American and Russian oil wells and bituminous shales.

The hydrocarbons represented in the above oils belong to the paraffin and olefine series, and are generally spoken of as parafine or mineral lubricating oils.

Parafines and olefines differ in that a parafine contains two atoms more hydrogen than the corresponding

number of the olefine group; for example, the lowest number of each group is respectively:

Marsh Gas.....CH_4

Olefiant Gas.....CH_2

Americans and Russians lubricating mineral oils contain a larger percentage of paraffins than olefines, while the shale mineral oils usually are the reverse, and contains more olefines than paraffins. As there is but little doubt that paraffins are better lubricants than can be obtained from shale oils, is easily accounted for. Mineral lubricating oils are not affected by high pressure steam or alkalies and these characteristics enable them to be used where other lubricants would be quite unfitted for the work.

Animal Oils:-- These oils are sometimes called fixed oils because they are not volatile without decompositions and are found already formed in certain tissues of animals and plants. They differ from mineral oils in containing oxygen as an essential constituent, the proportion ranging from 9.4 to 12.5 per cent. The distinction between fixed oils and fats is only a question of temperature. All fixed oils become fats at low temperatures, and all fats become oils at or below 150 F. A fundamental difference between fixed oils

and mineral oils exist in their behavior towards atmospheric oxygen. Mineral oils are indifferent to oxygen, but all the fixed oils combine with it, and most of them undergo, as a result of oxidation, changes which convert them sooner or later into solid elastic varnishes.

Animal oils are usually either colorless or yellow; vegeatable oils are of various shades of yellow and green, the green color being due to the pressure of chlorophyll, which is characterisitc of this class of oils. Fixed oils very seldom present a fluorescent appearance, unless adulterated with mineral oil.

Some of the animal oils chiefly employed for lubrication are tallow, tallow oil, lard oil, neatsfoot oil, whale oil, sperm oil, and porpoise jaw oil.

In practical use both animal and mineral oils are found in use separately and often one adulterated with the other. The reason for which, brings up the theory of lubrication.

The following bearing directly upon the theory of lubrication is abstracted direct from the chapter on "Theory of Lubrication" of Archbutt and Deeley, "Treatise on Lubrication and Lubricants."

It has been pointed out previously in this treatise that if it were not for the properties certain

liquids and soft solids possess of keeping the relatively moving surfaces apart, and thus reducing the frictional resistance between them, it would be impossible to carry on many important processes. In the case of properly lubricated bearings, the laws governing the friction between contaminated surfaces do not hold true, for at the high speeds and often at comparatively low speeds also, such well lubricated surfaces are wholly separated from each other, and the friction depends upon the thickness of the lubricating film, the area of the surfaces in contact, their relative velocities, and the viscosity of the lubricant.

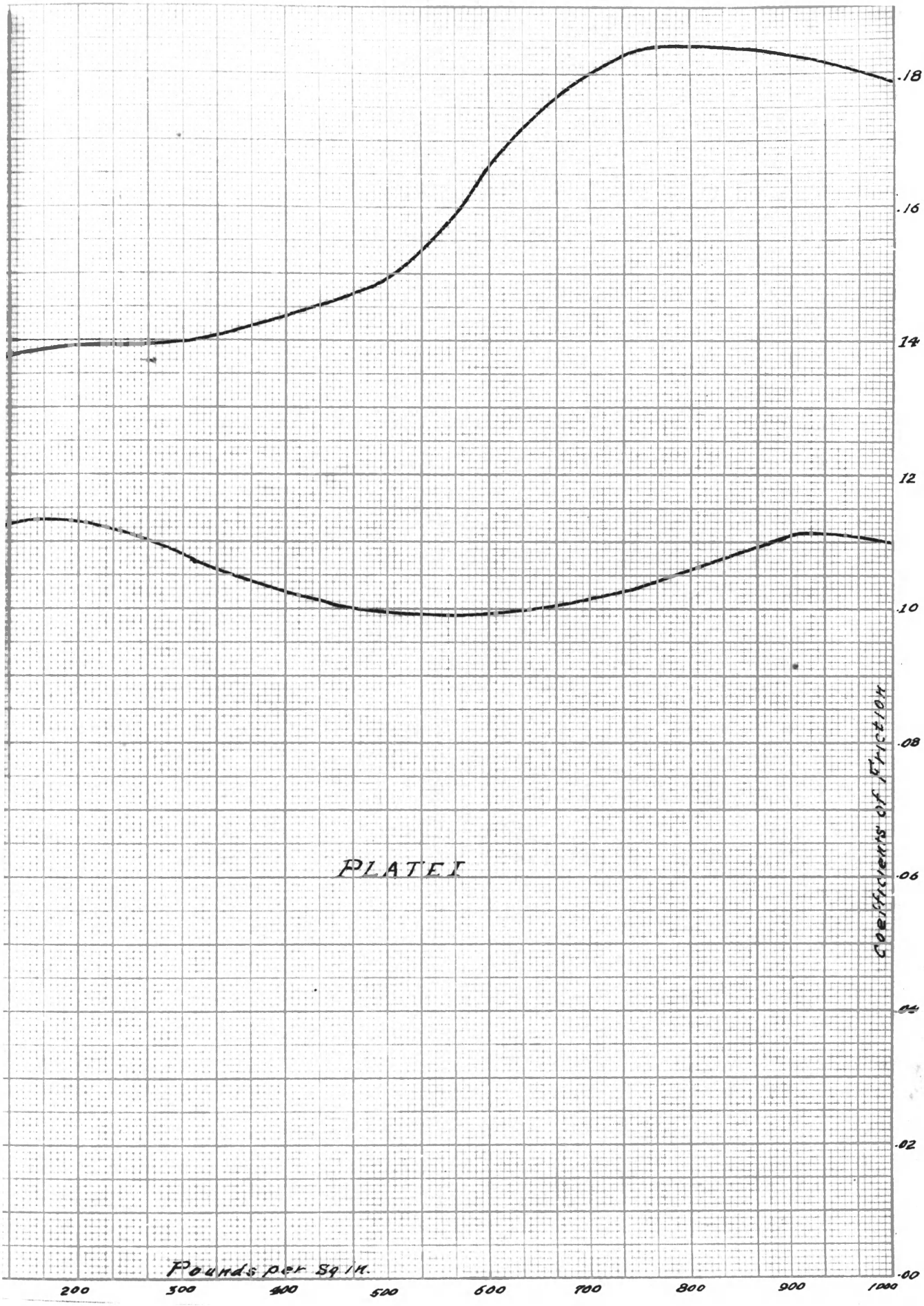
The experiments of Mr. Beauchamp Tower proved that a great deal depends upon the way in which the lubricant is applied to the bearing. Most of the recorded experiments were made in such a manner that the lubricant was what he called 'perfect'; i.e., the bearing was flooded with oil. Taken in connection with other experimental results, they have, mainly due to the work of Prof. Osborne Reynolds, enabled a fairly complete theory of high speed lubrication to be formed.

The same cannot be said of static or low speed friction of lubricated surfaces and on this part of the subject only those conditions which seem of importance will be pointed out and experimental data listed.

Theory of Static Lubrications:- When two surfaces, between which there is a viscous lubricating fluid, are pressed firmly together, the lubricant is slowly expelled, the faces approaching each other very closely, and in order to cause them to slide over each other, considerable force has sometimes to be exerted. If the solid surfaces were perfectly smooth, and the film had an appreciable thickness, any force however small would cause relative motion, for in the case of a viscous fluid film, the resistance is proportional to the speed, so that when the speed is zero, the resistance is zero.

But bearing surfaces are never perfectly smooth, and, as in the case of so-called solid friction, the projecting portions interlock somewhat, even though they may not actually be touching each other, being separated by the presence of the intervening lubricant. It is to this interlocking that the static friction must be mainly attributed; therefore the extent to which the film of any particular liquid can be reduced in thickness by the pressure urging the faces together, is a very important consideration.

Plate I taken from Archbutt and Deeley's book on lubricants and lubrication shows graphically the results obtained by Thurston with his mechanical oil-test-


.18
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Coefficients of Friction
PLATE I
Pounds per Sq. In.
200
300
400
500
600
700
800
900
1000

ing machine, when sperm oil and lard oil were used. With sperm oil the static coefficient rose very rapidly until a pressure of about 70 lbs. per square inch was reached. It then increased less rapidly to 500 lbs., and more rapidly again to 750 lbs., when at the last observation, 1000 lbs., a further falling off was experienced. Lard oil, having about twice the viscosity of sperm oil, gave a much lower coefficient of friction, but it behaved very much the same as the sperm oil, the coefficient rapidly rising with increase of load up to 100 lbs. per square inch. Above this load, however, it decreased, passing the maximum at 150 lbs., reaching the minimum at 500 lbs., and rising to a second maximum at 900 lbs. The general cha racter of the curve is the same as that for sperm oil, but with the terminal portions depressed.

Lard oil must, therfore, remain, in spite of the load on the bearing, as a much thicker film between the surfaces, than the sperm oil, and must keep the irregularities from interlocking to a greater extent. Here the smallest coefficient of friction is given by the liquid having the greatest viscosity. The higher pressures stated are much greater than it is found possible to employ in actual practice when the load is intermittent.

Coefficient of Friction at Low Speeds: It has been found that friction is much less dependent upon the system of lubrication at low speeds than at higher speeds; indeed, with a short cord of bearing surface, and therfore with heavy loads per square inch, the friction is about the same, whether the pad, oil bath, or other system of lubrication be adopted. The following results were obtained by Goodman:-

TABLE I

COEFFICIENT OF FRICTION

SPEED OF 7.8 FEET PER MINUTE.

SYSTEM OF LUBRICATION	LENGTH OF CHORD OF BEARING SURFACE (INCHES)				
	2.0	1.75	1.5	1.0	0.5
	COEFFICIENTS OF FRICTION				
Oil Bath	0.92	0.70	0.64	0.48	0.47
Saturated Pad	1.13	0.92	0.72	0.48	0.47
Oily Pad	1.87	1.25	0.94	0.57	0.51

It formerly was supposed that there was a sudden change in the value of the coefficient of friction when the surfaces in contact came to rest. This was proven to

be wrong by Professors Jenkins and Ewing by some tests made during the year 1877. They found that the frictions of rest and of motion have often very different values, and in no case was there a sudden or abrupt change in value, the kinetic coefficient gradually changing as the speed decreased until the static coefficient was reached. Their object was to measure the friction at such low velocities as 0.0002 feet per second, and from the results obtained by them we may safely conclude that the coefficient gradually changes when the velocity becomes small.

Later experiments have confirmed this view. Still our knowledge of the variations which kinetic friction undergoes with change of speed is far from being complete.

Theory of Low Speed Lubrication - It has already been pointed out that if the irregularities of the surfaces did not interlock, the friction would be much smaller than it really is at very low speedx, and the friction at rest, if the viscous lubricating films were sufficiently thick to prevent interlocking would be nil. Therefore it can be assumed that the surface irregularities are large as compared with the thickness of such a lubricating film, and that as the speed increases the film becomes thicker. As the surfaces move over each other the elevated portions alternatively approach and recede, and the greater the relative speed the more r apid is this

action. Each time two opposite elevated points approach, the liquid is expelled from between them, and then drawn in again as they recede from each other. But the viscosity of the lubricant resists this alternate squeezing out and drawing in of the film more and more powerfully as the speed of rubbing increases.

The more rapid the movement the smaller is the volume of lubricant forced from between these elevated points, and the thicker the film becomes. At very low speeds this thickening of the film is assisted by the lubricant which is trapped at the front edge of the bearing. To this fact must mainly be ascribed the low friction of even flat surfaces at high speeds. The thickening action occasioned by the elevated points, or roughness of the surfaces passing over each other can only occur to a very limited extent, and would seem to prevent the kinetic friction at very low speeds, from exceeding the static friction by so much as it otherwise would.

The resistance due to the viscous friction of the film is proportional to the speed of rubbing, and inversely proportional to its thickness. Up to speeds somewhat below one centimeter per second we must therefore assume that the lubricating film does not thickan with sufficient rapidity to prevent the speed from increasing the fric-

tional resistance. At greater speeds, however, the film thickens so rapidly, as compared with the increase in speed, that the coefficient soon falls to a small fraction of its original value.

High Speed Lubrication: Lubricating Film - At a speed depending greatly upon the load and the nature of the lubricant, the rubbing surfaces commence to separate, and a comparatively thick pressure film forms between them and carries the load. The extent to which the thickness of this film increases with the speed, varies not only according to the load, the viscosity of the lubricant, the area of the bearing and the speed, but also according to the shape of the surfaces and the relative positions they assume. As illustrations of the conditions under which pressure films are produced, three different cases will be considered, namely as follows: First, the case of plane parallel surfaces; second, plane surfaces inclined to each other; third, those of cylindrical surfaces, such as journals.

Figure 1 illustrates the action of a lubricating film upon plane parallel surfaces.

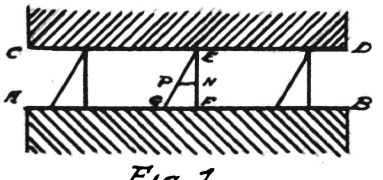

Fig. 1

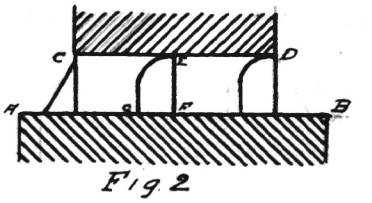

Fig. 2

It shows two parallel planes of unlimited length and breadth separated by a viscous film of thickness r_0. The upper plane CD is supposed to be fixed, while the lower plane HB moves in the direction of the arrow with a velocity v_1. Then from the definition of viscosity $n = \frac{r_0 f}{v_1}$, which will be taken up later, there will be a tangential resistance to motion $f = n\frac{v_1}{r_0}$, n being the viscosity of the liquid, f the force per unit of area when one plane has an area H, the total resistance to motion will be $F = n\frac{v_1 A}{r_0}$,

The tangential motion varies uniformly from v_1 at HB to nil at CD. Thus if the length FG be taken to represent v_1 then the length of the line PH will represent the velocity at P.

When first the planes are set in motion the inertia of the viscous liquid prevents it from at once assuming this condition of flow, but in a comparatively short period all irregularities of motion subside, and the velocity of the liquid at any plane is strictly proportional to its distance from CD.

When the planes are of considerable area and are close together, the viscosity of the oil powerfully resists its escape at all points, and at the same time, by tending to make the rates of distortion everywhere

equal, causes it to accumulate and force the planes apart until the forces are in equilibrium. This action is going on at all points between the surfaces when their length, measured in the direction of motion, is not very great. The force tending to throw the planes apart is, therefore, distributed over them much as is the force resisting the approach of two surfaces separated by a viscous medium.

In this way when the opposing surfaces are constrained to remain parallel, a pressure film is maintained between them, and a considerable load may be supported by CD, so long as AB is in rapid motion.

No doubt, at low speeds, as the load urging the faces together is increased, the volume entering decreases more rapidly than does the distance separating the rubbing surfaces, until finally the marginal pressure of the imprisoned film prevents the liquid from entering at all, and the faces close together; but when the speed is great and the load is increased, there is reason to suppose that the liquid fails quite suddenly to get between the surfaces, which thereupon "seize". It can therefore be assumed, that, except at low speeds, the volume entering is approximately proportional to

$$v_2 \, \zeta \, b = V$$

the factors v and k_0 being the same as mentioned above and b the length of the orifice at D.

Lubrication of Inclined Plane Surfaces - When the rubbing surfaces are plane, the opposing surfaces are free to adjust themselves according to the position of the load they carry and the distribution of the pressure of the film keeping them apart, this is not always the case but has very few exceptions. The conditions under which the lubricant acts then differ somewhat from the case of plane parallel surfaces. In the latter case it is the inertia and viscosity of the lubricant that we must attribute the presence of the pressure film; but when the faces are free to become inclined, the liquid wedges itself between them and forces them apart.

In figure 3 HB is a plane surface of unlimited length and breadth moving in the direction of the arrow beneath the inclined surface CD.

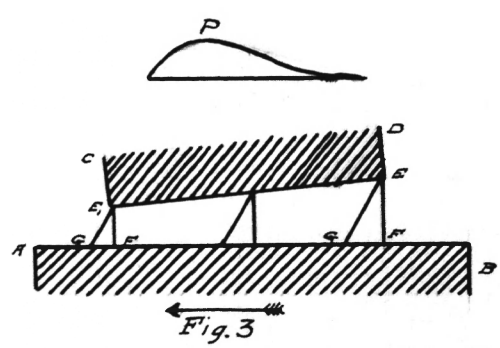

Fig. 3

which is of limited length and very limited breadth, in a direction perpendicular to the paper. Omitting the

effects which would be produced by the inertia of the liquid, the conditions of flow are as follows:

At the lower edge of D where the liquid enters between the surfaces, the volume introduced is proportional to $\frac{EF \times GF}{2}$.

However, owing to the inclinations of the surfaces, the volume passed out at the edge C is only proportional to $\frac{EF \times GF}{2}$.

When the surface CD is of considerable area and the planes are very close together, the excess of oil or other lubricating fluid introduced at the edge D must excape; but this is opposed at all points by the viscosity of the liquid. The oil therefore, tends to accumulate, and a pressure is set up which forces the surfaces apart until the load is sufficient to prevent further recession. This action is caused by the inertia of the entering fluid.

Professor Osborne Reynolds has made a study of these effects, and according to him the effect reaches a maximum for bearings of such dimensions as are used in practice when EF= 2.2 E F and owing to the greater freedom with which the lubricant can escape at the end D and the sides near it, the point of maximum pressure p is somewhat nearer C than D. The curve at the top of the figure indicates, approximately, the pressure at differ-

ent points of such as film, tending to force the surfaces apart. The direction in which the load acts, normal to AB, does not necessarily coincide with p but with the centre of the area enclosed by the curve and base line of the pressure ordinates.

The pressure exerted by the film must always be equal to the resultant external force which, neglecting the obliquity of CD, is perpendicular to HB, and tends to force the surfaces together. When the surfaces are free to assume any position, the pressure of the film, inclination of surfaces, etc., adjust themselves to suit the load and its point of application, and the nearer the surfaces are caused to approach each other the greater is the friction and consequent pressure for the same velocity.

The relations obtaining between the load, pressure, etc., under these circumstances have been determined by Professor Osborne Reynolds. These relations are gone into in detail in the Philosophical Transactions of the Royal Society, 1886, page 173.

The next theory to be taken up will be lubrication of cylindrical surfaces. This is the most common form of bearing surface, and the one with which most of the experiments have been made and recorded. Until Mr.

Beauchamp Tower's results were published little was really known concerning the effects produced by varying the method of applying the lubricant. With properly sloped brasses resting upon well lubricated journals, he succeeded in obtaining results which Prof. Osborne Reynolds demonstrated were in accordance with hydro-dynamical theory.

A brass which has been running for some time upon a lubricated cylindrical journal wears in such a way that the radius is always slightly greater than that of the journal. When forced into contact the brass and journal do not touch over the whole of their surfaces. On the other hand, when the journal is in rapdi motion and the weight on the brass is not too great, the surfaces are separated from each other by a continous oil film. Figure (4) shows a section of such a brass and journal, the latter rotating in the direction of the arrow, the surfaces being respectivley AB and CD.

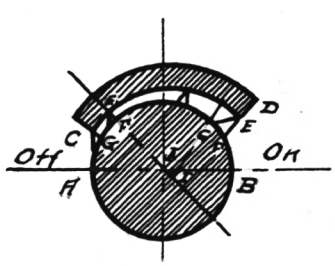

Fig. 4

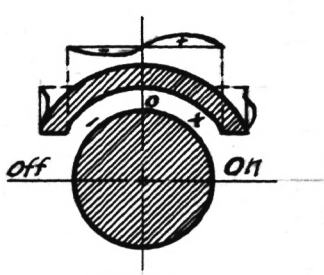

Fig 5

The curved surfaces AB and CD have their

centers at I and J, a line drawn through which indicates the position of nearest approach E_1F. Between EF and E_1F the distance separating the surfaces decreases, the triangle E_1FG is consequently smaller than EFG, and during its passage the lubricant is compressed and tends to force the surfaces apart. As we go from E_1FG toward C the faces recede, and were it not for the flow of oil from the portion where the liquid undergoes compression, the pressure might be and, sometimes does, become negative.

When there is no load the conditions are as shown in Figure (5). The vertical pressures are shown by the curved lines on the top of the figure. On the right-hand or "on" side the pressure is positive, while on the left-hand or "off" side it is negative, and the vertical components of these pressures balance each other. On the other hand, the horizontal component of the pressures to the left and right, indicated by the curves at the sides, will both act on the brass to the right, and as these will increase as the surfaces approach, a distance corresponding to JI, figure (4) must be actly such that these components balance the resultant diction.

The thickness of the film at different points

and the variations of the position of E F with different loads, have been worked out mathematically by Professor Osborne Reynolds, and for full treatment of the subject, the original papers should be consulted.

