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METHODS FOR THE DETERMINATION OF THE LUBRICATING ABILITY OF LUBRICANTS.

The problem of the exact determination of those properties of an oil or other lubricant which go under the name of "lubricating ability" is occupying technicians in increasing measure. The chemical industry, both with and without the use of natural oils, develops by synthetic processes lubricants of widely varied structure, viscosity, stability etc. and renders them suitable for special purposes. As soon as it was recognised that both in the laboratory and in industry it was possible to be independent of the control of natural oil resources and thus of their inherent properties, an abundance of new products appeared. These products possess the general physical properties of a lubricant, but their applicability to technical lubricating processes must yet be evaluated.

There is an equally great number of physical and chemical processes for producing or removing particular important properties. The literature on the subject is so extensive that it is impossible to tabulate it even approximately. The most comprehensive survey can be found in the two-volume report of the Institution of Mech. Engineers on the lubrication conference in London, 1937. (Lubricants and Lubrication). In the meantime new methods developed in Germany have made their appearance. Undoubtedly some physical properties are not accurately measured with all these methods. The question is only to what extent any conclusions thus arrived at are important for the technical lubrication effect proper or are applicable to it.

I. Technical lubricating ability.

The fact cannot be overlooked that lubrication always constitutes a technical process involving the kinematics, design and technological characteristics of the machine parts and mechanisms involved; and this irrespective of whether the minute pin of a high precision gear or the heavy shaft of a large engine are considered. It is always a question of transmitting forces from a moving to a stationary part through the oil film; i.e. a process of transmission of energy which, considering the small quantities and dimensions of the actual loaded oil film, constitutes quite an exceptional strain on these small elements. This results in turbulent thermal, chemical or other intermolecular reactions. So many difficult problems, all covered by the concept of "lubricating ability", arise, that from the practical standpoint division into wide groups is necessary. It is thus possible to cover the most typical cases as first approximation, without neglecting more accurate work on special subjects. The two most frequent fundamental cases are sliding friction in cylindrical shafts or flat contact surfaces, and rolling friction between rolling cylindrical surfaces, with or without superimposed sliding friction. The former is further characterised by continuity of action, the latter by periodicity.

Consideration of these covers the majority of technical lubrication processes. If tests are designed to cover the above two classes, then they will also include the most important varieties of practical conditions, and - what is particularly important - the prevailing type of flow process. It must not be overlooked that all these and in particular the two practical groups seem always to be connected with certain flow processes so that they must also be defined as hydrodynamic. The kind of hydrodynamic stress is of great significance for the carrying capacity and the ultimate stress of lubricants. This applies in particular to the region of so-called

"hydrodynamic friction" and even into the region of boundary friction, though other factors appear in this case.

If the test method is too remote from the type of flow prevailing in practice, the results are not applicable. This is the reason why methods interesting in themselves, as e.g. the four ball method of Boerlage, Thoma-Voitländer or those of Vieweg and Kluge in the P.T.R. and many others give very interesting results, only comparable with the equipment used. They do not however allow a generally useful determination of the practical lubricating ability. They are mainly useful only for purely scientific purposes.

The conventional viscosity measurement in the viscometer has on the other hand a far greater practical significance. In the hydrodynamic equations this is the only physical quantity; and therefore its effect in the region of hydrodynamic floating friction establishes the lubricating ability in this flow condition. Unfortunately, however, the viscosity depends not only on temperature but also on pressure, a fact which leads to insurmountable mathematical difficulties. Moreover, in narrow spaces the hydrodynamic pressure is largely overshadowed by surface effects and variations in the structure of the oil film. This condition of Boundary Friction, at the limit of the carrying capacity of the oil and of its molecular stability, is however in many cases decisive for working security. By analogy with mechanical tests on metals, the principle of obtaining the ultimate stress in order to ascertain the characteristic properties, can be applied to testing the carrying capacity of an oil film. The hydrodynamic lubrication theory shows that two sliding or rolling surfaces get closer to each other, i.e. the film thickness decreases, the greater the load and the lower the relative velocity. For a given load the film has a minimum velocity at which it collapses, i.e. at the limit of its carrying capacity. An accurate appreciation of this condition is worth while. It lies deep in the region of boundary lubrication, i.e. in that phase of lubrication at which no metal wear has yet occurred, though there already appears stiffening of the oil film and a variation in the molecular structure. As proved in numerous tests, the fundamental differences between lubricants and between metal surfaces appear then with particular clarity.

