THE INSTITUTION OF MECHANICAL ENGINEERS

General Discussion on Lubrication & Lubricants

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GROUP III. INDUSTRIAL APPLICATIONS

LUBRICATION IN THE COLD PRESSING OF SHEET STEEL

By J. C. Arrowsmith, M.Met.*

The omission of a lubricant between the sheet and the dies in the cold pressing of steel is usually immediately evident in the resulting damage to either the pressing or the dies or both, or in a torn pressing. Whilst the use of simple lubricants such as mineral, vegetable, or animal oils has often been sufficient, there have always been cases where such lubricants have proved inadequate under conditions of extreme pressure. The development of improved pressing steels and the increase in the speed of press operations have increased the necessity for improved drawing lubricants. The literature on the subject, however, is very scanty, though within the past few years a few papers have been published (Jevons 1936, 1937; Montgomery 1933; Reswick 1933) which show that this subject is beginning to receive serious study.

In the cold pressing of sheet steel a piece of flat sheet (the blank) is usually forced by means of a solid tool (the punch) into the space within a draw ring. The flow of the metal into the draw ring is usually controlled by means of pressure applied to the periphery of the blank which is supported between the draw ring and another ring known as the blank holder. The metal of the blank takes up its new shape partly by extension, partly by compression and partly by sliding over the surface of the die. The pressure applied to the blank by means of the blank holder is seldom sufficient to hold it rigidly, the function of the blank holder usually being to *restrict* the flow of the metal rather than to *prevent* it. In order to obtain additional restriction one or more beads may be applied to the surface of the blank holder and corresponding grooves cut in the surface of the draw ring, the blank having to flow around these obstructions in its passage into the die.

Whilst there is some movement of the metal over the surface of the punch as a result of the metal stretching, the greatest movement takes place over the inner edge of the draw ring. Owing to the pressure of the blank holder and the additional pressure due to the action of the punch and also to the sliding of the metal of the blank over the die,

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* **The Pressed Steel Company, Ltd., Cowley, Oxford.**

there is a tendency for any lubricant to be squeezed out and wiped off.

In addition to the high working pressures, another factor which tends to bring about metal-to-metal contact is the roughness of the finish on the dies. Whilst the tools for small press work are usually polished to a very high degree of smoothness, this is less practicable for large dies. Should the film of lubricant break down, the friction increases rapidly, as the condition is one of metal in direct contact with another metal. The result is frequently the fracture of the blank. As a result of the excessive friction, the speed of the flow of metal into the die is reduced and since the movement of the punch continues unchecked the change of shape of the blank by plastic deformation continues. In some cases the sliding movement of the blank is so restricted that the plastic deformation proceeds to the breakdown of the material and a split pressing is produced. In other cases, the blank continues to move into the die, but owing to the breakdown of the lubrication, "scoring" of the pressing occurs. Such scoring is usually caused by particles of the metal of the blank becoming attached to the draw ring or blank holder, with consequent damage to the surface of that and all subsequent pressings until the die surface is reconditioned.

The difficulty of removal of these fragments of the blank from the surface of the die has led to the term "cold welding", since it was realized that something akin to welding had taken place. Only recently have Bowden and Ridler (1936) shown that temperatures in the neighbourhood of the melting point of the metals in question are reached merely as a result of sliding friction between metal surfaces. The condition of lubrication during cold pressing is thus one in which an extremely thin film of lubricant is operating. "Boundary" conditions are said to exist where lubrication depends upon the film of molecular dimensions which remains upon the metal surfaces long after the bulk of the lubricant has been mechanically removed. In order to meet these conditions, and at the same time to overcome the disadvantages of the most useful single lubricants, it has become general practice to make use of compounded lubricants or "drawing compounds". The composition of these has been governed by the properties desirable in such a material, the most important of which are outlined below.

(1) *Film Strength.* This, the most important property, may be obtained by the use of tallow, wool grease, lard oil, or castor oil. Tallow, however, is not easy to apply, and soon turns rancid. Wool grease has an objectionable odour, and like the others is fairly expensive. It is preferable to use them in the form of soaps, as they are then more easily miscible and their subsequent removal from the work

is simpler. Chlorinated and sulphurized oils will also confer high film strength on a compound.

Maintenance of the film of lubricant can also be obtained by incorporating in the compound a fine powder such as chalk, talc, or asbestine, the idea being to prevent the oil film from being squeezed out under conditions of extreme pressure. Such powders probably also serve to prevent metal-to-metal contact and thereby obviate seizing. In addition to the above advantages, flake graphite when added to a drawing compound further assists lubrication by its own slippery nature.

(2) *Slipperiness.* This property is essential in any lubricant in order to reduce friction. However, the maximum degree of slipperiness may be not only unnecessary but in fact undesirable. In many instances the production of a satisfactory pressing depends upon a suitably *restricted* flow of the metal of the blank between the blank holder and draw ring. This restriction is achieved by the use of draw beads in the blank holder and by the application of pressure on the blank holder. If the restriction is inadequate, then a wrinkled pressing results. Restriction is reduced beyond the minimum required for a good pressing by the use of an excessive quantity of drawing compound or by the use of one possessing too great slipperiness. A change to a more slippery drawing compound may have a detrimental effect on the quality of the pressing. Satisfactory lubrication may be obtained in certain cases with an oil of very limited slipperiness, as is proved by the production of many pressings without the use of drawing compound, from blanks bearing merely a very thin film of light mineral oil such as is used to prevent rusting. Slipperiness is a factor of greater importance, however, where it is necessary to increase the slip of the blank over the surface of the punch. Where deep draws are being made with a punch of a spherical contour, fracture of the blank may often be prevented by applying an extra slippery drawing compound to the punch. Flake graphite is a useful addition in such cases.

(3) *Ease of Spreading.* The necessity for this property is twofold. First it is essential that by virtue of its "wetting" power, the lubricant shall be able to distribute itself over the surface of the metal. Then, as the blank slides over the surface of the die the lubricant will be brought into intimate contact with those parts at which lubrication is most vital. The application of the drawing compound to the blanks is assisted by this property. Application may be by means of rollers, in which case ease of spreading will make possible a uniform application with the maximum economy. If, as is more usual, the compound is applied by brushing, the operation will be faster and more economical

if the material spreads easily. The commonly used emulsion type of compound is very satisfactory in this respect, whereas thick mineral oils are much less so.

It should be noted that ease of spreading is not incompatible with adhesiveness, another desirable property which prevents the lubricant from running off the blanks on to those parts of the die where more restricted flow is desirable.

(4) *Ease of Removal.* Since most steel pressings have to be painted after fabrication, it is necessary for the drawing compound to be removed. Any constituent which is liable to give trouble in removal should be avoided, and it is for this reason that flake graphite is generally avoided. The method of removal must be considered when formulating the compound. An alkali wash will suffice for most of the emulsion type compounds, but difficulty will be experienced if too high a proportion of mineral oil has been incorporated. Trichloroethylene degreasers will remove uncombined oils with ease, but will have little action upon soaps and will leave most of the solid fillers behind on the work. Where electric resistance welding is to be employed to assemble the pressing, the use of a compound containing a high proportion of filler is to be avoided, since most of the fillers used are non-conductors, and may give rise to faulty welds where the compound has been incompletely removed.

(5) *Non-corrosiveness.* The compound should have no corrosive action upon either the blank or the dies. The free fatty acids present in the commoner type of lubricants are seldom strong enough, even in the presence of water, to cause much damage to steel. Rusting seldom takes place even where the water content of a drawing compound may be quite high. The water apparently evaporates and leaves a protective film of oil on the pressing. Some of the modern extreme-pressure lubricants, notably the sulphurized and the chlorinated oils, are distinctly corrosive and where these are used, they must be removed from the dies and pressings as soon as possible.

(6) *Heat Resistance.* In the production of deep pressings at the high press speeds employed to-day very high temperatures are set up momentarily. The lubricating value of certain materials is seriously impaired by this rise in temperature. An oil of high film strength might thus provide perfect lubrication if the temperature were kept low and yet fail when the temperature rose above a certain limit. Since the solid fillers which are frequently incorporated in drawing compounds are practically unaffected by the temperatures met with in drawing it is reasonable to expect that there will be some considerable advantage to be derived by the use of a compound containing filler.

(7) *Stability.* All possible steps should be taken during the manufacture of a drawing compound to prevent changes taking place in the material during storage, such as settling out of the suspended solids, separation of the emulsion, and decomposition by fermentation. When such changes occur it is rare that any remedial measure is entirely satisfactory. Mechanical separation may be reduced to a minimum by attention to the particle size and the consistency of the liquid portion. The stability of the emulsion is more difficult to control, but fermentation may be prevented by the addition of a preservative such as benzoic acid or cresylic acid.

(8) *Freedom from Objectionable Physiological Effect.* No lubricant should be used which may have any harmful effect upon the operators who have to use it. The sterility of the compound should be assured by means of additions of antiseptics, otherwise the compound may give rise to an outbreak of skin complaints or boils among the operators. Foul-smelling ingredients should be avoided as well as those that are troublesome to remove from the operators' hands.

(9) *Economy.* Although this feature has been placed last, it is not the least important. Prices have nowadays to be justified by careful economy tests under shop conditions. The material is not valued solely on its unit price, but the price is considered together with economy in operation. Where a more efficient drawing compound is used, there will be a saving not only in material used, but also in the cost of application and possibly in the press operation cost.

Composition of Drawing Compounds. The best drawing compound must be distinctly in the nature of a compromise. In practice a mixture can be obtained which will give satisfactory results under most conditions. For special conditions it is usually possible to make modifications in the composition which will improve the lubricating properties. For the pressing of steel sheets such as are used for automobile body parts the most popular type of compounds comprises those with compositions roughly as follows :—

	Per cent
Fatty oil (both free and combined as soap)	33
Chalk powder	33
Water	33

According to the severity of the draw, this may be used without any further addition or with the addition of flake graphite, mineral oil, or water. The two latter additions are, of course, made with a view to economy.

There are many variations of the above type of compound, depending on the use of different fatty oils, soaps, and fillers. Extreme-pressure lubricants are sometimes incorporated, but do not appear to be particularly effective. In many press shops the use of these comparatively modern compounded lubricants is still regarded with mistrust, as very good results are obtained with the older and more simple lubricants such as castor oil and tallow.

Evaluation of Drawing Lubricants. The identification of the constituents of drawing compounds by chemical and physical methods of analysis is often extremely difficult, and an absolute analysis is usually an impossibility. The value of such a compound must therefore be judged by its performance as a lubricant, having in mind also the other desirable attributes which have been outlined above. It is usually impracticable and unnecessary to consider a special lubricant for each press operation and the popular demand is for a "general purpose" lubricant. The comparison of different lubricants under working conditions is rendered difficult by the fact that it is usually impossible to standardize the test conditions. The condition of the dies, the setting of the dies in the press and variation in the surface characteristics, ductility, and hardness of the steel being pressed are all liable to interfere. Again, since any reduction in lubricating properties may result in damage to the pressings being produced, workshop tests may involve considerable expense. On the other hand, there is at present no laboratory test which would approach closely enough to the conditions of lubrication prevailing in the shop, for such a test to be absolutely comprehensive. It is possible, however, to make small-scale laboratory tests employing heavy loads which give a useful indication of film strength, and thereby reduce the amount of shop testing to a minimum.

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STEAM TURBINE LUBRICATION SYSTEMS AND FIRE RISK

By S. F. Barclay, D.Sc., Ph.D.*

In modern turbine practice, the steam temperature is well in excess of the self-ignition temperature of the lubricating oil, so that oil which escapes from the lubricating system or from the governor relay can take fire if it comes into contact with any metal surface at steam temperature. This undesired effect of raising steam temperatures was not generally anticipated and it came as a shock to many engineers to learn from disastrous fires that part of the price to be paid for the present-day efficiency of the turbo-alternator set is a fire hazard unknown a generation ago. A number of disastrous fires and many more outbreaks which, happily, were extinguished in the early stages, have shown the need for taking active steps to combat the fire risk.

SELF-IGNITION OF LUBRICATING OIL

The temperature at which self-ignition of lubricating oil takes place is influenced to a certain extent by the conditions under which the test is conducted. If, for example, a typical lubricating oil is heated in a container, a temperature of approximately 750 deg. F. has to be reached before the oil will take fire, but if the same oil is poured over a hot pipe then ignition may take place at a temperature as low as 560 deg. F. The ignition of the oil is preceded by two phenomena: first, the oil cracks under the rising temperature, with the formation of lighter and of heavier fractions than the oil itself; and second, as these newly formed vapours come into contact with atmospheric oxygen, they combine with it. In other words, before the oil on the hot pipe takes fire, a flameless combustion has been in progress in the atmosphere around the pipe. For this reason, ignition never takes place at the surface of the pipe, but always at some point in the atmosphere above it, usually at a distance of a few inches; on occasions, the author has seen combustion taking place at a height of more than 2 feet above the pipe.

By condensing the vapours developed as oil falls on to a superheated steam pipe, a fair idea can be obtained of the nature of the flameless combustion. In a typical example, the condensate consists of approximately 60 per cent of oils and of oil-soluble substances and approximately 40 per cent of water and of water-soluble substances. The oil phase contains about 85 per cent of unsaponifiable oil, about 9 per cent

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* Mather and Platt, Ltd.

of saponifiable oil, and about 6 per cent of bituminous matter. The water phase is composed of about 89 per cent of water, 6 per cent of soluble bitumens and gums, 3 per cent of acetic acid, and 2 per cent of aldehydes. Whereas the flash point of the original oil is rather more than 400 deg. F., the flash point of the condensed vapours is only 174 deg. F.

It can now be seen how oil takes fire when it falls on to a superheated steam pipe. The oil cracks, with the formation of vapours which begin to "burn" with heat evolution but without flame as they rise in the air, At the same time lighter fractions are developed with a flash point much lower than that of the original oil and this continues until the temperature within the rising vapours is sufficient to cause them to ignite, when the flame flies back to the pipe to cause the oil lying on it to burn. This phenomenon shows how lagging and other pipe coverings may fail to prevent an outbreak of fire, since the vapour rising from oil slowly percolating into imperfections of the pipe covering can take fire in the air well above the pipe itself.

Tests made with a typical turbine lubricating oil show that ignition is certain with steam pipe temperatures of 600 deg. F. or higher, while it is possible down to a minimum temperature of about 560 deg. F. Such figures are materially influenced by the volume of oil used, by the manner in which it falls on to the steam pipe and by the ambient temperature: the situation can, however, be summarized by saying that the steam temperatures employed in the modern power station are in excess of the ignition temperature of the lubricating oil. There is apparently little difference in the self-ignition temperatures of different makes of lubricating oil.

The Burning of a Jet of Lubricating Oil

Lubricating oil burns quite freely once it has been ignited, whether it be in bulk, as a thin layer on a floor surface, or as a film on metal surfaces. If, however, the oil is sprayed into the air, the issuing jet cannot be ignited unless delivered at high pressure through an "atomizer". When the issuing jet of oil falls on to a heated surface, the situation is quite different; even a coarse jet of oil will then burn freely and as the hot surface is further heated by the burning oil, the conflagration assumes an astonishing degree of intensity. The intense burning of a jet of oil escaping from a defective pipe has been the main destructive agent in recorded fires, as the intense heat developed affects the other oil pipes and oil-containing gear in the neighbourhood, more oil being released to add to the conflagration.

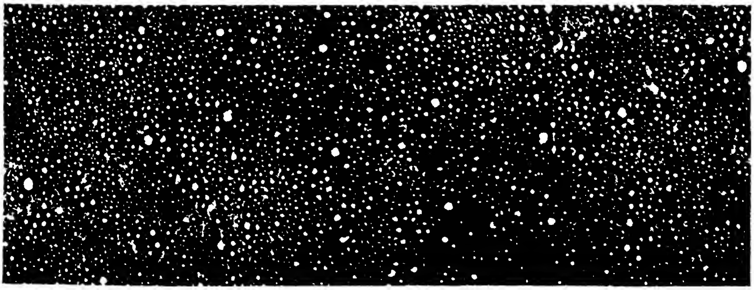
The exterior of lagged steam pipes, whether coated with metal sheath-

Fig. 1. Ignition of Turbine Lubricating Oil as it Falls on a Superheated Steam Pipe

Fig. 2. Experimental Equipment Reproducing a Portion of an Actual Modern Turbo-Alternator Set.