Effects of Load and Speed - In case of plane surfaces, the frictional resistance is proportional to the square root of the load. In the case of cylindrical surfaces, however, the rubbing faces, owing to their curvature, cannot separate sufficiently, while still sustaining a load, to give this result, for when the load is increased E_1F decreases, the lubricant is prevented from escaping on all sides as freely as before, and EF, i.e., the distance between the brass and journal on the one side, increases. Therefore, as the load is increased, the positive vertical component to the right increases (figure 5) and overbalances the negative component to the left, which decreases, and E_1F , the point of nearest approach, moves to the left until the load reaches a particular value; above this load the point of nearest approach moves toward O. During this change in the position of E_1F the thickness of the film at different points alters in such a way that the viscous resistance which it offers to the motion of the journal remains nearly constant, and the friction is practically inde-

pendent of the load when the speed is sufficient to maintain a pressure film between the two surfaces.

According to this reasoning the friction appears to be approximately proportional to the area of the contact surfaces, the speed of the journal, and the viscosity of the lubricant, and is nearly independent of the load.

At the outset, as with plane surfaces, when the oil film has scarcely established itself, the friction, other things being equal, varies with changes of load. However, owing to the curvature of the surfaces, the film very quickly reaches a maximum thickness.

TABLE (2)

BATH OLIVE OIL-TEMPERATURE 90 F.

Journal 4" wide by 6" long. Chord of Arc of Contact 3.92"

Nominal Load lbs per sq in	Nominal Frictional Resistance Per Sq. In. of Bearing. 105 ft. per min	157 ft. per min	209 ft. per min	262 ft. per min	314 ft. per min	366 ft. per min	419 ft. per min	471 ft. per min
520		.416	.520	.624	.675	.728	.779	.883
468		.514	.607	.654	.701	.794	.841	.935
415		.498	.580	.622	.705	.787	.870	.995
363		.472	.580	.616	.689	.725	.798	.907
310		.464	.526	.588	.650	.680	.742	.835
358	.361	.438	.515	.592	.644	.669	.747	.798
205	.368	.430	.512	.572	.613	.675	.736	.818
153	.351	.458	.535	.611	.672	.718	.764	.871
100	.360	.450	.550	.630	.690	.770	.820	.890

The nominal load per square inch is the total load divided by (4X6)

Table (2) gives the results obtained by Mr. Beauchamp Tower with an oil bath lubrication. The loads range from 100 lbs. to 520 lbs. per square inch, nominal, and yet for each speed the frictional resistance is nearly a constant. Above speeds of 100 ft. per minute low speed effects do not show themselves, and the resistance will be seen to be nearly proportional to the square root of the speed instead of to the actual speed. Professor Osborne Reynolds considers this as due to the fact that when the speed is increased the rate of shear which the film undergoes is likewise increased, the film becomes heated, its viscosity is decreased and less resistance to motion is encountered.

Effects of Area:- With parallel plane surafces it was found that the friction does not increase much when the area is increased, for an increase of area brings about an increase in the thickness of the film. But a cylindrical form of bearing gives a different result, the frictional resistance being more nearly proportional to the area owing to the fact that when the speed is moderately high the film has a fairly constant effective thickness under all loads. The frictional resistances at different loads does not vary

very much. On the other hand, a long, narrow brass offers a very much smaller resistance to the motion of the journal than does a brass having a wide chord of contact. The area of the surfaces is in all cases proportional to the length of the arc of contact. If the films were of even thickness throughout the length of arc, the friction would be simply proportional to its length. The film, however, is much thicker on the "on" side where the lubricant enters than it is nearer the center of the brass. The resistances offered to the motion of the journal by those portions of the brass near the "on" and "off" sides are, therefore, less than near the middle of the bearing, and as the brass is cut away at the sides, the thickness of the film is slightly reduced. The reduction of the resistance is, therefore, not quite proportional to the reduction of the area.

In a great many cases the loads upon bearings are by no means constant, for the faces often alternately approach and recede from each other. In such cases the alternation is very rapid, the bearing will carry a very great weight, for at each alternation the pressure is completely relieved and the oil trapped cannot be expelled during the short time the load rests

on the bearing. The reason the lubricant cannot escape while the pressure lasts is doubtless that the volume of oil which can be squeezed out by any particular load is proportional to the cube of the thickness of the film.

Until quite recently, full advantage was seldom taken of the viscous lubricating properties possessed by lubricants. Sometimes it is difficult to apply the lubricating fluid in the most effective manner, while in many case the more perfect the system from a friction point of view, the greater the waste of oil.

Methods of Lubrication - Proper lubrication is a matter requiring considerable experience and good judgement as has been pointed out before in this treatise. There are no standards of efficiency that can be adopted. It is necessary for each engineer to establish his own standards of comparison. The fact that a bearing runs cool is not of itself proof that it is as nearly frictionless as possible nor that the cost of lubricating it is as low as it can be made. The engineer must experiment and establish a standard of lubrication, first being satisfied that the condition of the bearing is favorable to cool running with the least lubricant. Under these conditions he is able

to change lubricants and ascertain their relative merits and the cost.

To reduce the cost of lubrication to the lowest possible figure it is necessary to buy as little new oil as possible. Changing lubricants may result in reducing the quantity required and still not affect the total cost, or it may result in using a greater quantity of cheaper lubricant, which also will have no effect on the cost. Economy is measured by the cost of obtaining the same results.

In proportion to this ability to reduce the friction in a plant, by more suitable lubrication, the following results will be secured:

1. (a) Reduction in the total horsepower load of the plant.

(b) Reduction in the power of transmission and the power of each part of the plant.

2. Reduction in the amount of coal required for developing the power necessary to operate plant.

3. Reduction in the amount of feed water used for power purposes.

4. Reduction in temperature of all bearings and spindle bases.

5. (a) Increase in the speed of machines and spindles.

(b) Increase in the speed of the engine if the load upon the engine has been excessive.

6. (a) Increase of production of the plant, if the production is related to the speed.

(b) Increase in production due to less stoppage.

7. (a) Decrease in repairs.

(b) Decrease in the number of belts required for driving spindles.

(c) Decrease in the wear on belts.

The sole guide to the efficiency and economy of lubricating oils must be their results in use. If the buyer once starts to measure results and to buy his lubricating oils, not on a first cost basis, but on a basis of efficiency and ultimate economy, he will have little to fear in putting his problem frankly before an oil manufacturer of standing.

Cause for Large Friction Losses - Friction in bearings cannot be eliminated entirely but it can often be lessened. This can be done in one or more of three ways; first, by putting the bearings in line; second, by putting the rubbing surfaces in good condition; and third, by using a proper lubricant. These three conditions or causes of friction trouble are applicable to all bearings whether for shafting, in

machines or in engines. Very little is known about cylinder friction and lubrication, an entirely different problem being presented in such cases of lubrication.

Figure (6) represents a main bearing with quarter-boxes:

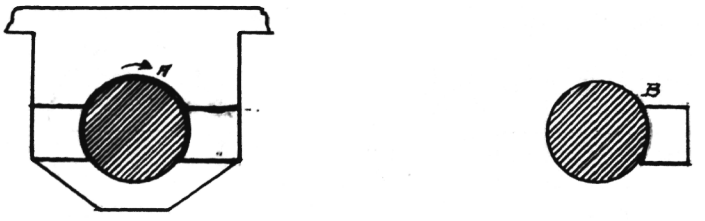

Fig. 6

One very general cause of bearings heating is that the edge of the bearing becomes very sharp and strips the oil from the shaft or pin, as shown in Fig. (6). This of course prevents the oil or other lubricant from clinging to the shaft and being carried around with it. This condition can be easily removed by scraping the bearing surface near the edge so as to form a shallow tapered pocket or recess for the lubricant.

This same principle may be applied to shaft bearings, and oftentimes the relieving of the edge of the bearing will not only cure a bothersome box,

but will result in a saving of lubricant.

The cutting of oil grooves and the shape of grooves is a matter on which there is a wide difference of opinion among engineers. Some have obtained good results with one arrangement of grooves, and some with another, and others believe grooves to be entirely unnecessary.

Wherever grooves are cut in bearings or brasses, they should be made shallow and moderately wide, instead of narrow and deep, and the edges should be rounded so as to reduce the tendency to strip the lubricant from the shaft or pin. The object of the groove is to collect any surplus oil near the initial point of application and conduct it along the shaft to the center and ends of the bearing, thus facilitating an even distribution of the lubricant.

J.H. Spoor of Madison, Wisconsin, has worked up some very interesting results on cylinder lubrication, and has plotted the results in order to check the data, and also furnish a means to make those data applicable to any engine. To read the curve for any engine it will be necessary to find the circumference of the bore and multiply that by the length of stroke in inches (surface lubricated = diameter times 3.1416

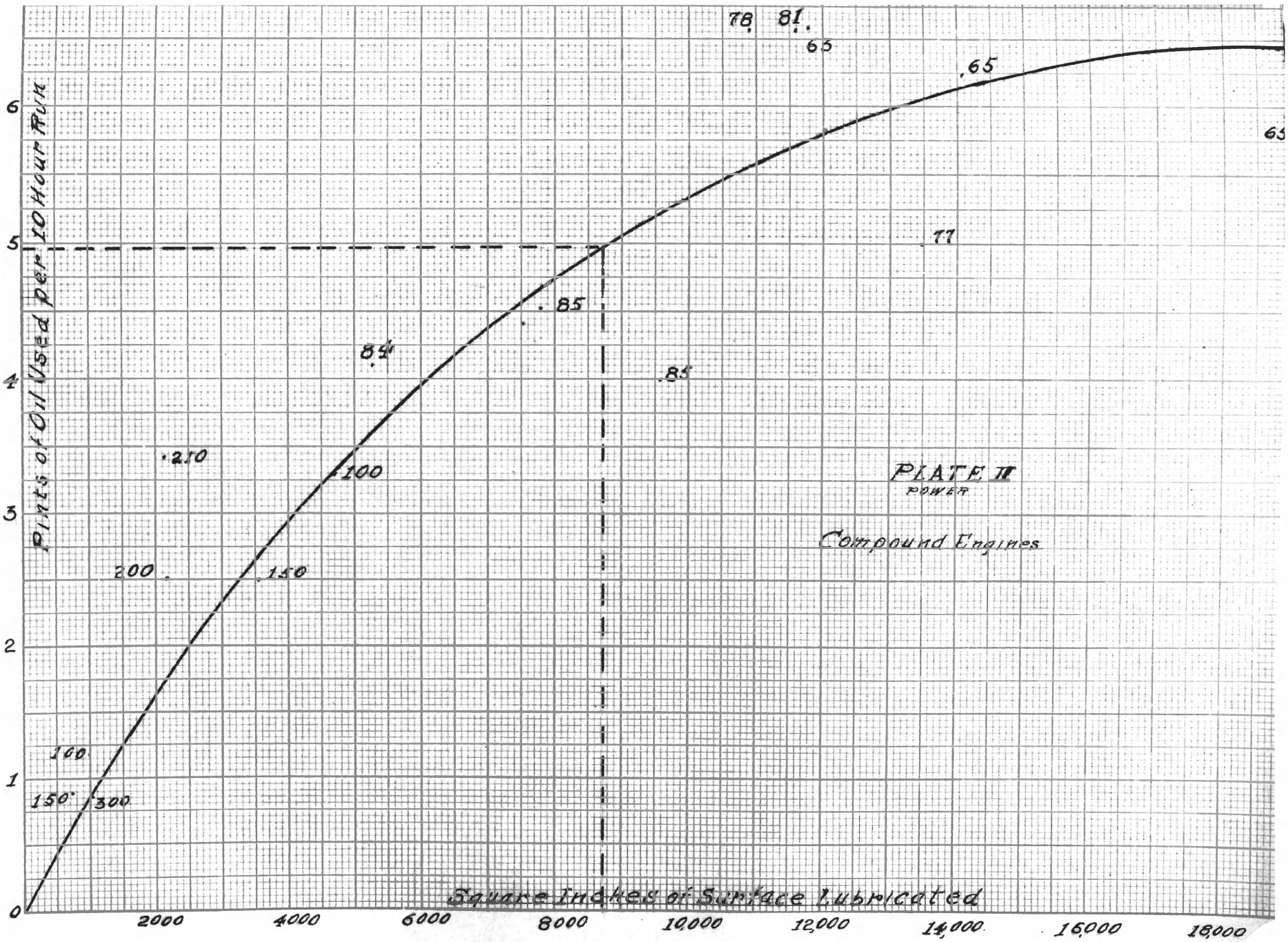
PLATE II
POWER
Compound Engines
Pints of Oil Used per 10 Hour Run
Square Inches of Surface Lubricated
0
1
2
3
4
5
6
2000
4000
6000
8000
10,000
12,000
14,000
16,000
18,000
.78
.81
.65
.65
.65
.77
.85
.85
.84
.210
.100
.200
.150
.160
.150
.300

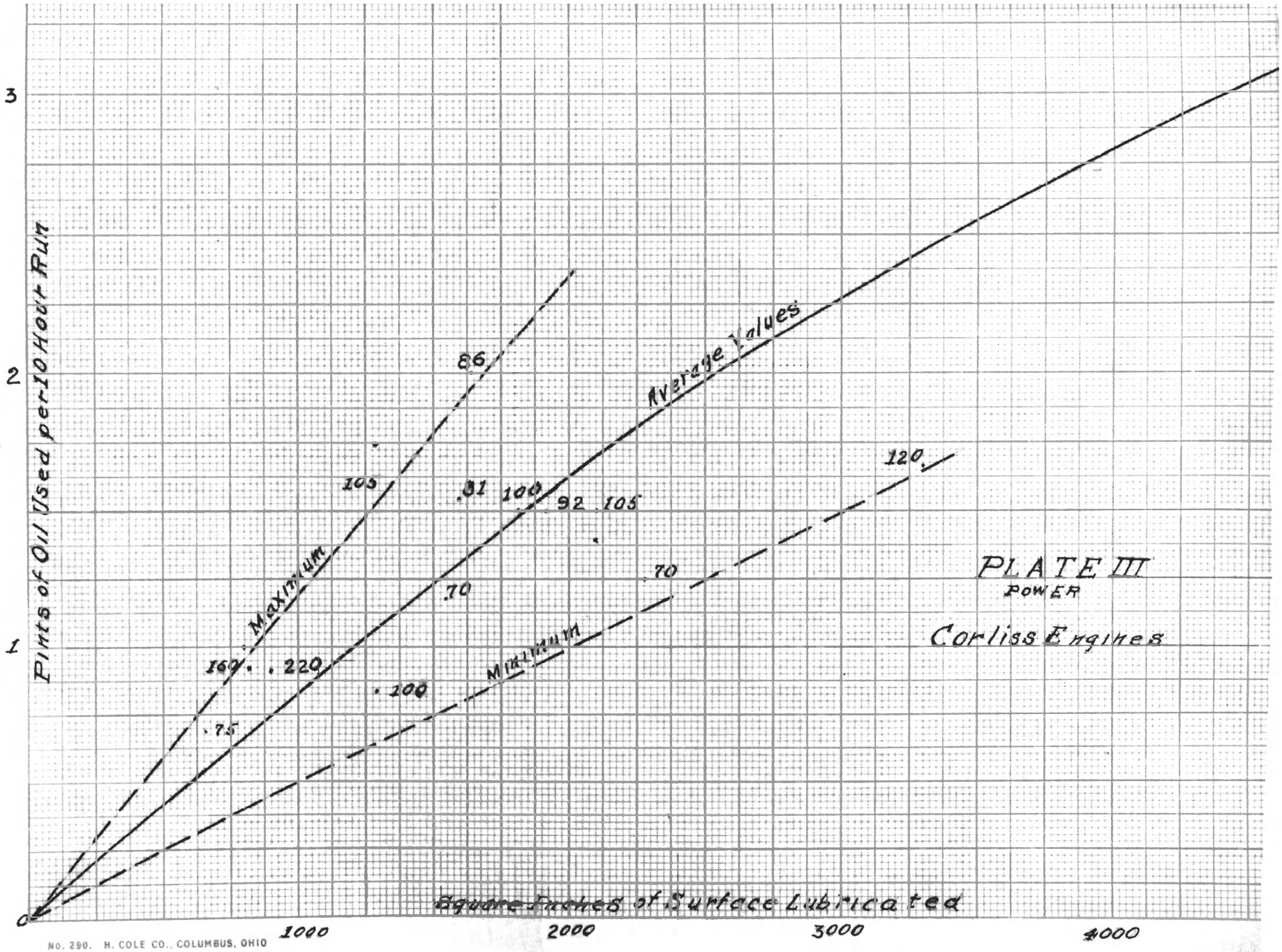
PLATE III
POWER
Corliss Engines
Average Values
Maximum
Minimum
Pints of Oil Used per 10 Hour Run
Square Inches of Surface Lubricated
0
1
2
3
1000
2000
3000
4000
86
105
.81
100
92
105
120
70
.70
160
220
.100
.75
No. 290. H. COLE CO., COLUMBUS, OHIO

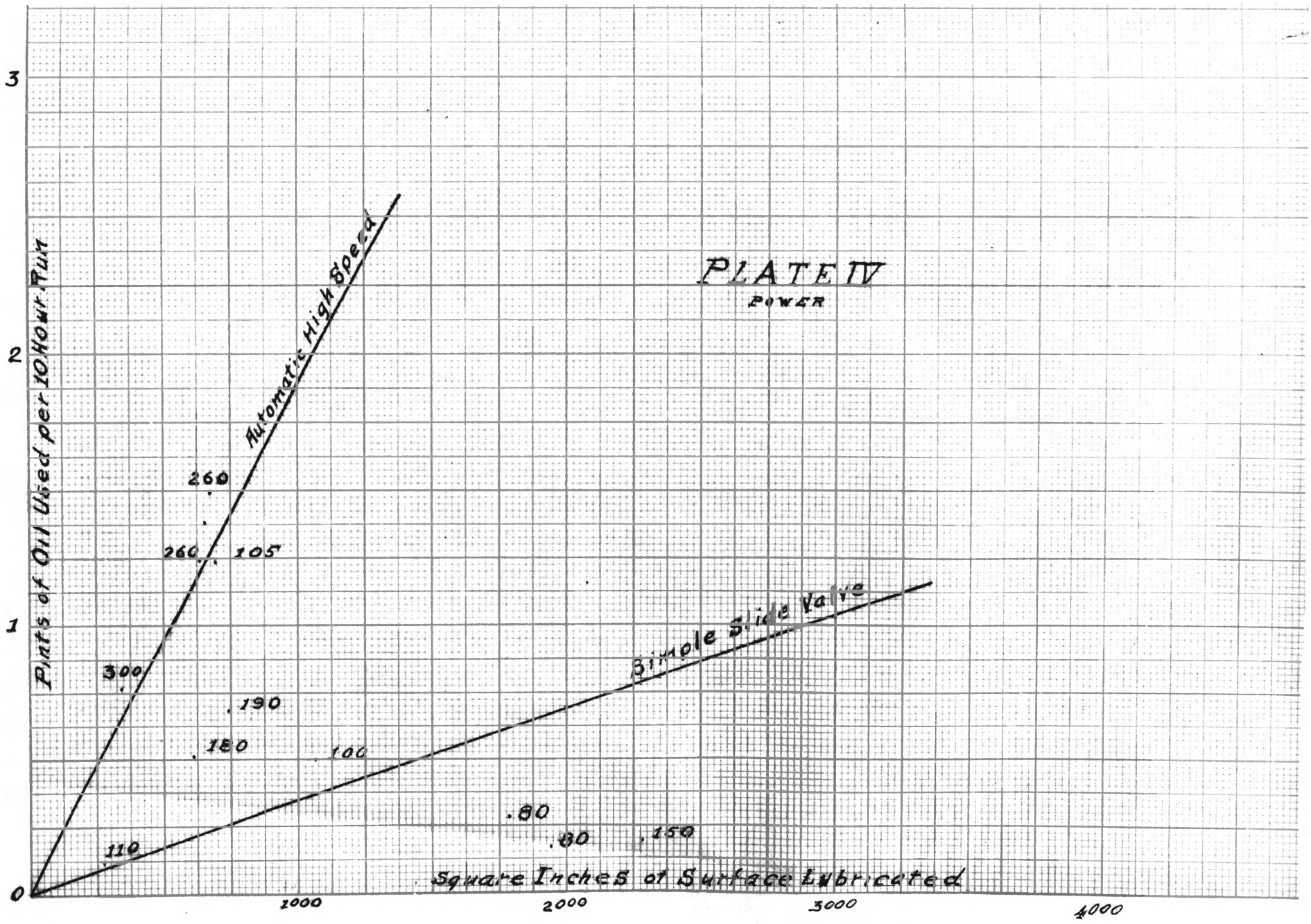
PLATE IV
POWER
Pints of Oil Used per 10 Hour Run
Square Inches of Surface Lubricated
Automatic High Speed
Simple Slide Valve
0
1
2
3
1000
2000
3000
4000
260
260
105
300
190
180
100
80
80
150
110

times the stroke). In case of the compound engine the area lubricated would be the sum of the surfaces of the two cylinders.