The curvature of the sliding or rolling surfaces is also important. Here the cylindrical shaft in a plain bearing represents one extreme, the tooth flank and the roller bearing the other. The latter allows very high Hertz specific pressures, of 10,000 kg/sq.cm. and more which however occur only periodically at each point; on the other hand in plain bearings, where 1,000 kg/sq.cm. are rarely exceeded, a local state of equilibrium can always be attained.

II. Machines for realistic tests of lubricating ability.

The above considerations led to the development of: 1) Bearing test machine; 2) Gear test machine. Both, originally designed for other purposes, have proved, when correctly used, excellent machines for testing lubricating ability in practical cases.

1) Bearing test machine.

A design of bearing test machine is used, similar to that described in Nücker's report "Über den Schmiervorgang im Gleitlager" (No.352), but of a reduced size for shafts 60 mm. diameter and 400 mm. long (Fig. 1). For maximum rigidity the test shaft is mounted in 3 bearings; the test bearing is fitted in a steel casing, to which the loads are applied by means of weight through a 1:100 lever system. The friction moment on the bearing is measured by an accurately calibrated friction scale. This machine produces without deformations

bearing loads up to 300 kg/sq.cm. and over, referred to the projected area of the shaft. It is driven through a Leonard converter and a Flender gear box, so that the circumferential speed can be varied between 0.05 and 9.5 m/sec. As boundary friction rarely operates at speeds in excess of 3 m/sec., the transition stage into the range of hydrodynamic friction can be easily attained.

The oil is drawn by a gear pump from a tank at a given temperature and pumped under pressure into an oil pocket (Longitudinal groove). Inlet pressure, temperature and quantity are measured constantly; the operating temperature by a thermocouple inserted in the bearing surface close to the pressure peak, which indicates directly the temperature of the oil film.

The Tests are always made in a state of equilibrium after preliminary running-in. According to the kind of bearing material different loads are applied and at constant load the tests are run with varying speed. It is generally sufficient to start with a top shaft speed of 3.14 m/sec. subsequently reducing it to a minimum. The moment of the film collapse is clearly marked by the seizure of the machine and the sudden jumping up of the friction torque. This is often preceded by an uncertain region in which the scale shows signs of unrest. With sufficient foresight it is however possible to cross this region and record the point of minimum film thickness at which the film carrying capacity has not entirely disappeared.

The shape of the temperature curves gives the upper limit, i.e. towards the high speed region. The top limit load, i.e. maximum carrying capacity is that at which the temperature rise (within the measured speed range) shows a definite tendency towards an equilibrium value below the stipulated upper limit of 80°C. If the speed is further increased, the load must be reduced, due to the proportionate increase in friction work done.

Fig. 2 shows a typical graph for a metal bearing with the laboratory reference oil, Rhenania Ossag BC 8. The oil inlet pressure was kept constant at 3.5 atm. and 5 loads of 44.4, 86,2, 127.5, 169 and 210 kg/sq.cm. referred to the projected area of the carrying surface. The bearing clearance is 0.25 mm. on the diameter.

At the minimum load of 44 kg/sq.cm. the μ curve is a good example of the typical shape according to the hydrodynamic laws, i.e. a drop till an inversion point is reached, then a steep rise at the boundary friction up to a minimum final velocity of 0.08 m/sec. If the load is increased, the μ curves become flatter; the boundary friction region expands further and the limiting speed increases. The uncertain region also expands. At the maximum load of 210 kg/sq.cm. the use of low speeds is doubtful.

If it is a case of comparing a group of oils, inlet temperature, oil pressure and bearing clearance are kept strictly constant, and the same shaft is used. Great care is taken to avoid seizure, i.e. actual wear and the same bearing is used throughout. If however the influence of different bearing materials is to be considered, it is advisable to choose two extremely different kinds, e.g. W.M.80 and plastics.

In principle each combination oil-bearing material gives a different diagram; it is however impossible to combine e.g. 8 oils with the much greater number of bearing materials and one or two only are considered.

The measurement of the oil throughout, so called "oil-slippage" (Schlüpfigkeit) of the combination, is also important. The measured oil quantities are plotted in a special diagram (see Fig. 2); they rise nearly linearly with the speed and give almost always the characteristic result, that high carrying capacity combinations exhibit a small oil flow, low carrying capacity combinations on the

other hand a large oil flow.

