LABORATORY FOR LUBRICATION RESEARCH

POLYTECHNIC - DRESDEN.

NEW METHODS FOR THE EVALUATION OF THE

LUBRICATING ABILITY OF OILS AND GREASES.

by

E. Heidebroek VDI

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CONTENTS:-

The concept of "Lubricating Ability".

The concept:- "Lubricating Ability" of a lubricant, in particular of the usual lubricating oils and greases, has not been clearly defined yet, and there is not any clear description yet of how it can be measured numerically. Many papers, however, have been published describing oil-testing machines which were supposed to serve this purpose and which allow mainly a mechanical test of lubricants.

It is believed that, apart from chemical values, these mechanical tests would make it possible to make comparisons and draw conclusions as to the

operational behaviour of lubricants. The most comprehensive survey of these methods is given in the report entitled:- "General Discussion on Lubrication and Lubricants".(x) [1]. It is necessary to mention this report here because of the wide range of the sources quoted in it.

During the last few years the following tests have become particularly popular:- The "four-ball machine" devised by Boerlage, the "SAE" test, the Voitländer method [2] and some others. These methods work on the principle that a lubricant is put under load between rotating spherical or cylindrical surfaces. The load or the speed is then increased until seizure takes place and the torques and temperatures etc. are then measured. Others again use slide-blocks with a plane surface which are pressed against rotating discs testing the friction values of the lubricant and the limit loads as well as the other behaviour of the material. But the results of these tests can only rarely be applied immediately to practical conditions. Many methods work on principles used in the sphere of hydrodynamic fluid friction, others again prefer to use only limit friction and the effects of wear. It is obvious that this alone may alter the physical conditions fundamentally. But in practical application one or the other or both states are already established by the operational conditions themselves, and the lubricating ability must prove itself in them. Thus e.g. it is found that the numerous bearing test machines are in a certain sense also oil testing machines, but it is difficult to isolate the specific characteristics of the lubricant, since so many other factors are involved.

When working in the sphere of so-called pure fluid friction, it is found that there appears only one physical quantity concerning the lubricant in the hydrodynamic equations which were used to calculate the friction etc. This one physical quantity is the viscosity η (kg x s/m^2). All other quantities are geometrical ones referring to shaft and bearing. In order to carry out the calculation, it is necessary to assume η to be constant. But it is well known that this is not even remotely correct. Thus we can only estimate a mean η-value if the curve representing η as a function of the temperature is known. The dimensions of the viscosity are determined by Newton's fundamental equation:- $\tau = \eta \times dv/dy$ as kg x s/m^2 in practical units. It is measured in the usual way with a viscometer.

Though the hydrodynamic method of calculation has proved itself and though its assumptions have been confirmed in principle under certain conditions, in the last few years it became very doubtful whether the physical concept of viscosity alone - viz. the shearing coefficient - allowed a satisfactory treatment of the factors which characterize lubricating ability. For this quantity is apparently only determined from flow measurements and could not be conclusively related to the molecular-physical properties of the lubricant. Thus new mysterious concepts were coined such as "oiliness", "onctuosité" etc. and so an attempt was made to describe the relationship between viscosity and pressure in particular, as well as other observations, e.g. the fact that oils of the same viscosity but of a different chemical constitution may have very different "carrying capacity", friction coefficients etc. [1].

Recent researches in the field of physical chemistry of which here only those made by Debye, Kyropoulos, K.L.Wolf, Harms, [3, 4, 5, 6] and others are quoted, leave no doubt that the chemical structure of the hydro-carbon groups in question decisively affect their internal friction, the effect of temperature and pressure, but most of all, their capacity to orientate themselves to a more or less orderly flow. It is also beyond doubt that this internal order of the molecules is decisively influenced in the neighbourhood of metallic or other surfaces by the free surface forces which extend from them. The question is only, from what proximity these factors become sufficiently effective to disturb or even destroy completely the assumptions which are the basis of the hydrodynamic theory. To solve this question is the task of technical research. Physical chemistry is at present engaged in investigating

(x) The numbers in square brackets [ ] refer to the Bibliography at the end of this report.