As plotted, the points along the curves show quite a uniformity, and when the great variety of conditions is considered from which they come, the results, especially in the case of the compound engine, show considerable regularity. On account of this regularity it is quite evident that there are some very necessary conditions which must be fulfilled in order to have good lubrication. If there were no such conditions which must be fulfilled, the points would fall all over the paper.

The average curve as plotted for the Carliss engines, comes remarkably close to the curve. See plates II and III for the compound engines. However, a compound engine usually operates with a four valve gear and, as far as lubrication goes, is practically a double Carliss and, hence, does not require any different lubrication. In plate IV are shown the results for the slide valve engines, the automatic engines. There seems to be a kind of shading off from one to the other and there are hardly enough points to establish any definite curve. The simple slide valve en-

gine being simple and hard to get out of order receives less attention and is subject to many more abuses than the more expensive engines.

A comparison of the curves seems to show some important results but nothing that will help solve lubrication difficulties.

On nearly all the points plotted the revolutions per minute are given, and in the majority of cases the speed does not seem to play a very important part. Although, as a rule, the higher speeds seek the part above the curve, showing that the higher speed needs a little more lubrication.

Methods of Lubrication - Bath or "perfect" lubrication is obtained by allowing the under side of the journal to dip into a bath of the lubricant. In this way the journal picks up and throws against the brass a thick film of oil and the result is "perfect lubrication", i.e., the thickest film is secured which the bearing can automatically maintain.

Pad lubrication is obtained by pressing a woolen or felt pad soaked with the lubricant against the journal. When the pad rests in a bath of the lubricant and is kept inferior to those given by the bath. When, however, the supply of oil to the pad is

deficient, the difference is marked, as was shown by some results tabulated by Goodman, which are shown in the following table:

TABLE III

Method of Lubrication	Width in Inches of Chord of Bearing Surface				
	2.0	1.75	1.5	1.0	0.5
	Frictional Resistance in Lbs.				
Oil Bath	2.89	2.42	2.14	1.18	0.82
Saturated Pad	4.47	3.98	2.49	2.10	1.06
Oily Pad	7.97	6.62	5.80	3.80	2.70

Ball and roller bearings have come into very extensive use, having been found very suitable for motor cars, cycles, dynamos, and other machines, the bearings of which have to carry moderate loads.

In case of ball bearings the design should be such that the lubricant used can escape readily on each side of the balls as they roll in their races, with rollers, however, the lubricant must be pushed along in front of them. On this account the viscosity of the lubricant used must have an appreciable effect upon the

friction in the latter case.

The methods of oiling that are used extensively in present practice may be divided into two classes or systems, viz., mechanical and hand lubrication. The latter needs no explanation. Mechanical lubrication comprises two methods, the direct and indirect method. With the direct method a pump is used to force oil directly through the bearings, while with the indirect method the pump forces the oil, through a system of pipes, the latter terminating at or in the oil cups, the flow of oil into the cups being regulated by small stop-cocks. In one system water or air pressure is employed to force the oil through the pipes, the latter terminating slightly above or within the oil cups. In still another known as the gravity system, the oil is pumped into an elevated tank from which the system of pipes is supplied.

The splash system is a direct method of oiling the crank pins of high speed engines. It is extensively used and is a good plan when the oil is kept free of water. The greatest objection to this system of oiling is the presence of water in the oil. The water of condensation from the stuffing box will often find its way to the crank chamber or pump well, and

water being heavier than the oil, it will be taken by the pump and discharge into the bearings which is likely to cause hot bearings and pins. The oil and water form an emulsion in the crank chamber and this reduces the lubricating effect of the oil. It is necessary therefore to keep a lookout for such conditions and occasionally draw off the water which will usually prevent troublesome results.

In Figure (7) is shown diagrammatically the air or water pressure system of oiling in which H is the oil or pressure cylinder, B, the oil storage tank, and C, the filter. Cylinder H may be filled with oil through plug D by hand, although it is more convenient to first fill it with water and allow the water to run out through a valve H the receding water forming a vacuum and drawing the oil into the tank. After this valves H and F are closed and the water valve E opened. The water pressure then forces the oil out through pipes P P thence through suitable piping to the points where it is to be used. A modification of this system is used in a great many cases, instead of relying altogether on water pressure, a pump attached to the reciprocating parts of the engine is used to circulate the oil and return it to the storage tank.

Figure (8) illustrates a simple form of gravity

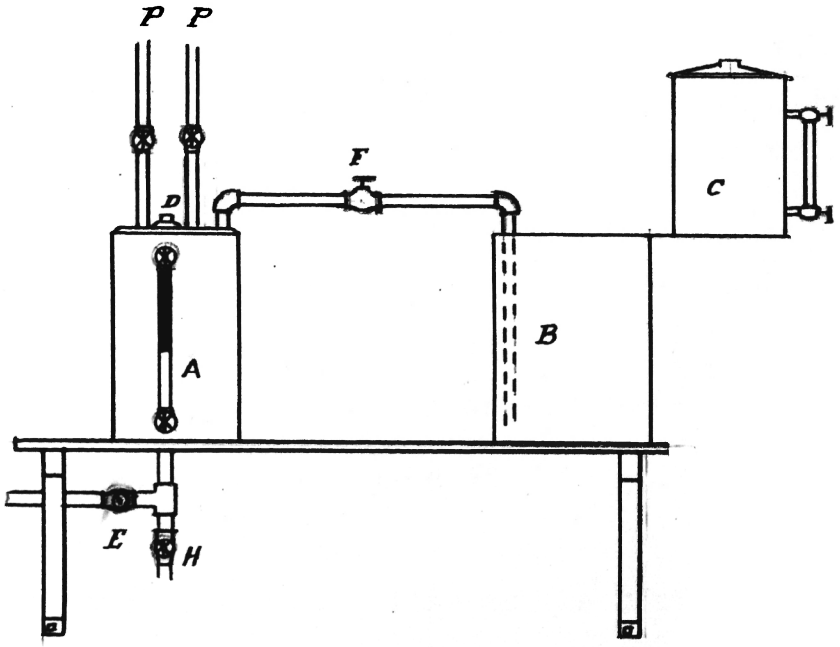

Fig. 7
AIR or WATER PRESSURE OILING SYSTEM
SOUTHERN ENGINEER

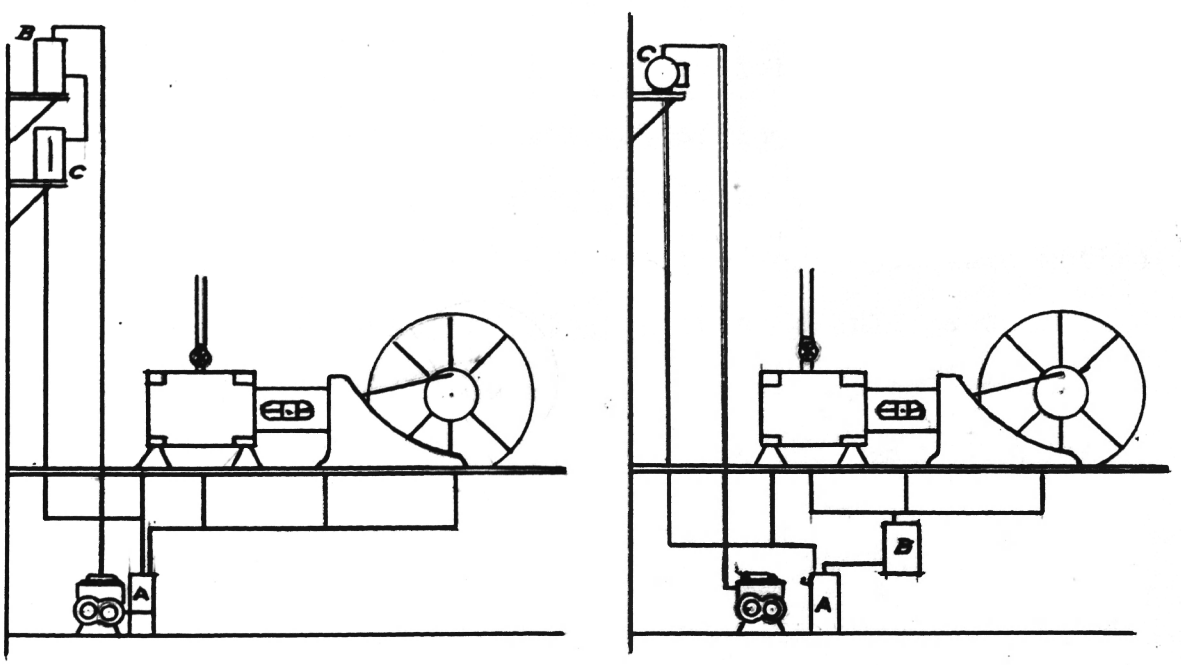

Fig. 8
TWO ARRANGEMENTS OF GRAVITY OILING SYSTEMS
SOUTHERN ENGINEER

system. Waste oil from the different engines is led to tank A. The pump connected to the tank discharges the oil into a filter B, placed at a considerable heigtht above the engines. The oil flows from the filter B into the storage tank C, from which it flows through suitable piping to the oil cups. The flow of oil is regulated by suitable cocks over each bearing or cup.

The hand system of oiling is the most wasteful method there is of oiling, and it should be replaced as soon as convenient by a mechanical system along the principles described above.

In all the systems described the object that is being sought after is perfect lubrication, with as little waste of the lubricant as is possible under such conditions. The method of applying the oil at the top of the bearing is undoubtedly the worst possible method that can be employed. The oil bath should be used in every case, but in some cases such a method cannot be used on account of locating, shape and use for which the bearing is used.

Chapter III

Means of Studying the Coefficient of Friction

Coefficient of Friction: As has been pointed out in chapter one the relationship borne by the frictional resistance to the load or force pressing the surfaces together has been the subject of much speculation and discussion, and although numerous experimental researches have been carried out, there always has been much conflict of opinion. From what has been stated concerning the action of solid surfaces in close contact have upon each other, it will be clear that, although it may be possible to formulate somewhat general laws of solid friction, such laws cannot be expected to hold under extremes of pressure or speed; and they will be affected to some extent by the nature of the materials in contact and the surface conditions.

Friction renders results very deceptive and for this reason an engineer might congratulate himself on having reduced friction losses to a minimum when in reality he may be burning tons of coal a day to make good, or rather overcome, preventable losses due to friction. This, being the result of not knowing what the losses of friction amount to.

Friction is alright where it belongs, and all

wrong where it does not belong. It is, of course, impossible to remove and prevent friction altogether. It may be compared to air in a condenser; it can be eliminated to a great extent, but not entirely, and friction should be treated much in the same way. We know what a perfect vacuum is and try to attain it, and the gage shows how nearly we approach it. We know what a frictionless bearing is, and we should endeavor to obtain it, but unfortunately we have no easily applied method of indicating continuously how nearly a bearing approaches the condition of being frictionless. Friction losses are measured with a high degree of accuracy but there is no convenient method of indicating day after day just what they are in any bearing or any set of bearings. This accounts very largely for ignorance of the true condition of affairs which ignorance is too often displayed in power plants and manufacturing establishments. It is largely a case of out of sight, out of mind, and no attempt is made to find out what is being done until the loss becomes excessive.

Objects of mechanical testing: Mechanical tests of lubricants may be made with two distinct objects in view, one being to measure the amount of frictional

resistance offered to free motion by bearings lubricated in different ways, and the other to determine the relative oiliness or greasiness of the lubricants. The mechanical testing of lubricants will be looked at under the above heading from the first point of view pointed out above.

The more closely the conditions under which the oil is to work in practice are approached, the more satisfactory will be the results obtained, for, from the point of view of frictional loss each change of load, alteration in the speed of the journal, variation in the method of supplying the lubricant, or change in the rubbing surfaces, has an important effect upon the frictional resistance. In order, on the other hand, to determine the oiliness or greasiness of a lubricant, it is necessary to eliminate, as far as possible, results due to viscosity. Imperfect methods of lubrication admit of this being done most satisfactorily, experiment having shown that at low speeds greasiness is an all-important factor, in perfect lubrication at high speeds viscosity plays the more important part.

Thin films and low speeds are also the most suitable conditions under which to ascertain the effects resulting from changes in the nature of the metals in contact, as at high speeds a brass is generally lifted

quite off its journal by the formation of the pressure film.

Useful hints concerning the endurance of oils may also be obtained with such machines, by observing the changes which the friction undergoes after the oil has remained exposed for some hours to the air.

The proper test for a lubricant is not then, merely a viscosity measurement, but a test for the friction under service conditions of temperature, speed, pressure and clearance.

For every lubricant (free from foreign and harmful ingredients), there is a set of working conditions, under which it will give a minimum friction coefficient. This minimum friction coefficient lies between the values of one half of one per cent and one per cent.

The apparatus for testing lubricants must therefore permit of the reproduction of any set of service conditions and, more important still, must make it possible to keep these conditions absolutely constant during the test or comparison.

These conditions are:

Adequate supply of lubricant.

Definite and constant values for the rubbing speed,

Definite and constant pressure.

Definite and constant temperature.

Definite and constant thickness of lubricatory film (or clearance between the rubbing surfaces)

Means of Studying Coefficient of Friction or Oil Testing Machines: The data available hitherto has been obtained chiefly from chemical and physical analyses, but these alone are not sufficient as has been pointed out in the preceeding pages, and they are also frequently misleading. It is not enough to know, for instance, the percentage of acidity and pitchy ingredients in an oil, its specific gravity, flash-point, and viscosity, etc. though these particulars are of value in their way; they do not however, afford a reliable basis on which to judge of the suitability of the oil or grease for any given purpose. Of far greater importance are practical tests under actual working conditions, but such tests involve considerable risk and great and unnecessary expenditure of time and trouble.

Beauchamp Tower's results were the first reliable results to be obtained on the coefficient of friction and were obtained by measuring the moment.

Tower's apparatus - Fig.(1) The journal ex-

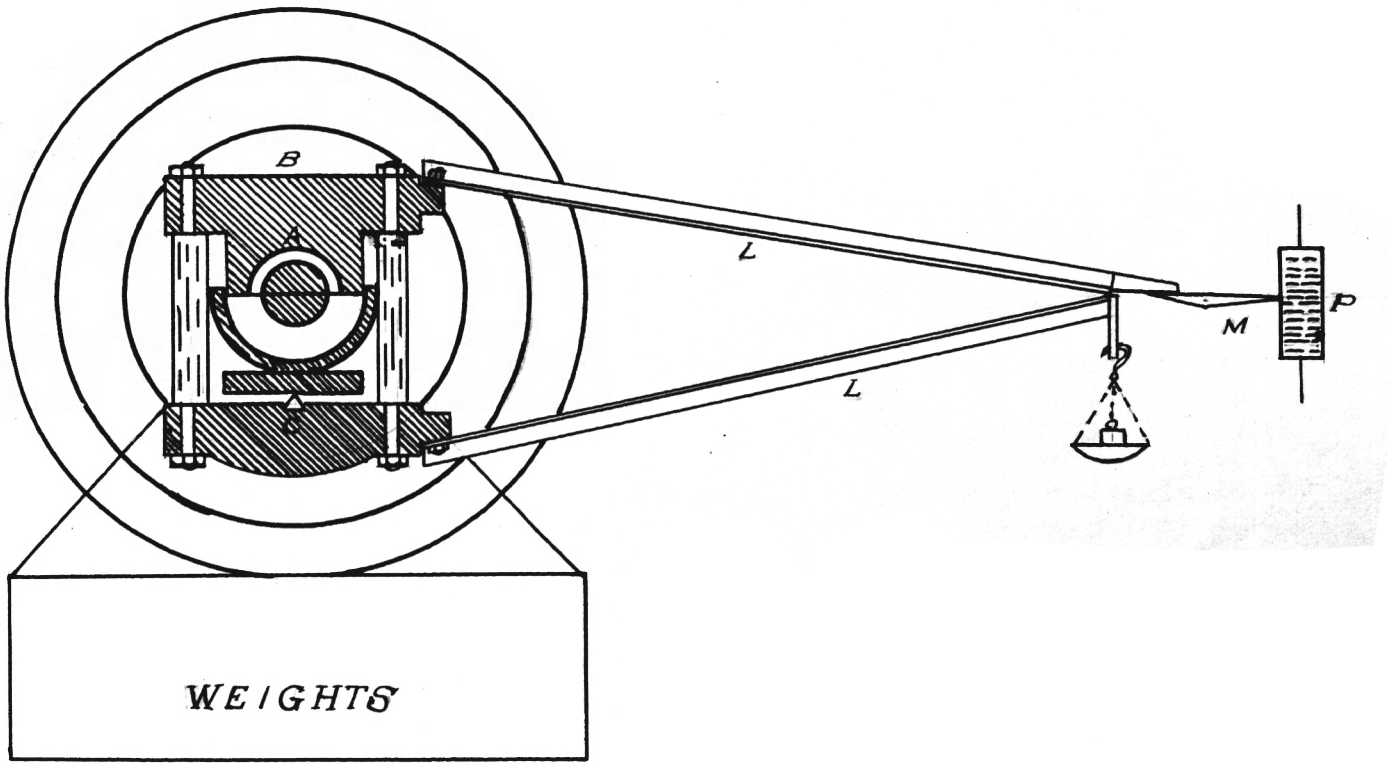

Fig. 1

perimented on was of steel four inches in diameter and six inches long with its axis horizontal. A gunmetal brass H embracing somewhat less than half the circumference of the journal, rested on its upper side. The exact arc of contact of this brass was a cast iron cap B, from which was hung by two bolts a cast iron cross-bar C, carrying a knife edge. The exact distance of this knife-edge below the center of the journal was five inches. On this knife edge was suspended the cradle D, which carried the weight applied to the bearing. The cap, bolts and cross-bar were put together in such a manner as to form a rigid frame, connecting the brass with the knife edge. If there had been no friction between the brass and the journal to tend to carry the brass, and the frame to which it was attached, round with it, until the line through the center of the journal and the knife edge made such an angle with the perpendicular that the weight multiplied by the distance from the knife-edge to that perpendicular, offered an opposing moment just equal to the moment of friction.

Suppose

r = radius of the journal.

s = distance of the knife edge from the perpendicular.

w = the weight.

Then

$$s \times w = \text{moment of friction.}$$

Now the friction at the surface of the journal

$$= \frac{\text{moment}}{r} = \frac{w \times s}{r}$$

$$\text{Coef. of Friction} = \frac{\text{Friction at surface of journal}}{w}$$

$$= \frac{s}{r}$$

In order to read the value of the coefficients thus obtained, a light horizontal lever L was attached to the frame connecting the brass to the knife-edge. It was 62-1/2 inches long or 12-1/2 times the distance between the center of the journal and the knife edge, so that at the end of the lever the chord indicating the coefficient of friction was magnified 12-1/2 times. As a chord 4 inches at the knife edge represents a coefficient of 1.0, a chord fifty inches at end of lever also represents a coefficient of 1.0, while 5 inches represents a coefficient of 0.1, one-half inch 0.01, and one-twentieth inch 0.001. The position of the end of the lever during each experiment was recorded by a tracing point attached to the end of the lever, and marking on metallic paper, carried on a revloving vertical cylinder P. The distance

between two lines obtained by running the axle both ways, when measured on the above sclae, indicated the value of the coefficient. Tower's results obtained by the use of this apparatus were the first reliable results obtained on the value of the coefficient of friction. By using pad and bath lubrication, and running the journals at moderately quick speeds, results were obtained which were closely in agreement with each other, and demonstrated that under some conditions of running, the friction of a journal is extremely small, and is independent of the nature of the contact surfaces.

Thurston's Oil-Testing Machine - These machines are made by a company in England, and are frequently used in England and America. They are convenient, and are easily kept in order; for scientific work, however, they are hardly suitable, but serve for a quick test. Two sizes are made.

A description of the larger size will be given. It has a cylindrical journal against which work two brasses, one above and the other below. Slung from the upper brass is a hollow pendulum containing a strong spring, the upper end of which lifts the bottom brass against the journal, while the lower end depresses the pendulum and helps to weigh the upper brass.