Fig. 3 shows the results of tests on a series of oils; the μ-values, oil quantities and v_{min}, measured at the minimum speed are compared. This comparison gives characteristic data. A group of 8 oils of similar composition but very different viscosity are compared; fluid ones on the left, viscous ones on the right in order of increasing viscosity (Fig. 6 shows a viscosity curve). It appears that the minimum velocity and the oil quantity fall off as the viscosity goes up, whilst the friction coefficient μ varies comparatively little. It must be noted that it is a question of measurements in boundary friction and the numerical value of μ cannot always be accurately determined. In the present design of bearing test machine the sensitivity of the friction balance mounted on knives depends to a large extent on the load. If however tests are carried out at the same load throughout, the μ-values can well be compared without using the absolute numerical value. Often undue importance is laid on the numerical value of μ both in tests and calculation; the insufficient comparability of the test apparatus and the practical value do not justify this importance.

The comparison contained in Fig. 3 shows qualitative comparative values, from which with some experience the relative properties of this group can be easily deduced. Quantitative data can also be obtained.

Quantitative oil characteristics

If one starts from the assumption that the hydrodynamic laws of the friction theory apply also to the region of boundary friction, as long as the temperature prevailing at the measuring points and the corresponding viscosity are taken into account, it is possible to calculate the width of the minimum clearance resulting from the displacement of the shaft, as well as the displacement angle α_0. If δ is the bearing clearance, they are connected by equation:

$$h_0 = \frac{\delta}{2} (1 - \cos \alpha_0)$$

At the limiting speeds the displacement angles are always very small; here and in the calculation of the minimum clearance, microscopic superficial roughness must naturally be neglected. Another factor which is neglected when the test point lies in the rising branch of the μ curve, is the manifest structural change of the oil film in very small clearances, i.e. the variation of viscosity with pressure. These effects cannot be calculated in advance, which however does not seem necessary as results will show later.

Table 1 contains the measured values of n_{min} and the calculated values of α_0 and h_0 (= width of minimum gap), as well as the kinematic viscosities η at 30°C for a series of 8 oils of given structure. The values of h_0 and α_0 were calculated on viscosities corresponding to the temperatures measured at n_{min}

The following expression was used:-

$$\alpha_0^3 \cos \alpha_0 = \frac{6 \eta R_1^2 \cdot U \cdot (I \text{ const})}{c^2 - P/b}$$

where c is the clearance number $= \frac{\delta}{2}$ and I is the integral of a geometrical function of the displacement angle, which for the region in question assumes a constant value. Instead of the absolute clearance $c = \frac{\delta}{2}$ the relative bearing clearance c/R can also be introduced. This equation is taken from the method developed by the author in book 4 of "Forschung im Ingenieurwesen" 1935, p.161 et seqq.;

it allows the calculation, with comparative ease and accuracy, of the values of α_o or h_o.

Fundamentally small values of α_o and h_o should correspond to small values of n_{min}. In reality however owing to the large η values of the oil group, both quantities increase with increasing viscosity, i.e. wider gaps are permitted by the higher viscosity. E.g. in the lighter load group for W.M.80 at p = 44.4 the bottom limit of the carrying capacity of the film is not attained; with the available driving gear technical difficulties prevent a further speed reduction.

Even for v = 0 and α_o = 0 the film thickness never reaches zero value as in theory; there remains always a definite film thickness, partilularly for oils of high molecular weight and great chain length.

The carrying capacity of the oil film at the critical bottom limit can be expressed in terms of

h_o = minimum gap and

n_{min} = minimum practical speed

as follows:

$$F_t = \frac{\text{Const} \cdot h_o}{n_{min}}$$

where the value of the constant depends upon the apparatus used. If we assume:

$$F_t = \frac{1{,}000\ h_o}{n_{min}}$$

the relative carrying capacity of the oils can be expressed in terms of or referred to a reference oil.

Table II collects the F_t values for the oil group in question; Fig. 5 gives a graphic survey.

This characteristic however does not explain the above-mentioned fact that the oil quantities measured with the various combinations of materials are widely different for given geometrical dimensions of the bearings. They decrease with increasing oil viscosity or molecular weight. It is therefore obvious that the specific oil flow must be included in the coefficient.