the relationship between the so-called viscosity characteristics and the molecular structure, and also in exploring the effects of the surface forces. Under the assumption that in thin oil films these effects may become noticeable from 1/1000 mm downwards, it will be found that this state of the so-called limiting friction is much more frequent under practical conditions than is usually thought. Assuming that the cylinder surfaces are absolutely even and inelastic and the loads not at all high, the clearance for limiting friction calculated by the hydrodynamic laws were far narrower with many journal bearings, with almost all gears, with roller bearings, rolling surfaces, etc. than those obtained experimentally. This is due to the fact that the speeds are variable, rounding off is not as it should be, specific loads are high, etc. Therefore it is no exaggeration to say that limiting friction occurs more frequently than pure fluid friction.

This fact makes it necessary to investigate what happens in the region of limiting friction more closely. Tests with journal bearings have been started and were reported in the interim report No. 71.

In order to extend the results of the tests showing the importance of the combinations of shaft, bearing-material and lubricant used and also to explain especially the effects of the latter, two further groups of tests were made in the author's laboratory, viz:-

        I. Break-off tests on thin oil films.
       II. Tests on the lubricating ability of various lubricants for gears.

These tests are described in the following paper:-

I. Break-off tests on oil films.

(1) Test arrangement for the determination of the break-off Viscosity.

The main problem for physical-chemical research at present is the determination of the adhesive force by which the molecular chains of the lubricants are held fast on the surfaces. In order to solve this problem by means of technical measurements or methods, the following test arrangement was developed (Fig. 1).

A plate, highly polished on the surface (a cast steel flat) ("Tuschierplatte") is put into an open vessel maintained at constant temperature in an oilbath with heating coil. A thin film of the lubricant to be tested is put on the polished surface and a 50 mm^2 polished counter-plate of the required material placed upon it. This test-plate is suspended from the arm of a decimal balance with even distribution of load.

The test is started by exerting sufficient pressure on the test plate (b) to overcome the upthrust of the balance (p), by means of a weight acting through a lever (e). On moving the plate slightly to and fro for one minute, under pressure, it will be noticed that it moves with increasing difficulty and eventually almost stops with a jerk, so that it can be shifted only by force. Then "contact" between the metal plates and the lubricating film is made. Now the weight is quickly lifted off and the upthrust of the balance, loaded with a certain weight (p), is allowed to act freely on the test-plate. A certain time passes until the upper plate breaks off, and this time and also the balance load are measured. The time may vary from a few minutes up to many hours according to the load or the type of the lubricant. (Translator's note. Break-off time will also depend on the metal from which the test plate is made.)

In spite of the make-shift device it is found that these break-off tests, when repeated a hundred times, give surprisingly uniform results. Let $\mathfrak{G}$ be the tension in kg/cm^2 referred to the area of the breaking-off plate and let t_A be the time in secs. which passes until it breaks off, then the formula is always (fig. 2):-

$$\mathfrak{G} \cdot t_A = \text{Const.} \quad \left[\frac{\text{kg} \times \text{s}}{\text{cm}^2} \right] = \mathfrak{N}_z$$

or:- $t_A = f(\frac{1}{\sigma}) = $ Const./σ. This means that t_A, plotted against σ (fig. 2), yields always with great accuracy a symetrical hyperbola, when σ is changed (by altering the load on the balance) and when the other experimental conditions (type of oil, temperature, material of the break-off plate) were left unchanged.

The constant thus found has the dimensions of "viscosity" $[\text{kg} \times \text{s/cm}^2]$

This important and fundamental result allows the use of the test apparatus for various purposes. The following problems were investigated and tested with it:-

(1) The magnitude of the constants on using combinations of various materials with the same lubricant.
(2) The same quantity with the same materials but with various lubricants.
(3) The influence of temperature.
(4) The influence of the pressure load previously applied.
(5) The influence of the nature of the surface.

This important constant will be called in this paper always η_z and named "time-strength" or break-off viscosity". The tests showed that this constant represents a quantity equivalent to that measured by viscosimeters, and the apparatus itself is a new kind of viscometer of great simplicity.