Fig. 1 shows that the ignition of the oil does not begin at the pipe surface, but in the rising vapours above it.

When an oil pipe running above a group of steam pipes (Fig. 2) is allowed to leak at the rate of only 6 gal. per min., the small outbreak of fire which ensues rapidly attains great intensity, the flames rising to a height of 30 feet. Within one or two minutes, the heat evolved is sufficient to deform surrounding metal work and to bring about failure of adjoining oil pipes.



× 125 dia.

Fig. 3. Extinction of Oil Fires by Emulsion Formation

The oil-in-water emulsion begins to separate within a minute or two of turning off the water and is complete after about five or six hours, leaving the oil entirely undamaged.

[I.Mech.E., 1937]

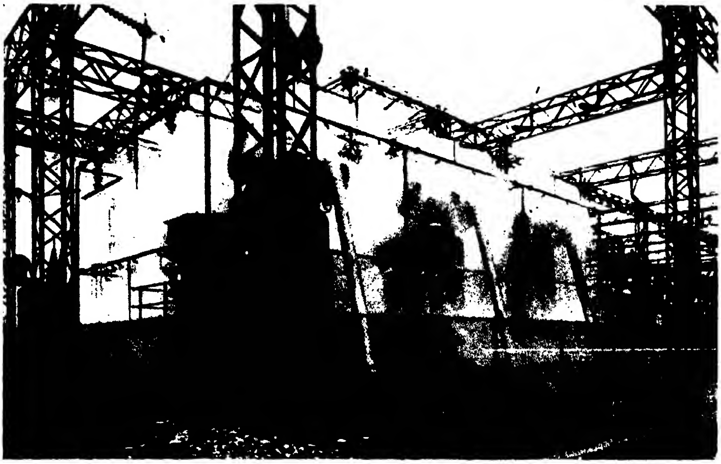


Fig. 4. Application of an Emulsifying Spray for the Protection of Outside Transformers

A similar type of spray is provided to protect the oil-containing equipment of turbo-generators (reproduced by courtesy of the Detroit Edison Company).

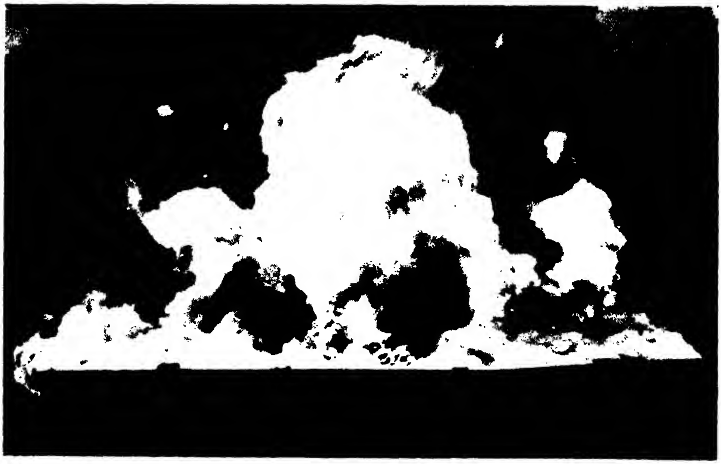


Fig. 5. Extinction of an Oil Fire by Emulsion Formation

A shallow tank, 2 feet deep, with a superficial area of about 600 sq. ft., contains 6,000 gallons of light fuel oil. Emulsifying sprayers are mounted above the oil surface. The conflagration is extinguished in two or three seconds from the moment of turning on the water. Only a thin surface layer of the oil needs to be emulsified to extinguish the flames and the huge fire illustrated can be put out with a discharge of not more than 40 gallons of water.

ing or not, constitutes an almost ideal surface for promoting intense burning of a jet of oil by virtue of the heat insulation and the ease with which the surface temperature is raised by the burning oil.

FIRE PROTECTION

Any effective scheme of fire protection embraces three equally important factors, namely (1) the minimization of fire outbreak; (2) the localization of the fire zone; (3) the provision of fire-extinguishing apparatus.

Minimization of Fire Outbreak. The fire risk with steam turbines would disappear if a non-inflammable lubricant could be produced. This, however, is a remote probability, having regard to the severe bearing duty of the modern turbo-alternator set. The governor relay system is different, as the lubricating properties of the fluid used are not so important and a substitute for oil of some non-inflammable fluid may be found. Already, chlorinated hydrocarbons, such as "Aroclor", have been employed on a small scale and, according to published accounts, with success. The elimination of fire risk in this way is, however, not achieved without some sacrifice: in particular, chlorinated hydrocarbons tend to cause corrosion.

The piping of the lubricating oil system must be isolated as far as possible from the steam piping. The welded joint is to be preferred to a threaded connexion: the pipes should be robust but with sufficient flexibility to prevent expansion troubles. The practice of encasing the oil pipes in outer protecting pipes is so excellent that it should receive wider adoption.

It has been suggested by several authors that the main oil tanks should be located in situations remote from the possible fire zone. Such a precaution is vitally important in the absence of effective fire-extinguishing apparatus. A remote position of the tank is, however, inconvenient and it is unnecessary when other precautions (referred to later) are exercised.

The lagged steam pipes should be encased in metal coverings and, in addition, should be shielded by guards, drip trays, etc., at all points where oil pipes run over or near them. The shielding should be as thorough as possible, since the lagged steam pipe, whether cased in metal or not, represents an ideal surface for promoting the burning of oil, but the limitations of such precautions must be realized. Thin metal shields, etc., heat so rapidly that they are little inferior to the steam pipe itself in promoting intense combustion of oil jets. If the joints of the steam pipes are not lagged, they should be carefully shielded as they will constitute the main danger of igniting escaping oil. Whether the lagging be metal-cased or not, it is desirable to give it an oil-resisting finish.

There seems to be little hope of introducing marked improvement in the arrangement of the governor relay so long as lubricating oil is used to actuate it. There have been oil fires just as serious when the valves have been placed in separate steam chests at the side of the cylinder as when they have been mounted on the cylinder. The proposal to operate the valves by mechanical linkage from remotely placed pistons is apparently not viewed favourably by turbine desginers.

There is often considerable congestion of pipes beneath the turbine staging, as the lubricating oil pipes run repeatedly close to the steam pipes. In such cases, the precautions against fire outbreak cannot be so thorough as is possible with new equipment: any deficiency in precaution should be compensated by increased effectiveness of the fire-extinguishing equipment. Trouble from the possible overflow of drip trays should be prevented by fitting them with drain pipes leading to safe discharge points.

Localization of the Fire Zone. A precaution which is rarely employed is to provide dwarf walls on the basement floor around the turbine foundations to restrict the flow of escaping oil. Such walls may be quite low and inconspicuous. The area enclosed by these dwarf walls should be drained with pipes leading through a flameproof trap to a safe sump. If the drainage is really adequate, the dwarf walls may be only a few inches high and so give rise to no inconvenience.

When practicable, it is preferable to locate the oil auxiliaries on the basement floor rather than mount them at some height above. The proposal to run off the lubricating oil to a safe sump in the event of fire will commend itself to few, particularly as the other precautions, in conjunction with the installation of effective fire extinguishing equipment, will ensure freedom from serious fire damage. The damage to the turbo-alternators caused by lack of lubrication may well be greater than that caused by the fire, as has been proved in at least one case.

Fire-extinguishing Apparatus. Hand appliances are useful for dealing with any minor outbreak, but for any major conflagration, they are all but useless. *Any effective fire-extinguishing equipment must be of the fixed installation type with remote control.*

The cubic capacity of the average power station is so considerable that it would be economically impracticable to achieve flame extinguishment by the introduction of inert gas. With the probability of burning jets of oil and of burning oil films on metal surfaces above the floor level, foam is not a suitable extinguishing agent for such a task.

Within recent years a system of fire extinguishing has been evolved which is suitable for reliably combating any forms of oil fires which may occur in turbo-alternator sets. The principle of this system is to convert the oil from an inflammable liquid into one which is incapable of burning. This conversion is accomplished by emulsifying it with water. Nozzles are provided for discharging water on to the oil surface. The water issues in the form of a well-distributed strong driving spray which strikes the oil with considerable force, giving rise to sufficient mechanical agitation to churn the oil surface into an emulsion of the oil-in-water type. The nozzles are made in patterns to give an emulsifying power at close range, medium range, or long range; they may be mounted within a few inches of the surface to be protected or at a distance of as much as 40 feet from it. The emulsifying nozzles, supplied with water under pressure, are mounted so as to command the whole of the danger area. Some nozzles are mounted horizontally to discharge along each of the oil pipes: more nozzles are mounted to command the whole of the exterior of the oil-containing auxiliaries and further nozzles are mounted to command the whole of the floor area where the burning oil could collect. Immediately a fire-extinguishing installation of this kind is put into use, any oil which has escaped is instantly rendered non-combustible. The emulsified oil not only does not burn, but on flowing over burning oil will quickly extinguish the flames. The emulsifying spray discharging on to the oil surface carries some entrained air with it; thus the resulting emulsion is somewhat aerated and so tends to remain floating on the surface of any oil on to which it may flow. Any oil under the action of the emulsifying sprayers is effectively emulsified whether it is escaping as a jet from a defective pipe, whether it is in bulk in a tank, or whether it is in a thin layer or film on a metal or other surface.

This system of extinguishing oil fires by emulsion formation is dramatic in its effect, as the flames die down instantly when an emulsion of paper thickness is formed on the surface of the oil and this happens within a second or so after the water is turned on. The emulsifying discharge from the nozzles is controlled by one or more remotely placed valves and since the water can be turned off almost as soon as it has been turned on, with the assurance that all flames will have been extinguished, the inconvenience caused by the water discharge is small.

The long-range emulsifying nozzle is of particular interest in its application to the governor relay system: such nozzles may be mounted in remote situations on the power station walls to extinguish any burning oil above the turbine staging, without interfering with access to the turbine and without the introduction of unsightly equipment.

CONCLUSIONS

The proper approach to the fire hazard is first to take all reasonable steps to reduce the chance of an outbreak; second, to take all reasonable precautions to limit the area of the possible conflagration; and third, to install effective fire-extinguishing equipment. Some fires have revealed obvious shortcomings in precautionary measures, the conclusion sometimes being that the fire hazard can be eliminated by removing proved weaknesses of arrangement or of equipment. This is not correct. So long as a big volume of inflammable oil is distributed under the turbine staging in containers of various kinds and in an elaborate pipe system, interlaced with the means of ignition in the form of the pipes, cylinders, etc., charged with superheated steam, so long will the fire hazard remain. Wise precautions can reduce the probability of fire, but they can not eliminate the need for effective fire-extinguishing equipment.

The modern power station with its potentialities of disastrous fire is too recent a development for statistics to show usefully the degree of the fire risk. Much, however, can be learnt from a study of the fire records of analogous risks. These records show that buildings containing highly inflammable substances such as oil along with sources of ignition are prey to a regular incidence of fire. In assessing the value of precautions, stress is too often laid on the normal, whereas fire records reveal that in all but a negligible percentage of cases the conflagration has its origin in some abnormal happening that could hardly have been foreseen. In a modern power station with a generating capacity of say 250,000 kW., the total volume of oil in the station (in transformers, circuit breakers etc., in addition to the turbine oil) can exceed 200,000 gallons. If the means of igniting the oil could be banished from the station, there would still be a remote fire risk, but with the means of ignition constantly maintained in proximity to the inflammable oil, it can only be a matter of time before some abnormal happening will defeat the interposing barriers.

In the average industrial situation, the probability of fire becomes the determining issue in justifying the cost of installing fire-extinguishing equipment. There are, however, special situations where the thought of disastrous fire cannot be tolerated, even though the probability of fire may be small. Statistics show that the fire risk aboard passenger vessels is all but negligible, but instead of the leading steamship companies ignoring the fire risk, they extend their provisions against fire, from the realm of most thorough precaution, to that of installing the most up-to-date fire-extinguishing equipment. The inference is clear that the degree of probability of fire does not count: the question of material loss hardly arises since it is ordinarily covered by insurance: the bald issue is that the possibility of serious fire aboard a passenger-carrying vessel cannot be tolerated in the face of the relatively small cost of installing equipment which can surely prevent it.

It is submitted that the same state of affairs prevails in principal power stations. The probability of fire, however reduced by the exercise of precautions, is always present and the possibility of disastrous fire cannot be tolerated, having regard to the relatively low cost of installing fire-extinguishing equipment which would surely prevent such a disaster. In point of fact, the cost of the most thorough fire-extinguishing equipment is inconsiderable in relation to the sums expended in other directions calculated to insure continuity of supply.

The author is indebted to the directors of Messrs. Mather and Platt, Ltd., for permission to publish this paper.

MEASUREMENT OF TEMPERATURE FLASHES ON GEAR TEETH UNDER EXTREME PRESSURE CONDITIONS

By H. Blok*

Various considerations render it desirable to know the temperatures occurring at the surface of contact of the teeth of highly loaded gears. The author has published (1937) theoretical calculations showing that these temperatures are often very high and are of great importance for wear and seizure.

During meshing each tooth face will show a temperature "flash" due to the frictional heat developed at the surface of contact which moves over the tooth face; this temperature flash is restricted to the immediate neighbourhood of the instantaneous surface of contact and will disappear very rapidly (partly due to the conduction of heat into the mass of the gearwheel) as soon as the tooth face is out of mesh, so that the tooth face considered will have reached its initial level of temperature again towards the following meshing period. It is the object of this paper to describe a method for the measurement of these temperature flashes and to check the accuracy of a previous calculation.

To measure the temperatures at the surface of contact the thermo-electric method is the only one that can be successfully applied. It is therefore necessary to make the gearwheels of steels of different compositions so that they will show a sufficiently high thermo-electric force; obviously the materials have to be adequate as gearwheels. After numerous trials a good combination of steels was found.

For simplicity, spur gears were investigated. Spur gear testing machines, designed according to the well-known energy circulation method, have already been built by many other investigators (Rikli 1911, Lewis 1923, Ulrich 1932, Soden Fraunhofen 1934). For the present purpose, these designs have the disadvantage that all test gears are running, which makes it difficult to measure the thermo-electric forces between the wheels as gliding contacts have to be used which may introduce disturbing electric forces.

A new design was therefore developed, using a planetary construction in which two narrow sun wheels remain stationary while a wide satellite, meshing with both, revolves around them. This arrangement makes the thermo-electric temperature measurements very easy. It also enables the electrical resistance at the surface of

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* Delft Laboratory, Royal Dutch Shell Group.

contact to be measured, which may give some information about the nature of the contact.

For a better understanding of the problem Fig. 1 may be considered. Teeth 1 and 2 of the driving wheel *x* and the driven wheel *y* are in mesh; Fig. 1*a* relates to the initial, Fig. 1*b* to the so-called central, and Fig. 1*c* to the final stage of the meshing period of these teeth. Owing to the load the tooth faces will be locally flattened. The indentation area AB (surface of contact) is shown enlarged under Fig. 1*b*; its width 2R is very small compared with its length, which is equal to the face width. It will be seen that the surface of contact AB moves from the root towards the tip of tooth 1 and from the tip to the root of tooth 2, so that these movements are in the same direction; in the case of involute spur gear teeth these are uniformly accelerated or uniformly decelerated movements.

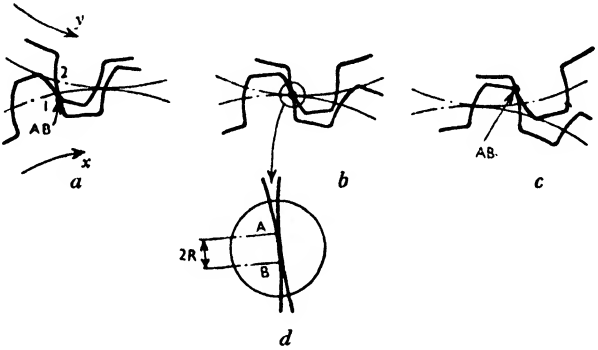


Fig. 1. Meshing of Two Teeth

a Initial position. *b* Central position. *c* Final position.
d Indentation area AB (surface of contact), magnified.