Once the minimum gap h_o has been calculated, it is possible to find by a simple conversion the gap width in which the oil pressure attains its theoretical maximum and dp/dy = 0. Pure laminar flow can be assumed, which on the shaft equals U, the peripheral velocity of the shaft and falls off to 0 linearly towards the bearing surface. The oil quantity pumped into the actual carrying gap is:

$$Q^x = \frac{h_o^x \cdot U \cdot b}{2}$$

where h_o^x is the gap width at the pressure peak and b the bearing width. For $h_o^x = 1.226 \times h_o$, b = 4 cm, $U = \frac{6\ \pi\ n_{min}}{60}$ cm/sec = 0.314 . n_{min} (shaft dimensions 60 mm. diameter, 40 mm length) it is

$$Q^x = \frac{h_o^x \cdot 0.314 \cdot n_{min} \cdot 4}{2} = 0.77 \cdot h_o \cdot n_{min}$$

If we now relate the above factor F_t to this expression for oil quantity and form the expression:

$$F_q = \frac{F_t}{Q^x}, \text{ it is}$$

$$F_q = \frac{1000 \cdot h_o}{n_{min} \cdot Q^x} = \frac{1000 \cdot h_o}{0.77 \cdot h_o \cdot n_{min}^2} = \frac{1300}{n_{min}^2} = \frac{Const}{n_{min}^2}$$

if the constant 1000 is maintained.

Thus the calculated quantity h_o is eliminated and the new factor $F_q = \frac{1300}{n_{min}^2}$ is based only on the measured quantity n_{min}; it is inversely proportional to its square. It expresses the carrying capacity characteristic reduced to the specific oil consumption. Naturally it has only a relative meaning in comparison with a given reference oil. Its structure however is exceptionally simple and it is based on a measurable value. This clarifies the importance of the minimum speed, directly connected with the film failure, for the carrying capacity of the oil.

It is therefore independent of the total oil quantity flowing through the bearing, which is also measured, though it is difficult to relate to the geometrical dimensions owing to the three-dimensional character of the flow.

In Table III are collected the calculated F_q values. The figures marked with an asterisk indicate that at the prevailing low loads the carrying capacity could not be fully evaluated because it was technically impossible to reduce the speed further. With W.M.80 and for p =127 kg/sq.cm. on the other hand, the characteristics are fully expressed.

In Table IV all figures for W.M.80, p = 127.5 and synthetic resin plastics with p = 52.5 are referred to oil 2 as reference oil; the difference in carrying capacity is clearly shown.

The difference in results between the combinations white metal-steel and synthetic resin-steel is due to the different "affinity" of each bearing material for the shaft, which can only be explained by molecular physics and not by hydrodynamics. Synthetic resin plastics have a much lower load limit owing to their bad thermal conductivity and temperature sensitivity.

Fig. 4 shows by how much the oil "slippage" of the combination synthetic resin-steel exceeds that of white metal-steel. For synthetic resin the oil pressure had to be reduced to 1 atm. against 3,5 atm. for white metal, in order to attain approximately the same oil quantities with the same clearance and temperature. As the oil quantity pumped in the bearing gap proper, i.e. between h_o and h_o^x amounts only to a small fraction of the total, it can be assumed that the total flow is roughly proportional to the square root of the inlet pressure. If we reduce the oil quantity for W.M.80 by 3.5 at p = 44.4 we obtain the bottom curve of Q_{min} which shows how widely the oil slippage of the two combinations differs. This applies to all oils, as well as to alloys other than W.M.80. This fact observed in numerous tests cannot be explained hydrodynamically. It is presumably based on the fact that the orientation of the oil molecules after

pumping under pressure into the narrow gap has a stronger effect than the boundary surface forces (adhesion forces) which, at least in free flow, were so far assumed effective to a distance of only 1 or 2 molecule lengths.

Summing up it can be said: the above quantities F_t and F_q, considered as comparative quantities, give a faithful picture of the carrying capacity measured with realistic and closely approximate conditions of sliding friction.

If we plot quantities F_t and F_q, after conveniently reducing their scale, on a graph for the various oils (Fig. 5) and the corresponding viscosity, we find that the variation of the characteristics for this type of stress follows very approximately the viscosity values. (These are taken at 30°C, i.e. at the same inlet temperature; the temperatures obtaining at n_{min} differ only slightly from it).