The following gives a more detailed summary of the knowledge gained.

(2) Combinations of various materials with the same

lubricating oil.

The following examples will be selected from the tests. Plates of 50 mm^2 each were broken off from the following materials:-

No.	Material.	Type.
1	White metal	W M 80 normal
2	Bronze ("Rotguss")	Unknown.
3	"Lautal"	Vereinigto Aluminiumwerke A.G. = United Aluminium Works, Ltd.
4	Synthetic resin (No. 1)	"Dekorit F" (Raschig), filler-free moulded resin.
5	do. (No. 3)	Gerohlit (Römmler) irregular textile reinforcing.
6	do. (No. 4)	Gerohlex (Römmler) coarse texture textile.
7	do. (No. 6)	"Z," "B" (Nowack) pressure-moulded.
8	do. (No. 8)	"Gerohlex" (Römmler) fine texture.

Fig. 3 shows the measured values of η_z plotted against the load for the bronze ("Rotguss") plate. There are clearly no extensive variations in spite of the make-shift test arrangements. Fig. 4 shows a diagram of the results. They show first of all that each material gives a different constant η_z with the same base plate. Variations in temperature have a strong effect. But

this test result shows that apart from the properties of combinations of various materials, the condition of the surface has doubtless some effect, i.e. the surface roughness is important. All plates were generally processed with the same emery cloth of the finest grain but this does not prove in itself that the degree of roughness is exactly the same. When a given plate was later processed with coarser emery, it was found that the viscosity value was at once substantially reduced. Since no instruments for the exact measurement of the surface condition were available, test plates were ordered from (Messrs. Carl Zeiss-Jena) who measured their degree of roughness accurately by the interference method. These plates will be used as standards. It is important to settle this problem in order to determine exactly the effect of various combinations. It can be assumed that the degree of smoothness of the surface affects the number of the molecular chains which are bound vertically to the surface and thus influences the degree of resistance offered to breaking-off. The question whether a more morphous surface (Beilby - layer) or a more crystalline layer has a greater adhesive strength, is still a matter of great controversy in physico-chemical research. The test method described here will probably allow these problems to be approached again successfully. Surface conditions and the structure of the bulk metal can, however, hardly be separated, as in the processing methods in question the former will always be a function of the latter.

Still more conclusions may be drawn from Fig. 2 in which the tensile stresses are plotted against the break-off time. The course of the hyperbola shows that the greater the tensile stresses the shorter the break-off times. Theoretically the time 0 would correspond to $\sigma_z = \infty$. For reasons of equilibrium, however, these tensile stresses must affect both the surface of the material and the oil film. In other words:- when short break-off times are kinematically determined, break-off stresses may occur to such an extent that they may cause - particularly if recurring frequently - fatigue breakages of the surface particles, and may lead to destruction of the surface. Such conditions present themselves e.g. with tooth profiles at the pitch circle where there is pure rolling motion, and with rolling curves, etc. This would explain in a novel but not at all improbable way the occurrence of the so-called pitting, which, as already frequently supposed, depend upon the presence of a lubricating layer. Such causes may perhaps account for some corrosion phenomena. Since contact is made only after the surfaces have been previously pressed against each other, this would amount to an alternating tension - pressure strain of the material to which brittle, crystalline materials yield more easily than highly elastic or plastic ones.

(3)   Combination of the same break-off plate

with various lubricating oils.

Using a circular and carefully polished break-off plate of 50 mm diameter made from St.38, the following lubricating oils were examined in order to find out their η_z values:-

Table I of tested lubricants. (Oil table).

No.	Code name in Research Report.	Supplier and Name of Lubricant given by the firm.	Condition of Lubricant at Room Temperature.
1	Oil A	Rhenania Ossag B C 8	Clear yellow oil of low viscosity.
2	Oil B	I.G.Farbenindustrie Oil I	Clear dark-brown oil of low viscosity.
3	Oil C	I.G.Farbenindustrie Oil II	Clear red-brown oil of low viscosity.
4	Oil D	I.G.Farbenindustrie Oil III	Cloudy yellow-brown oil with a viscosity higher than that of the oils A, B and C.
5	Oil E	I.G.Farbenindustrie Oil IV	Clear honey coloured oil. Viscosity higher than that of oil D.
6	Oil F	I.G.Farbenindustrie Oil V	Clear yellow-brown oil of very high viscosity and of a glue-like consistency. (Great adhesive effect.) The oil can be drawn out into long fibres.
7	Oil F_1	I.G.Farbenindustrie Oil VI	Appearance and behaviour similar to oil F, but of considerably higher viscosity.