The author has shown (1937) that the field of the surface temperature rise on the tooth faces is shaped according to the curve in Fig. 2, assuming that there is no temperature jump from tooth 1 to tooth 2 for any point in the surface of contact; this field is superimposed on the mean levels of temperature of the body of the teeth.

At a certain moment during the meshing period let v_1 cm. per sec. denote the tangential velocity to tooth face 1 of the surface of contact AB on tooth face 1, while v_2 denotes the corresponding velocity on tooth face 2. The sliding velocity between these teeth will now be equal to the difference between v_1 and v_2, while the rolling velocity will be equal to the smaller of the two velocities v_1 and v_2.

It has further been shown (Blok 1937) that the maximum temperature rise at the surface of contact, $\theta_{\max_t}$, in degrees centigrade (see Fig. 2) is equal to

$$\theta_{\max_t} = 0{\cdot}83 \frac{f P_{l_t}(v_1 - v_2)}{(b_1\sqrt{v_1} + b_2\sqrt{v_2})\sqrt{R}} \quad . \ . \ . \ . \ . \quad (1)$$

where f denotes the coefficient of sliding friction, P_{l_t} the specific line load on the teeth in kilogrammes per centimetre face width at the moment considered; $(v_1 - v_2)$ = the absolute value of the sliding velocity in centimetres per second; R = half the width of the surface of contact in centimetres; and where $b_1 = \sqrt{\lambda_1 \gamma_1 c_1}$ and $b_2 = \sqrt{\lambda_2 \gamma_2 c_2}$, in which λ is the thermal conductivity in kilogramme-centimetres per centimetre per degree centigrade per second, γ the specific gravity in kilogrammes per cubic centimetre, and c the specific heat in kilogramme-centimetres per kilogramme per degree centigrade, while the subscripts 1 and 2 refer to the material of teeth 1 and 2 respectively.

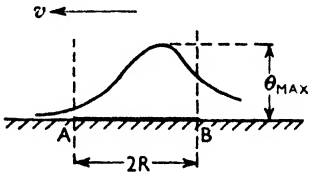


Fig. 2. Temperature Flash on the Area of Contact AB Superimposed during Meshing on the Mean Level of Temperature of the Teeth

For gears with line contact, R, half the width of the surface of contact, will vary proportionally to $\sqrt{P_{l_t}}$ (according to Hertz), and as both v_1 and v_2 are proportional to the number of revolutions, n, of one of the meshing gears, it may be deduced from equation (1) that for a given gear transmission

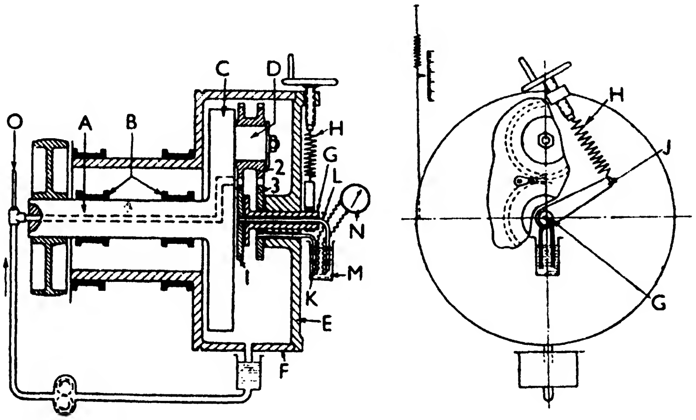
$$\theta_{\max_t} = C_t \cdot f\sqrt[4]{P_{l_t}^3 n^2} \quad . \ . \ . \ . \ . \ . \ . \quad (2)$$

in which the value C_t, characteristic for the given transmission, changes only with time during the meshing period. The value P_{l_t} for a single pair of teeth will vary considerably during meshing, as the total number of pairs of teeth in mesh fluctuates and as the distribution of the total load over all these pairs of teeth changes. The mean temperature flash at a definite moment t is approximately two-thirds $\theta_{\max_t}$. Assuming that the gear meshing is theoretically correct it follows that the time average of the mean temperature flashes, θ, is equal to

$$\theta = C_t \, . \, f\sqrt[4]{P_l^3 n^2} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (3)$$

in which C is a constant, characteristic for the gear transmission considered, while P_t is the total circumferential load in kilogrammes per centimetre of face width, and may be deduced from the transmitted torque and the size of the gears.

Description of the Testing Machine. In the planetary gear-testing machine (Fig. 3) a wide satellite spur gearwheel 2 revolves around the crankpin D fixed to the disk C on the shaft A; this gearwheel meshes with two stationary narrow spur gearwheels 1 and 3; wheel 3 is bolted to the cover E of the housing F and wheel 1 is mounted on a pivot G. By exerting a pull with a calibrated spring H (Fig. 3*b*) on the end of the lever J fixed to pivot G, a load of known magnitude can be created between the gear teeth of 1 and 2; the same load will then exist between the wheels 2 and 3. Obviously the load on the gears can easily be controlled during running, since pivot G and lever J are stationary. The gears are lubricated by oil circulated through the hollow shaft A and sprayed through a nozzle, fixed on the rotating disk C, on to the gears.



a *b*

Fig. 3. Planetary Gear-Testing Machine

Measurement of Friction and Surface Temperatures. The tooth friction occurring on the two sun wheels 1 and 3 can be found very accurately by measuring the reaction torque on the housing F, which is supported in bearings in such a way that it can oscillate about the principal axis of the machine. The friction of these bearings, which influences the measurement of the torque, amounts to only a few units per cent of the tooth friction, ball bearings being used throughout.

For measuring the mean temperature flash at the contact surfaces of the gear teeth, the thermo-electric method is applied by making the sun wheel 1 and the satellite 2 of the same steel *a*, while the sun wheel 2 is made of another steel *b*. Wheel 1 is electrically insulated from pivot G, and wheel 3 from cover E. A rod K of steel *b* is soldered to wheel 3, while a rod L of steel *a* is soldered to wheel 1. The cold junction M is connected to a millivoltmeter N so as to indicate the time average of the mean temperature of the tooth faces. Actually this time average of the temperature, denoted by x, does not relate to only one pair of teeth during its meshing period, as for the gears used in the particular test during the greater part of this period the load is divided between two pairs of teeth. The millivoltmeter was calibrated up to 300 deg. C., using two rods made of the steels *a* and *b*; the thermo-electric force was found to be proportional to the temperature difference between the cold and hot junction. In calculating the time average x of the temperatures of the tooth faces from the indication of the millivoltmeter it has been assumed that the thermo-electric force was independent of pressure on the surfaces of contact. Further it may be assumed that metallic contact occurred everywhere on the surfaces of contact, as the mean pressures were so high (10,000 kg. per sq. cm. and higher); the millivoltmeter will then give a true indication of the time average x of the temperature. The level of temperature, y, of the body of the teeth was found by means of a chromel-alumel thermocouple soldered to one of the sides of a tooth of sun wheel 3. By subtracting y from x the mean temperature flash θ will be found; it will be shown later that θ is proportional to $f\sqrt[4]{P_l^3}$ (the value of n from equation (3) being kept constant throughout the tests).

Test Conditions. A test was carried out, at a constant speed of 1,470 r.p.m. of the main shaft, on hardened steel gears with 30 ground involute straight spur teeth, with a pressure angle (angle of obliquity) of 20 deg. and a modulus of 3 mm. (diametral pitch=8·47 per inch); the addendum of the teeth was taken as equal to the modulus. The face width of the stationary sun wheels amounted to 4·5 mm. The oil temperature was kept constant at 65 deg. C. After running-in for half an hour at a specific line load P_l of 31 kg. per cm., in order to warm the whole machine, the load was increased every 60 seconds by steps of 31 kg. per cm. until a specific load of 310 kg. per cm. was reached, which was also maintained for 60 seconds. The specific load was then decreased again in similar steps to 124 kg. per cm. At each load the measured mean temperature flash θ attained a practically constant value after applying this load and allowing a few seconds for the millivoltmeter to reach its indication. The friction, as indicated by the reaction torque of the housing, appeared to be constant for each step of load.

Results. The coefficient of sliding friction f was calculated for each load by means of the well-known formula for the efficiency η of straight spur gear transmissions

$$\eta = 1 - \frac{\rho}{r \cos \theta} \cdot f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which ρ denotes half the effective line of action, r the radius of the pitch circle, and θ the pressure angle. For the transmission considered this gave

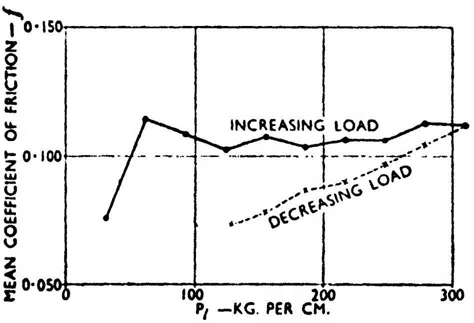


Fig. 4. Coefficient of Sliding Friction f on Tooth Faces as a Function of the Specific Line Load P_l

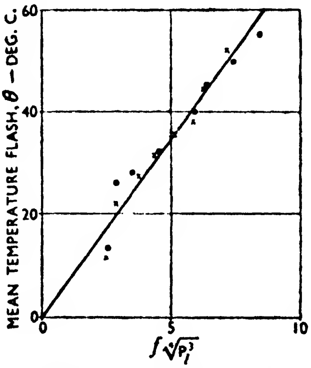


Fig. 5. Measured Mean Temperature Rise θ on the Tooth Faces Plotted against $f\sqrt[4]{P_l^3}$

$$\eta = 1 - 0{\cdot}342f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Let M_{f_1} denote the moment of friction of wheel 1, M_{f_3} of wheel 3, and M_d the known torque exerted on the gears, then from equation (5):

$$M_{f_1} + M_{f_3} = 0{\cdot}342f \, . \, M_d \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

in which f is the mean coefficient of friction of the two gear transmissions and $M_{f_1} + M_{f_3}$ are found by measuring the reaction torque on the housing. Other tests showed that the coefficients of friction for the two different combinations of steel used, *ab* and *aa*, were practically equal, so that the coefficient of friction between 2 and 3 could be computed directly from equation (6); for the particular test considered, it is plotted against the specific load in Fig. 4. Thus f remained at a nearly constant value of 0·11 from $P_l = 62$ kg. per cm. up to 310 kg.

per cm., but after decreasing the load again, f decreased continually; this decrease may be due to running-in effects.

The measured mean temperature rise on the tooth flanks of the wheels 2 and 3 is plotted against $f\sqrt[4]{P_l^3}$ in Fig. 5. It is evident that there is a proportional relationship between θ and $f\sqrt[4]{P_l^3}$, so that equation (3) is confirmed with regard to the influence of P_l. Other tests gave the same relation between θ and $f\sqrt[4]{P_l^3}$. Further tests will be carried out to see whether the relationship between θ and the number of revolutions n, also indicated in equation (3), holds good.

It is also intended to measure the thermo-electric tension between the wheels 2 and 3 by means of an oscillograph so that a more complete picture may be obtained of the variation of the temperature flashes on the tooth faces during meshing and in order to see whether equations (1) and (2) hold good. As a rough estimate the highest temperature flash occurring on the teeth will be two or three times larger than the measured average of the flashes, so that at the highest load the highest temperature will be about 200–250 deg. C. as the mean temperature level of the body of the teeth was about 75 deg. C.

Remarks. These temperature calculations can be applied to other gears which show line contact or even oblong surfaces of contact (e.g. helical spur gears, spiral bevel, or hypoid gears). It has already been shown (Blok 1937), by applying equation (3) to some experiments on automobile spiral bevel gears described by Almen (1935), that the temperatures occurring on the tooth faces are an important factor in seizing.

For a better understanding of the extreme-pressure lubrication problem the temperatures at the surfaces of contact should be studied further on actual gears as well as on machines for testing extreme-pressure lubricants.

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CUTTING FLUIDS

By Professor O. W. Boston *

Cutting fluids comprise liquids, solids, and gases, which are applied to the tool point to facilitate the cutting operation. They may be used to accomplish any one of a number of purposes, singly or in combination, such as to increase the tool life for a given cutting speed, prevent distortion of the work during machining, reduce power consumption, break up the chips and help to remove them, and, at low speeds and light cuts, improve the smoothness of the machined surface. The performance of a cutting fluid, therefore, must be based on its success for each particular application to a specific metal, tool, and cutting operation. The properties desired in any cutting fluid may be summarized as follows:—

(1) High heat absorption, i.e. high thermal conductivity.

(2) Good lubricating qualities.

(3) High flash point, i.e. not liable to spontaneous combustion and the hazard of fire.

(4) Freedom from separation of solids at ordinary working temperatures.

(5) Stability, so as not to oxidize in the air or give rise to gummy deposits on the sliding surfaces of the machine.

(6) Components do not easily become rancid.

(7) No unpleasant odour when heated or after continued use.

(8) Not injurious to the skin of the operator either directly as from high acidity, or indirectly by contamination.

(9) Not injurious to the bearings of the machine, even if mixed with the machine lubricants or applied directly to the bearings.

(10) Non-corrosive to the work and machine.

(11) Fairly transparent so that the cutting action of the tool may be observed.

(12) Low viscosity to permit free flow from the work back to the storage tank, and to drip from chips.

(13) Low price and readily obtainable.

Types of Cutting Fluids Used. The many types of cutting fluids in use may be classified as follows:—

(1) Air used as a blast or suction.

(2) Aqueous solutions, such as plain water or water containing a small percentage of an alkali.

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* Director, Department of Metal Processing, University of Michigan, Ann Arbor, Mich., U.S.A.

(3) Emulsions of a soluble oil or paste. The soluble oils usually consist of a mineral oil, an animal or vegetable oil such as lard or tallow, plus an emulsifier such as soap. The mineral oils employed in the United States are usually Gulf-coast or mid-continent products. In some products, the fatty oils are sulphurized with flowers of sulphur, or sulphur monochloride. The emulsifiers consist of Twitchell base (turkey red oil), diethylene glycol, saponified phenol, saponified naphthenic acids (up to 15 per cent), or alcohol. In some cutting pastes a saponified mineral oil very high in soap content is used to give a grease-like consistency.

(4) Oils include straight oils, sulphurized oils, and chlorinated oils. Straight cutting oils include mineral oils, straight fatty oils, and compounded oils. Lard oil is the most commonly used fatty oil. The compounded oils consist usually of a mineral oil plus from 10 to 30 per cent of fatty oil.

Sulphurized oils consist of a sulphurized straight mineral oil with an active sulphur content not over 3 per cent. They are used as prepared but may be blended with a light mineral oil for certain work. Sulphurized base oils, in which the fatty oil, such as lard oil, is sulphurized with from 8 to 12 per cent of active sulphur added to it, form a base. For use, the highly viscous base oil is blended with 5 to 20 parts of a straight mineral oil of a viscosity such as 110 Saybolt sec. at 100 deg. F. to suit requirements. This resulting blend is quite transparent and has a low viscosity.

Chlorinated mineral oils of a stable variety have been found to have properties favourable to metal cutting operations.

The Performance of Cutting Fluids. The influence of cutting fluids on the *torque* (Boston 1933) of a 1¼-inch diameter twist drill, when operating at a speed of 60 ft. per min. and a feed of 0·015 inch per revolution, in each of nine metals is shown in Fig. 1. From this it is clear that the performance of one cutting fluid on a given metal is not a criterion of its comparable performance on another material. The *thrust* curves for the same materials plotted over the cutting fluid numbers are shown in Fig. 2. Again, no similarity is seen between the curves of any two metals. It also has been found (Boston 1932) that the *quantity* of the cutting fluid delivered to the point of the drill will influence the value of the torque and thrust. The torque is increased as the cutting fluid is reduced for all of the cutting fluids. The thrust, however, is increased for cutting fluids 5, 6, 7, 8, and 9 (Fig. 1), but is reduced for cutting fluids 2, 3, 4, and 11. It remains practically constant for cutting fluid No. 10.