This agreement confirms the accuracy of the measurements and of the methods used; it also proves that qualitatively the laws of the hydrodynamic flow theory seem to apply far into the region of so-called boundary friction, though this is restricted to the typical plain bearing flow and to the present oil group which has the same structure with different molecular weights, with the exception of reference oil No.2.

It is surprising on the other hand that, although the μ values measured at the limiting speed do not follow the viscosity or molecular weight grading, they vary but little from one another (compare Fig.3). The μ value expresses the internal fluid friction, i.e. the internal drag of the molecular chains oriented in the direction of flow, which slide along between the adhering boundary chains. It is therefore possible or even probable that the group tested is similar in respect of that physical effect (not necessarily viscosity) on which the magnitude of the internal friction depends. This is quite possible for the oil group in question; as a similar investigation leads, however, to certain rather obscure problems of physical chemistry, we shall not go further into it. Alternative test methods are available for the determination of other specific properties of the oils, which should be known in order to achieve a complete investigation of the lubricating ability under different working conditions.

The method is quite effective in the region of typical plain bearing loads. It can be applied to the comparison of different metal combinations, which has been done successfully by keeping the same oil and has proved to agree well with practical experience on a large number of substitute materials. A report on this subject will be given later. Also the influence of different surface conditions, clearances etc. can be explained in this way. The multitude of combinations is however so great, that the scanty means and equipment available at present allow only some of the problems to be dealt with.

The above tests bring up also the following consideration. In several other methods of determining the lubricating ability a friction coefficient μ is often found to be characteristic of the oil properties and is used as a reference value. This friction coefficient μ derives from the old Coulomb's Law $R = P \cdot \mu$ or $\mu = \frac{R}{P}$, where R is the tangential friction force and P the outside load perpendicular to it.

In reality R always stands for the integral of the very variable shear stresses occurring over the friction gap, alternating between the stationary and moving surfaces, which are different. This integral is however a complicated function of the form:

$$R = \frac{r \cdot \eta \cdot U}{q_o \cdot c} \; (f\varphi)$$

where c is the clearance number and f (ϑ) an involved trigonometrical function of the angle covering the actual lubrication gap. The value of R or μ varies with variation in these values. It is the difference between a slide or a pin pressed against a revolving pulley and two cylinders, parallel or at right angles, rolled against each other. Other geometrical relationships may be cited. Each gives a different value of μ. All these values, even if measured most accurately, are never mutually comparable. They cannot therefore give equal results for the oil properties, apart from the fact that boundary friction can also be expressed differently.

The friction coefficient μ can be referred only to the apparatus in question; this renders evident the above mentioned need for adapting the test method as accurately as possible to practical conditions. Only when the thickness of the layer falls off to mono molecular dimensions, i.e. it becomes a surface film, does the friction coefficient assume again a constant value according to Coulomb's law.(5)

2) The carrying capacity in the gear testing apparatus.

A test apparatus was developed in the writer's laboratory some years ago which allows the recording and photographing on an oscillograph of the variation of the friction forces tangential and perpendicular to the tooth flank on running gear wheels under load. Books 25 (1) and 59(2) of the "Deutsche Kraftfahrtforschung" contain a detailed report on this subject. Originally developed to investigate the general loading effects on tooth flanks, and noisiness, this instrument very soon proved to be an exceptional device to test not only the accuracy of tooth cutting but also the action of different lubricants. Dietrich (1) has already mentioned it and established the remarkable fact that the friction variations decrease as the viscosity increases. This contrasts with the popular view that the function is a direct one. These tests were further developed by the inclusion of metered lubrication, on which Pietsch (2) reported in detail. If an accurately measured oil quantity, 1 to 5 cu.cm., is spread quite uniformly on all tooth flanks after cleaning them thoroughly, the shape of the friction curve shows the time "wear" of the oil film. The picture in the freshly lubricated condition is the same as with immersion lubrication. After a certain time, i.e. after a corresponding number of load fluctuations, the oscillogram shows very irregular friction jumps on some and later on all tooth flanks; moreover these are marked by the appearance of fretting corrosion marks on the flanks. The times involved vary between 2 and 60 minutes so that they can be measured with sufficient accuracy. They are characteristic for each oil and to a certain extent represent the "endurance" of the oil film.