Oil A is the standard oil B C 8 of "Rhenania-Ossag" which is always used in the laboratory. Its viscosity-temperature curve was repeatedly determined by viscometers. (fig. 5). When the measured values of the break-off viscosity η_z are plotted against the temperature the result is, as shown in fig. 6. The equivalence between the two diagrams is very obvious. On plotting one ratio $\psi = \eta_z/\eta$ a straight line is obtained which is parallel to the temperature axis, i.e. ψ is independent of temperature and represents the constant conversion factor for a certain oil. With this constant conversion factor we can determine the $\eta_z - t_A$ curve from the η curve. In other words:- the factor ψ only affects the measurement. The same viscosity-curve may be plotted by using both scales, e.g. as shown in fig. 8 for oil A.

The position of the points found by the η_z measurements agree very well indeed with the old viscosity measurements, and so prove that the apparatus described here is a correctly working viscometer. In order to make a comparison, fig. 9 shows the same representation for Oil V. The η curve was placed at our disposal by the oil laboratory of the I.G.Farbenindustrie A.G.", Ammonia Works Merseburg, Director Dr. Zorn. The accuracy of the break-off method is clearly proved by it. It is impossible to foresee yet what advantages in future viscosity measurements will generally result from this fact. It is enough to think only of the difficulty of viscometer measurements at low and very high temperatures!

Fig. 10 compares the η values for 22° with the ψ values for oils where oil A is taken as unity. From this again we discover that oils of a different

chemical structure have ψ values, i.e. η_z - values which need not be at all related to each other in the same way as the η values. The same constant conversion factor ψ applies to η and η_z for each oil, but it is different for each different oil. This fact proves clearly that the usual η value, determined in capillary flow tests, does not at all describe THE physical factor which characterizes the oil clearly but that there must be still others depending on the molecular structure which have some influence. If this were not so, the comparison of η values should give the same factor as comparison of η_z values.

The ψ values of the tested oils are shown in the following table:-

Oil:-	A	B	C	D	E	F	F^1
	0.568	1.63	1.00	1.36	1.54	13.5	9.75 x 10^6

or arranged according to the magnitude of :-

Oil:-	A	C	D	E	B	F^1	F
	0.568	1.00	1.36	1.54	1.63	9.75	13.5

The dimensionless coefficient ψ shall first be used as a "quality factor". When the oils are arranged according to their η value at 30°, we have:-

Oil:- A C B D E F F^1

The sequence is thus different from that of the ψ values. η values at 50°C give the same order. When Oil A was taken as unity it can be seen that the relative values are mutually displaced a little, since each oil is associated with a different function of $\eta = f(t_A)$. It remains to be seen if there is another, and at any rate more complicated function and relationship between the ψ and η values. Since the two quantities η_z and η_1 and thus also the ψ value, are conditioned at any rate by the molecular structure of the oils, their inner relationship can be explained only from this angle. This scale of ψ values will be returned to later.

(4) Influence of the temperature.

As mentioned above, the relation between the break-off viscosity η_z and temperature is the same as for the normal viscosity η. It decreases with rising temperature. The law $\sigma \times t_A = \eta_z$ = constant remains valid for each temperature, keeping in mind the fact that for each oil the ψ value = η_z / η remains constant and independent of the temperature.

(5) Influence of the contact pressure.