Other experiments (Boston 1933) (Fig. 3) illustrate for the same eleven cutting fluids that the tangential *cutting force* in pounds is a function of the shape of the tool. Tool A had a 15-deg. back rake angle and no side rake, while tool B was similar to A except that it had 8-deg. back rake and 14-deg. side rake angle. The cutting speed was

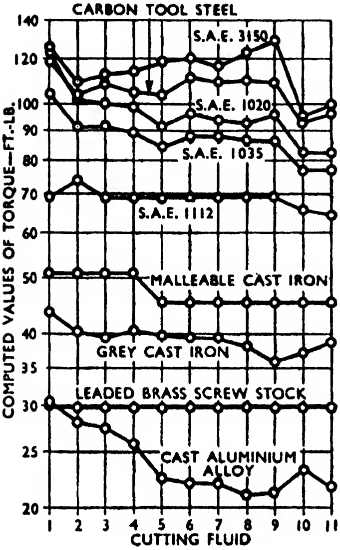

Fig. 1. Values of Torque for 1¼-inch Diameter Drill

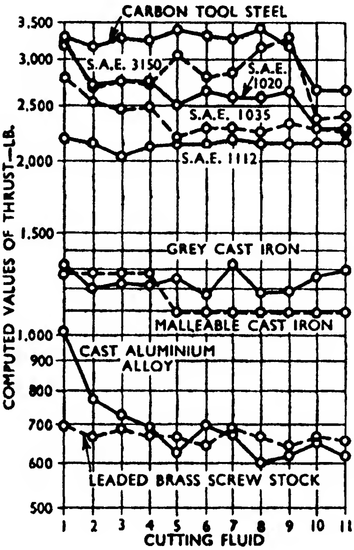

Fig. 2. Values of Thrust Corresponding to the Torque Values shown in Fig. 1

Drill operating at 0·015 inch feed and 60 ft. per min., when cutting metals with eleven commonly used cutting fluids. The cutting fluids are as follows:—

No. 1 Dry cutting.
„ 2 Water containing 1·5 per cent borax.
„ 3 Emulsion of 1 part soluble oil to 50 parts water.
„ 4 Emulsion of 1 part soluble oil to 10 parts water.
„ 5 A No. 2 lard oil.
„ 6 A light mineral oil.
„ 7 A heavy mineral oil.
„ 8 Oil No. 6 plus 10 per cent lard oil.
„ 9 Oil No. 6 plus 5 per cent oleic acid.
„ 10 A sulphurized mineral oil.
„ 11 A sulphurized lard-mineral oil.

66 ft. per min., the feed 0·0069 inch, and the depth of cut 0·100 inch. The stock was 1·8 inches in diameter. This shows that the saving in power through the use of cutting fluids is greater for tool A than for tool B.

Fig. 4 (Boston 1934) shows the time in minutes to *cut off* a bar of metal 1½ inches square in a power hacksaw with a blade 1 inch wide, 0·065 inch thick, with six teeth per inch, producing a kerf of 0·082 inch. The set of the teeth was right, left, two straight, etc. The feeding pressure was 119 lb. The cutting fluids, indicated by number, are the same as those described in Fig. 1. It is seen that dry cutting, cutting fluid No. 1, requires the greatest cutting time for all metals. The water compounds, Nos. 2, 3, and 4, give nearly equal values, slightly lower than those for dry cutting, for all of the metals. The two sulphurized oils, Nos. 10 and 11, caused the saw to cut fastest on all metals.

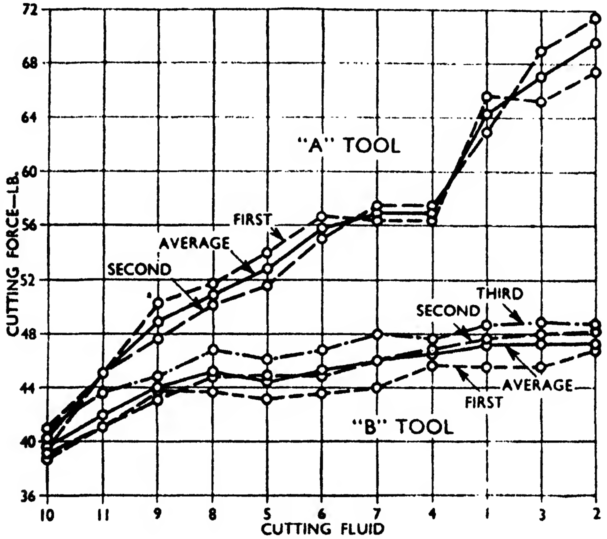

Fig. 3. Tangential Cutting Force in Turning Annealed S.A.E. 3150 Steel

The cutting fluids (set out under Fig. 1) are arranged in order of increasing force on tool A.

The same eleven cutting fluids were used further to lubricate a bearing in a bearing testing machine in order to determine the value of each as a *lubricant* (Boston 1932). Dry lubrication was replaced with clear water. A new combination of journal and bearing was used for each test. The journal diameter was 1·00 inch, and the bearing was 1·007 inches in diameter and 1 inch long. The journal was rotated at 100 r.p.m. The load was increased at a rate of 1,000 lb. every ten seconds. The beam load on the 15·75 inch long lever arm was plotted over the applied pressure for each cutting fluid and it was found that the lowest beam load was for water until a pressure of about 9,000 lb. was reached, when the beam load rapidly increased and bearing seizure took place at about 10,000 lb. pressure. Lard oil (No. 5) gave a low beam load, but seizure occurred at a pressure of approximately 13,000 lb. The maximum beam loads occurred for oils 6, 9, 7, and 10. Seizure occurred for cutting fluids 6, 9, and 7 at relatively low pressures. For No. 10, seizure occurred at the highest beam load of 223 lb. and a pressure of 14,000 lb. The sulphur in the plain mineral oil permits the generation of considerable friction and heat, but prevents seizure of the bearing. It is interesting to note that the beam load for a given pressure for the sulphur-base oil, No. 11, was less for most pressures than were those for No. 10. The fat of this oil presumably reduces the friction, although seizure occurs for a beam load of only 200 lb. at a pressure of only 11,000 lb. This characteristic represents the main difference between the blended sulphurized base oils and the sulphurized plain mineral oils.

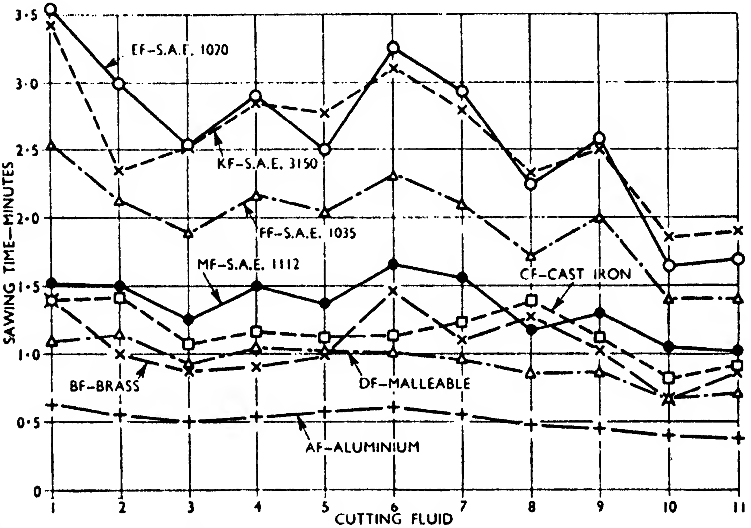

Fig. 4. Time required to Saw off Sections 1½ inches square of Each of Eight Metals in a Power Hacksaw with the Cutting Fluids set out under Fig. 1

Results of careful tests have shown that the straight line curve representing *cutting speed* as a function of tool life for a given set of conditions in turning, as plotted logarithmically, may be changed in its vertical position on a scale, as well as in slope from the horizontal, if one cutting fluid is replaced by another, with all other factors remaining constant. These factors cover the type and material of tool, tool shape, size and shape of cut, cutting process such as turning, drilling, broaching, grinding, milling, threading, and the analysis and structure of the material cut. Data are given below to substantiate this statement.

When turning 3·5 per cent nickel steel (Boston 1936) in an annealed condition, with high-speed-steel tools having 8 deg. back rake, 14 deg. side rake, 15 deg. side-cutting-edge angle, and $\frac{3}{64}$ inch nose radius, with a depth of cut of 0·050 inch and a feed of 0·0255 inch, it was found that the cutting speed for a 60-minute tool life was as follows:—

TABLE 1. CUTTING SPEEDS FOR A 60-MINUTE TOOL LIFE

Depth of cut, inches	0·050	0·100
Feed, inches	0·0255	0·0125
	Cutting speed, ft. per min.	
Cutting fluids:—		
Dry cutting	83	97
Sulphurized plain mineral or sulpho-chlorinated oil	89	120
Plain mineral oil (viscosity 110 deg. at 100 deg. F.)	97	112
Emulsion of 16 : 1 strength	103–105	120
Borax water	105	125
Increase over dry cutting	26·5% (22 ft.)	22·4% (28 ft.)

The increase of speed for dry cutting gained by changing the shape of cut is 17 per cent.

These various lines for the cut of 0·050 inch by 0·0255 inch were represented by the equation $VT^{1/9\cdot6}=$ a constant, for all cutting fluids except the sulpho-chlorinated mineral oil, which was represented by the equation $VT^{1/7}=163$. For the cut of 0·100 inch by 0·0127 inch, the lines were represented by the equation $VT^{1/7\cdot3}=$ a constant for dry cutting, water, and the emulsions, $VT^{1/7\cdot8}=192$ for plain light mineral oil, and $VT^{1/8\cdot4}=193$ for the sulpho-chlorinated mineral oil.

It has been observed in many tests that the cutting fluid which gives the longest tool life does not necessarily give the lowest tangential *cutting force* at the same time. Some tests give definite data to the contrary.

To obtain a superior *finish* on any metal cut surface, the size of the built-up edge should be reduced. The machinist accomplishes this by increasing the cutting speed, reducing the size of cut, grinding the tool to keener cutting edges and greater rake, and by applying a cutting fluid. The results of numerous tests suggest that the cutting fluid itself has the least influence on finish of all factors named for normally high cutting speeds. For low cutting speeds, as in threading, the cutting fluids do produce superior surface finishes.

The *temperature* developed at the tool point while turning various metals varies almost directly with the speed for a given cutting condition. When a cutting fluid is applied in turning steel, however, all other factors remaining constant, the temperature when cutting dry is highest, when cutting with an oil somewhat lower, and when cutting with water the lowest. It is believed that tool life in turning is a direct function of temperature so that it would be expected that the cutting fluid having the highest cooling property would produce the longest tool life. At the same time, it is reasonable to expect that an oil, as a result of its good lubricating properties, might reduce the heat generated at the tool point and, even in spite of its lower thermal conductivity, keep the temperature at the tool point lower than when the better refrigerant is used, and, as a result, provide longer tool life. Experimental data confirm this belief in some specific tests, but fail to confirm it in others. Presumably, lubrication and thermal conductivity are both of importance. It is believed that the ratio of lubrication to cooling varies for different specific applications. At the present time sufficient data are not available to establish a satisfactory rule governing this relationship.

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MODERN PROBLEMS IN GREASE LUBRICATION

By H. G. Catterall and J. Maitland *

Lubricating greases are often chosen for the lubrication of parts of machinery difficult of access during operation. The method of application has changed somewhat and to-day grease nipples, grease guns, and mechanical grease lubricators, etc., are used. These appliances are very welcome in that they tend towards cleanliness, tidiness, and economy, but themselves present problems which may not at first seem apparent.

Grease Nipples. There is a tendency to fit grease nipples on most types of machinery. The design and size of these nipples is not standardized, so that various types of grease guns and special fittings have to be employed. The hand grease gun generally operates with a pressure of 5,000 lb. per sq. in., and usually forces the grease past some type of ball valve through very fine clearances. Such a pressure often separates the lubricating grease into its constituent parts, i.e. mineral lubricating oil and soap. As a result the grease manufacturer to-day has to produce lubricating greases which will not separate under these pressures.

Grease gun application often necessitates the provision of a grease to suit the method of application rather than the lubrication of the bearing, and greases of softer consistency than normal have to be used. This softening of consistency can, to a degree, be countered by the use of a mineral oil of higher viscosity, but the fault still exists, namely, softer consistency of grease than is desired; increased consumption is a direct result.

The lack of standardization in grease gun nipples and guns gives rise to unnecessary difficulties. Thus, on a hot saw recently installed in a modern steelworks, there were five different types of grease nipples, so that five grease guns would have to be employed for that one machine. It is desirable to standardize on fewer designs of grease nipples with the ball valve as large as possible so as to provide an ample orifice ensuring the easy flow of grease to the bearing.

Spring Pressure Grease Cups. Prolonged pressure on both lime and soda base greases tends to separate such greases, releasing the mineral oil and leaving behind the soap. The oil released is often incapable of lubricating the bearing satisfactorily, whilst the residual soap usually

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* The Vacuum Oil Company, Ltd.

clogs up the grease cup outlet orifice. Many spring pressure cups are wrongly designed. With the flat-bottomed cup an appreciable quantity of grease towards the bottom of the cup remains stationary, and as a result, is under pressure for a long time, whereas the cone-shaped bottom allows continuous movement of grease.

Some grease cups are fitted with a grease gun nipple through which the grease is introduced, and the operative in this case pumps the grease through the nipple against the pressure of the spring behind the piston, the spring being in compression when the cup is fully charged. This is a typical condition for separating grease. In addition, the grease is materially softened while the cup is being filled, with the result that finally a grease softer than desirable is applied to the bearing.

The conditions described above make it difficult for the grease supplier, who is always limited by the conditions of application, rather than the mechanical requirements.

Application of Grease to Ball and Roller Bearings. Manufacturers of ball and roller bearings often recommend grease replenishment after a period of three or six months, and some state that the ideal quantity is one-third the capacity of the bearing. Some ball and roller bearings can only be lubricated through grease nipples. The operative tends to use the hand gun until no more grease can be introduced, i.e. until the bearing is filled. As a result the grease will churn considerably and may thin out materially. Churning is attended by heating, so that the problem is to provide a grease which will maintain a suitable texture and consistency at the operating temperature. To this end various-lime base greases are marketed, but have a limited application owing to their relatively low melting points. Soda-base greases, however, usually have higher melting points and so find a wider application.

All hand-operated lubricating devices, however, have certain drawbacks. Thus, lubrication depends upon the attention and goodwill of the operator. In refilling some lubricant is usually lost. The conveyance of the lubricant is effected at irregular intervals and generally in quantities which greatly exceed the correct amount for any given period with the result that much of the grease leaves the ends of the bearings in a totally unused condition.

Mechanical Grease Lubricators. The fundamental conditions of a modern lubricating method are maximum safety of working, combined with most economical consumption of lubricant and reduction in power cost. These conditions can best be met by a mechanical grease lubricator which ensures that every bearing of whatever size, load, or speed receives just the exact amount of grease necessary for its lubrication. The application of grease by means of mechanical lubricators has been advocated because of: (*a*) regularity of feed; (*b*) reduction in consumption and consequent economy; (*c*) reliability; and (*d*) increased bearing life as a result of (*a*).

Force feed grease lubrication has undoubtedly come to stay. Certain mechanical lubricators are open to criticism, chiefly owing to their inability to deliver per feed sufficient grease to cater for the exacting conditions encountered in the lubrication of the necks of rolling mills. Cases are on record, however, where even the smaller and cheaper mechanical lubricator satisfactorily delivers a suitable grease to bearings almost 500 feet away from the lubricator. Thus the force feed system of grease lubrication gives satisfaction in the quarry and crushing trade. To-day the largest jaw crusher in England (72 inches in the jaw) is satisfactorily lubricated by the force feed system; the slightest overheating in the Pittman bearing would have serious consequences.