This is a realistic test, in the conditions characteristic of tooth load. It covers a wide region of practical lubrication technique, and a very important one. The stress has a periodic character as each tooth is engaged for short periods. The prevailing pressures, calculated on the basis of Hertz pressures, reach very high values (up to 10,000 kg/sq.cm.). They must therefore be absorbed in very narrow gaps, i.e. in the distinct boundary friction region of the oil film. The extent to which hydrodynamic laws still apply in these narrow gaps of molecular order is uncertain; Peppler has already dealt, in an earlier paper, with tests on this subject.

From the kinematics point of view the relative motions between tooth flanks are interesting because a rolling motion is always superimposed on the sliding motion. Pure rolling motion prevails at the pitch-circle. In his work (p.14) Pietsch (2) gives a clear picture on the peculiar form of rolling paths. According to him in the course of rolling the oil layers subjected to high pressure break off continuously because the point of application of the load moves continuously. The oil layer is subject to a kind of "

stress" which unaoubtedly affects the molecular cohesion of the layer in a different manner than pure sliding motion.

If we compare the endurance ("life") of an oil series against a reference oil, the resulting values give a new characteristic value for this type of stress. Pietsch's investigations have shown that quantitatively these values are not in accordance with the η values. This is confirmed by similar tests on the first oil series which was used for plain bearing tests. As the quantities F_t or F_q showed a very clear relationship with the viscosity, the comparison of the (absolute) life values Ro with the viscosity (Fig.6) shows a rather independent behaviour. Only the last two highly viscous oils 7 and 8 show a clear rise if in large quantity. It should however be noted that they have a greater adhesive strength and they cannot be removed so easily from the flank.

Likewise the measured friction variations with a gradually falling tendency show as little dependance on viscosity as the μ values at n_{min} of the plain bearing tests, for the purpose of comparison.

It can be concluded that this test method encounters physical properties of the oils other than those met in the sliding test. An oil group subjected to sliding action can give characteristics exactly corresponding to the viscosity, without giving the same rating when subjected to a different kind of action under "break-off" conditions, i.e. without affording conclusions as to the carrying capacity in the gear or similar cases. The fact that these oils react practically in the same way to tooth flank action, and according to their viscosity grade to sliding action, affords the conclusion that they possess a similar structural property, which affects the sliding action but not the rolling action, although their molecular weights and viscosities are widely different. The following comparison shows that the viscosity or related characteristics of the sliding action do not have the same importance for the carrying capacity in the break-off state. An oil of entirely different chemical structure, having a viscosity nearly equal to that of oil 4 (oil X) was tested with an oil dose of 1 cu.cm. at 100 r.p.m. and 47 kg. tooth load on the same test gears. Its life time was 55 min. against 13 for oil 4, the friction amplitude T_{max} being 3.5 mm. as against 13 mm. for oil 4.

Fig. 6 shows the point in question for Ro or T_{max} for the oil of equal viscosity. The difference under the same test conditions is self-evident. It is even clearer if we examine the oscillograms. The oscillograms for all oils of series 1 to 8, which are not reproduced here, are all of the same type, analogous to the oscillograms 4 and 5, p. 18 of Pietsch's paper. Only the maximum amplitudes T_{max} differ slightly, as shown in Fig. 6. Oil X on the other hand gives an oscillogram similar to No.15 on p.20 of the above mentioned paper, with small deflections and damping of the oscillations. This proves that a structural effect in this oil allows the combination of comparatively low viscosity with the high endurance of an oil of ten times the viscosity.

No other technical method has so far been discovered which emphasizes these effects; it is clear that they can supply valuable indications as to the structure of new synthetic oils, under realistic operational conditions. The idea of using gears for oil testing has been repeatedly applied, among others by the Friedrichshafen gear factory. They always depend however upon the measurement of the energy expenditure of complete gear units. The lubrication process proper on the flank in the carrying gap is strongly over shadowed by splash effects, pump effects etc. Only the use of the oscillograph permits the isolation of the friction proper with the equipment used here. These tests are only a beginning and can certainly be

improved upon. In particular they could be extended to higher loads and speeds. The load applied here to the gear pair used corresponds to a Hertz pressure of about 3.000 kg/sq.cm.

The same method also permits the easy determination of the degree of deterioration of the lubricating ability, as can often occur in the sump of an engine when the oil is thinned by gasoline vapour. A normal oil, similar to No.4 of series 1-8 (top curve) and a new product of different structure (bottom curve) are compared in Fig.7. The first oil is clearly considerably affected by the gasoline dilution whilst the second seems almost completely unaffected. The above can provide valuable information on synthetic oil production.