In carrying out the break-off tests, it was found that the measurements were the more uniform the longer the initial pressure, which serves to bring about the final "contact", was applied. Since the break-off process was a function of time, it may be assumed that the approach during the contact pressure is also a function of time. Therefore, different break-off times will result from different times of "contact pressure". This is e.g. clearly shown by fig. 11 where the break-off times $t_A 1$ are plotted against the times of contact pressure $t_d 1$. The oil B of table I was broken off with a steel plate at a temperature of 21° and with a contact pressure of 10 kg. The diagram proves that after a certain time of contact pressure the curve of the break-off times becomes asymptatic, and thus after this time the final contact is made. This occurs earlier or later according to the magnitude of contact pressure (fig. 12). Times which exceed the "point of contact" have no practical effect on the break-off process.

When the various times of contact-pressure $t_d 1$ which are associated with the various break-off times, are plotted against the contact pressure $P_d 1$ (kg) all the curves obtained are hyperbolas (fig. 13). Thus a "contact-pressure

viscosity" $\eta_{d}l$ can be determined from the law $\sigma_{d} \times t_{d}l$ = constant = $\eta_{d}l$ in exactly the same way as was done during the break-off tests and this "contact pressure viscosity" therefore gives a constant value for each load (fig. 14). That leads to the conclusion that during the application of the contact pressure the intramolecular process is subject to the same laws as during the breaking off. In other words the "material" oil behaves the same in thin layers under either tension or compression load. But the values $\eta_{d}l$ and $\eta_{z}l$ are - at the same temperature of 21° - not numerically equal as shown in fig. 15. (Notice the scales!) The $\eta_{d}l$ - values are always larger than $\eta_{z}l$ - they increase at a quicker rate and tend to infinity, whilst the $\eta_{z}l$ value remains constant.

It would of course be most interesting to relate the break-off - or contact pressure forces to the distance apart of the plates at any moment, and so obtain a sort of pressure-tension diagram. But it is very difficult to measure the distance between the plates accurately because of the minute distances involved. (2 μ and less). Fig. 16 shows results from a rough orthotest.

In this test the increase of the distance of the plates is plotted against the time and a parabolic function seems to be obtained. It is intended to make further tests on a larger scale with improved measuring instruments. It would be of great importance to find out if results analogous to those of creep tests on metal rods could be observed, especially at higher temperatures. This would support the theories advanced by Debye and others about the quasi-crystalline structure of viscous liquids.

(6) Influence of the condition of the surface.

It was mentioned in Section I that the break-off force decreases with increasing roughness of the surface. It was not possible to carry out exact measurements. Schmaltz [7] quotes the methods which are available up to the present in full and they can be found there. Accurate optical measurements e.g. by the interference method, can only be obtained with a surface roughness greater than about 2 μ. There was no evidence available that the instruments, developed by the Americans for the so-called "Superfinish" process and based on a stylus system, really enable measurement of a roughness below one μ. Tests in Germany did not prove that the smallest size of the feeler points which both cause and are susceptible to changes of form, copy the surface more accurately than the interference method. All these methods reproduce merely a small strip. Only a very uniform distribution of uneven spots would allow conclusions to be drawn regarding the whole surface. Tests will be made in this direction. A really accurate knowledge of the surface structure can probably only be obtained by using electron-optics. This line of investigation is pursued by the institutes of physical chemistry. They are particularly engaged in solving the question of whether there are quasi-amorphous surface characteristics similar to the Beilby-layer. Finch [1] and other research workers believe that they have found such characteristics. Furthermore, research is undertaken to find out if in this case they are important for the adhesion effect of lubricants.

Anyhow the test methods described allow already the carrying out of comparative measurements for technical purposes. Those interested are therefore requested to send, as required, sample-plates made and processed according to the various methods in order to have the break-off viscosity determined by means of a reference oil or for certain lubricating oils. The example in the following chapter shows that there is a relationship between such a measurement and the practical state of lubrication.

II. Tests on the Lubricating Ability
of various Lubricants on Gears.

The lubricating ability of any lubricant must affect the friction forces which occur on cylindrical rolling and sliding surfaces during lubricated motion

and these effects can be investigated if they can be measured directly. In the case of the continuous sliding motion of a bearing this is relatively easy. This has been initially achieved with running/gears by means of the test-arrangement developed in the author's laboratory under closest approach to practical conditions, as reported by G. Dietrich [8]. In this paper the behaviour of various lubricants was investigated for a certain pair of gears. Because of uneven load distribution one tooth-flank carries the load alone for a relatively long time, and therefore the magnitude of the friction tangential to the tooth-flank could be reliably determined. The oscillogram obtained is shown in fig. 17 (No. 26 of Dietrich's report).