Fig. (2) is a side elevation with the brasses and pendulum in section. The hollow shaft H, driven by a cone pulley B, is mounted on a strong cast-iron stand, the forked ends of which form the bearings upon which it runs. One end of the shaft projects beyond the bearing and is fitted with a brush C, on the outside surface of which the oil tests are made. This enables various metals to be used as bearing surfaces, for the brush can easily and quickly be removed and another put in its place. The box EE, forming the head of the pendulum, holds the two bearing brasses D, on which the pendulum is slung from the journal. In order to enable the operator to vary the pressure of the brasses against the journal at will, the pendulum is constructed as follows: A wrought iron tube H is screwed into the axle box E, which passes round the journal and holds the bearings D in position. In the upper part of this tube is loosely fitted a distance piece O, which, by the action of a strong spring below it, forces the under brass against the journal. The end of, the spring does not press directly against the distance piece, but upon a screwed washer, K threaded upon a rod M, which has a square end, by means of which it can be rotated. At its lower end the spring rests upon a cap I, screwed to the tube H, and by it the stress on the

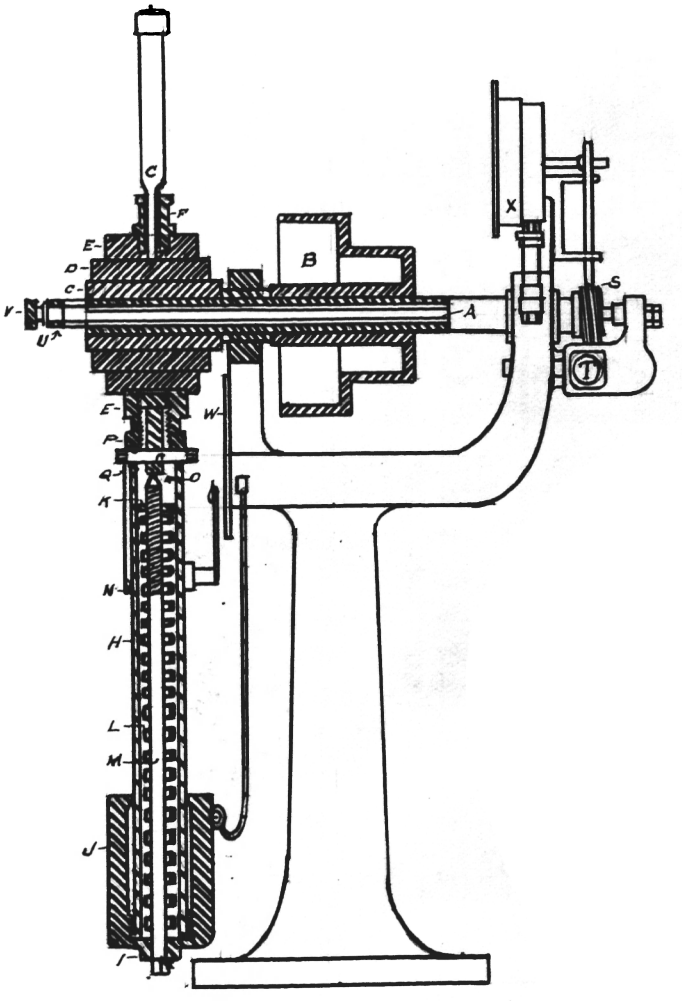

Fig. 2

spring is brought to bear upon the upper bearing D, as well as on the lower one. To compress the spring and increase the load on the bearings, the square end of the rod M is rotated, which moves the washer K, to which is fixed the index finger. A scale, screwed to the tube H, shows the weight per square inch and the total weight on the brasses for every position of K. A ready means of taking the pressure off the brasses is furnished by the nut P, which can be screwed down until the key R, against which it presses, separates the distance piece O from the bearing.

Any friction between the blocks D and journal C will cause the pendulum to move in the direction of the revolution of the shaft and the greater the friction, the further will the pendulum swing.

The lubricant is supplied to the journal by means of pads, which are fixed in the sides of the axle-box. A thermometer C, the bulb of which is let into the metal of the bearing, furnishes a means of ascertained the temperature to which the latter is raised by the friction.

Several methods of calculating the areaof contact of a brass have been adopted. The plan commonly followed had been to multiply the length of the brass (b) by the diameter of the journal ($2r_o$); $2 r_o b$ is then the

projected area of the bearings on a diametrical plane, or as it is commonly called, the nominal area. It is impossible except with light loads, to make a bearing run satisfactorily, when the brass embraces as much as one-half of the journal. In practice the sides have to be cut away until the actual arc of contact is only one-fourth, or one-fifth of the circumference, it is better to substitute actual for nominal areas by multiplying the length of the arc of contact (a) by that of the brass (b).

Olsen's Improved Oil Testing Machine: This machine (See fig. 3) is Olsen's latest improved oil testing machine. The test journal is three inches in diameter and six inches long. The bearing has a projected area of 24 aquare inches, and total pressure of 6000 lbs. may be applied. The machine is equipped with revolution counter, pressure and friction indicator and thermometer, and the bearing has a reciprocating motion similar to other types of oil testers.

To operate this machine, the bearings are first placed in position and adjusted by hand wheel below the bearing and the machine set in motion. The desired pressure is then applied by operating the hand wheel to the right, as indicated by the lower dial. The friction is indicated by the upper dial. The lubricant may be sup-

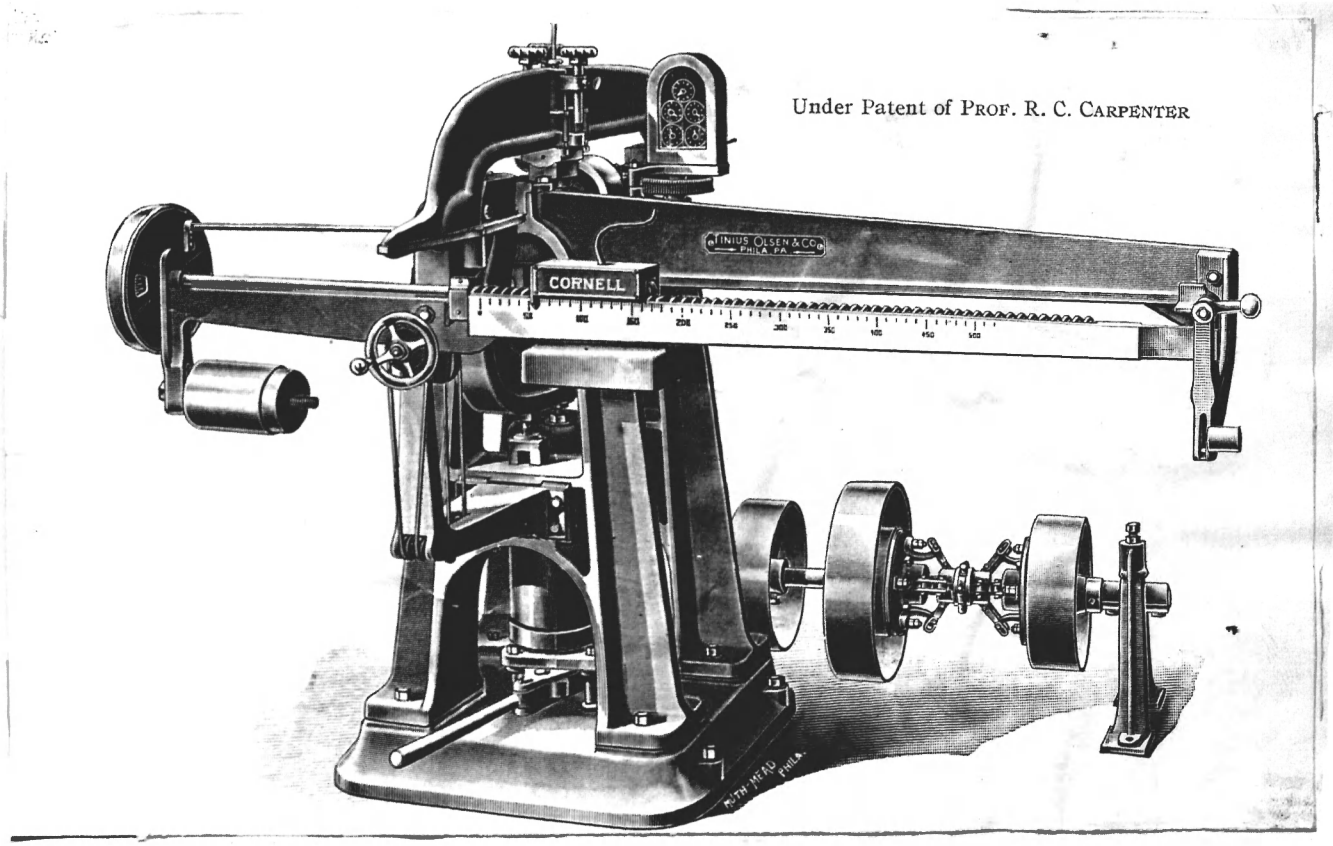
Under Patent of Prof. R. C. Carpenter
JNIUS OLSEN & CO.
PHILA. PA.
CORNELL

Fig.3

plied by drops, or a feed oil cup may be readily attached if so desired. The machine is operated to the best advantage in connection with a two horse power variable speed motor.

The bearing boxes are made hollow and provided with pipe connections for either heating or cooling the bearings, as may be desired, for testing cylinder oils.

The Improved Cornell Oil Testing Machine: This machine is under patent of Prof. R.G. Carpenter (See Fig. 4). Numerous advantageous points are covered by this design, so that either friction, durability, or wearing capacity may be determined, under varying conditions, and a complete test thus obtained.

The journal is 3-3/4 inches in diameter, by 3-1/2 inches long, and the bearing block two inches wide, thus providing a projected area of testing bearing of 7 square inches, on which a maximum of 700 pounds per square inch, may be applied. The journal has a reciprocating motion to prevent cutting and to distribute the lubricant.

The bearing block is only on the upper half of the journal, thus providing the best of lubricating facilities, and of duplicating results. The lubricant may be either fed to the bearing, by drop, or by flooding, or any special lubricating device may readily be connect-

Under Patent of Professor Rolla C. Carpenter

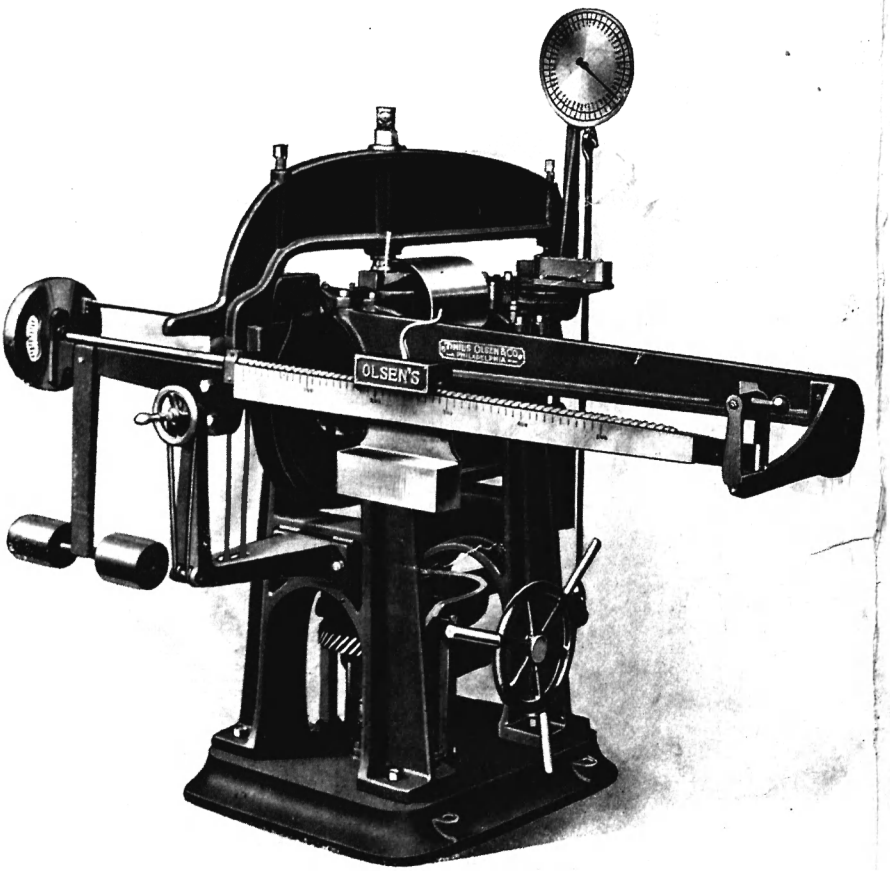

Fig. 4

ed, thus giving a test of the lubricator, as well as of the oil, or grease.

Bearing metal, such as bronzes, babbits, etc., may either be made into standard bearing blocks, or more simply in strips, fitted in the bearing block, furnished with the machine. These strips are so arranged that after the test they may be removed from the block, filed away, and tested again at any later date, in direct comparison with any other metal under the prevailing condition of the journal and lubricant at that time.

A wearing test is made by weighing the strips after having them carefully worn to a perfect bearing, then running them 1,000,000 revolutions, and reweighing them. The loss in weight gives the wear, from which the rate of wear of that metal can readily be computed.

In operating the machine, the bearing is first inserted, after which the upper central screw is adjusted by the hand until the bearings on the end of the spindle are flush with the surface of the lower yoke. The roller bearing below the journal shaft should then be so adjusted as to roll and thus support the journal shaft. The pressure is applied by hand wheel at base, of any amount as may be noted on the dial at the top of the machine. The lever system is balanced, so that any reading from zero

to 5000 pounds may readily be read from the dial. The friction is measured directly from the dial vernier screw beam to a maximum of 250 pounds, with great accuracy and sensitiveness. The temperature is taken from the thermometer placed in the bearing and the number of revolutions read from a direct connected counter. The machine is set in motion after bearing has been inserted, pressure is applied, and data taken of the temperature, friction, and number of revolutions every five minutes, the results tabulated, from which the coefficient of frictions may be obtained, and from which curves may be plotted, showing the relation between the friction and the temperature, and the number of revolutions or time . The machine works best when operated in conjunction with a five horse power variable speed motor.

Riehle U.S. Standard Machine for Testing Oils and Bearing Metals: This machine is a modification of the oil testing machine originally designed and patented by Prof. John Goodman, who in turn received his idea from the form of apparatus used by Mr. Tower, which was discussed at the beginning of this section of the treatise.

Description and Operation: The Riehle U.S. Standard (See Fig. 5) machine can be used either to test the wearing qualities of different bearing metals, or the

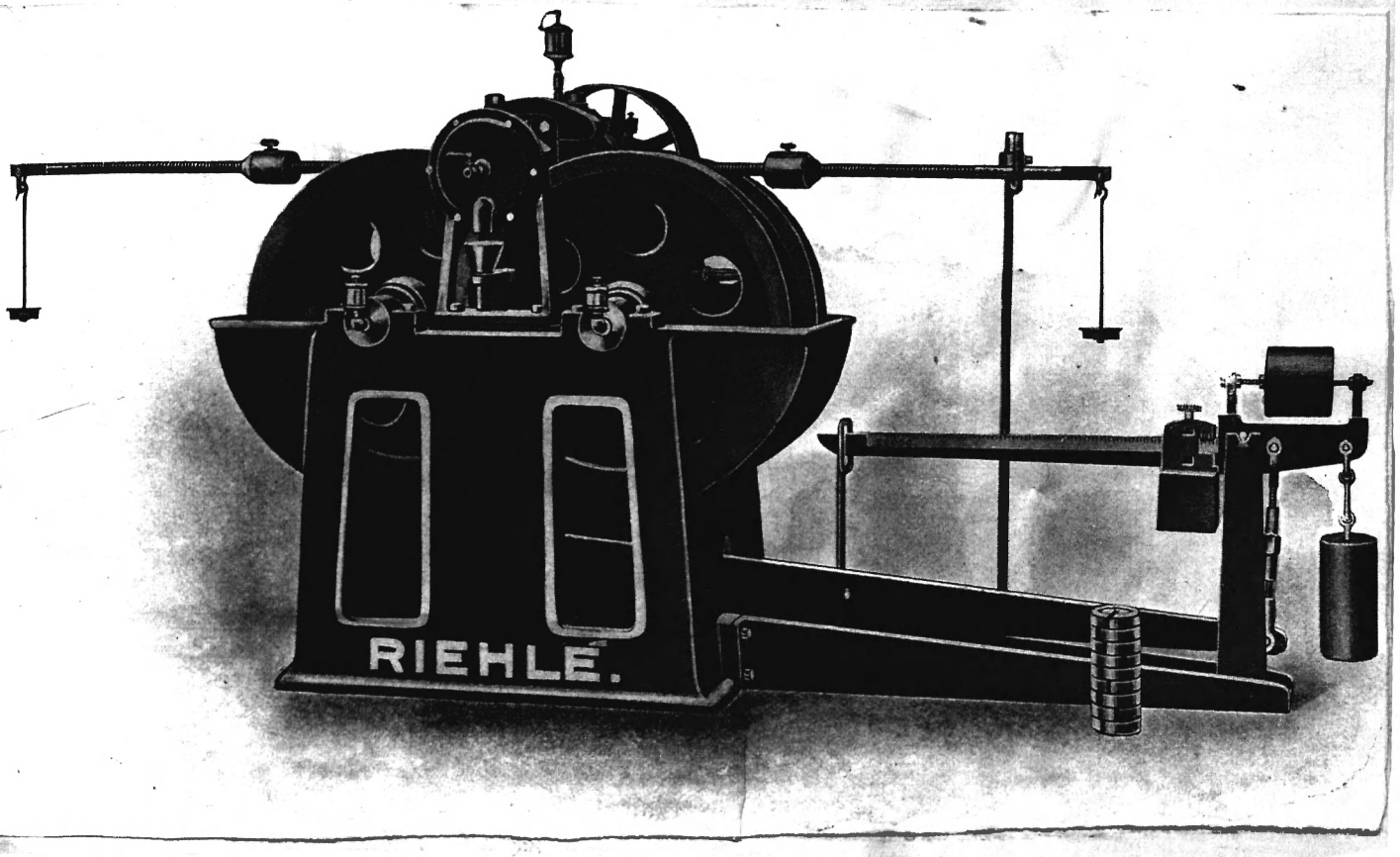

Fig. 5

lubricating properties of various oils. The load on the bearing is applied by means of a turnbuckle connection between the beam and lower lever, and is weighed on beam by large poise. The friction in pounds on the periphery of the journal is indicated by a poise on the upper or friction beam, reading by increments of one pound. The journal of the machine is mounted on four large rollers, which reduce the friction and prevent its heating, which would affect the results of temperature tests. Ball thrust collar bearings prevent side motion of the journal, and take any thrust in this direction which would cause friction.

The bearing to be tested fits in a cap to which the yoke frame is attached: this yoke frame is fitted with two knife-edges equidistant from the centre of the shaft; two clevises join these knife-edges with similar knife-edges in the equidistant lever below, from which connection is made to the intermediate lever and load beam. The yoke frame is thus perfectly free to rotate about the journal, and any tendency to do so will show on the friction beam. The machine is arranged to allow the pulley to drive in either direction.

Riehle Improved Oil Testing Machine: (See Fig. 6) This machine can be used either to test the wearing

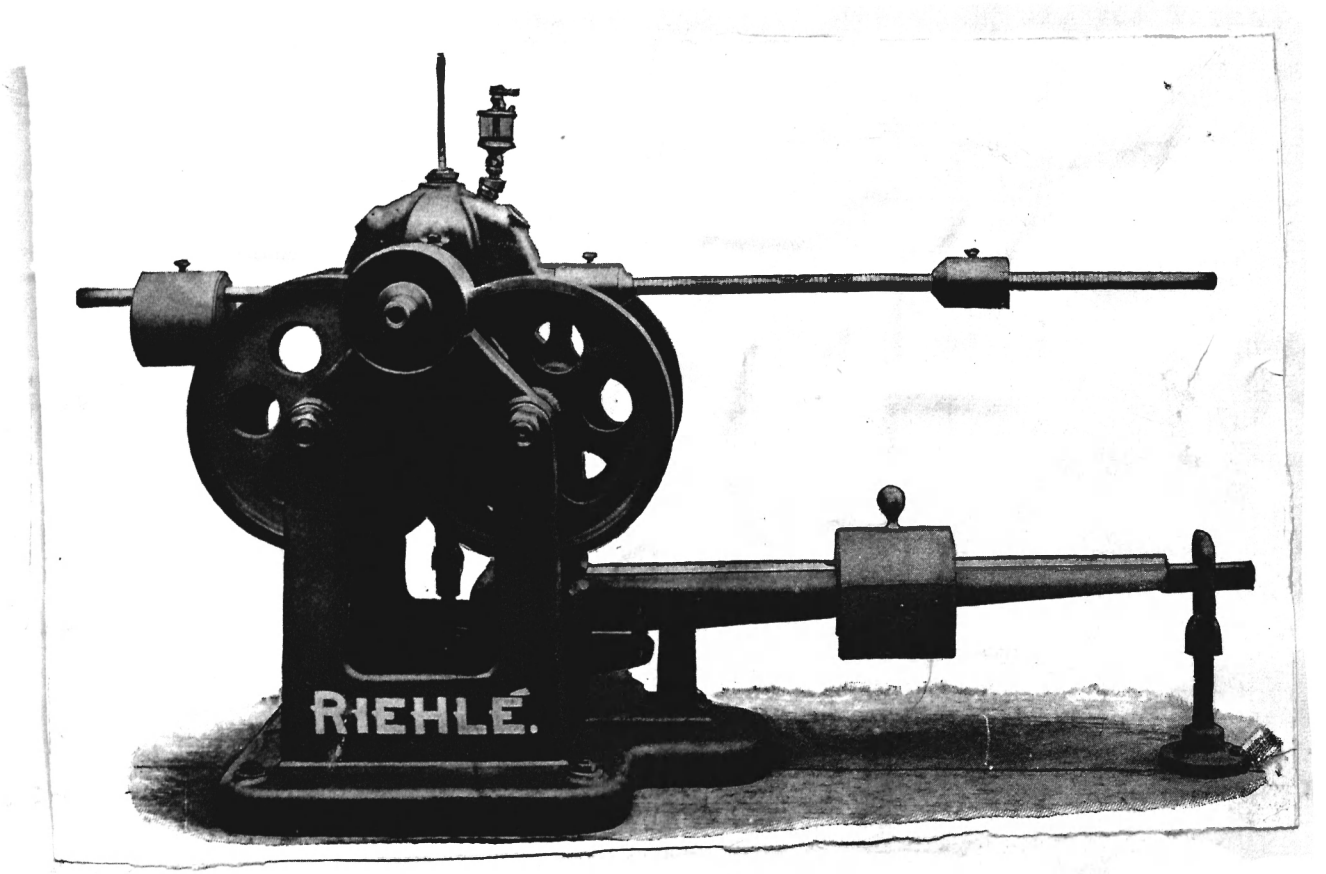

Fig. 6

qualities of bearing metals or the lubricating properties of oils. The load on the bearing is applied by 33-1/3 pound poise acting through two levers, the friction being measured by moving two small poises along the rod which extends from the yoke frame holding the bearing block. The main axle of the machine is mounted on four large rollers, which not only reduce friction, but also prevent any local heating affects the temperature of the axle when temperature tests are being made. The bearing under test fits into a yoke frame having two knife edges, at equal distance from the center of the shaft. Two links join these knife edges with two similar knife edges in the cross head below, from which connection is made to the levers and poise weights. The yoke frame is thus perfectly free to rotate about its center, the friction on the bearing being measured by the movement of the two small poise weights about the center of the axle.