Lubrication of Rolling Mills by Mechanical Lubricators. The observation that a softer grade of grease than is desirable has to be recommended becomes very serious when considering the lubrication of the roll necks on rolling mills. Here a lubricant has to maintain a film in the presence of large amounts of cooling water, and under severe bearing pressure and shock loads. Normal greases for such a purpose are usually in block form, and often contain mineral pitch, which provides the necessary adhesiveness and tenacity of film. These hard greases cannot be dealt with by any mechanical lubricator, while softer greases tend to be washed off the roll neck. This can be countered to a degree by the incorporation of different soaps, but there is plenty of scope for improvement in mechanical lubricators so as to make them capable of delivering more appropriate greases.

The usual type of mechanical lubricator will develop a pressure of 5,000 lb. per sq. in. Such a pressure can be encountered where small diameter piping is employed for conveying the grease from the lubricator to the bearing. Piping of at least $\frac{3}{8}$ inch bore should be employed.

With rolling mills, it is essential to admit the grease in the bearing at a point just before the pressure area. The supply of grease to a suitable grease groove is then maintained by means of the lubricator. This groove merely acts as a reservoir which has to be kept full. With the exception of roll necks of sheet mills, it is usual to apply copious amounts of cooling water to the roll necks of bar, rod, wire, section, and blooming mills, etc., up to as much as 18,000 gallons of water per hour for four roll necks. Under these conditions, and employing force feed lubrication, it is fairly easy to provide a lime-base grease of suitable texture and consistency which, with this amount of water, will keep the roll necks cool. Obviously a soda-base grease would be unsuitable as the grease film would be quickly washed away.

On the other hand, if a shortage of water arises, the lubricating problem becomes more acute. The lime-base greases used in force feed lubrication have melting points around 200 deg. F. Normally a roll neck operates at a maximum temperature of 120 deg. F., but with water shortage, the temperature rises to as much as 160 deg. F. It is then that the normal lime-base greases fail to give efficient lubrication.

For certain rolling mill applications, and provided a suitable mechanical lubricator is available, extra bearing life is obtained. As an example, there is in this country a 32-inch 2-high reversing mill which rolls 150,000 tons of steel before the whitemetal bearings are changed.

Lubrication costs in rolling mills are reduced by the introduction of the force feed system, usually with an increase in efficiency, despite the fact that the lubricating cost may be reduced by three-quarters. There is, however, a tendency to reduce grease feeds beyond a safe margin, though cases are recorded where increased but not excessive feed has almost doubled bearing life.

American practice, however, has concentrated more on systems depending upon compressed air or hydraulic power for operation. They are usually very successful, but again with greases of soft consistency. These systems are in vogue for the lubrication of semi-continuous wide strip mills, for both hot and cold rolling. Here, however, the 4-high roll stands are usually fitted with roller bearings which may have to be grease lubricated. The main problem is reliability and regularity of feed to keep the bearings full, although the lubricator may be delivering grease 1,000 feet distant. The systems are elaborate yet reliable, and are usually fitted with electric alarms which indicate when a fault develops.

The lubrication of roll necks is still largely effected by means of block greases. These are extremely hard and usually contain soda soaps; in consequence the greases emulsify somewhat in the presence of the cooling water. These greases, however, often contain mineral pitch, which prevents too rapid an emulsification, and ensures a more reasonable consumption.

Sheet Mill Lubrication. The usual European practice is to produce high-class steel sheets on 2-high mills with roll temperatures up to 950 deg. F., and resultant roll neck temperatures of 350–600 deg. F., the latter temperature usually depending upon whether water is applied to the roll necks, although roll turning and the resultant "shape" of the rolls also play a large part.

Lubrication is usually carried out by applying grease blocks to the roll necks. It is not commercially possible to manufacture greases with sufficiently high melting points and of suitable texture to meet these operating temperatures, and lubrication is best obtained by collecting grease after use ("returns") and reboiling on site these returns with a proportion of new grease which varies from 10–30 per cent, according to the quality of the grease and the operating conditions. The reboiled mixture is cast into blocks and, after storage, used on the necks. In effect, during their lubricating life, the lighter volatile fractions are driven off from the percentage of new grease employed and as a result the "returns" have a much higher melting point than the new grease.

As the operating temperatures cause the grease to carbonize, lubricants must be used which have little tendency to do so. Much assistance is given in this respect if cooling water is applied to roll necks, despite the fact that the water must of necessity be small in volume. Greases which carbonize readily give rise to increased bearing wear, as well as an increase in power consumption. Furthermore, the increase in carbon leads to a dry lubricating film, and this dryness of film is almost wholly responsible for the imperfections which occur in sheets due to so-called "splashing" of lubricant, but which is actually a transference of the dry carbonized grease, usually in powder form, from the roll neck on to the sheet during hot rolling, resulting in depressions on the sheet surface and causing many rejections when the sheets are inspected after cold rolling.

Cold Drawing of Steel Tubes. Cold-drawn steel tubes provide a difficult problem, as it is necessary to lubricate both the outside and inside of the tube, thereby lubricating the die and the mandrel while effecting the drawing operation. Mineral oils are not satisfactory for this purpose, as they have not sufficient film strength, and normal lubricating greases and "soluble" oils are unsuitable. However, there are types of soda-base greases the water emulsions of which give very suitable tube drawing compounds. Their use prevents "pick-up" on either the die or mandrel, and results in excellent finish on the tube. Lubrication is obtained by immersing the tubes in a solution of the compound, thereby coating both the inside and outside of the tube with lubricant, so that both the die and inner mandrel are eventually well lubricated. It is now possible to produce steel tubes with a 40 per cent reduction in cross-sectional area with only one draw.

Grease Specifications. These are often drawn up to ensure economy in buying, whilst attempting to protect the quality of the grease to be supplied. Usually the specifications are based upon an analysis of a grade which gives satisfaction. Therefore the presentation of a specification to a manufacturer of greases is practically the same as presenting him with a formula. A grease manufacturer's experience, coupled with a sound knowledge of the practical requirements, can ensure that the final product will give satisfactory performance, so that the specification will be met and possibly even an improved grade submitted.

A prominent feature of most grease specifications is melting point (method often unstated). Usually, melting point is not a determining factor in the lubrication, for this undoubtedly ***depends upon the provision of a lubricant which has correct texture and consistency under its operating conditions***. Again, many specifications stipulate the amount of mineral oil to be used in the grease without even mentioning the viscosity of the oil. This might result in the submission of a grease made, say, from spindle oil, one of the cheapest mineral oils available. A supplier, however, who knew the practical requirements would possibly employ an oil with a higher viscosity, and costing more initially. His final product might not be considered competitive; nevertheless it might represent the best value of the samples submitted against the specification. For grease manufacturers to quote slavishly against, and for buyers continually to purchase on specification, does retard progress.

CHARACTERISTICS OF GEAR LUBRICANTS REVEALED BY THE FOUR-BALL EXTREME-PRESSURE LUBRICANT TESTING APPARATUS*

By D. Clayton, B.Sc.†

Apparatus and Method of Test. A description of the four-ball apparatus, supplied by the N.V. de Bataafsche Petroleum Maatschappij, has already been published.‡ A paper describing the use of the apparatus for tests of ordinary lubricants appears under Group IV,§ and can be referred to for further details of the present method of test and of the general results obtained. Briefly, a ½-inch diameter hard steel ball is rotated under load in the cavity formed by three other similar balls clamped in a cup containing the lubricant to be tested. The mean diameter of the closely circular impressions worn on the three clamped balls, the time to seizure, and the coefficient of friction at the various stages of the test are found; the first two are plotted against load. If seizure does not occur, the impressions on the clamped balls are very closely equal in diameter to the values calculated by the Hertz formula (see, for example, the Hertz line in Fig. 2).

Extreme-Pressure Lubricants. Five extreme-pressure lubricants, obtained in November 1935, have been tested. Photographs of typical impressions on the clamped balls, formed when seizure did and did not occur, are reproduced at *a* and *b* in Fig. 1. The magnification for *b* is higher than for *a*, to reveal the appearance of the impression more clearly. The mean diameters of the wear impressions and the times to seizure for three oils, A, B, and C, are plotted in Fig. 2. (In the seizing-time–load diagrams the arrows near the top of the diagram indicate that in these tests there was no seizure within the 60 seconds of the test; where a number is added, this shows the number of seconds to seizure, the point falling above the limit of the diagram.)

A notable feature of the seizing-time–load diagram is the rather wider scatter of the observation points than is usual with other types of lubricants. In view of the consequent uncertainty of the positions of the mean curves, the differences cannot be regarded as significant,

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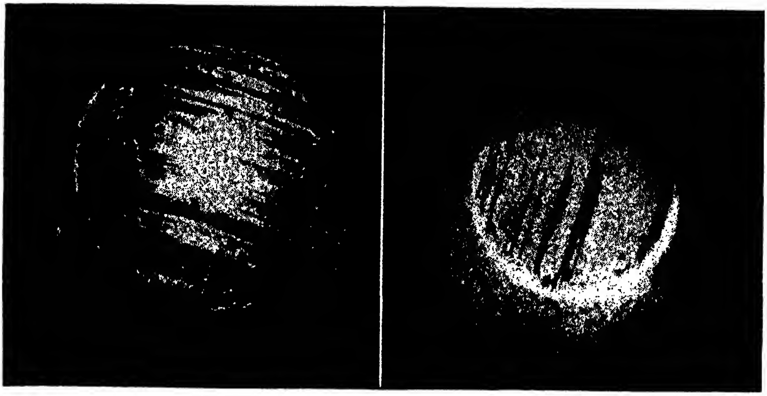
* Work performed for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† National Physical Laboratory, Engineering Department.

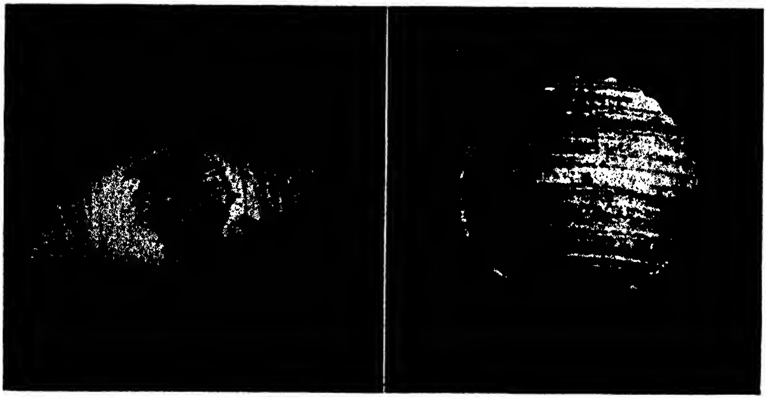
‡ Boerlage, G. D., *Engineering*, 1933, vol. 136, p. 46.

§ See the author's paper in Group IV.



a Extreme-pressure oil A. 130 kg. Seizure. ×20.

b Extreme-pressure oil A. 50 kg. Non-seizure. ×80.



c Extreme-pressure oil C. 130 kg. Seizure. ×20.

d Oil X (zinc oxide). 82·5 kg. Seizure. ×20.

e Mineral base oil of oil X. 81 kg. Seizure. ×20.

Fig. 1. Wear Impressions

except as regards oil B, which did not take such a high load before seizure occurred, but did not seize in any shorter time than the other oils if the load was sufficiently high. Apart from oil C, the wear results were remarkably consistent. The results for two other extreme-pressure lubricants were similar as regards time to seizure to oils A and C; the breakdown loads were respectively 100 and 110 kg. and the wear was the same as with oils A and B at corresponding loads.

Extreme-pressure oil C was peculiar in its seizure characteristics. Whereas the friction-time charts of the other oils were always of the same type, that at *a* in Fig. 3 being typical, those for oil C varied. Some had a well-defined peak, as at *b* in Fig. 3, which differed from those of the other oils in being sharper and having a lower maximum coefficient of friction (μ); the wear was accordingly less, the diameters of the impressions plotting as the portion *a* of the wear-load curve, Fig. 2. The other friction-time charts were of the type shown at *c* in Fig. 3, where only a slight rise of friction occurred, and a longer time elapsed before recovery to the final constant friction value; the wear was less than when the previously described type of seizure occurred, the diameters of the impressions plotting as the portion *b* of the wear-load curve, Fig. 2. The impressions in most of the latter cases had

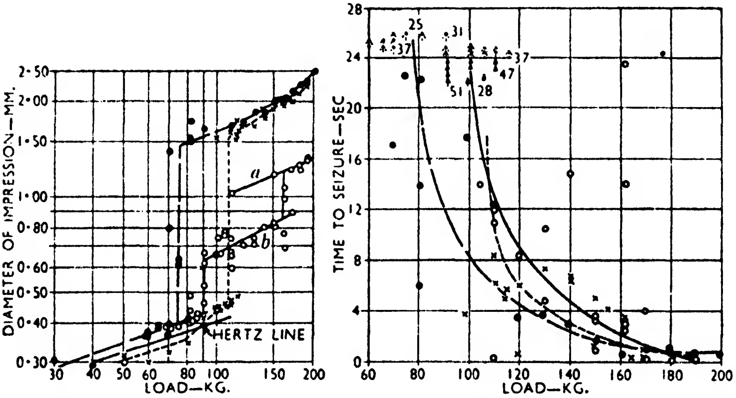


Fig. 2. Mean Diameters of Wear Impressions and Times to Seizure for Three Extreme-Pressure Lubricants

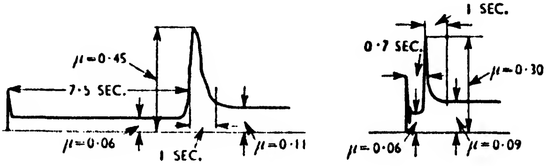
×---×---× Oil A.

●—●—● Oil B.

○———○———○ Oil C.

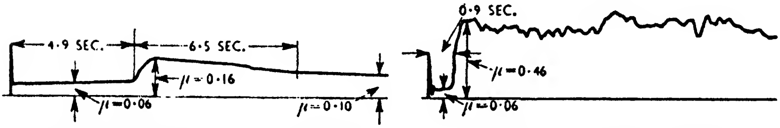
irregular, not circular, boundaries (Fig. 1*c*), as though cohesion between the balls had occurred only over part of the contact area; as the shape varied from test to test, the values for the size of impression do not plot so regularly.

The quicker recovery from seizure (where a well-defined peak was obtained), the lower maximum coefficient of friction, and the smaller wear, which was doubtless related with the two other features, must be counted as important advantages of this oil. For should the operating circumstances in a mechanism in practice lead to a condition of incipient seizure between two contacting surfaces, there would be a better chance of recovery: the amount of material removed would be less, and, the heat generated being less, the progression to full seizure would be less likely to occur. Similarly with high point contact during running-in, the local cohesions and breakdowns would be much less likely to develop into general seizures. Against these advantages must be counted the rather lower breakdown load than for three of the other extreme-pressure oils tested.



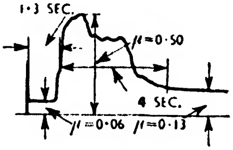
a Extreme-pressure oil A, 130 kg.

b Extreme-pressure oil C, 180 kg.



c Extreme-pressure oil C, 130 kg.

d Base oil of oil X, 97½ kg.



e Oil Y (containing ZnO), 110 kg.

Fig. 3. Friction-Time Charts

The maximum coefficient of friction at seizure increased with load from about 0·4 to 0·5 in all cases except with oil C with which, where there was a well-defined peak, it varied from 0·2 to 0·35, and, where there was no peak, it averaged about 0·15. The time to recovery from seizure averaged rather more than 2 seconds with oil B, and was about 1½ seconds in all other cases; this was the time taken to approach the constant friction stage, and did not in all cases express the sharpness of the peak.

Oils Containing Zinc Oxide. Tests were carried out on a gearbox oil X; a heavier, back-axle oil Y, and on the plain mineral base oils (Fig. 4). Castor oil containing zinc oxide was also tested. The striking feature about the tests of the mineral base oils was the long, ragged seizure. Fig. 3*d* is the first portion of a typical friction-time chart for the higher loads; at lower loads the major irregularities were rather less pronounced. The addition of the zinc oxide had the effect of greatly shortening the time of recovery, and of making the seizure peak much smoother (cf. Fig. 3*e*); in fact these oils were only slightly inferior in these respects to castor oil.*

Although the viscosity of one was three times that of the other, the two mineral base oils behaved generally in a similar manner, the X base oil being slightly the inferior. The breakdown load was just over 60 kg. in each case, though the X base oil gave two seizures at lower loads; this was exceptional and was disregarded in drawing the mean curves. Over the range of loads causing seizure the wear tended to be rather higher for the X base oil, thus corresponding with the generally longer time to recovery from seizure; with the Y base oil recovery occurred after 20–40 seconds, whereas with the X base oil the range extended from 20 seconds to beyond the full 60 seconds of the test. In these last cases the motor was switched off before recovery had occurred, and the wear would have been rather greater had the test been continued.