III. Break-off method tests.

As the tests above described are rather length and increasingly large oil quantities are involved in plain bearing tests, it is desirable to find quick test methods which allow rough testing of some of the large number of combinations. The method described at p.74 of book 2, 1941 [4] serves this purpose. After initial pressure (up to metallic contact) the oil samples are tested for their tensile or shear strength between two plates by breaking-off in the vertical or tangential direction. This is done by measuring the break-off time t for a given break-off tension σ. A simple relationship was always found of the type:

$$\sigma_z \cdot t_z = \eta_z \quad \text{or} \quad \sigma_s \cdot t_s = \eta_s = \text{Const.}$$

The so-called break-off tenacity η_z or η_s has the dimensions of viscosity (kg.sec/sq.cm.) and it represents a quantity equivalent to the usual viscosity measured by the viscometer. If however one refers it to the kinematic viscosity and ostablishes the quotients $\psi_z = \frac{\eta_z}{\eta}$ or $\psi_s = \frac{\eta_s}{\eta}$, they represent a non-dimensional quantity, equal for all temperatures, and having a definite value for each oil. These values represent no absolute physical quantity, as they are largely dependent on the microstructure of the plate surfaces. Under the same test conditions they can only be used as comparative values, by forming the quotients $\psi = \frac{\psi_z}{\psi_s}$ i.e. by determining the ratio of tensile to shear strength.

The physico-chemical meaning of these characteristics is being thoroughly investigated in the appropriate institutes; the concepts adhesive capacity, surface tension and cohesion pressure, as well as orientation effects, all affect the results.

In oil series 1 - 8 (with the exception of reference oil 2) the values ψ_z and ψ_s remain at almost the same level; they seem therefore to be independent of viscosity grading exactly as do the values Ro, μ_{min} and T_{max}.

The following example shows very interesting developments of this kind. An oil, which is chemically well defined and soluble in water, i.e. produces no emulsion, was tested for lubricating ability in the bearing test machine; the results were outstanding, with figures roughly corresponding to its viscosity though rather higher.

If the pure oil is diluted in increasing proportion with distilled water, the figures obtained in the gear or the break-off test vary as shown in the following graphs. In Fig. 8 the viscosities at 30°C are plotted against the volumetric dilution ratio, also values

ψ_z and ψ_s as well as their quotient $\upsilon = \psi_z/\psi_s$ referred to pure oil assumed = 1. It appears that whilst the viscosity falls off steadily with increasing proportion of water, the ψ_z values ($\psi_z = \frac{\eta_z}{\eta}$)

as well as the ψ_s values at first rise steeply, then remain constant for a certain range and and then again rise steeply with clear sharp bends at about 35 and 50% water additions. As the ψ values represent the ratio of break-off tenacity to (viscometer) viscosity, the decrease of η appears in the first part of the range and not in the middle and subsequent portions, because in the region of 35 to 65% water dilution the viscosity decreases only slightly.

Each mixture ratio represents a clearly defined oil; it is therefore quite clear that break-off tenacity cannot be a simple function of viscometric viscosity, but it follows other intermolecular effects.

Fig. 9 shows the same discontinuities; in it the endurance values Ro measured in the gear test are plotted in absolute values against the mixture ratio. Here again distinct breaks appear at 35 and 50%.

Fig. 10 shows the friction deflections corresponding to the average friction coefficient for various kinds of oil dosing. A steady rise of the friction values appears so clearly to go together with a rise in the water content, i.e. with a decrease of the viscosity, that the above-mentioned fact is definitely confirmed. This proves again that in the state of boundary friction prevailing in the gear test the hydrodynamic laws are reversed; also that high viscosity oils produce better friction conditions in gears than low viscosity oils, irrespective of their chemical structure.

It is difficult to make comparisons of this kind with naturally occurring oils, which have an exceptionally involved chemical composition rendered even more complicated by blending. They are not soluble in water and produce emulsions. On the other hand gear and break-off testing allows the easy comparison of a large variety of synthetic oils of different molecular structure, in which definite changes of the chemical structure take place. This should considerably ease the choice of the large number of possible compounds.