In operating the tester it is important to see that the belt runs away from the weight beam and to the left of the machine, must be free in their action under load, if there is any sluggishness of action of specimen beam, something needs adjusting. All that is necessary after regulating flow of oil, the speed recorded, and weight applied for pressure on specimen, is to balance specimen

beam by moving out small poises; these will indicate the coefficient by the graduations on the beam.

The axle being hollow, water can be circulated through it during test, thus maintaining a constant temperature.

One of the latest inventions along this line is the Sternol patent oil-testing machine, and is being placed on the market by the Stern Sonneborn Oil Company, Limited Royal London House, Finsbury Square, London, E.C., also Glasgow, Paris, etc.

It has been constructed to meet the needs of the most widely varying plants, and to produce the conditions obtained in actual practice. It determines and automatically records the properties which govern the values of different lubricants when in actual use, and it is claimed that it is the first machine really available for the purpose and that it has made clear many hitherto obscure points about lubrication. The speed of the machine can be varied to produce the equivalent of from 50 to 3000 revolutions per minute, that is,a speed of the frictional surfaces of about 5 inches to 25 ft. per second. The pressure can be varied from one pound to 750 pounds per square inch and the temperature can be raised from that of the frictional surfaces when at rest to about 450 de-

grees centigrade. This is more than equivalent to the heat of dry steam produced under the highest pressures.

A special feature of the apparatus is that, by using it with a high pressure steam boiler or superheater cylinder, oil can be tested under actual working conditions. By exposing the oil to the influence of superheated or saturated steam for any period, it determines to what extent, if any, disintegration of the oils and formation of sediment takes place. After testing, the cylinder oil is, by means of steam or hot air, blown on to a slip of paper, on which is clearly seen the condition of the oil, and the changes it has undergone through the treatment to which it has been subjected.

After this the oil is again tested in the machine itself, in order to determine what change has taken place in its lubricating value, through the influence of heat. On a slip of paper the machine indicates automatically the degree of friction and the temperature of the frictional surfaces, and ascertains the absolute , as well as the relative values of the lubricants. In this way a standard is established which enables the values of the frictional curves to be exactly expressed in figures.

A general view of the apparatus in perspective is shown in Fig. (7) and details in Figures (1) to (19)

MACHINE FOR TESTING LUBRICANTS.

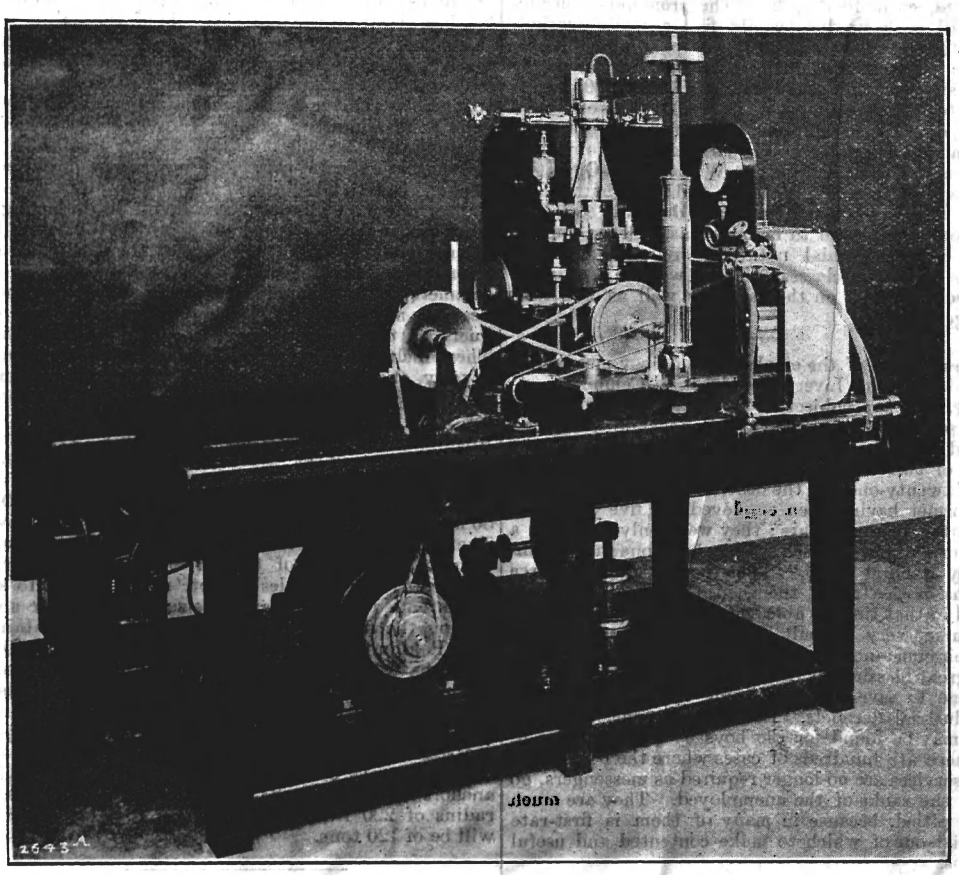

FIG. 1.

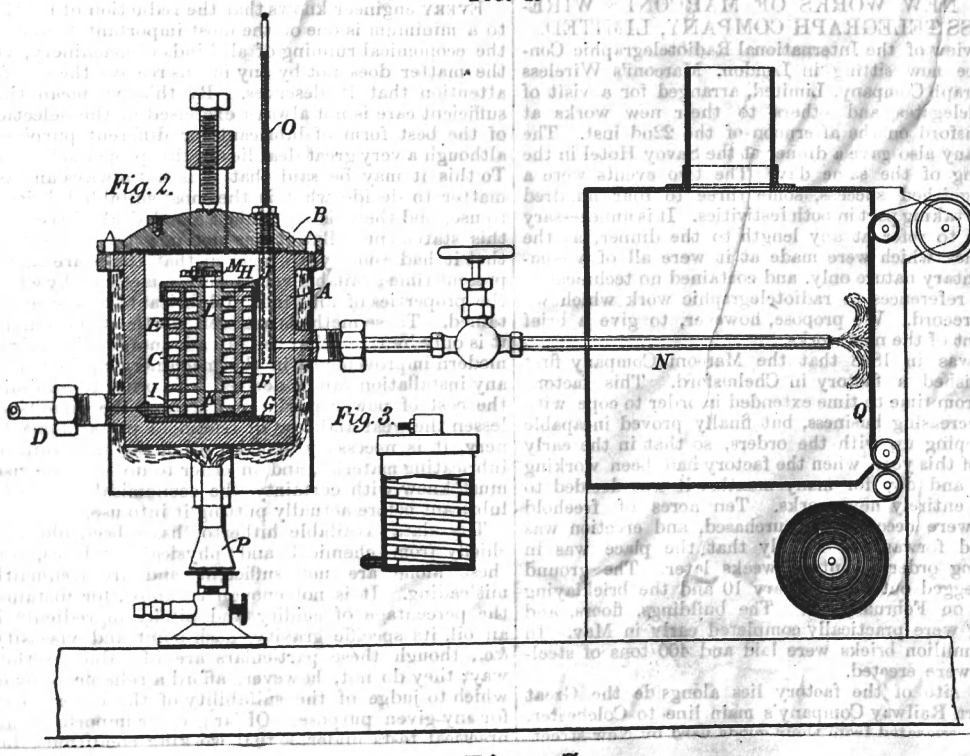

Fig. 7

which are taken from "Engineering". The attachment for testing cylinder oils under dry heat and wet and dry steam is shown in Fig. (2), consists of a vessel H with a cover B, the vessel H containing another vessel C, which i is in communication at the bottom with the pipe D, as shown through which steam is admitted, when the oil has to be tested in this way. Inside the vessel C are two cylindrical parts E and F, which form two spiral ribs on the outside, E being made to fit closely within the vessel C, and F within E. The vessel C stands upon an asbestos pad G, and has on it a cover H, which is screwed on. The vessel C is shown in part section with its cover in Fig. (3). The oil under test is fed into the outer spiral chamber at I, passes up the spiral and through the opening J into the spiral F, which it descends, going out at the bottom to the opening K by the central bore-hole L, whence it excapes into the chamber H, through the opening M, and out of the chamber H into the pipe N. According to the nature of the mixture of steam and oil any constituent from the latter tending to deposit settles on the surface of the spirals, and the amount and natureof such deposit may be ascertained by removing the covers B and H, and withdrawing the spirals, while at the same time, the length over which such deposits ex-

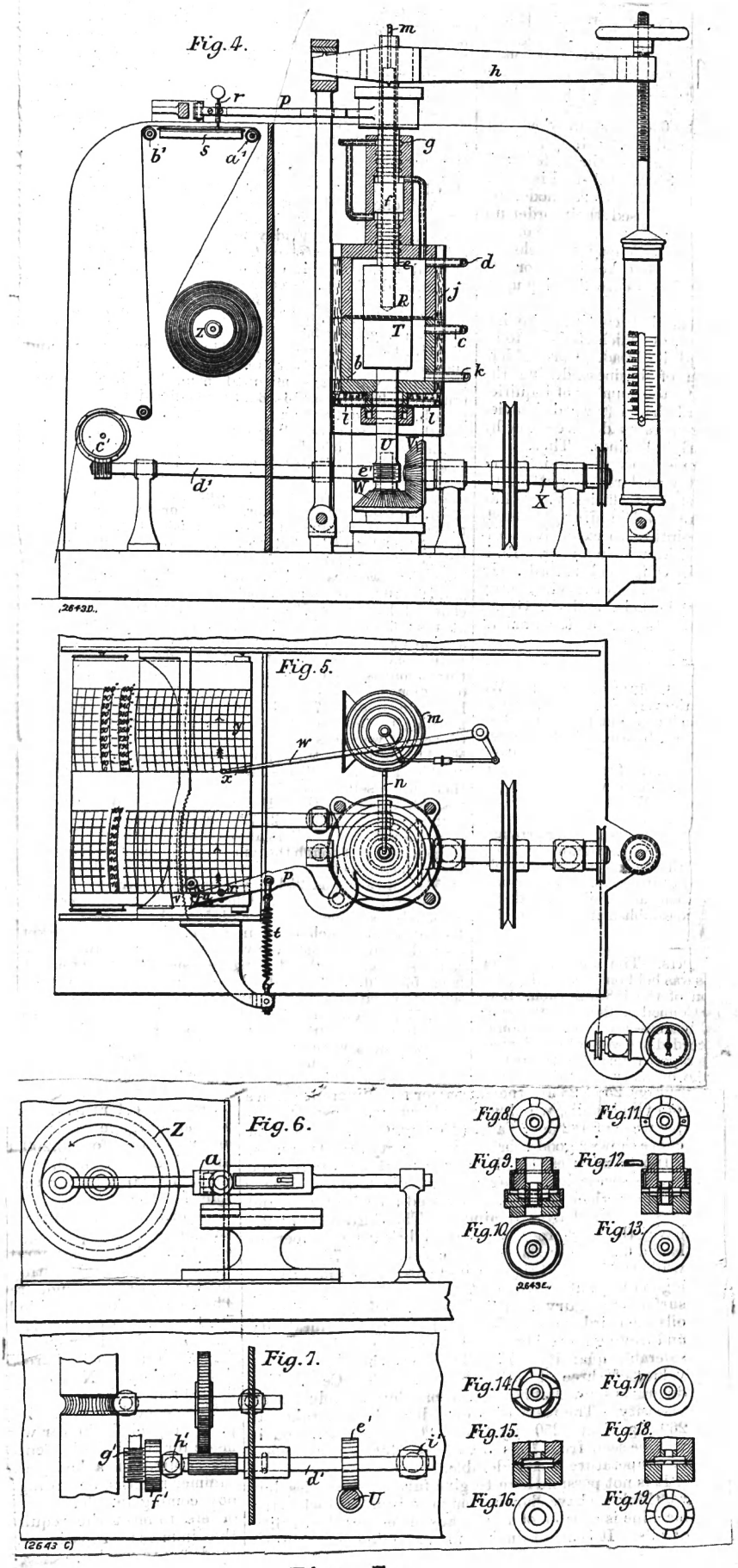
Fig. 4.
Fig. 5.
Fig. 6.
Fig. 7.
Fig. 8
Fig. 9.
Fig. 10
Fig. 11
Fig. 12
Fig. 13
Fig. 14
Fig. 15.
Fig. 16.
Fig. 17
Fig. 18.
Fig. 19
(2643 C)

Fig. 7a

tend may be readily seen, and, if desired, the deposits may be removed and subjected to further tests. The temperature of the steam is indicated by the thermometer O, and any desired temperature may be kept up by means of the burner P. After passing through the vessel H, the steam and oil can be blown through the tube N on to a slip of paper O , on which is shown the condition of the oil after treatment.

The part of the machine in which the frictional tests are carried out is shown in Fig. (4). The two parts R and T form the two friction members. They have annular surfaces, between which lubricating oil or fat is fed in any suitable manner, such as by raising the upper member R and supplying oil between the surfaces, or by providing an oil pan R from which oil may be fed to the surface through a small hole. The lower friction member T is mounted on a vertical spindle driven by the bevel wheels W and V from the pulley, which is driven by same source of power, means are also provided whereby the spindle U can be made to rotate a short distance one way or the other by means of the mechanism shown in Fig. (6) and in the background in Fig. 1, which is driven by the rope pulley Z. From this pulley reciprocating motion is given to the cross head a, which motion is transmitted

to the end of a lever keyed on the spindle U, Fig. (4), but not shown in that figure. The friction members R and T are surrounded by a casing b, in which there are the steam inlet and outlet connections c and d. The upper friction member is carried on a vertical rod e which passes through a gland on the casing. This rod has an enlarged part on it at f, forming a piston, and a small pipe connects the steam space in the casing b with a small annular space g above the piston, as shown in Fig. (4). The rod e is loaded by means of the lever h held down at one end by the spring balance, by means of which a varying load can be put upon the frictional members, H, jacket j surrounds the casing b, and means are provided for cooling the lower friction member T with water, if desired, for which purpose a pipe k leads to the interior of the casing b, and there is an overflow pipe which is not shown.

The pipe k may, if desired, be used as a run-off pipe for condensed steam. When required, the casing b can be heated by means of the Bunsen Burner C. The instrument is also fitted with a recording apparatus, which is shown in figure 5, on which there is a spiral spring thermostat m, the thermometer tube n leading therefrom through the rod e to the friction members. The rod e is connected by means of a lever p with the pencil r,

which works over the recording arrangements s, Figs. 4 and 5. In connection with the lever p is a spring t, which is put in tension by the movement of the lever. On the lever p there is also a pawl u, which engages with a tooth rack v, and prevents a return movement of the lever p until released by the disengagement of the pawl. The thermometer is connected with a pencil x, carried on the lever w, which works over a recording device y. The strips on which the records are so marked are moved simultaneously and uniformly by mechanisms as shown; and, if desired, the records can be taken on one broad strip. The strip is fed from the roller z, and passes over the guide rollers a' and b' and the winding-up roller c', which is driven by worm-gearing as shown. When the spindle U has a to-and-fro motion given to it, the spindle d'(Figs. 4 and 7) is shifted sideways, so that the worm wheel e' is out of engagement with the worm on the spindle U (see Fig. 4) while the worm-wheel f' comes into engagement with the worm g' on the driving spindle of the rope pulley Z, Fig. 6, for which purpose the spindle d' is adjustably mounted in bearings. The operation of the mechanism is as follows: When the spindle X is put in gear with the spindle U, the testing of the lubricating oil between the friction members R and T can take place under the continuous rotation

of the member T. Owing to friction the member R is caused to rotate with the member T, and through the lever p acting against the spring t (Fig. 5) the recording pencil r describes a curve on the recording strip. The friction member R will, of course, be rotated more or less with the member T according to the nature of the lubricating oil being tested. If the spindle d' is moved as before described, and the rope pulley Z be put in gear with the spindle U, an ascillatory movement is given to the friction member T, through the mechanism shown in Fig. 6, and the properties of the oil under test can be ascertained under this motion, the member R being rotated, more or less, according to the friction, the return movement of R being prevented by the pawl u being allowed to engage with the rack V. By this means the lubricant can, if desired, be tested both for rotary motion and for oscillatory motion, and the pressure can be varied by means of the spring balance. In addition to this, steam pressure can be applied during the test, the steam being enclosed in the casing b or passed through the same. Tests may also be carried out while the lower member T is cooled by water admitted by the tube k, the water running off by the tube C. This method is adopted if tests are made at a constant temperature.

As a machine for testing the quality of oils, it is claimed it has no equal. A full discussion of the above machine may be found in "Engineering", from which the above discussion was taken.

Another machine of foreign make is Hislop's Oil-Testing Machine. The machine is in use at the works of the British Oil and Turpentine Corporation, Limited, Excelsior Refinery and Wharf, Hayes, Misslesex. The machine is very much on the same principle as those manufactured by the Riehle Bros. Testing Machine Co. and the Tinus Olsen Manufacturing Co., so that a description of it is unnecessary.

In conclusion of this subject of methods of testing or means of studying the coefficient of friction, there is no machine that has been devised so far that is capable of testing lubricants under actual conditions. In the first place, the conditions under which bearings run are so diversified that it would be impossible to get one machine to correspond to all these conditions. In the second place, the necessity of measuring the friction makes it difficult to construct a machine which will operate as an ordinary bearing.

CHAPTER IV.

INVESTIGATION OF THE PROPERTIES OF LUBRICANTS.

As has been stated before, the object of lubrication is the reduction of friction between moving surfaces, and a lubricant should show therefore, the following characterisitcs:

1. Sufficient "body" to keep the surfaces, between which it is interposed, from coming into contact.

2. The greatest fluidity consistent with (1).

3. A minimum coefficient of friction.

4. A maximum capacity for receiving and distributing heat.

5. Freedom from tendency to "gum" or oxidize.

6. Absence of acids and other properties injurious to the materials in contact with it.

7. High vaporization and decomposition temperatures and low solidification temperature.

8. Special adaptation to the conditions of use.

9. Freedom from foreign matters.

The properties which ought to be taken into consideration, when it is desired to form an opinion as to the suitability of a given lubricant for a particular

class of work, which are as follows:

1. Their identification and adulteration.

2. Density.

3. Viscosity.

4. Gumming.

5. Decomposition, vaporization, and ignition temperatures.

6. Acidity.

7. Coefficient of friction.

A common proceedure, when making a selection of a lubricant is to invite tenders to a certain specification, some of which are extremely vague.

The following is a good average: A pure hydrocarbon oil, steam refined and charcoal filtered, and of a good body. Flash point not less than 500° F., viscosity at 60° F., say 3.00 Carpenter. The oil must be free from acid, wax, fatty matter, and light hydrocarbons. Evaporation at 350° F. for two hours not to exceed 0.2 per cent and no change in the oil to be apparent after twenty hours exposure.

Out of possibly twenty oils submitted fifteen are up to the specification, the most exceed it. Why do they then not all give the same results in actual work?

It will be seen that in the specifications not

one test refers directly or indirectly to friction or friction reducing qualities, not one refers directly or indirectly to steam, not one refers in any way to friction reducing properties under steam, but this is just what is wanted. Oils with the best friction reducing properties for steam engines under various conditions.

The question then that comes up is, what then is the distinctive quality, property or feature which gives to an oil its individuality or essential value as a lubricant for a steam engine under certain conditions.

The tests that usually influence the choice or purchase of lubricating oils are the specific gravity, viscosity, cold test and flash tests, the most importance being given to the first two properties.