The time to recovery with oil X was 3–4 seconds and with oil Y 4–7 seconds. Corresponding with the reduction of the time to recovery, the wear was also reduced, rather more with oil X. In this seizure region the wear was very closely the same as with castor oil.

There was a marked difference in the appearance of the wear impressions. Whereas with the base oil they were very scored and coloured debris had collected round the edge, with the zinc oxide the impressions were much smoother, little debris had collected, and there was no coloration. Photographs of typical impressions for oil X and its base oil are shown in Fig. 1*d* and Fig. 1*e*.

The breakdown load for oil Y was almost the same as for the base oil, though two seizures at higher loads were mild, with small wear

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* See the author's paper in Group IV.

With X, however, the breakdown load was 75 kg., higher than with the base oil and equal to that of castor oil. A peculiarity with X was that with the low loads not causing seizure, the wear impressions were unusually large, the mean curve for this portion of the wear-load diagram coming well above the Hertz line. The appearance of the impressions was similar to that when a mild seizure occurs, but there had been no rise of friction. It was thought that the zinc oxide must be acting as an abrasive, but there was no evidence of this when the oil Y and the castor oil containing zinc oxide (below) were tested.

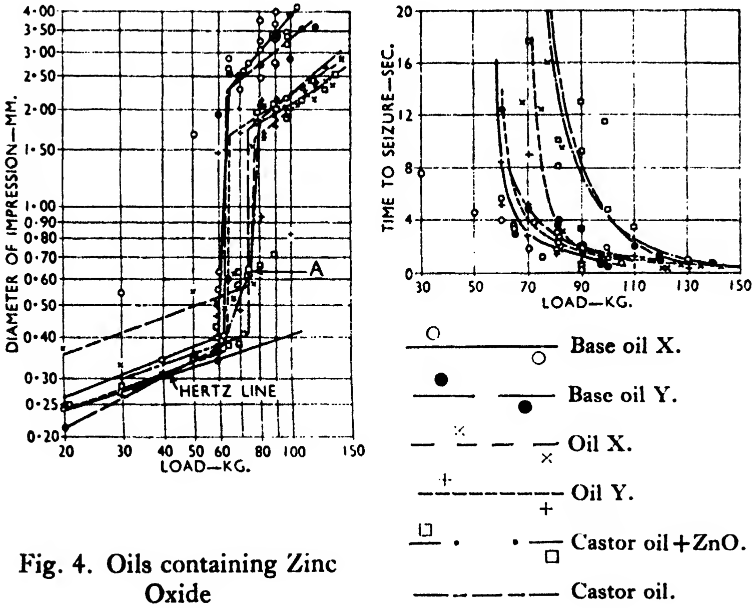

Fig. 4. Oils containing Zinc Oxide

Both base oils showed the characteristic common to practically all the ordinary mineral oils tested, of generally not giving a normal seizure within the one minute of the test if seizure did not occur within about 5 seconds of the start. This is in marked contrast with the fatty oils and the extreme-pressure oils, which will give a normal seizure peak after as long as 15 seconds or more. With this limitation it is difficult to draw the upper part of the seizing-time–load curve for mineral oils, and in comparing these curves this feature should be taken into account.

The coefficients of friction were rather variable and there was no noticeable difference between the values for the oils and their base oils. The maximum at seizure increased with load from 0·40 to 0·55 approximately.

As the addition of the zinc oxide had the effect of making the mineral oils like castor oil in some respects, it was naturally interesting to see whether the addition to castor oil itself would have any effect. The results included in Fig. 4, on which the mean curves for castor oil from the other paper (Group IV) have been superimposed, show that it has very little effect; mild seizures after long runs appeared, however, giving wear values in the region A, and the wear at the highest loads tended to be rather less than for castor oil. Generally, however, the wear was of a similar order, and the times to seizure, the breakdown load, the times to recovery, and the coefficients of friction were all practically identical for the two oils.

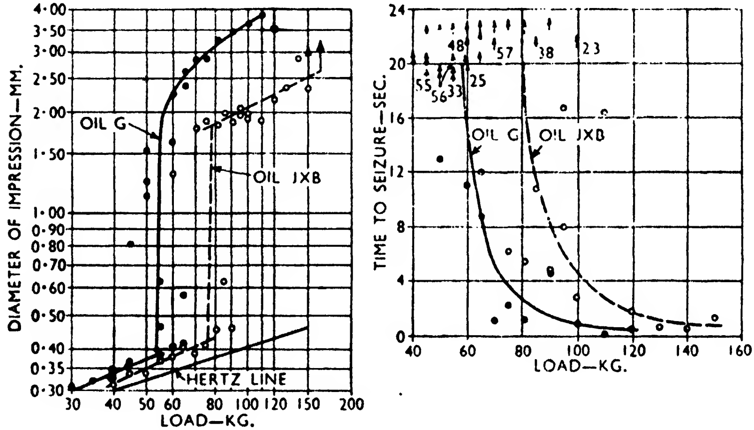

Fig. 5. Mineral Gear Oils

Two Mineral Gear Oils. Of two additional gear-type mineral oils, one was a marketed oil G, and the other was an oil which had only been refined to a limited extent and was not marketed; it is used as a reference oil with the identification mark "JXB". Oil G had a breakdown load (Fig. 5) of about 55 kg., rather less than the base oils of X and Y referred to above. The wear with seizure was of the same order as that of the base oil of X, but the curve was not quite so steep. The oil JXB had a breakdown load of about 75 kg., equal to that of castor oil and extreme-pressure oil B; the wear with seizure was equal to that of castor oil at low loads and less at the highest loads, but above that of extreme-pressure oil B. This low wear was doubtless related to the quick recovery from seizure; this occurred in 3–4 seconds, whereas with oil G it took 20–30 seconds. The maximum coefficient of friction at seizure with oil G varied from 0·35 at breakdown load to 0·5 at the highest loads, and with oil JXB from 0·45 to about 0·55; the values were about the same for the same loads.

The superior qualities of oil JXB are interesting because the oil probably contains a good proportion of unsaturated constituents.

Viscosity. As described in the paper in Group IV, it is inferred that no load-carrying fluid film is formed between the balls. To the extent that viscosity does play a part in gear tooth lubrication, the apparatus is under a disadvantage as regards direct tests of gear lubricants. To be able to investigate the other qualities without interference from viscosity is, however, regarded as a great advantage for research purposes.

Coefficients of Friction. The maximum coefficients of friction at seizure have been given above. The values before seizure and after recovery are not very precise owing to variance in the recording mechanism. Generally, however, the coefficient before seizure was about 0·06–0·07 for the mineral oils; it was about 0·06 for the extreme-pressure oils. The latter figure is not quite as low as for castor oil, so that these oils do not owe their good qualities to an abnormally low coefficient of friction. A low value is desirable, however, from the point of view of heat generation, which must play a part in breakdown. The coefficient of friction after recovery was generally 0·10–0·12 for the mineral oils and those containing zinc oxide; it was rather less for extreme-pressure oil C, but generally about the same for the other extreme-pressure oils.

The Significance of Various Characteristics. The importance of high breakdown load is obvious. The qualities of the oils which result in low maximum friction at seizure and quick recovery from seizure are probably of importance for running-in, apart from the amount of damage which would be done should breakdown occur in practice. For, as suggested above, when high point contacts occur, with local cohesion, the removal of material is much more likely to remain local. Some support for this view is found in the smoother appearance of the impressions on the balls with extreme-pressure oils than with the mineral oils at loads not causing seizure; in fact with some of the extreme-pressure oils it was extremely difficult to find the impressions at low loads. It is understood, also, that a number of firms are using extreme-pressure oils for running-in gears, and then allowing the use of ordinary gear oils subsequently.

The slope of the seizing-time–load curve is important in that within the range of loads which would lead to seizure, the oil which takes the longer time to seizure will allow a longer time for the contacting parts of the surface to change and so avoid a seizure. This is particularly true when there is movement of both elements as in gears. It will also confer a greater immunity from the effects of odd shock loads.

Conclusions. In tests of the seizure characteristics under conditions of high pressure between hard steel balls, the order of increasing quality as regards breakdown load and wear with seizure is: ordinary mineral oils, oils containing zinc oxide and castor oil, and extreme-pressure lubricants. The time to recovery from seizure, which is probably related with the wear, is long with the ordinary mineral oils (20–60 seconds), intermediate with castor oil and oils containing zinc oxide (3–6 seconds) and very short with extreme-pressure oils (1–2 seconds). There is some overlapping, one of the extreme-pressure lubricants having a breakdown load no higher than castor oil. Further, a moderately refined mineral oil has a breakdown load and wear of about the same order as castor oil.

PRACTICAL CONSIDERATIONS OF LUBRICATION AND LUBRICANTS IN REGARD TO ANTI-FRICTION BEARINGS

By T. W. Cooper*

Anti-friction bearings require very little lubrication to enable them to run satisfactorily, as it is merely necessary to maintain oiliness of the relatively moving components. The ideal for this purpose is a dense oily mist. Quite apart from lubrication, however, a matter of almost equal importance for consideration is the protection of the highly finished and hardened steel surfaces from moisture and other erosive agents, and for this purpose grease forms the most suitable medium. It is simple to apply and retain, and requires a minimum of attention, and is to be preferred wherever conditions of high temperature and, or alternatively, high speed or other special circumstances do not preclude its use. The first consideration when arranging for the lubrication of anti-friction bearings is therefore to determine whether oil or grease is better suited to the operating conditions.

OIL LUBRICATION

Types of Oil. When oil is used, it should preferably be a straight mineral grade to avoid the development of acidity in service, and where high speeds and, or alternatively, high temperatures are involved, it should be selected with a viscosity to suit the operating conditions. For high speeds and moderate temperatures a spindle oil, with a viscosity of about 300 sec. Redwood at 70 deg. F. is satisfactory, whilst if the operating temperature is in the range 140–220 deg. F., a medium-grade engine oil, with a viscosity of the order of 1,500 sec. Redwood at 70 deg. F. is to be preferred. For temperatures over 220 deg. F. a steam-cylinder oil should be used. In selecting the oiling system and viscosity of the oil, it should be remembered that, other things being equal, the bearing temperature increases with the quantity and viscosity of the oil in the housing. The quantity and viscosity of the oil should, therefore, be kept as low as possible on high-speed applications.

Method of Application. The method of supplying the oil to the bearing must be chosen with due regard to operating conditions, and

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* The Hoffmann Manufacturing Company, Ltd.

a brief description of some of the systems in common use may be of interest.

(*a*) Splash lubrication from a sump in the bearing housing with the oil level such that each ball or roller dips into it in turn as the bearing rotates, or, in the case of an enclosed mechanism, from a sump in the machine casing, from which oil is splashed and allowed to drain on to the bearings.
(*b*) Circulation of a small quantity of oil by means of a rotor attached to the shaft, drawing oil from a small reservoir, usually in one of the bearing housings, and delivering it to all the bearings on the shaft.
(*c*) Circulation of a larger quantity of oil by means of a separately driven pump, drawing oil from a separate storage tank.
(*d*) Slow drip feed, with gravity supply or pump circulation.
(*e*) Wick feed.

(*a*) is usually confined to slow or medium speed horizontal shafts, (*b*) to high-speed light-duty vertical spindles such as are used in certain woodworking machinery, and (*c*) to high-temperature, and, or alternatively, large high-speed applications where it is desired to utilize the oil as a cooling medium also. System (*d*) is obviously applicable to a wide variety of conditions, whilst (*e*) is usually confined to high-speed lightly loaded bearings.

Grease Lubrication

Types of Grease. Lubricating greases for anti-friction bearings are normally made with about 80–85 per cent of suitable mineral oil solidified by the addition of a lime or soda soap, and must be free from particles of solid matter or any "fillers," as these cause wear. The consistency of a grease must not be sufficiently low to permit escape from the housings and not so high as to cause channelling under working conditions. The range between these extremes is considerable, but in general terms it may be said that for low speeds and temperatures the consistency should be a little greater than that of vaseline, increasing as the conditions become more arduous.

The behaviour of a grease in service also depends on the structure or texture imparted to it during manufacture. The texture may vary from smooth or buttery to a high degree of stringiness, greases with the latter characteristic often being termed sponge or fibre greases, and described as short, medium, or long fibre, to indicate the degree to which they possess stringiness. Both extremes should be avoided, as a smooth grease tends to channel and leave the bearing dry, whilst excessive fibre results in severe mechanical agitation of the grease, particularly at high speed, causing a high running temperature and difficulty in retaining the grease in the housing.

Lime-soap base greases should be of low to medium consistency with short fibre, and a melting range of 200–220 deg. F. Such greases are suitable for normal speeds and continuous operation at temperatures up to 100–110 deg. F.

Soda-soap greases fall broadly into two grades, one intended primarily for high speeds and the other for high temperatures. Both have a melting range of about 300–350 deg. F., generally with greater consistency than the lime-soap greases, and preferably rather short fibre, the high-temperature grades being suitable for continuous operation at temperatures up to about 220 deg. F. The high-speed grades have usually rather lower consistency than those for high temperatures, and are suitable for sustained temperatures up to about 160 deg. F. They also have an increased tendency for oil separation to occur under working conditions. This separation must only be slight, but an amount sufficient to provide a film of oil over the working surfaces of the bearing is obviously beneficial for high-speed applications. Excessive separation results in loss of oil from the housings, leaving only a dry soap residue which cannot lubricate effectively, and thus causes wear. Such a property does not readily lend itself to precise definition, but for practical purposes a guide may be obtained by the condition of a grease after standing for some weeks in its container. The surface of a grease of this type should then be sensibly oily, but without sufficient free oil to produce visible flow when the level is altered.

Method of Application of Grease. Given a suitable grade of grease, and housings arranged so that it is adequately retained, the efficiency of the lubrication depends very largely on the care with which the initial charge is applied. To maintain continuous lubrication it is essential that a body of grease shall be in intimate contact with the sides of the cage so that rotation of the bearing will disturb the surface of the grease and draw a supply from it to the working surfaces. The optimum amount of grease to employ will vary a little with the grade used and the speed, since overfilling, particularly with the more fibrous greases, will cause overheating. But with most greases the bearing should be completely filled, and the space in the covers lightly filled, flush with the sides of the bearing. On starting up, some of the grease in the bearing will be expelled from it, and must be accommodated in the covers, which should for this reason not be packed tightly, but, as already stated, only lightly filled. It is essential that the initial charge should not be entirely smoothed down into the back of the covers, as if this is done, no disturbance of the charge takes place during running, and as it is then impossible for it to lubricate the bearing, wear will quickly follow.

If the foregoing procedure is followed, some temperature rise will occur immediately on starting, due to churning, but will subside in the course of a few hours. A peak temperature immediately after starting of, say, 110–120 deg. F. with a lime-soap grease and 140–160 deg. F. with a soda-soap grease at a higher speed at normal atmospheric temperature, is quite in order and need occasion no alarm. Prolonged running at temperatures above those already indicated as suitable for the various types, will cause a grease to break down into its constituents of soap and oil, thus rendering it unfit for further service, and necessitating a complete recharge with new grease. Should the temperature show no signs of falling after an hour or two, the machine should, therefore, be stopped and allowed to cool, after which, assuming that the initial peak was due to grease churning, it will probably be found that excessive heat is not developed on restarting.