SUMMARY

The purpose of this report was not to submit numerical results on the technical lubricating ability of any particular oil, but to demonstrate technical methods with technical quantities by means of which those results can be obtained in test conditions very close to the actual conditions of technical lubrication. Thus purpose was largely attained in bearing and gear tests; the break-off tests are designed to ascertain certain structural properties of the oils. Thorough physical fundamental research is needed to relate them to lubricating effect. Their statistical determination would afford certain relative possibilities of comparison. An exact or absolute definition of the concept "Lubricating ability" in the physical sense does not exist and probably never will. The engineers would derive little help even if further research should lead to an agreement on this concept, as practical lubrication technique will always bear reference to the special requirements of the application. Herein lies the significance of the above described methods.

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Table I

Oil	1	2	3	4	5	6	7	8
WM 80 (p=44.4)								
n_{min} =	55	44	40	35	20	20	20	20
h_o =	1.63	1.68	1.36	1.96	2.07	2.38	4.64	6.4 µ
α_o =	9°13'	9°26'	8°29'	10°9'	10°26'	11°12'	15°40'	18°20'
WM 80 (p=127.5)								
n_{min} =	435	130	240	135	80	50	30	20
h_o =	2.0	1.39	1.96	1.98	1.80	1.81	2.04	2.74 µ
α_o =	10°20'	8°34'	10°10'	10°8'	9°45'	9°47'	10°20'	12°
Gerohlex (p=52.5)								
n_{min} =	275	150	175	90	55	35	20	20
h_o =	2.64	2.5	2.4	2.93	3.26	3.26	3.5	6.0 µ
α_o =	12°40'	12°30'	12°10'	12°35'	13°	13°	13°30'	17°50'

Table II

F_t - Values:

Oil	1	2	3	4	5	6	7	8
WM 80 p=44.4	29.5	38	39	56	103	119	230	330
WM 80 p =127.5	4.6	10.7	8.15	14.7	22.5	35.5	68	137
Gerohlex p = 52.5	9.6	16.6	13.7	32.6	59	93	175	300

Table III

1) WM 80, p = 44.4

Oil	1	2	3	4	5	6	7	8
n_{min} =	55	44	40	35	20	20	20	20
n_{min}^2 =	3025	1936	1600	1225	400	400	400	400
$F_q = \frac{1300}{n_{min}^2}$ =	0.43	0.67	0.81	1,06	3.25	3.25÷)	3.25÷)	3.25÷)

2) WM 80, p = 127.5

Oil	1	2	3	4	5	6	7	8
n_{min} =	435	130	240	135	80	50	30	20
n_{min}^2 =	189225	16900	57600	18225	6400	2500	900	400
$F_q = \frac{1300}{n_{min}^2}$ =	0.0069	0.077	0.0225	0.071	0.203	0.517	1.44	3.23

3) Gerohlex, p = 52.5

Oil	1	2	3	4	5	6	7	8
n_{min} =	275	150	175	90	55	35	20	20
n_{min}^2 =	75625	22500	30625	8100	3025	1225	400	400
$F_q = \frac{1300}{n_{min}^2}$ =	0.017	0.0572	0.0423	0.16	0.397	1.05	3.23÷)	3.23÷)

(The final values marked thus ÷) show that the minimum r.p.m. could not be reached for technical reasons)

Table IV

Relative values of F_q referred to Oil 2 = 1

Oil	1	2	3	4	5	6	7	8
WM 80 p = 44.4	0.635	1	1.18	1.57	4.82+)	4.82+)	4.82+)	4.82+)
WM 80 p = 127.5	0.09	1	0.292	0.92	2.64	6.7	18.7	42
Gerohlex p = 52.5	0.297	1	0.74	2.8	6.95	18.4	56.7	56.7+)

Fig. 1 - Prof. Heidebroek's bearing test machine.

Fig. 2 -

Fig. 3 - Oil comparison in plain bearings.

Fig. 4 - Comparison of the oil slippage between W.M.80 and synthetic resin bearings at lower limit speed.

Fig. 5 - Comparison of plain bearing characteristics F_t and F_q with the viscosity η (30°C)

Fig. 6 - Comparison of the break-off characteristics, viscosity and friction coefficients for n_{min}. bearing metal WM 80.

Fig. 7 - Oil deterioration due to fuel effect on oil x and another oil of similar viscosity.

Fig. 8 - Relative graph of lubrication characteristics and viscosity in terms of mixture ratio.

Fig. 9 - Life as a function of the mixture ratio for different oil quantities and load.

Fig.10 - Increase in the friction amplitudes (friction coefficients) on dilution - oil x and distilled water.