The viscosity as ordinarily determined by determining the rate of flow of 100 c.c. of the liquid through an orifice, and then comparing this rate of flow with that of water at 60° F. gives only the relative viscosity, which is not the true viscosity of the lubricant according to the latest research on this property of lubricants. Viscosity is discussed more fully in Chapter V of this treatise.

Mr. Hearts, an expert chemist, had a special viscosimeter designed so as to test the viscosity of oils

up to 500° F., this being 200° F. higher than any commercial viscosimeter in general use, and this temperature being considered as high as it was safe to go. It was found that at 400° F. the difference in viscosity between widely different oils became less marked and at 500° F. the viscosity became about the same, and they were tried at 600° F., at which temperature the viscosity became practically uniform, there being only a few points or seconds difference between machinery oils, and the heaviest cylinder oils. Therefore, it was thought that there must be some better test than viscosity to differentiate, between qualities of oils. Accordingly the vaporizing temperature and flash point were examined. In all the oils tried, the vaporizing temperature was found to be very much below the flash point, and before the flash point is reached the oil must be heated in most cases at least 150° F. higher than its vaporizing temperature. It is frequently assumed that the vaporizing temperature is an index of the volatility of an oil. But such an assumption was not borne out by these tests. The facts that were established from these tests were that the vaporizing temperature, and the loss by evaporization, are both useless as indices of quality in an oil. Deduction from open flash point and viscosity tests upon these properties bring out results counter to preconceived ideas. The

viscosity curves clearly prove that in spite of such wide variations at 212° F. they all drop very perceptibly at 250° F., at 500° F. they are all practically uniform. The results of these experiments tended to show that no law of high flash following great viscosity existed. The tests also show that all oils lose their viscosity very quickly after 212° F. and approach a minimum viscosity at a certain temperature, after which their viscosity remains fairly constant. It was noticed that all the oils, at 600° F. have a particular uniform viscosity of about 30. It is evident that viscosity at 212° F. is a most unreliable guide and it would be difficult and useless to take the viscosity of an oil as determined in these tests, as an index of its value as a lubricant.

Flash Point and Boiling Point: It might be taken for granted that the higher the flash point, but here again the results of tests run counter to such assumptions, there being no uniformity in the results obtained. Some results showing oils, with the same flash point, having different boiling points and vice versa. These tests were all made from a large variety of cylinder oils.

Formerly cylinder oils were compounded in order to obtain a high viscosity. The refiner had to rely on mixtures of animal and vegeatable oils separated from

petroleum in order to obtain a high viscosity.

But within the last ten years, other varieties of petroleum have been found to yield lubricating oils with superior viscosity and wearing qualities which make it no longer necessary to rely on compounded oils either for use on bearing or in cylinders. This is of special importance with reference to cylinder oils, for it is well understood that the conditions of high temperature and highly heated steam in cylinders lead to saponification of the animal or vegeatable oil used in compounding with consequent corrosion of the cylinder.

Mabery and Mathews in obtaining the results shown in the following pages on the viscosity in hydrocarbon compounds, avoided the errors in methods in which differences in specific gravity, and accurate observations of temperatures are neglected. It was evident that not many of the commercial methods could be used, since the data afforded by those methods are merely empirical, and with no definite relations to a common standard. The Ostwald method was at best suited for the determinations which they desired to make.

In Table One it will be observed that the viscosity increases somewhat irregularly with every increment of CH_2 and that the change is greater with the increase in

TABLE I (20°)

Hydrocarbon.	B.P.	SP. GR.	Sp. Vis.
C_7H_{16}	98° - 100°	.724	.51
C_8H_{18}	125°	.735	.60
$C_{11}H_{22}$	172° - 173°	.747	.96
$C_{10}H_{22}$	174° - 175°	.753	.95
$C_{10}H_{22}$	163°	.745	.89
$C_{12}H_{26}$	209° - 210°	.762	1.25
$C_{12}H_{26}$	212° - 214°	.769	1.49
$C_{15}H_{32}$	158° - 159°	.793	2.79
$C_{15}H_{32}$	155° - 158°	.796	2.75
$C_{16}H_{34}$	174° - 175°	.799	3.55
$C_{18}H_{38}$	199° - 200°	.813	5.97

molecular weight.

In Table II are given the values for hydrocarbons with the same boiling points but members of different series. The greater viscosity of the hydrocarbons, poorer in hydrogen, is clearly shown. In comparing the viscosities of the two hydrocarbons boiling at 294 and 296 it will be observed that the difference is greater than the difference between the viscosities of the two hydrocarbons boiling at 274 - 276 . This shows the influence of a decreasing percentage of hydrogen since in the first set the change is from 2 n plus 2 to 2 n minus 2, whereas in the second set, the change is only from 2 n plus 2 to 2 n. Both viscosity and specific gravity increase with the decreasing hydrogen. Another possible influence, namely, the internal structure of the different hydrocarbons. It is reasonable to assume that the straight or open chain structure of the paraffin hydrocarbon $C_n H_{2n}$ plus 2 behaves differently under the stress of internal forces on which the viscosity depends, from the ring or cylic structure, which must be accepted for the other series, until more is known.

This is plainly shown in lubircation, where the paraffin hydrocarbons are of comparatively little value.

If then, the lower series furnish lubricants

Series	B.P.
$C_n H_{2n+2}$	294° -
$C_n H_{2n+2}$	294° -
$C_n H_{2n+2}$	274° -
$C_n H_{2n}$	274° -

TABLE II (60°)

	Sp. Gr.	Sp. Vis.
296°	.781	10.88
296°	.841	21.23
276°	.775	8.51
276°	.835	15.63

with greater viscosity, the addition of a member of a higher series should give a mixture lower in viscosity.

In Tables III and IV the amounts of solid paraffin hydrocarbons added, were all that the oils could hold in solution at that temperature. No noticeable changes appear in specific gravity, there were material changes in viscosity. Table IV shows that a dimunition in viscosity still holds at a higher temperature, but in a less degree even when a larger portion of the paraffin hydrocarbon is introduced. It is evident that neither specific gravity nor boiling point can be depended on for lubricating value unless the source of oil is known . The method of manufacture has also much to do with relation of specific gravity and lubricating value.

General Conclusions: One conclusion is very clear; that even taking the ordinary tests as reliable standards, no one oil excells in all tests. It is then impossible for a chemist or engineer to say from tests of a sample bottle, "This is a perfect lubricant", for a steam engine of any type, or that a high class oil is quite unsuitable for engines of several types.

Another conclusion seems to be clear: That these physical tests by themselves are not reliable standards or guides, they do not give any real clue to the

TABLE III (20°)

Hydrocarbon	B.P.	Sp. Gr.	Sp. Vis.
a. Penn. distillate filtered (C_nH_{2n-2})	312° - 314°	.868	87.42
b. Same cooled to -10 and filtered	312° - 314°	.868	88.16
c. b + 2.35% solid paraffin			
C_nH_{2n+2} 2% of same B.P.	312° - 314°	.868	82.30
d. Penn. distillate C_nH_{2n} cooled to			
-10° and filtered	276° - 278°	.861	37.57
e. d + 2.5% solid paraffin			
hydrocarbon C_nH_{2n+2} %			
same B.P.	276° - 278°	.860	36.39.

TABLE IV (60°)

Hydrocarbon	B.P.	Sp. Gr.	Sp. Vis.
a. $C_n H_{2n-2}$	294° - 296°	.841	21.23
b. $C_n H_{2n+2}$	294° - 296°	.781	10.88
c. Pa. hydrocarbon $C_n H_{2n}$	274° - 276°	.838	15.63
d. Pa. " $C_n H_{2n+2}$	274° - 276°	.775	8.51
e. C + 5% of d		.831	15.16

TABLE V.

Oil Treated.	B.P.	Sp. Gr.	Sp. Vis.	Time of Test	Temperature Start	Temperature 2 hrs	Temperature Break	Coef. of Friction Start	Coef. of Friction 2 hrs.	Coef. of Friction Break
Hydrocarbon $C_n H_{2n}$	274°-276°	.861	37.57	120"	76°	107°	124°	.02	.01	.015
" $C_n H_{2n-2}$	312°-314°	.868	88.16	150"	80°	130°	185°	.01	.02	.02
" $C_n H_{2n-4}$	228°-230°	.923	94.3	210"	75°	113°	164°	.03	.01	.01
Castor		0.97	104(210)	150"	70°	165°	225°	.04	.025	.03
Sperm		0.94	192(90)	140"	70°	120°	160°	.02	.01	.01
Rope		0.91	108(150)	120"	80°	160°	230°	.03	.02	.05
Cylinder Oil Compounded		0.92	123(212)	75"	80°	180°	210°	.05	.03	.06
		1.89	135(210)	200"	80°	185°	225°	.07	.03	.06

behavior of oils under steam or when in use at high temperatures and pressures. Specific gravity is an unfailing index to many oils if the true standard be known. But it by no means follows that another oil, of the same specific gravity, as sperm oil possesses the same properties, or the same nature, or is suitable for the same work.

Viscosity is generally accepted as a standard of value in classifying lubricating oils, but it is not certain that it is relaible in indicating the durability and wearing qualities of oils differing widely in composition. There is little doubt that a confirmation of viscosity by chemical data and frictional durability tests may be depended on to give accurate information for commercial use.

The real test of lubricating capacity depend on temperature, measure of friction, and time the oil continues to lubricate after it ceases to flow on the journal. It is interesting to observe that the life of the individual hydrocarbon increases with the decrease in hydrogen, and in a similar ratio to the increase in specific viscosity. The most valuable quality of an oil is its ability to reduce friction to the smallest value; of the hydrocarbons compared in the above tables, the one

with the least hydrogen of the series $C_n H_{2n-2}$ seems to show the best efficiency, as it also shows the greatest durability.

The coefficient of friction as used in above tables represents the friction of one pound for each pound of load on the journal which is sustained by the oil in use.

CHAPTER V.

VISCOSITY.

In distinguishing between solid and fluid matter, it is customary to define fluid as a state of matter incapable of sustaining tangential or shearing stress. The definition is only true as applied to actual fluids when at rest. The resistance encountered by water and all known fluids flowing steadily along parallel channels, affords proof enough that in certain states of motion all actual fluids will sustain shearing stress.

Using the term distorition to express change of shape, apart from change of position, uniform expansion, or contraction, the viscosity of a fluid is defined as the shearing stress caused in the fluid while undergoing distortion, and the shearing stress divided by the rate of distortion is commonly called the coefficient of viscosity, or commonly, the viscosity of the fluid.

All liquids exhibit viscosity, although in a varying degree. If a vessel of water be tilted, and then brought back quickly to its original position, so as to set the water in swaying motion, it will be observed that with each movement the amplitude of the movement becomes smaller, and that in a very short time the movement dies

away, and the surface of the liquid comes to rest in a horizontal position. If a similar experiment be made with sperm oil, or castor oil, a much greater resistance to movement will be observed, the swaying movements will not only be much slower, but will also be fewer in number. In all cases the movement ceases sooner or later, and it is the internal friction or viscosity of the liquid which arrests the motion.

A relationship between viscosity and cohesion have been assumed by some, but no connection has yet been clearly shown.

Theory of Viscosity: On account of the value of an oil for lubricating purposes depending very largely upon its viscosity, the laws governing viscous flow will be discussed.

When a fluid lubricant is interposed between two solid surfaces, one of which is in tangential motion and the other at rest, that portion of the fluid which is in contact with and adherent to the moving surface is constrained to move with it, while that protion which is adherent to the surface which is at rest remains motionless. Between the two surfaces the fluid may be regarded as consisting of a series of superposed layers, each moving at a speed proportional to its distance from the solid

fixed surface. This is the simplest form of viscous flow which is illustrated in Fig. 1.

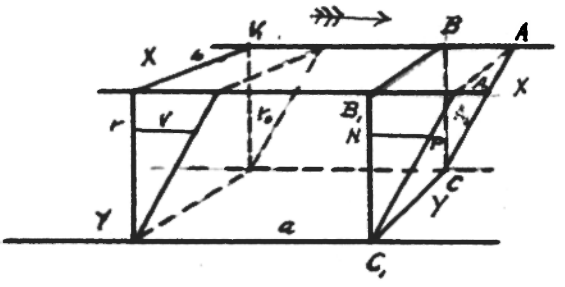

Fig. 1

A stratum of a viscous medium is enclosed between two parallel planes xx and yy , the upper of which is supposed to be moving with uniform velocity in the direction of the arrow, while the lower remains fixed.

The force required to maintain continuous relative motion between opposing plane surfaces, such motion being of the nature of a shearing stress, is measured by the stress per unit area of either of the planes. Thus we may write F = fA, where F is the total force, f the force per unit area, and A the area of the planes over which the stress acts.

Since no stresses other than those transmitted by the shear act on the mass of liquid abr$_0$, any section through it parallel to the two bounding planes is exposed

to the same stress per unit of area as is the liquid in contact with the planes. If A_1C_1 becomes the position after the lapse of one second of a normal line of section, such as B_1C_1 then the inclination of the line A_1PC_1 will be the same throughout its length, and it must be a straight line. Also the length of B_1A_1 represents the velocity of the upper plane, and the length NP the velocity at a distance B_1N from xx. Therefor since

$$\frac{NP}{N_1C_1} = \frac{B_1A_1}{B_1C_1}, \quad \frac{v}{x} = \frac{v_1}{r_0} \text{ ----- } (1);$$

may be written, where v is the velocity at a distance x from yy_1 and v_1, and r_0 are their values at the surface XX.

The stress f is, therefore, proportional to $\frac{v}{x}$ which may be called the rate of distortion, and thus make

$$f = n\frac{v}{x} \text{ ----- } (2);$$

where n is a constant which varies with the temperature for any liquid, and is known as the coefficient of viscosity of that liquid.

$$F = fA = f\,a\,b \text{ -------} (3);$$

$$= n\frac{v}{x}A \text{ ----------} (4);$$

$$= n\frac{v_1}{r_0}A \text{ ----------} (5)$$

$$\therefore n\frac{Fr_0}{v_1A} \text{ ----------} (6)$$

and $v_1 = \frac{Fr_o}{nab}$ ----------(7)

Also the rate of distortion------- $= \frac{v_1}{r_o} = \frac{F}{nA}$ ---(8)

If v_1, A_1 and r_o, are each made unity, then $n = F$, and the viscosity n is measured by the tangential force per unit of area of either of two horizontal planes at the unit of distance apart, one of which is fixed, while the other moves with unit velocity, the space between the planes being filled with the viscous substance.

By the establishment of the above simple working definition equations are formed applicable to all problems involving viscous flow.

Both in the measurement of the viscosity of various liquids, and in the problems which have to be solved in the theory of lubrication, it is necessary to find the volume of the liquid displaced under varying conditions. This is most easily accomplished by calculus. It will be attacked geometrically in this case.

The volume V swept through by any cross section B_1BC_1C in moving to the position $A_1AC\ C_1 = \frac{v_1\ br_o}{2}$.

and this is a measure of the volume of liquid displaced.

Substituting for v_1 its value from equation (7)

$$V = \frac{Fr_1^2}{2\ an}$$

and for F its value from (3)

$$F = \frac{fr_o^2\ b}{2n}$$

When the theory of lubrication is discussed the conditions of viscous flow between planes are all important. Still such conditions are not those under which the coefficient of viscosity n is most accurately and easily measured. The most concordant values have been obtained by measuring the rate of flow through capillary tubes in which, as between plane surfaces where the flow results from the pressure of a given head of liquid, the rate of shear is greatest at the bounding surface.

In a portion of tube of length a, the whole pressure on the cross section tends to shear the liquid at the bounding surface, and the area over which this pressure acts is equal to $2\pi r_0 a$. Now the area of the cross section of the tube $= \pi r_0^2$. Therefore, the total pressure over the cross section $= p\pi r_0^2$ and from $\frac{F}{nH} = \frac{v_1}{r_0} =$ rate of distortion, the rate of shear at the boundary

$$= \frac{p \times \pi r_0^2}{n \times 2\pi r_0 \times a} \text{ which resolves itself into } \frac{pr_0}{2na}$$

In the case of a small cylinder of liquid concentric with the tube and of radius r_1 the rate of shear at its boundary $= \frac{pr_1}{2na}$.

Therefore the rate of shear is directly proportional to the distance from the center line of the tube, and the curve indicating the rate of distortion is, a parabola; and the volume passed in unit of time is the

volume of a paraboloid of revolution, having a base of radius r_0 and a height $= \frac{pr_0^2}{4na}$. The value for the height can be explained by the following figure.

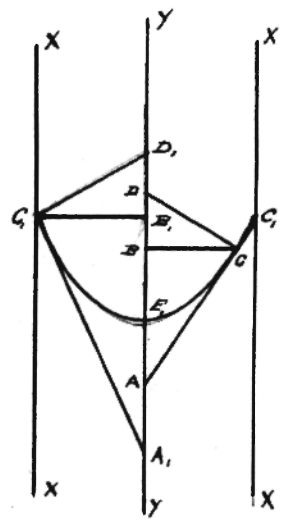

Figure 2

Since the rate of shear is directly proportional to the distance from the center line of the tube or the plane yy in the figure. That is to say, at any point in the liquid the rate is proportional to BC. Now $\frac{AB}{BC}$ is the rate of schear, CA being tangential to the curve at C. Draw CD perpendicular to CA to cut the center line in D. Then $\frac{BD}{BC} = \frac{BC}{BA}$ But $\frac{BC}{BA}$ is proportional to $\frac{1}{CB}$ (see above) Therefore $\frac{BD}{BC}$ is proportional to $\frac{1}{BC}$, or BD is constant. But BD is the subnormal of the curve C, E, C, which is a parabola, and therefore the volume of fluid passed on each side of yy in unit time, is half the

volume of a parabolic prism $C_{,}E_{,}C_{,}$ of length b.

Now since the curve is a parabola $B_{,}E_{,} = 1/2\ B_{,}A_{,}$ and $\frac{B_{,}A_{,}}{B_{,}C_{,}} = \frac{pr_o}{2na}$ from the former discussion,

then $B_{,}A_{,} = \frac{pr_o^2}{2na}$

and $B_{,}E_{,} = 1/2\ B_{,}A_{,} = \frac{pr_o^2}{4\ na}$

Now the volume of a paraboloid of revolution = 1/2 the area of the base times the perpendicular heigtht. Therefore the volume passed in unit time

$$= 1/2\ \pi r_o^2 \times \frac{pr_o^2}{4na}$$

$$\text{or} \quad \frac{\pi}{8} \cdot \frac{pr_o^4}{na}$$

and the volume v passed in any given time

$$= \frac{\pi}{8} \times \frac{pr_o^4\ t}{na}$$

An expression for the viscosity of the liquid can be found from the above, by substituting for p its equivalent in terms of the head and density of liquid, and the force of gravity, i.e.

$$N = \frac{\pi gphr_o^4\ t}{8Va}$$

This is known as Poesenille's formula and may be used, after making a small correction for h, for the purpose of calculating absolute viscosities.

Physical and Mechanical Viscosity. Prof. Reynolds in his treatise on lubrication points out that it by no means follows that for each particular liquid a fixed value of n need necessarily exist. Experiment

shows that liquids have apparently two viscosities; in large pipes the resistance to flow varies as the square of the velocity, and not directly as the velocity. When such is the case n is not constant in value.

Carefully conducted experiments, have proved conclusively that the resistance under certain conditions is proportional to the velocity, and that n has then a constant value. It has also been proved that the volume passed varies as the fourth power of the radius of the tube. This being the case the flow of liquids through tubes of very small bore may safely be regarded as being controlled by the physical viscosity of the liquid and the use of Poisenille's as a means of calculating the value of n when the other values have been determined experimentally. But the fact must not be overlooked, that to obtain a correct result, the size of the tube, the intensity of the pressure producing the flow, and the viscosity of the liquid must bear certain relations to each other. The reasons for which have been successfully worked out by Prof. Reynolds.

Methods of Measuring Viscosity: Before mineral oils were introduced as lubricants, engineers had little reason to study the theory of viscosity or to specify any particular fluidity of the oils they wished to use. But

since the introduction of mineral oils, which can be prepared of any desired viscosity, the measurement of this property and the study of its influence upon lubrication have assumed great importance.

A great variety of instruments have been devised to measure viscosity, but none of the commercial instruments placed on the market, obtain the true viscosity, but only the relative viscosities of the lubricants tested. Such methods are purely empirical.

A true viscosity can be obtained by using capillary tubes and using the relation set up by Poisenille's formula or by the Ostwald method. In Ostwald's method a definite volume of liquid flows through a capillary tube under a definite head. In the calculation, the pressure under which the liquid flows through the capillary, is corrected for its density in the Ostwald formula, which is as follows:

$$N = n_o st / s_o t_o$$

When N equals the viscosity of the liquid examined, S equals the density of the liquid, t the time of out-flow of the liquid, n_o the viscosity of the standard liquid, S_o equals the density of the standard, and t_o the time of outflow of the standard liquid.