A bearing thus set to work should require only very infrequent replenishments, all that is necessary being to make up the slight loss due to shrinkage, and to disturb the body of grease in the covers to guard against any tendency for channelling to occur. The intervals at which this is necessary vary a little with the operating conditions, but generally from 6 to 12 months is satisfactory. Some indication may be obtained from the temperature rise of the bearing housing above that of the surrounding parts, particularly on high-speed applications. As a rule complete absence of temperature rise denotes complete lack of grease agitation and therefore a risk of insufficient lubrication. Under such circumstances small additions, at, say, daily intervals, should be made until a temperature rise of about 30 deg. F. is produced. Conversely, grease should not be added to a bearing which is running with a high temperature rise, but is otherwise satisfactory, since this will only cause further heating.

Maintenance of Supply. The design of the housing has an important function in respect of supply. Wherever possible, bearings should be completely enclosed, and where the shaft protrudes it is advisable to spigot the end cover in order to maintain concentricity. Under normal conditions it is sufficient to bore the cover with a fine running clearance of the shaft and turn a series of annular grooves in the bore (Fig. 1*a*).

With vertical shafts, no special precautions are necessary when grease is used, as it will be maintained in contact with the bearing by gravity. With oil, however, some arrangement for circulation is necessary, and this can be effected either by capillary action or by agitation set up by the rotating parts themselves, such as a flinger arranged below the bearing which dips into a reservoir formed in the end cover. In order to prevent the oil from leaking past the lower cover, an overhanging thrower can be arranged to fulfil the dual purpose of pumping and retention (Fig. 1*b*).

With horizontal shafts, should there be a tendency for the lubricant to be forced against the end cover and so cause leakage, a rotating flinger should be arranged to cover the aperture. By this means the lubricant is flung outwards towards the periphery of the housing and any oil which drains down the end cover is led away from the aperture by means of the channel or gutter formed on the lip thereof (Fig. 1*c*).

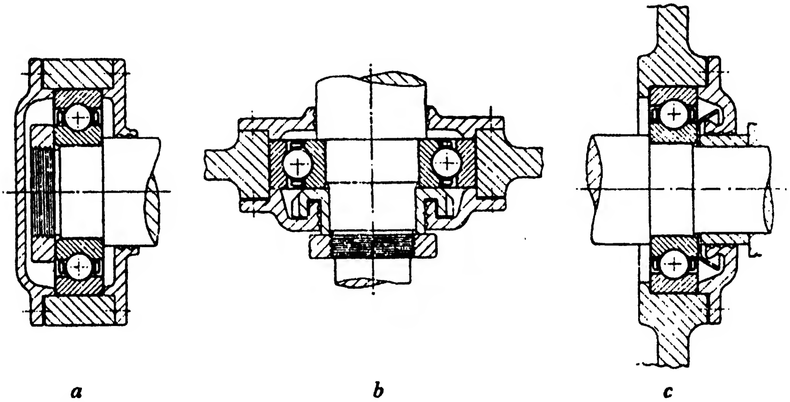

Fig. 1. Design of Housings for Horizontal (*a* and *c*) and Vertical (*b*) Shafts

THE TESTING OF LUBRICANTS

Tests are required for two purposes, firstly to determine if a previously unexamined grade will satisfy the service requirements and, secondly, to determine if further supplies of an approved grade are similar to the original.

Testing of Oils. The laboratory tests usually carried out on lubricating oils are as follows:—

(1) Viscosity at temperatures selected to embrace the working range of the bearings for which the oil is required.

(2) Freedom from corrosive tendencies. This involves the determination of any free acidity, both organic and inorganic, usually supplemented by exposing polished steel and brass specimens to the oil for a standardized period, during which no staining should occur.
(3) Analysis to detect and estimate compounding ingredients such as vegetable or animal oils, metallic soaps, graphite and sulphur compounds.
(4) With compounded lubricants it may also be necessary to check oxidation tendencies by measuring the rate of oxygen absorption under standardized conditions.
(5) Certain supplementary tests such as specific gravity and flash point tests are of value in checking routine supplies.

If after such tests uncertainty exists as to the suitability of an oil, a practical running test simulating operating conditions is always advisable, after which the test bearing should be carefully examined for any signs of corrosion or lapping or tendency of the oil to form gummy deposits or sludge.

Testing of Greases. Mainly because of the influence of the texture of a grease, laboratory examination alone is not sufficient to prove a sample satisfactory, although it will detect a completely unsatisfactory sample; and a practical running test is always to be recommended on a previously untried grade which satisfies the requirements of the laboratory tests. The laboratory tests usually carried out on greases are as follows:—

(1) Consistency. This is tested in the unworked and worked conditions, any marked change after working being regarded with disfavour.
(2) Melting range.
(3) Type and proportion of soap used, the content being usually about 15–20 per cent.
(4) Determination of free acidity, both organic and inorganic, or alkalinity, experience showing that as a rule a free organic acidity up to 0·5 per cent is preferable, but should not be exceeded on account of possible corrosive tendencies. This should be supplemented by the exposure of test specimens to a thin film of the grease.
(5) Determination of moisture content, which should be as low as possible.
(6) Detection of any fillers or solid matter.

(7) Examination for storage stability by actual storage test, sometimes supplemented by measurement of the rate of oxygen absorption under standardized conditions.

The practical running test should be carried out on a size and type of bearing in common use, at a high average speed for that size, and in housings of good representative design. Facilities should also be available for artificially maintaining the temperature at any desired level for simulating various operating conditions. The duration of the tests should be as long as possible, but critical examination of the parts after a period of about 500 hours will usually enable anyone with experience to forecast the results likely to be obtained in service. If a grease is satisfactory in other respects, a shorter period at elevated temperatures will usually give sufficient indication. During the running tests, careful record should be made of the temperatures developed, particularly during the initial period, as the rate at which the initial peak temperature falls is of considerable importance, and a record of the leakage from the housing, if any, should be made. After the test the parts should be carefully dismantled and the grease examined for signs of channelling, excessive change of consistency and oil separation, and discoloration. The test bearing should be taken apart and the working surfaces examined for signs of lapping, deposits, and corrosion. The working surfaces of the separating cage should also be examined for any signs of wear due to insufficient lubrication.

THE LUBRICATION OF CHAINS, WITH PARTICULAR REFERENCE TO TRANSMISSION CHAINS OF THE ROLLER TYPE

By R. Coulson *

An examination of any one of the various types of power transmission chain will reveal that the number of parts forming bearing surfaces is a very high proportion of the whole: in point of fact the proportion reaches 100 per cent in certain non-precision chains, e.g. the malleable cast type. Particularly, therefore, for high-speed, high-duty precision chains will the necessity for adequate lubrication be appreciated if efficient performance and long life are to be obtained.

Chains may be divided broadly into two classes: (1) chains used for conveying materials or articles; (2) chains for the transmission of power. Included in the latter are chains used for the driving of camshafts on internal combustion engines of all sizes from the 350 cu. cm. capacity motor cycle engine to the 12,000 h.p. Diesel engine for marine propulsion.

The lubrication of conveyer chains is in some ways different from that of the lubrication of power transmission chains, since questions of temperature may be involved and the material handled, if of an abrasive nature and liable to come into contact with the chain, may, unless precautions are taken, mix with the lubricant and form a very efficient grinding compound. The most satisfactory way of dealing with such a condition is to arrange for lubrication outwards from the centres of the bearing pins, by fitting grease nipples in suitable drillings in the pins and forcing grease into them, by either a hand- or an automatically operated grease gun. Failing the adoption of such a scheme, it may be preferable to run the chains entirely free from lubricant. The lubrication of transmission chains of the roller type is, however, the more particular object of this paper and the general problem involved—though also present with conveyer chains—is the lubrication of a number of bearings in which the surfaces to be separated by the film of lubricant are composed of case-hardened steel. The motion of these surfaces in relation to one another is intermittent and oscillatory and lubrication has to be carried out while the whole system of bearings is in motion.

Certain empirical values have been established as desirable maxima for pinion speeds and with these the rotational speeds of the parts comprising the chain joints are bound up. Taking into consideration

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* The Renold and Coventry Chain Company, Ltd., Manchester.

the diameters of the bearing pins for the various pitches of chain, the normal maximum rubbing speeds are in the neighbourhood of 100 ft. per min. The pressures under which the joints operate may be as high as 3,000 lb. per sq. in., calculated on the projected area of the bearing, though specific pressures much lower than this are more general; in fact, where the service required from the drives is continuous, or the application is of a special nature, pressures of 1,250 lb. per sq. in., again based on the projected area, are aimed at.

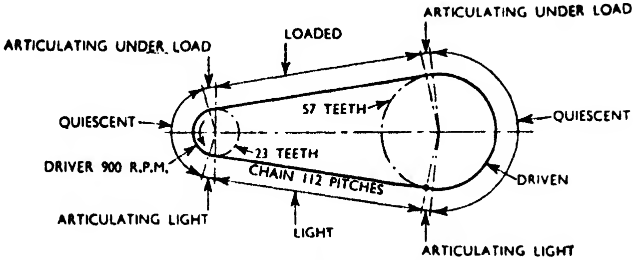

Fig. 1. Typical Roller Chain Drive

It has been mentioned above that the motion of the bearings is intermittent and it may be well at this stage to trace the conditions under which a joint in a roller chain operates during one complete transit. Fig. 1 illustrates a typical drive consisting of a 23-tooth driver pinion and a 57-tooth driven wheel, the chain being 112 pitches in length: the pitch of the chain is immaterial to the argument, but should not normally exceed 1 inch for the speeds shown. Fig. 2 shows a development of the joint path and indicates that, starting with the point at which the particular joint under consideration engages with the driven wheel, there is first of all a short period during which articulation takes place, the section of chain embracing this joint being at the time unloaded or at least being under load due to centrifugal and catenary effects only. Next there follows a period of quiescence during which the joint gradually comes under load and finally has to

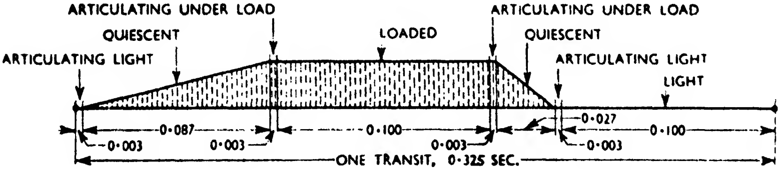

Fig. 2. Development of the Joint Path in a Roller Chain Drive

articulate under load. In this condition it passes along the free length on the driving side and when it reaches the pinion has again to articulate under load. A shorter period of quiescence then follows during which the load is gradually released; articulation light then takes place, and is followed by the joint passing along the unloaded free length, after which the cycle commences again.

For the rotational speeds, numbers of teeth, and chain length in question, the time of a complete transit for one joint is 0·325 second and of this time approximately 0·012 second is theoretically spent in articulation, the exact proportion being four times the reciprocal of the chain length in pitches, during only half of which the joint is under load.

This analysis shows that the conditions under which the joints of a chain operate are not so severe as might be anticipated from a cursory glance at the subject, and to complete the picture it may be pointed out that the amount of angular movement taking place at a joint as it passes from the free length on to a wheel or pinion, and vice versa, is in inverse ratio to the number of teeth. Since a pinion should have not less than 19 teeth it follows that the maximum articulation will not exceed approximately 19 deg., the product of the two being 360 deg.

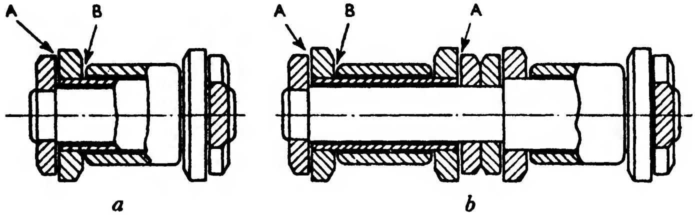

Fig. 3. Sectional Views of (*a*) a Simple Chain and (*b*) a Duplex Chain

Before dealing with suitable lubricants and the methods of applying them to a chain, attention may be drawn to the sectional views of simple and duplex chains shown in Fig. 3, *a* and *b* respectively, since the actual paths the lubricant must follow in order to reach the bearing surfaces are not always appreciated. The bearing formed by the bore of the bush and the pin is of major importance and because the bush is a force fit in the inner plates it is not possible, in chains of normal construction, to reach this otherwise than via the clearance between the inner and the outer or inner and intermediate plates; these paths are indicated by the arrows at A. The oil required to lubricate the bearing formed by the bore of the roller and the periphery of the bush can enter via the clearances between the roller and the inner plates, i.e. at B in the illustrations.

Turning now to the actual lubricants to be used for roller transmission chains, it should be emphasized in the first place that the application of grease externally is useless. Grease is suitable only if melted and used as a bath in which to immerse a chain which cannot be lubricated in any other manner. The rear transmission chains of motor cycles and similar exposed drives where it is impracticable or inconvenient to arrange for continuous lubrication by means of oil, are instances where this method may be employed. The grease bath method of lubrication requires certain precautions in application, such as thorough cleaning of the chain after removal from the wheels and before immersion in the bath; the wiping off of the surplus before replacement, and so forth.

The correct lubricant for power transmission chains is oil, which should be a mineral-base oil of medium body, free from any tendency to gum. No special brand is required, but an oil having a viscosity of 600–900 sec. Redwood at 70 deg. F. is suitable for normal conditions. It follows, however, that if oil is applied to a chain moving at any considerable speed, a portion only of it penetrates to and lubricates the bearing surfaces, the remainder being thrown off by centrifugal action. It is therefore necessary, unless the chain transmission forms part of a machine or mechanism where the contour of the adjacent parts forms a casing, to provide one. In short, it is no less necessary to ensure suitable running conditions for chain transmission than it is for, say, spur or worm gearing, if the best results are to be obtained. Granted, therefore, that a chain case is provided, it should be noted that in addition to acting as a collector for the oil thrown off the chain it also protects the drive from dirt, dust, or any foreign matter which may be present in the surrounding atmosphere, and moreover acts as a guard and so complies with the requirements of the Factory Acts.

While the delivery of a quantity of oil on to a chain is desirable on account of the cushioning effect which it produces at the points of engagement with the wheels, it will be apparent from a study of Fig. 3 that the best places at which to deliver the oil are on to the edges of the plates: this gives the lubricant the greatest opportunity of reaching the bearing surfaces and at the same time the surplus oil provides the cushioning effect already mentioned. Another point is that the oil should be delivered to the inside of the chain so that it has to pass through the chain before being thrown off by centrifugal action.

From the foregoing it follows that oil should be delivered to a chain from a pipe, furnished with nozzles or orifices in line with the plates, the pipe being arranged to pass within the circuit of the chain close to the point of engagement with the pinion or smaller wheel. If the machine or mechanism to which the chain is fitted already has a supply of oil available, a branch or tapping from this supply may be connected to the spray pipe. If, on the other hand, the drive is isolated and is enclosed by a sheet metal case, it is usual to provide a sump, which may be either integral with the case or separate from it, and to feed oil to the spray pipe by means of a simple type of pump which is driven usually from the slower-speed shaft by means of a light belt, or, with large horse-power drives, by a roller chain of small pitch. This method of lubrication is economical and efficient; it is recommended for all drives where powers in excess of 50 h.p. have to be transmitted.

Certain dispositions of drive with the large wheel below can be lubricated by means of an oil bath, oil being maintained in the case at such a level that part of the chain is completely submerged. An elaboration of this scheme may include gutters to catch the oil thrown up by the chain and a chute to deliver it to the inner face. If applied to chains running at high linear speeds, however, this scheme results in considerable disturbance of the oil and calls for care in designing the chain case, in order to ensure that it is completely oiltight.

For drives transmitting less than 50 h.p. and running at medium speeds it is common practice to provide an automatic drip-feed lubricator fed by a small pump: this has the advantage over the ordinary type of drip-feed lubricator that oil delivery ceases when the drive is stopped and is resumed automatically when the drive starts: it is also economical, since the collected oil is recirculated through the drip feed.

For slower-running drives, or where simplicity and low initial cost are desired, drip-feed lubricators of normal type may be used, but the qualities mentioned are offset very quickly by the increased cost of maintenance, while the amount of oil delivered to the chain varies with the amount present in the reservoir.