Based on these results, the tests were now extended to a larger number of gear-oils and greases. It is well known that the various oil firms manufacture special high pressure oils or greases which meet the demands of gears better, and guarantee, in particular, a better protection against excessive wear.

The substances tested are shown in table II; they consist of a group of pure oils (the same as in Part I) and greases. The first oil is the standard "Rhenania-Ossag" B C 8, generally used in the laboratory. The oils B to F^1 were kindly given by the oil research laboratory to the I.G. Farbenindustrie, A.G." Ammonia Works, Merseburg, Director Dr. Zorn. They are a series of high molecular weight hydro-carbon oils which were developed for special purposes and some have a very high viscosity. The greases are products of Rhenania-Ossag, made by the usual saponification method. They are the standard commercial types manufactured for use with gears. I would like here to express my thanks to both firms for their readiness to help us in this matter.

Fig. 17:- Oscillogram 26. Frictional force independent of the relative sliding velocity of the flanks at a low oil viscosity ($\approx 10°$ Engler). Course of the frictional force and of the normal force. Pair of gears 1/23. Speed 75 r.p.m. Oil F. Temperature 80°C.

(1) Test arrangement and method.

Whilst Dietrich always worked with "immersion lubrication", the method used here was "metered" lubrication as follows:- First the test gears are carefully cleaned and then a certain quantity of lubricant applied with a brush, 1 cc and 5 cc at a time. All teeth were painted over several times until it could be assumed that the required quantity was evenly distributed. The gears were then turned several times by hand and then driven as described in Dietrich's paper. 2 speeds (n = 100 r.p.m. and n = 500 r.p.m.) and 2 loads (torques of 280 or 120 cm kg corresponding to N=0.39 or 0.835 h.p.) were used. The test-gears came from Z.F. Friedrichshafen, gear pair 1/23 finished by the Maag grinding process. (See [8]).

The run was recorded by the oscillograph and constantly observed until compared with the initial state - a clean change, i.e. "frictional jumps" could be seen. Previous experiments with degreased dry flanks showed even after a few revolutions under load the beginning of frictional oxidation by distinct rust-brown stripes. It is interesting to note that the region of the pitch-circle is not affected, as is clearly shown in fig. 18. Therefore it may be said that frictional oxidation occurs only in conjunction with sliding friction.

The oscillogram shows the beginning of frictional oxidation, i.e. dry friction, very clearly by the appearance of large deflections (frictional jumps). On applying a measured amount of lubricant as described above, and running the gears for a fixed number of minutes, 2 characteristic running times could be defined, viz:- the time when:-

(a) the first frictional jump occured (point R_{o1})
(b) frictional jumps were noticeable on almost all flanks (point R_{o2})

Since the individual flanks are processed in a slightly different way, the frictional oxidation characteristics are not only recorded by the oscillogram

but are also clearly visible on the flanks themselves, at first only on individual teeth and then on all. Fig. 18 shows how clearly they show on the flanks. It is only an extremely thin film which can be wiped off easily.