The values thus obtained, express the ratio of the viscosity of the liquid under examination to a stan-

dard liquid used for reference. Water is the standard liquid commonly used, and the values of N are referred to as "Specific Viscosities".

The specific viscosities obtained at different temperatures are not comparable, since they express only the ratio at the particular temperature chosen, and take no account of the change in volume of the apparatus, especially in size of the capillary. To compare the results obtained at different temperatures it is necessary to convert the values into absolute units, using the known values for the viscosity of water at the temperature used.

Of the commercial instruments on the market and in use for determining the relative viscosities, the Engler viscosimeter is recognized as a standard in Germany and also in this country by the government.

The Engler viscosimeter is made of metal copper and consists of a vessel for holding the liquid. The vessel that contains the liquid has an orifice at the bottom to let the liquid out by removing the stopper. A water bath is kept around the liquid vessel by means of another vessel, which encloses the liquid vessel.

In using this instrument the viscosity of an oil is stated in seconds required for 200 c.c. of the oil to run into the flask. Water usually requiring from 50 to

53 seconds at 20 C. The viscosity is expressed as a ratio. Thus, if it takes 140 seconds for 200 c.c. of the oil to run out and 52 seconds for water, the ratio is $\frac{140}{52} = 2.69$, the oil thus having a viscosity of 2 x 69 times that of water.

The Carpenter viscosimeter is used quite frequently in this country and differs from the above, in that the liquid flows out so that it is under a constant pressure head all the time.

There are several other types of instruments in use around over the country, which results in a good deal of confusion when the viscosity of an oil is stated, unless it is known what standard they refer to.

CHAPTER VI.

SURFACE TENSION.

Surface Tension: By forces determining the structure of bodies we mean molecular forces. In liquids and solids there must be a force of the nature of attraction holding the molecular together, and a force equivalent to repulsion preventing actual contact. This attractive force is called cohesion when it unites molecules of the same kind and adhesion when it unites molecules of different kinds. The phenomena resulting from such forces are often classed under the heading of superficial tension or more commonly in the case of lubricants the term surface tension is used.

The phenomena resulting from superficial tension are among the most interesting and important that can engage the attention of the engineer and they show themselves in a great many ways. Capillarity, oiliness, greasiness, wetting and emulsification, are either wholly or in part phenomena resulting from superficial tension. The apparent attraction or repulsions exhibited by small floating bodies on the surface of a liquid are due to surface tension. If small fragments of camphor are placed on a clean-water surface they dart about in a very life-like manner. This is due to the fact that the camphor

dissolves slowly in the water, and that the surface tension of a solution of camphor in water is less than that of the pure water. Therefore, if the camphor dissolves a little faster at one side of the floating fragment than at the other side, the surface tension at the first side is reduced most, and the greater surface tension on the one side draws the fragment away.

The surface tension of different liquids vary very considerably, that of mercury being greater than that of water, while the surface tension of water is greater than that of oils.

Theory of Surface Tension: Some reasons have already been quoted in support of the view that molecules of solids powerfully attract each other when brought into close contact. This is equally true of the molecules of liquids, which, in spite of their fluidity, require the expenditure of a considerable amount of energy to separate their molecules completely. It has been calculated that such attractions become perceptible when the distances separating the molecules are reduced to $\frac{50}{1,000,000}$ millimetre. There is every reason to believe that the atoms and molecules never come into actual contact, this attractive force, which is operative over a limited range, must be balanced by a repulsive force of suitable inten-

sity, operative over a different range; therefore the molecules approach each other until their attractive and repulsive forces are in equilibrium. The attractive forces may be accounted for without assuming any attractive force other than that in accordance with Newton's Law.

Conditions of Surface Molecules: It is evident that the molecules in the interior of a liquid are all in the same condition of stress. But the molecules at or very near the surface are in a different position, for the attractive forces are then by no means equal in all directions.

The unbalanced stresses are constantly tending to reduce their number and, therefore, to decrease the surface area. Any increase of the surface area involves an increase in the number of molecules whose attractions are not satisfied, and to effect this, force must be exerted. We thus have all liquid surfaces tending with considerable force to contract their areas and exhibiting the phenomena of superficial tension. Thus a mass of liquid left entirely to the action of cohesive forces, assume a sherical figure, but the fact should not be lost sight of, is that the spherical form is the result of an endeavor of the particles to get as near to each other as possible.

It might be thought that it is impossible to

experimentally demonstrate the existence of surface tension in solids, the behavior of liquids and gases in contact with solids is in accordance with theory. A certain liquid will not act towards all solids in the same way. Water will wet and spread over a clean sheet of glass, but when a drop of water is placed upon a sheet of solid paraffin wax it draws itself up into a bead. Thus it may be concluded that water has a surface tension greater than that of paraffin wax, but less than that of glass. All lubricants have surface tensions much smaller than those of the metallic surfaces to which they are applied, therefore, they spread over them and wet them.

When oils and water are brought into contact with metals the liquid of least surface tension is not necessarily easily displaced by the liquid having the greatest surface tension, for everything depnds upon which liquid touches the metal first.

This agrees well with what is known concerning the actions of water in steam cylinders, for when a large amount of condensation takes place, and the water cannot get away freely, efficient lubrication is almost an impossibility.

Measuring of Surface Tension:-- Consider any liquid bounded by a plane surface, of which the line mn fig. (1) is the trace

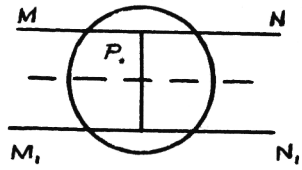

Fig. 1.

And let the line m'n' be the trace of a parallel plane at a distance A from the plane mn. The liquid is then divided into two parts by the plane of m'n'. The general mass of the liquid, and a sheet of thickness A between two planes. Then if we imagine a plane passed through any point within the general mass , it is clear that the attraction of the molecules on opposite sides of that plane will give rise to a pressure normal to it, which will be constant for every direction of the plane; for the number of molecules now acting on the point is the same in all directions. Let, however, the point chosen be P, situated within the shell. With P as a center, and with radius A, describe a sphere. It is plain that the number of molecules active in producing pressure upon the plane through P, parallel to mn, is less than that of those producing pressure upon the plane through P normal to mn. The pressure upon the parallel plane varies as we pass from the mass through the shell from the value which it has within the mass to zero which it has at the plane mn. From this inequality of pressure in the two direc-

tions parallel and normal to the surface there results a stress in tension of the nature of a contraction in the surface.

In every system free to move, movements will occur until the potential energy becomes a minimum; hence every free liquid moves so that its bounding surface becomes as small as possible; that is, it assumes a spherical form. This is exemplified by falling drops of water, and in globules of mercury.

If we call the potential energy lost by a diminution in the surface of one unit, the surface energy per unit surface, we can show that it is numerically equal to the surface tension across one unit of length.

Suppose a thin film of liquid to be stretched on a frame ABCD, see fig. 2.

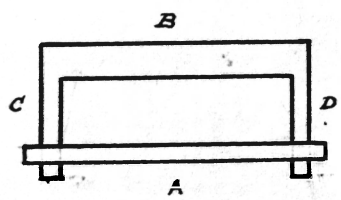

Fig. 2.

of which the part CBD is solid and fixed, and the part A is a light rod, free to slide along C and D. This film tends, as we have said, to diminish its free surface. As

it contracts, it draws A towards B. If the length of A be equal to a and A be drawn towards B over b units, then if E represents the surface energy per unit of surface, the energy lost, or work done is expressed by Eab. If we consider the tension acting normal to A, the value of which is T for every unit of length, we have again for the work done during the movement of A, Tab. From these expressions we obtain at once E = T, that is, the numerical value of the surface energy per unit of surface is equal to that of the tension in the surface, normal to any line in it, per unit of length of that line.

For practical use a metal rectangle is used by supporting it in the liquid by means of a jolly balance. Then pull the rectangle up until the film breaks.

The film in this case, it must be remembered, consists of two tension surfaces, which pass around and grasp the wire frame on all sides.

The relation that is used is:

$$F = T \times 2W.$$

where F = force in dynes that the spring pulls up.

T = surface tension.

W = width of the rectangle.

Calculations of Surface Tension by Rise in

Capillary Tubes: This method is employed quite frequently, the degree of accuracy, however, is limited by (a) the narrowness of the tube, (b) the shortness of the elevated liquid column, (c) the difficulty of measuring the mean height of the liquid, (d) irregularities in the bore of the tube, and (e) the difficulty of securing a clean surface. The method is accurate enough, however, for all practical purposes.

The relation that is set up being

$$2 \quad rT = \quad r\ hdg$$

$$\text{or} \quad T = \frac{rhdg}{2}$$

where r = radius of tube (found with a microscope)

d = density of the oil.

g = acc. of grav. = 980.

A third method is by the drop method, and according to Rayleigh, the weight of the drop equals Mg and Mg = 3.8 a T. where a is the radius of the neck of the drop and T is the surface tension.

CHAPTER VII

EXPERIMENTAL RESULTS.

The relationship between the coefficient of friction, viscosity and surface tension has received considerable attention from engineers all over the country only very recently.

The leaders in this line of research are men working in Cornell University Oil Testing Laboratory.

One of the first difficulties encountered was a quick and easy method, but very accurate method of determining the true viscosity of lubricants. The true viscosity is not given by any of the commercial instruments, now in use, due to the hydraulic friction entering into the results as ordinarily obtained.

Prof. Allan E. Flowers, now at the Ohio State University, but formerly a graduate student at Cornell, seems to have found a method of finding the true viscosity. He has turned his idea over to the Tinus Olsen Co. who are now designing the apparatus for the market. This apparatus is said to consist of a ball rolling down a tube, the ball having a fixed relation to the diameter of the tube.

This idea was used as a basis for the construc-

tion of an apparatus by the author in an attempt at getting at the true viscosity. The apparatus constructed by the author consisted of a brass tube twenty-one-thirty seconds of an inch, internal diameter, and nineteen inches long. This tube was surrounded by another tube about two inches, internal diameter, the ends being closed by slipping brass collars over the ends of the inner tube, and then, soldering these collars fast to both tubes. This outside tube served the purpose of holding a bath so that any liquid in the inner tube could be regulated to any temperature desired. By boring a hole in one of the brass collars, a place was made for getting the bath into the outer tube, and this hole also was used to insert a thermometer into the bath for observing the temperature.

A couple of brass trunnions were next soldered on the outside of the tube, at the middle and at diametrically opposite points. These trunnions served as a means of placing the tube in an iron frame so that the tube could be swung end for end.

A steel ball, a half inch in diameter, was obtained and used inside the inner tube. The idea being to fill the inner tube with whatever liquid that was desired to work with, and then drop the ball down through the

liquid and observe the time the ball takes to get from one end of the tube to the other end. The measurement of this time must necessarily be very accurate, if reliable data is to be obtained.

An electrical method of some kind seemed best suited to the problem that was thus presented. Accordingly for one end a magnet was made, which was to serve the double purpose of being a stopper or cork, and at the same time being a means of holding and releasing the ball as was desired. At the other end a rubber cork was fitted into the tube and two electrical contacts put through this cork. These contacts were connected to an electrical lighting circuit and an ordinary sixteen candle power bulb placed in the circuit. Thus when the inner tube was filled with the desired liquid, the ball inside the inner tube and held at one end by the magnet, the tube was then swung into a vertical position. The ball was then released by the operator throwing a switch, which broke the direct current circuit in which the magnet was in series. The time at the instant of throwing the switch was taken and the instant that it arrived at the other end was known by the lighting of the electric light bulb, the ball completeing the circuit.

Water at 60° F. was used as a standard and

TABLE I

BALL AND TUBE APPARATUS FOR DETERMINING VISCOSITY

Tube: Inside Diam. 21/32". Ball 1/2".

Water	Temp. F.	Time	Rel Vis	Turbin Oil	Temp. F.	Time	Rel Vis	Eng Room Oil	Temp. F.	Time	Rel Vis	Eng Oil	Temp F.	Time	Rel Vis.
18"-1/2	60°	4.5"		18-1/2"	60°	41"	9.1	18-1/2	60°	34"	7.56	18-1/2	60°	132"	29.3
					80°	27.5	6.11		80°	25"	5.56		80°	50"	11.1
					100°	23.5	5.22		100°	22"	4.89		100°	35"	7.77
					120°	16"	3.55		120°	17"	3.78		120°	29"	6.45
					140°	9.5	2.11		140°	9"	2.0		140°	26"	5.77
					180°	6"	1.33		160°	7"	1.55		160°	18"	4.0
					200°	5"	1.11		180°	6"	1.33		180°	13"	2.89
					220°	5"	1.11		200°	5"	1.11		200°	8"	1.77
					240°	5"	1.11		220°	5"	1.11		220°	7"	1.55
									240°	5"	1.11		240°	6"	1.35

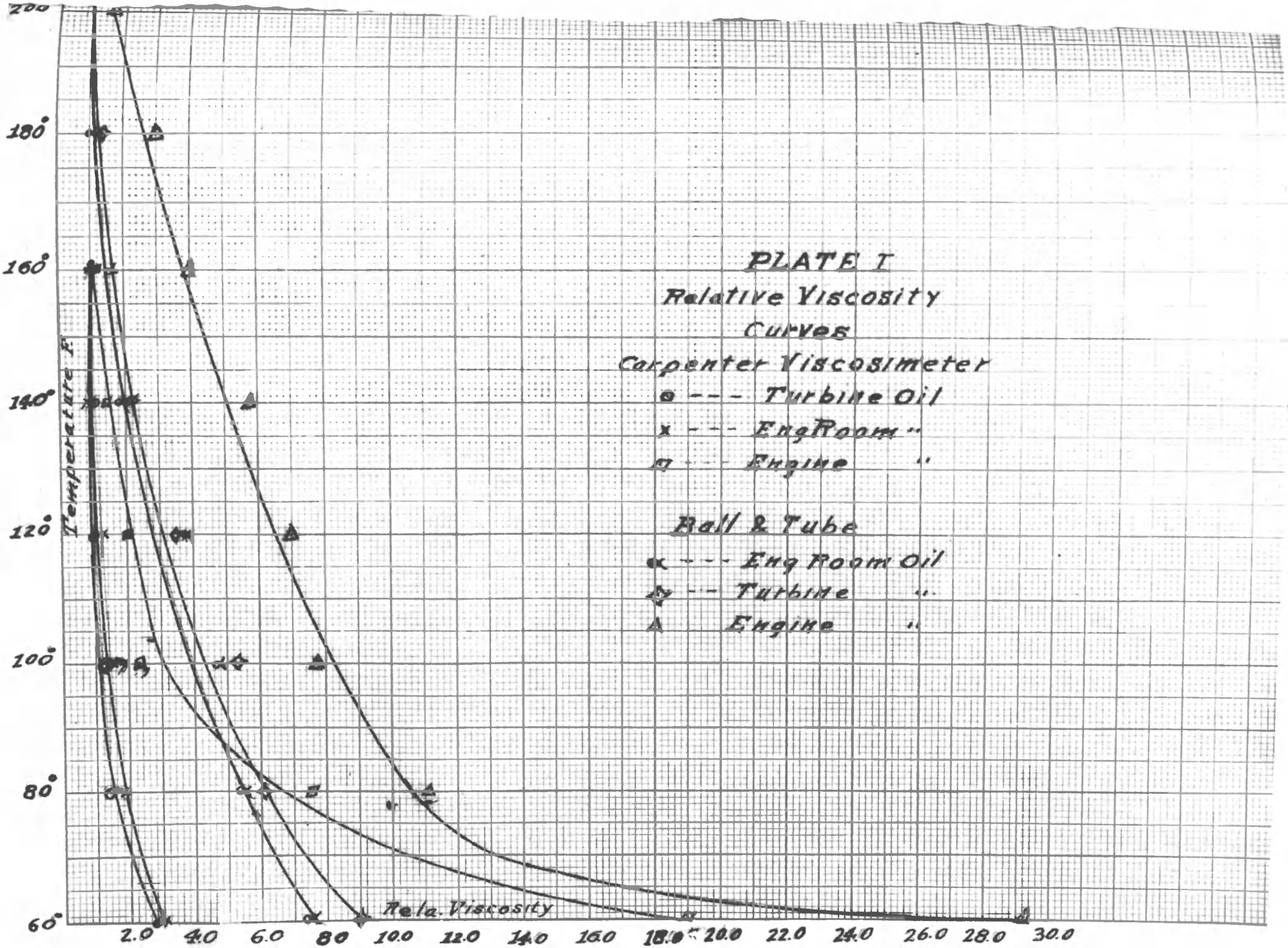
PLATE I
Relative Viscosity
Curves
Carpenter Viscosimeter
o --- Turbine Oil
x --- Eng Room "
□ --- Engine "
Ball & Tube
x --- Eng Room Oil
◇ --- Turbine "
△ Engine "
Temperature F.
Rela. Viscosity
200
180°
160°
140°
120°
100°
80°
60°
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
22.0
24.0
26.0
28.0
30.0

the time observed with the other liquids were referred to this, the ratio being used as the relative viscosity. Table I gives the results obtained by the ball and tube viscosimeter. Table II gives the results as obtained with the Carpenter viscosimeter and plate I shows the results of the two methods plotted and curves drawn in.

The relative viscosities as shown by the curves on Plate I give a much larger value to the ball and tube method, than were obtained by the Carpenter viscosimeter, which was contrary to what was expected. Different sized balls were next tried. It was noticed that in using the first ball, which was a half inch in diameter, that it could occasionally be heard rubbing against the sides of the tube as it dropped through the liquid. This rubbing or hitting against the side of the tube would tend to increase the time reading thus giving incorrect results. A quarter inch steel ball was next tried.

When attempting to time the quarter inch steel ball, when falling through the liquid, it was found almost impossible to do so accurately, as the ball dropped through the liquid at such a rate that timing was impossible. A three-eights inch steel ball was next tried. In this case timing was found to be possible and no hitting or rubbing against the side of the tube

TABLE III

Liq.	Temp. F.	Time	Rel Vis	Liq	Temp F.	Time	Rel Vis	Liq	Temp. F.	Time	Rel. Vis.	Liq.	Temp. F.	Time	Rel. Vis.
Water	60°	2"		Turbine Oil	60°	6.5"	3.25	Eng Room Oil	60°	6"	3.0	Eng Oil	60°	15"	7.50
					80°	5"	2.5	"	80°	4.5"	2.25	"	80°	9"	4.5
					100°	3"	1.5	"	100°	3"	1.50		100°	7"	3.5
					120°	2.5"	1.25	"	120°	2.5	1.25		120°	4"	2.0
					140°	1.75"	.875	"	140°	2"	1.0		140°	3"	1.5
					160°	1.5"	.75	"	160°	1.5	.75		160°	2.5"	1.25
					180°	1.25"	.625	"	180°				180°	1.75"	.875

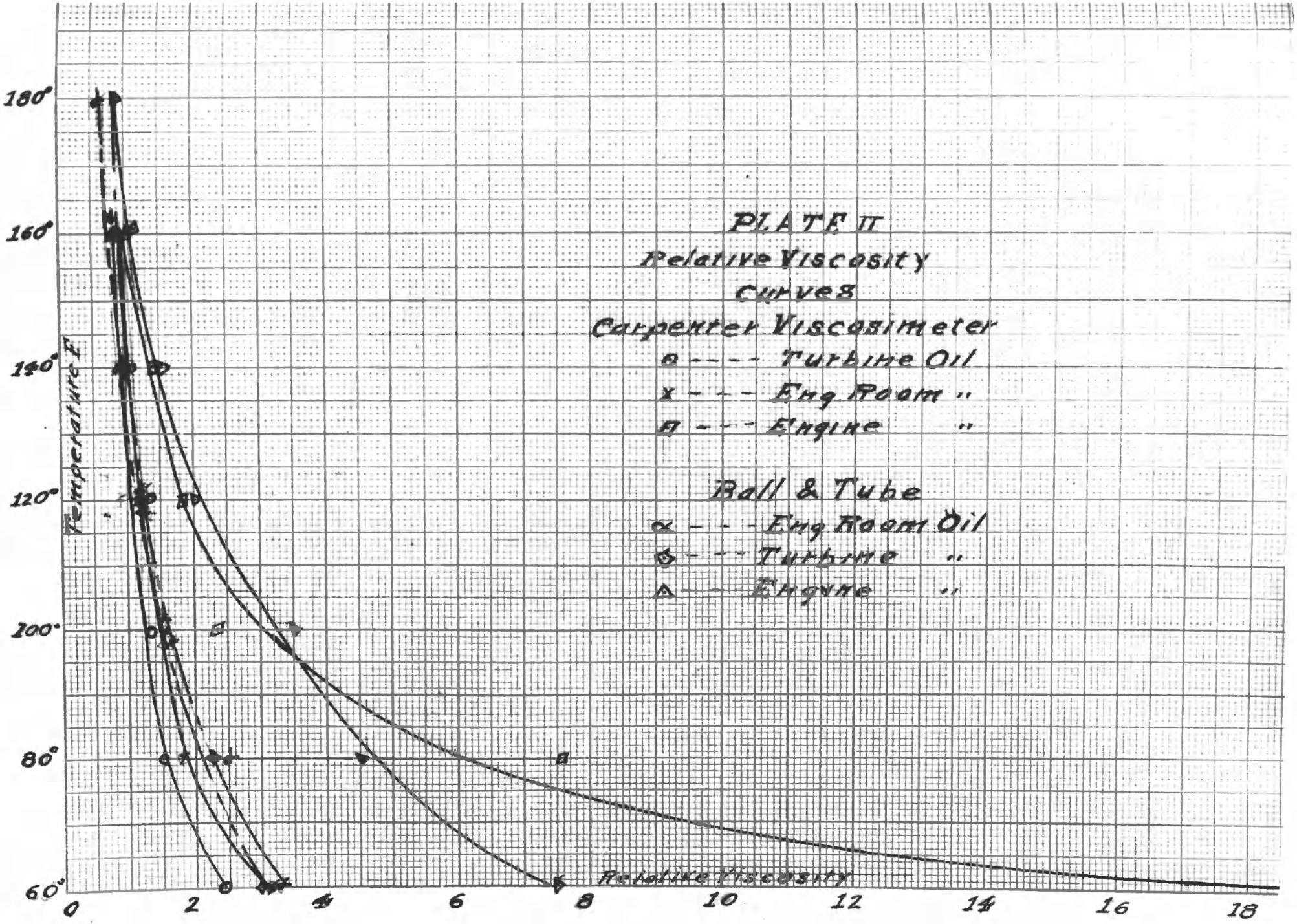
PLATE II
Relative Viscosity
Curves
Carpenter Viscosimeter
o - - - Turbine Oil
x - - - Eng Room "
□ - - - Engine "
Ball & Tube
α - - - Eng Room Oil
◇ - - - Turbine "
△ - - Engine "
Temperature F
Relative Viscosity
180°
160°
140°
120°
100°
80°
60°
0
2
4
6
8
10
12
14
16
18

was noticed.

Table III gives the relative viscosities as determined with the three-eights inch steel ball. The curves of Plate II show how these results compare with those obtained by the Carpenter viscosimeter. But since the relative viscosities as determined by the Carpenter viscosimeter cannot be taken as a true standard, some other method will have to be used in order to get a true standard, so that a correct relationship can be fixed between the diameter of the tube and the diameter of the ball.

The Ostwald Capillary tube method was resorted to but trouble was encountered by the oils worked with being too viscous for the size of the capillary, that was possible to obtain. Then a research was made to find the true viscosities of some liquids at different temperatures, which could be obtained and worked with by the ball and tube apparatus. Some data were found in Landolt Börstein Meyerhoffers Physckalisch Chemische Tabellan. The true viscosities of water at several different temperatures were found and also the true viscosity of glycerine at three different temperatures.

Now from Poisenille's Formula ($n = \frac{gphr^4 t}{8Va}$) which is explained in Chapter V, it is seen that the

TABLE IV

Name of Liquid	Temp. F.	True Viscosity	Density	Time with Ball and Tube Out	Time for 100 cc Flow	Carpenter Vis.
Water	32.6°	0.01757				
"	41.8°	0.01494	.99998			
"	60.0°		.99907	2"		31"
"	69.5°	0.00991	.99806	1.6"		30.8"
"	71.6°	0.00955	.99783			
"	102.7°	0.00662	.99279			
"	131.9°	0.00501	.98565			
"	162.59°	0.003915	.97645			
"	194.0°	0.003165	.96556			
"	212.0°	0.00283	.95866			
Glycerine	37.04°	42.20				
"	60.00°		1.25237			
"	64.85°	10.69	1.25236	41.3"		
"	69.5°	7.779	1.25183	31.9"		469.6"

viscosity is directly proportional to the density, time factor, and a constant depending upon the viscosimeter used. Thus if the true viscosity is known for a certain temperature and the density of the liquid at this same temperature, a constant can be found for the viscosimeter by dividing the true viscosity by the product of the time and density. The constant thus obtained will be of value for only the liquid with it was determined. This method has formerly been applied to viscosimeters in which a given volume of liquid is timed in running through an orifice, but recent research has shown that there is another factor to be taken into account with such a method, - namely hydraulic friction.

The hydraulic friction varies and cannot be taken into account. But with the ball and tube viscosimeter there is no hydraulic friction to be taken into account. The true viscosity can be determined by the ball and tube viscosimeter after having determined a constant for such a method. This was the next step taken.

In Table IV are listed the true viscosities and densities of water and glycerine at several different temperatures. The values for the true viscosities being taken from tables in Landolt Börnstein Meyerhoffers Physical Chemistry tables. The densities for water being

taken from tables in Archbutt's and Deeley's book on Lubricants and Lubrication. The densities for glycerine being determined experimentally.

From the data collected as the data in table IV various objections and difficulties are encountered. The temperature at which the ball and tube viscosimeter was most accurate, the author was unable to find the true viscosity of glycerine. The true viscosities of water were found for quite a range of temperatures, but the time of fall for the ball through water was so short that for water at 60 F., for which the time was two seconds, an error of one fifth of a second in taking the time of fall would mean an error of ten per cent, which is too large an error to disregard.

Using the relation that the viscosity is directly proportional to the temperature, density and a constant, a constant was determined for the ball and tube viscosimeter with glycerine. For a temperature of 64.85, the true viscosity is 10.69, density 1.25236 and the time of fall of the ball 41.3 seconds, which gives the following constant:

$$K = \frac{10.69}{1.25236 \times 41.3} = 0.206$$

and for a temperature of 69.5 F. the true viscosity is 7.779, density 1.25183, and time of fall 31.9 seconds or

a constant:

$$K = \frac{7.779}{1.25183 \times 31.9} = 0.195$$

If we assume an error of nine-tenths of a second in the second case and say that the time is thirty-one seconds, the value for the constat would then be 0.2005, instead of 0.195. But there is just about as much chance for error in the first determination as in the second. An average of the first two determinations would give a value of 0.2005, for the constant of the tube and ball viscosimeter for glycerine. But with only two determinations and chances for large errors in the taking of time, the exact determination of the constant is not very probable with such data. With water the chances for error would still be greater than it was for the glycerine.

What is needed is some kind of a ball that will take about 500 seconds to fall down through the liquid, as then an error of one second will mean an error of only two-tenths of a per cent, which is sufficiently accurate for all work. The true viscosities of several liquids should also be determined by a specially prepared Ostwald viscosimeter.

Prof. F.V. Farragher, of the Chemistry Department of Kansas University has encountered some of the difficulties, that was met with in the ball and tube ap-

ratus, in attempting to time a glass bobin falling through a liquid. If these difficulties can be overcome, this method would be of great practical value to the engineer as well as to the chemist as it would afford an easy and quick method of determining the true relative viscosities.

Surface Tension: Very little seems to have been done in investigating surface tension of liquids. There is little doubt that the surface tension may under certain circumstances, with a very large clearance between bearing and journal have something to do with the oil film. Table V gives the results of a series of experiments carried on with a jolly balance and a wire rectangle to determine the surface tension of the three lubricating oils which have been used throughout, except when prohibited in studying the true viscosities. The curves of Plate III show graphically how the surface tension varies with the temperature. But here again the method of obtaining the surface tension did not seem accurate enough, as it was almost impossible to check results, using extreme care to avoid oxidation affects. The surface tension as obtained experimentally varies directly with the temperature the same as viscosity and it was shown mathematically in Chapter VI that the surface tension was

Name	Temp. F.	S.T. in Dynes	Name	Temp. F.
Eng. Oil	60°	81.4	Eng Room Oil	60°
	80°	77.5		80°
	100°	73.9	"	100°
	120°	73.0	"	120°
	140°	71.3	"	140°
	160°	70.4	"	160°
	180°	70.0	"	180°

TABLE V

S. T. in Dynes	Name	Temp. F.	S. T. in Dynes	Name	Temp F	S T in Dynes
67.3	Turbine Oil	60°	79.8	Water	60°	112.5
66.7		80°	78.6			
65.8		100°	78.6			
65.8		120°	77.8			
65.8		140°	73.2			
65.3		160°	71.3			
65.0		180°	72.2			

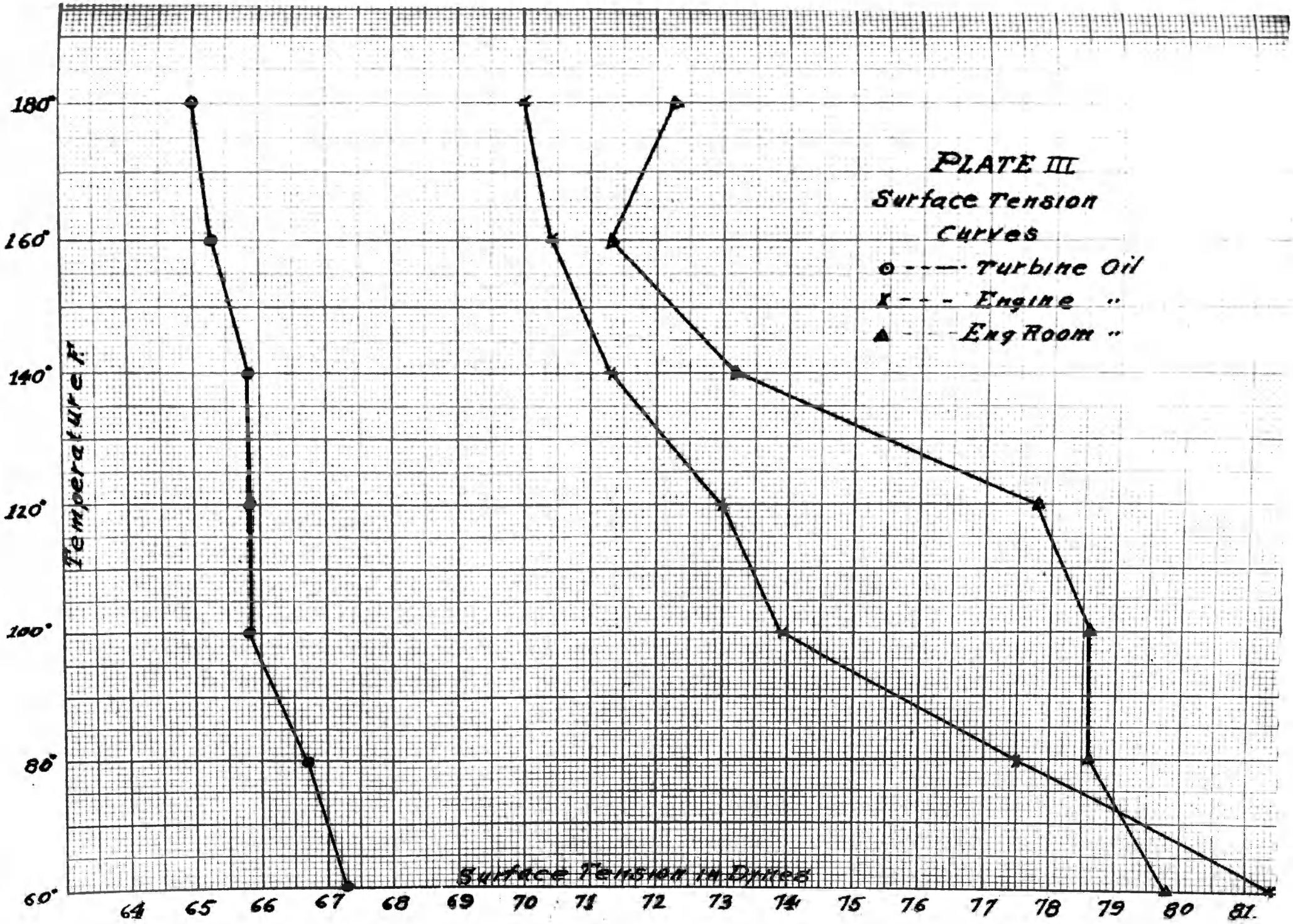
PLATE III
Surface Tension
Curves
Turbine Oil
Engine "
Eng Room "
Temperature F°
180°
160°
140°
120°
100°
80°
60°
Surface Tension in Dynes
64
65
66
67
68
69
70
71
72
73
74
75
76
77
78
79
80
81.

directly proportional to the viscosity factor. But what is needed is some relation between the coefficient of friction and the surface tension constant. These both depend on the molecular action and must therefore be related some way. If after once having the true viscosity some law might be set up by taking the ratio of surface tension to viscosity. If some empirical relation could thus be set up it would be very helpful in getting values for other cases, that is, if a relation exists between surface tension, viscosity and coefficient of friction, one would be able to calculate one of these values, knowing the other two.

Chapter VIII.

RELATIONSHIP OF COEFFICIENT OF FRICTION, SURFACE TENSION AND VISCOSITY

The task of selecting a lubricant for any particular engine or set of bearings, is one of great difficulty, for there seems to be no simple methods of determining some of the properties which it is very important to know in order to make a wise choice.

The tests which it is feasible to perform outside a well equipped laboratory are: viscosity, specific gravity, cold test, flash test, fire test, gumming test, and acidity. The cold test, flash test, fire test, gumming test and acidity test are of no value when it comes to deciding the friction reducing qualities of the lubricant in question. Specific gravity test is an important test when the true standard is known.

Mechanical tests should be made with two objects in view, one being to measure the amount of frictional resistance offered to free motion by bearings lubricated in different ways, and the other to determine the relative oiliness or greasiness of the lubricants.

Various forms of machines have been built and used for testing purposes as was pointed out in Chapter III. The information that these machines were desired

to convey were as follows:

1. Comparative oiliness or greasiness.
2. Frictional effects due to viscosity.
3. Effects of different methods of applying lubricant.
4. Effects of working different metals in contact.
5. Effects of temperature on friction.
6. Effects produced by different loads.
7. Effects produced by varying speeds.

Special designed machines furnished data which enabled Prof. Osborne Reynolds to place the mathematical theory of perfect lubrication upon a firm basis, with the result that the part played by viscosity in the reduction of the frictional resistance between cylindrical journals and their bearings is clearly understood.

What seems to be lacking at the present time is a clear understanding of the part played by the surface tension of the lubricants.

Bearing of Viscosity on Lubrication: As to the bearing that viscosity has on lubrication, there is no doubt that there is a very positive relationship between the working coefficient and the viscosity. The general relation has been shown experimentally from Mr. Tower's experiments and the theoretical relation has been investigated mathematically by Prof. Osborne Reynolds and a

also by Sommerfield. Sommerfield's treatment of the subject is an attempt to simplify the mathematics of the theory of lubrication.

The cylindrical form of bearing surfaces is the most common form and the one with which most experiments have been conducted. With properly shaped brasses resting upon well lubricated journals, Mr. Tower succeeded in obtaining results, which Prof. Reynolds demonstrated were in accordance with the hydrodynamical theory. This theory will not be gone into in this treatise, but may be found in the Philosophical Transactions Royal Society, 1886, p. 173. Only results that Prof. Reynolds and Mr. Tow Tower arrived at will be given.

The equation which Prof. Reynolds derived for this case is as follows:

$$\frac{\mu}{a} = .3635 \frac{M}{RU} \qquad (1)$$

where u is the coefficient of viscosity, a the thickness of the oil film, M the moment of friction, R the radius of the journal, U the velocity.

So long as a $\frac{L}{M}$ is not greater than zero n these approximate solutions are sufficiently applicable to any case; L is the load. For greater values of $\frac{L}{M}$ the solution becomes more intircate and difficult.

In cases of this kind the effects of viscosity

have been determined by Mr. Tower for various speeds above that at which the pressure film is fully formed. The lubricant used was lard oil, the viscosity of which was varied by heating the journal and brass. The viscosity was not specified but Prof. Goodman gives its comparative values at the temperatures stated.

Table I is taken from Mr. Tower's paper with the addition of a column giving the comparative viscosities as determined by Prof. Goodman. By studying the table the experimentally derived results will be seen to be entirely in agreement with the theory or the equation taken from Prof. Reynold's mathematical discussion of this condition of lubrication. The friction at each speed and temperature being as nearly proportional to the viscosity as can be expected, when the experimental difficulties to be overcome are borne in mind. With increasing speed, the temperature of the oil film is raised more and more above that of the metallic surface and the friction ceases to be proportional to the speed.

In the case of plane surfaces of unlimited length and parallel in the direction perpendicular to relative motion; the lower surface unlimited in direction of motion and moving with a velocity U. The upper surface fixed and extending from a distance 0 to a distance (a)

TABLE I

RESULTS OF MR. TOWER'S EXPERIMENTS.

Temp Fahr	Comp Vis	COEFFICIENTS OF FRICTION FOR SPEEDS AS BELOW							
		105 Feet per min.	157 Feet per min.	209 Feet per min.	262 Feet per min.	314 Feet per min.	366 Feet per min.	419 Feet per min.	471 Feet per min.
120	35	.0024	.0029	.0035	.0040	.0044	.0047	.0051	.0054
110	40	.0026	.0032	.0039	.0044	.0050	.0055	.0059	.0064
100	48	.0029	.0037	.0045	.0051	.0065	.0065	.0071	.0077
90	55	.0034	.0043	.0052	.0060	.0077	.0077	.0085	.0093
80	65	.0040	.0052	.0063	.0073	.0093	.0093	.0102	.0112
70	75	.0048	.0065	.0080	.0092	.0115	.0175	.0124	.0133
60	89	.0059	.0084	.0103	.0119	.0140	.0140	.0148	.0156

The formula derived is as follows:

$$F = .6572 \frac{u\ a\ u}{n} \qquad (2)$$

where F is the friction, u the viscosity coefficient, and h, the thickness of the oil film. No experimental data were found for a case of this kind.

It also has been found that the greater the viscosity, the greater a load the bearing will carry without undue friction, resulting from the failure of the pressure film to form properly.

The reduction in the frictional resistance which results from the lowering of the viscosity by the heating of the lubricating film is of no advantage, and accounts in a great measure for the failure of bearings to carry heavy loads at high speeds. The heating effect is the most severe when the film is thinnest.

Bearing of Surface Tension on Lubrication: Prof. Reynold's has proved that the results obtained by Mr. Tower with lubricated journals were due to the viscous nature of the liquids employed, there is yet some properties lacking which lubircation seems to depend upon. Engineers know that when the supply of oil is insufficient, the rate of friction very slow, or the load excessive, lubrication depends upon other less generally understood and somewhat obscure properties. Thurston

states that a liquid, to act as a lubricant must possess enough body or combined capillarity and viscosity to keep the surfaces between which it is interposed, from coming in contact. Although he does not substantiate this statement in detail, there is every reason for regarding it as true, if not taken too literally.

The capillary rise in tubes is a striking manifestation of surface tension and from Poisenille's formula for the flow of liquids through capillary tubes the volume passed in t seconds is

$$V = \frac{\pi r^4 gdht}{8\ na} \qquad (3)$$

where g is the force of gravity, d is the density of the liquid, h is the head of liquid, t is time in seconds, n is the viscosity, a is the length of tube, r is the radius of the tube, and V is the volume of liquid passed in t seconds. The volume which such tubes will supply depends thus, not only on the superficial tension or capillary of the liquid, but also upon its viscosity. The height to which it rises increases with the surface tension and the radius of the tube.

Now to determine the surface tension of a lubricant by the capillary tube method, which is given in Chapter VI of this treatise, is as follows:

$$2\pi rT = \pi r^2 hdg \qquad (4)$$

or $$T = \frac{rhdg}{2} \qquad (5)$$

T being the surface tension in dynes, the other letters having the same meaning as taken above.

Putting both equations (3) and (5) equal to the height of rise, density and gravity constant, we get the following equation:

$$hgd = \frac{8\ naV}{\pi r_o^4 dt} \qquad (6)$$

$$hgd = \frac{2T}{r} \qquad (7)$$

then from the above equations (6) and (7)

$$\frac{8\ naV}{\pi r_o^4 dt} = \frac{2T}{r} \qquad (8)$$

or $$T = \frac{4\ naV}{\pi\ r^3 t} \qquad (9)$$

This last equation shows that the viscosity is directly proportional to the surface tension of the lubricant, and as the results of the experimental work given in Chapter VII tend to show the same thing with the exception of the one lubricant worked with.

Now to set up a relationship between the surface tension and the coefficient of friction by eliminating some common constant from the equations, as was done to obtain equation (9), some mathematical research will have to be made with such an object in view.

That some mathematical relationship may be set up involving the coefficient of friction, viscosity,

and surface tension seems very probable, for it has been shown that in certain cases the coefficient of friction is directly proportional to the viscosity, and also that the viscosity seems to be directly proportional to the surface tension. Thus if the above is true, the surface tension is directly proportional to the coefficient of friction.

What is needed most at the present time is some way of determining the true viscosity of a lubricant in the mechanical laboratory without making an extended research to accompoish such a result, also a means of getting the surface tension accurately and quickly. Then some law by which, if knowing the viscosity and surface tension, the coefficient of friction could be derived, as the frictional reducing qualities is what is needed to be known of a lubricant, along with the other properties that can be readily determined in the laboratory.

Another very interesting problem would be the study of the viscosity, and surface tension of lubricants under different pressures.