II. Table of lubricants tested.

No.	Code name in Research Report.	Supplier and Name given by the firm.	Condition of Lubricant at Room Temperature.
1	Oil A	Rhenania-Ossag B C 8	Clear yellow oil of low viscosity.
2	Oil B	I.G.Farbenindustrie Oil I	Clear dark-brown oil of low viscosity.
3	Oil C	I.G.Farbenindustrie Oil II	Clear red-brown oil of low viscosity.
4	Oil D	I.G.Farbenindustrie Oil III	cloudy yellow-brown oil of a higher viscosity than the oils A, B and C.
5	Oil E	I.G.Farbenindustrie Oil IV	Clear honey-yellow oil. Viscosity higher than that of oil D.
6	Oil F	I.G.Farbenindustrie Oil V	Clear yellow-brown oil of very high viscosity and glue-like consistency. (Great adhesive effect). The oil can be drawn out into long fibres.
7	Oil F_1	I.G.Farbenindustrie Oil VI	This oil looks like and is similar to oil F. Its viscosity, however, is considerably higher.
8	Grease G	Rhenania-Ossag Shell Ambroleum	Black grease of medium consistency. Particular property:- It adheres very badly to polished metal surfaces. The internal cohesion of the grease particles is very good. The grease can be drawn out into thin films.
9	Grease H	Rhenania-Ossag Shell High pressure lubricant raw.	Red grease of medium consistency. This grease can be applied to the tooth flanks very well. Poor internal cohesion. The grease cannot be drawn out into fibres.
10	Grease J	Rhenania-Ossag Shell Grease FD 16 from Shell Mex.	Sweet smelling black grease of low consistency. Already at room temperature a liquid phase appears. The internal cohesion is approximately similar to that of Grease B. (this "B" might be a printing mistake, Translator).
11	Grease K	Rhenania-Ossag	Light-brown "dry" grease of higher consistency. It can be drawn out into small fibres and be applied to the flanks better than grease G but not so well as grease H and grease J.

(2) Test Results.

The results of such a series of tests are given below, viz:- for "metered lubrication" of 1cc and speeds of n=100 or n=500 r.p.m. respectively with the corresponding output given in the previous section. Fig. 19 and 20 give the results diagrammatically. The result is most interesting. A clear "efficiency-scale" of the individual lubricants is obtained, especially in the case of the R_{o2} points (lower columns). The fact that the saponified greases are inferior to the pure C-H-oils is particularly characteristic. It is quite possible to make reliable observations, since a considerable number of minutes elapse before the first or all the frictional jumps occur. This is shown in the oscillograms fig. 21-24. Fig. 21 represents the friction diagram with immersion lubrication, viz:- with a continual renewal of oil. Fig. 22 shows the state after a run of 2 minutes. No difference can be observed. The first frictional jumps (R_{o1} - point) appear after a 20 minutes' run in fig. 23; the general frictional oxidation (large deflections) after 35 minutes are given in fig. 24. The diagrams of the runs were recorded with the standard oil A. The oscillograms 25-27 show in contrast how quickly the saponified greases are used up due to the squeezing effect caused by the meshing of the teeth.

The initial smoothness of the friction-oscillation disappears after only one minute! After only 3 minutes' run a very fluctuating friction graph is obtained with very big deflections. In contrast to this the highly viscous oil F gives a very smooth graph (fig. 28-30). Even after 15 or 45 minutes' run the small overhead oscillations caused by the surface roughness of the tooth flanks are smoothed out almost completely. After 2 minutes' run even the kinematic ground oscillations are still completely suppressed. The wheels run here without making the slightest noise! These oscillograms were taken with 5 cc "metered quantity", i.e. with a layer thickness of approximately 0·5 mm. With a decreasing thickness of the layer (1 cc = 0·1 mm thickness of the layer) the smoothing effect can still be observed after 2 minutes (fig. 31).

When the tested lubricants are arranged in order of effective life, starting from the maximum, the following order is obtained:-

        Oils:- F E B D C A
        Greases:- J K H G
        according to the viscosity η at 20°:-
        Oils:- FE D B C A.

(The greases are omitted since a comparative viscosity measurement was not possible). Thus the viscosity scale corresponds approximately to that of the efficiency scale except for the order B D C or D B C. The mutual relation of these values becomes clear when as in fig. 32 both the viscosity - and the life values are referred to those of Oil A as unity, and then compared. It can be seen then that there are e.g. considerable numerical differences between the relative η values of oils D, C and B but that there are only small differences between their ξ values (= effective life values relative to Oil A). This makes it clear that the viscosity is not the only criterion for good lubrication. This becomes strikingly apparent when the real magnitude of the frictional force tangential to the tooth flank is obtained from the oscillograms of the different oils, under similar test conditions. The oscillograms from tests carried out with n = 100 r.p.m. and with 1 or 5 cc "metered" oil quantity were taken after a 2 minutes' run, (fig. 21 and fig.22) and the values of the average maximum frictional deflection read off (see fig.17 above). Diagram 33 shows the results. It appears that the viscous oils show the smallest, and the fluid oils the greatest, deflection. Exactly the opposite result should be obtained according to the hydrodynamic theory, i.e. the friction should increase with the viscosity. This seems to be usually looked upon as a sort of "damping" of the frictional throws because of the higher viscosity. Since, however, the kinematic conditions are not changed, this concept loses its physical meaning. Only the internal molecular structure of the lubricants has any effect here. The more fluid the oil, the narrower the gap, and the more apparent the boundary effects.

Dietrich's paper [8], by the way, contains similar observations (page 25). He found that the friction-coefficients increase with temperature.

This corresponds to a reduction of noise in the case of the more viscous oils. The wheels run almost noiselessly with oil F. This again confirms the relation between frictional deflection and noise produced, pointed out in Dietrich's paper. Finally the relation between the relative break-off viscosity (ψ values), determined in Part I of this report, and the ξ values (relative life) will be investigated. It may reasonably be assumed that there is a relationship between the two, as, according to the theory advanced in this report, they both depend upon the molecular conditions which occur in the region of limiting friction. Exact quantitative laws are, of course, only possible as a result of accurate data. Various mathematical hypotheses have shown up to now, that there is a relation between the ξ values (relative life) and $\sqrt[3]{\psi}$. Fig. 34 compares the ξ values for the various oils with $\sqrt[3]{\psi}$. The apparent regularity is anyhow noteworthy as a qualitative comparison. If this were fundamentally so, i.e. quantitatively, this would mean that by a simple determination of the break-off viscosity a comparable evaluation of the efficiency or of the life of the oils, i.e. their "lubricating ability" would immediately be obtained for gears tested under the operational conditions above. It should be remembered that in Part I of the report the value $\psi \eta_x / \eta$ was determined as a constant which applies to a certain oil at all temperatures.

SUMMARY.

New test-methods have shown that hitherto unknown technical properties of lubricants could also be determined by consideration of technical quantities from a molecular physical viewpoint. One of them is the concept of "break-off viscosity", and together with the "break-off method" used in its determination it is leading to very good results. For running gears there is also a comparative method of determining the efficiency or the effective life of the various lubricants. There is probably a close connection between the two methods, and they will allow a qualitatively comparable interpretation of the lubricating ability.

Further tests will show whether these results are applicable also to other sliding surfaces which work in the region of limiting friction (piston-friction, plain bearings, etc.)

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FIGURES.

Fig. 1.   Diagram of the break-off device.

Fig. 2.   Curve.

Fig. 3.   Curve (Bronze/cast steel).

Fig. 4.   Time strengths η_z for various materials.   Oil B C 8.

Figs. 5, 6, 7.   Curves.

Fig. 8.   Oil A (Curve).

Fig. 9.   Curve.

Fig. 10.   Referred to oil A as unity (Curve).

Fig. 11.   Curve.

Fig. 12.   Curve.

Figs. 13 and 14.   Curves.

Fig. 15.   Curve.

Fig. 16.   Variation of break-off time with thickness of oil film
Oil I, St 37.11 / Stg.

Fig. 17.   Oscillogram 26.

Fig. 18a.)
Fig. 18b.) Photographs.
Fig. 18c.)

Fig. 19. $n = 100$ (graph).

Fig. 20. $n = 500$ (graph).

Fig. 21.   Running diagram with "immersion lubrication".

Fig. 22.   Running diagram with 1 cc "metered lubrication".
Time:- 2 minutes.

Fig. 23.   Start of frictional oxidation (point R_{O_1}).
Record after 20 minutes' run.

Fig. 24.   Progressive frictional oxidation.   Time:- 35 minutes' run.

Fig. 25.   Record of the beginning of the run.

Fig. 26.   Record after a run of one minute.

Fig. 27.   Record after a run of 3 minutes.

Fig. 28.   Record after 2 minutes' run.

Fig. 29.   Record after 15 minutes' run.

Fig. 30.   Record after a run of 45 minutes.