Summarizing the matter, the position is that the roller chain contains a number of bearings, which, since they are of the plain type, require to be lubricated. In order that penetration to the bearing surfaces may take place, an oil of medium body is required. The oil should be delivered to the inner face of the chain and should be directed on to the edges of the plates, a simple type of pump being provided for this duty if no supply of oil is available. A casing should be fitted to protect the drive and to collect and return the oil to the pump. Chain transmissions correctly proportioned, erected with reasonable care and lubricated on the lines laid down in the foregoing, can be relied upon to transmit the power delivered to them with an efficiency of not less than 98 per cent.

EXTREME-PRESSURE LUBRICANTS

By E. A. Evans*

The introduction of the hypoid gear has deflected attention from the coefficient of friction to the film rupture strength of lubricants. It has presented a new problem. When it was lubricated with a mineral oil, "scuffing" or scoring of the teeth occurred; in fact, no oil, whether mineral or fatty, would prevent it. During the pioneer period, research was conducted on a full-size gear or on a complete rear axle, but it was soon realized that this method of attacking the problem was too expensive and not entirely satisfactory, with the result that several simplified testing machines appeared. Amongst these may be mentioned the Timken, Almen, and Cornell. The principle common to all of these machines is that a cylindrical steel rod is revolved against a stationary steel surface. The lubricant is applied to the rubbing surfaces while the load is being increased until the oil film ruptures. The results are essentially arbitrary, and there is no guarantee that if a lubricant will withstand a particular pressure and speed on the testing machine it will give satisfactory results in the hypoid gear in a car. It is true that an indication can be obtained, but road or track tests must be made.

The evolution of the hypoid gear has forced even a closer co-operation between the automobile industry and the petroleum industry. In the United States the Society of Automotive Engineers has issued a tentative design for a testing machine in which two revolving wheels are incorporated. The wheels can be pressed together to give line contact, and can be rotated independently at will to give sliding friction. This is known as the S.A.E. machine.

Appreciation is due to the designers of hypoid gears because they focused attention on the film rupture strength of lubricants, but it would be unwise to associate film rupture strength only with the requirements of the hypoid gear, because extreme-pressure lubricants are now being utilized for many types of gear and heavily loaded bearings.

It is admitted that petroleum lubricants fail to maintain a film under extreme-pressure conditions. It has therefore become necessary to incorporate in the oil a substance which has the property of increasing the film rupture strength. Film rupture strength is entirely dissociated from the idea of oiliness; it is generally associated with a chemical reaction between the lubricant and the bearing metal. Lead soap was supposed to produce a protective film on the bearing surface

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* C. C. Wakefield and Company, Ltd.

in time, without the addition of any other reactive body, but it is difficult to ensure a thick enough film to give the needed protection. Lead soaps have been used in lubricants for years without their true function being revealed. This idea of producing a protective film has led to the adoption of active substances containing either sulphur, chlorine, phosphorus, or selenium. It is supposed that these corrosive atoms result in the production of a layer of sulphide, etc., which keeps the metal surfaces apart.

If flowers of sulphur is heated with oil a lubricant is produced which increases the film rupture strength of the oil, but it also produces a lubricant which is highly reactive on copper-containing metals, and is not too kind to bearings. A less reactive substance being desired, fatty oil was treated with sulphur chloride, and the resultant substance after some purification was dissolved in mineral oil. Very considerable satisfaction was obtained. The complexity of the reaction left the maker in considerable doubt whether the interaction between the sulphur chloride and the fatty oil could be controlled with accuracy. The obvious procedure was to manufacture a technically pure organic compound having the desired properties.

Compounds such as carbon disulphide and carbon tetrachloride are too volatile to be considered for practical application. Thionyl chloride appeared to be applicable, but unfortunately it is completely hydrolysed by water, and when an oil containing it is exposed to the air at 60 deg. C. the whole of the thionyl chloride disappears. Numerous organic compounds containing sulphur were investigated, but many of them were abandoned either because of their limited solubility in oil or owing to undesirable characteristics. Throughout, these results have been obtained on the Almen testing machine. The probability is that the results would not be reproducible on other testing machines, but the usefulness of the results rests in the fact that the compounds are classified.

Practically all mineral oils, irrespective of viscosity, have a film rupture strength of 4,000–5,000 lb. per sq. in. The variation appears to be due to the type.

Fatty oils show marked differences:—

Fatty oils	Film rupture strength, lb. per sq. in.
Rape oil	15,000
Wool wax	15,000
Neatsfoot oil	11,000
Blown rape oil	11,000
Seal oil	10,000
Olive oil	9,000
Castor oil	9,000
Lard oil	9,000
Sperm oil	8,000

The film strength of greases containing lime soap depends upon the amount of soap present, although exceptions do occur. The following greases are arranged according to the concentration of soap:—

Lime-soap grease	Film rupture strength, lb. per sq. in.
A	4,000
B	8,000
C	5,000
D	13,000
E	15,000
F	15,000

Apparently all sodium soap greases have a film rupture strength of 3,000–5,000 lb. per sq. in. The effect of other soaps on oil has been obtained by dissolving 5 per cent of soap in an oil (125 sec. Redwood at 140 deg. F.):—

Other soaps in oil	Film rupture strength, lb. per sq. in.
Lead oleate	3,000
Sodium oleate	4,000
Tin oleate	5,000
Aluminium oleate	8,000
Chromium oleate	13,000

Unless otherwise stated the oil used in the previous tests was used in all the following tests.

In view of the widely different results obtained with fatty oils it was decided to ascertain the effect of esters:—

Esters, 3 per cent solution	Film rupture strength, lb. per sq. in.
Benzyl oleate	3,000
Glyceryl oleate	4,000
Amyl oxalate	5,000
Butyl phthalate	5,000
Butylene glycol diacetate	6,000
Amyl acetate	7,000
Dibutyl phthalate	8,000
Amyl benzoate	8,000
Ethyl benzoate	8,000
Ethyl phthalate	9,000
Methyl salicylate	10,000
5 per cent solution	
Ethyl phthalate	9,000
Methyl salicylate	12,000

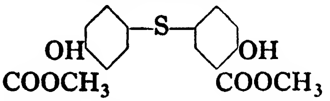
The alcohols and phenols which are soluble in mineral oil produced very little improvement in the oil. As tetraethyl lead can be used as an oxidation inhibitor in oil it was tried in various concentrations up to 1 per cent. The maximum strength obtained was 9,000 lb. per sq. in.

A few results are recorded to illustrate the behaviour of some sulphur compounds when dissolved in oil:—

	Film rupture strength, lb. per sq. in.
p-Chlorophenyl thiocarbimide	15,000
o- ,, ,,	9,000
p-Tolyl thiocarbimide	7,000
Sulphoricinic acid	5,000
α-Thionaphthol	3,000
2-Mercaptobenzothiazole (1 per cent solution)	5,000
,, ,, (5 ,, ,, suspension)	8,000

During this investigation compounds which formed a part of another subject of research were included as a matter of interest. Thiosemicarbazide has a solubility of less than 0·5 per cent, yet the film rupture strength was 8,000 lb. per sq. in. Diphenylcarbazone is still less soluble, but the strength was 8,000 lb. per sq. in., and dimethylaminobenzylidene-rhodamine, which is much less soluble, also gave 8,000 lb. per sq. in.

A series of thio-ethers was prepared. The most interesting was di (3-carbomethoxy-4-hydroxyphenyl) thio-ether.



When 0·5 per cent of this compound is dissolved in oil a film rupture strength of 15,000 lb. per sq. in. is obtained. An advantage of this compound is that at 100 deg. C. no visible corrosion is observed on steel, and with copper it is very slight. This thio-ether is being reserved for what are now termed mild extreme-pressure conditions. Since this research was completed the loading on hypoid gears has been increased. The maximum stress on the gear tooth surfaces may reach values as high as 350,000–400,000 lb. per sq. in., and the rubbing or sliding velocities may range from 0 to 1,800 ft. per min., or even higher. The stress is not applied over a large area of the tooth profile, but is transmitted through line contact. Scoring on the "drive" side of the gear teeth usually occurs at speeds slightly under the top speed of the car. Scoring on the "coast" side generally occurs when the clutch is suddenly dropped in when coasting at intermediate or high car speeds, or by a sudden release of the accelerator pedal when driving fast. This increase has called for a still higher film rupture strength of the lubricant. It is being achieved by reverting to the older procedure of dissolving sulphur in the oil and adding lead naphthenate. The author ventures to think that a lubricant made in this way can only exist until a more scientifically prepared article appears on the market.

The numerous references to extreme-pressure lubricants in the patent literature testify to the growing interest and importance of the subject. The vast array, and diversity, of compounds which have been patented give a clear indication that little is known about the chemical behaviour in extreme-pressure conditions, and how the additive agent attacks the metal surface.

The author wishes to record his thanks to Messrs. C. C. Wakefield and Company, Ltd., for permission to publish these results.

ROCK DRILL LUBRICATION AND PERFORMANCE

By S. F. Gimkey, M.Sc. (Eng.), A.M.I.Mech.E.*

The rapid progress in mining on the Witwatersrand owes much to the improved efficiency and construction of pneumatic hammer drills. The individual parts of these machines have been perfected to such an extent that drilling speeds of 10 in. per min. in hard quartzite rock of reef can be maintained with a minimum consumption of compressed air.

The co-operation and co-ordination of research between the efficiency departments of the various mining groups and the manufacturers of pneumatic drills have assisted in producing an efficient and well-designed tool to withstand the severe duty it has to perform. The human element plays an important part in the maintenance of high drilling efficiency. For this reason the weight of the drill has been reduced as much as the requirements for strength will permit. Its recoil has been reduced and a suitable profile has been adopted to facilitate drilling in awkward working places. The improved external form of pneumatic rock drills has been made possible by the introduction of the plate or sleeve valve in place of the piston or spool valve previously used. The former is housed in the cylinder itself in line with the piston and other reciprocating parts, whereas the latter had to be mounted on the side. Plate valves are lighter than piston valves, the wear is considerably less and lubrication of plate valves is of much less consequence than that of piston valves.

To conform with the Mining Regulations of the Witwatersrand, a water feed must be supplied at a minimum pressure of 30 lb. per sq. in. Generally a thin tube of small bore which passes from the back head of the machine through the rifle bar and piston conveys the water into the hollow drill steel and removes the chippings produced during the boring operations while helping to allay the dust created. The air pressure naturally tends to blow the lubricating oil out of the machine through the exhaust and the application of water through a rock drill further complicates the problem of efficient lubrication. Besides these difficulties, the proper lubrication of such machines underground when operated by natives presents difficulties not met with in the ordinary way.

A modern pneumatic hammer drill strikes from 1,700 to 2,000 blows per minute. It is a cold-working, high-speed machine, subject in the cylinder and exhaust ports to low temperatures; and therefore for

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* Senior Lecturer in Mechanical Engineering, University of the Witwatersrand, South Africa.

lubricating these parts a light oil with a low cold-test would probably be recommended. The mechanism which causes rotation of the drill steel will, however, require a lubricant which is somewhat different in character from that which is best for the cylinder and valve. In this respect the parts of a pneumatic hammer drill which often show considerable wear lie at the front of the machine and are contained in the front head.

Some machine operators pour an arbitrary quantity of oil into the air hose before connecting it to the machine, while others pour the oil into the air inlet port of the machine. These methods are crude and unsatisfactory, since most of the oil so supplied is immediately expelled into the atmosphere with the exhaust air. The small residue covers the moving parts of the machine. The film so formed remains only temporarily so that the machine might operate without lubricant for considerable periods.

Certain manufacturers of rock drills incorporate oil reservoirs in the bodies of their machines, but, in general, lubrication by hand is unsatisfactory for the following reasons:—

(1) The machine is not lubricated at regular intervals, thereby causing loss in drilling efficiency.
(2) Oil is used wastefully without ensuring constant lubrication.
(3) Oil plugs are occasionally lost, thereby permitting grit to enter the machine, with disastrous results to its mechanism and its efficient operation.
(4) Much time is wasted by frequent stoppages for lubrication.

The above difficulties have been surmounted by introducing the lubricant into the machine together with the compressed air by means of an air line lubricator which should atomize the oil. Since, therefore, only one kind of oil can be used for lubricating the cylinder and piston as well as the front-head chuck and ratchet mechanism, it is best to use a medium-bodied oil.

In the past the viscosity of the lubricant to be used has caused much controversy, some engineers advocating a fairly heavy oil so that when wear takes place inside the cylinder, there will be less tendency for the air to blow through, causing leakage, as may occur when using a very thin oil with the attendant increase in consumption of compressed air. Laboratory tests as well as trials underground appear to substantiate the view that an oil having a specific gravity of approximately 0·94 and a setting point below 0 deg. F., with a Redwood viscosity of approximately 2,780 sec. at 70 deg. F. and 240 sec. at 140 deg. F., is well suited for the purpose of rock drill lubrication when used in conjunction with suitable air line lubricators, which effect good atomization.

Several years ago experiments were begun in the Mechanical Engineering Laboratory at the University of the Witwatersrand in order to test the rock drill lubricators in use at that time. It was found that practically all lubricators suffered from one or more defects. With most lubricators it was only possible to use a very heavy oil, since, with the use of light or even medium heavy-bodied oils the consumption becomes excessive. The high oil consumption of light-bodied oils is caused by the unbalanced pressure obtaining in the lubricators during intermittent operation of the machine. Whenever the air is shut off at the drill the pressure in the oil reservoir of the air line lubricator rises to that of the static pressure of the air line. On opening the air cock, when starting the rock drill, the air line pressure drops by some 6 to 10 lb. per sq. in. Most air-line lubricators are so constructed that the unbalanced pressure between oil reservoir and air line, when starting the machine, causes a sudden rush of a large quantity of oil to the rock drill. Once the pressures are balanced, there is no further oil discharge from the lubricator. When a very heavy oil is used the quantity of oil discharged is the same, but the rate of discharge is considerably extended, so that if a particular lubricator of this type feeds a light-bodied oil to a rock drill for only a half minute during the starting period, the same lubricator will feed a very heavy oil for a longer period after the machine has been started. With such a type of lubricator used in the air line the rock drill will be flooded with oil whenever the drill is started and after that it is practically starved until it is stopped and then again restarted.

It was soon realized that a good air-line lubricator must give a uniform feed while the machine is running and under any conditions of running (the usual type of lubricator will soon empty itself when the drill steel frequently sticks in the hole while drilling). It must be compact and its weight must be a minimum and it must have no projecting parts, so that it can easily be pulled through an underground working place without catching on any timbering or debris. It is equally important that it must be able to work in any position and that the feed, when once adjusted, should not be susceptible to pressure variations. It should be adjustable, but the locking arrangement should be such that when underground, it cannot easily be tampered with. Lastly, a good air-line lubricator must be able to feed a lubricant of any viscosity in either direction without loss of air pressure.

Bearing these points in mind, work in the laboratory and underground lasting many months, resulted in a design of air-line lubricator in which a balancing piston was incorporated. This served to equalize the pressure in the oil reservoir when starting and stopping the rock drill. Although this air-line lubricator met with considerable success and satisfied many of the conditions already stated, a much improved type was recently evolved (Fig. 1). The air-line lubricator consists of two concentric tubes, the outer tube being readily coupled to the air line and the oil reservoir with filler cap being welded co-axially to this tube. The inner tube is free and provides an annular capillary passage for the lubricant. An outlet from the oil reservoir to the capillary passage is provided by means of the two holes of $\frac{1}{8}$ inch diameter in the outer tube. The inner tube is capable of endwise movement and oscillation, the amount of the latter being determined by wire clips placed at each end which also serve to prevent the inner tube from being pushed out completely. The concentric tubes are so made that the capillary action in the annular passage is insufficient in itself to feed the lubricant at the requisite rate into the air stream. To secure this result, the compressed air in the line is utilized. This air exerts a total pressure at the downstream end represented by "pressure head—velocity head" and it exerts a total pressure at the upstream end represented by "pressure head+velocity head". The resultant pressure head due to air pressure alone causing flow of oil into the air stream, is therefore $2(v^2/2g)$ and there is also the capillary attraction which ensures that the space between the tubes is always full of oil. The capillary passage is also a means of controlling the rate of flow by the choice of correct clearance between outer and inner tube; further, it serves to throttle the unbalanced pressure when the rock drill is started and stopped. This air-line lubricator fulfils all requirements for efficient lubrication of rock drills.

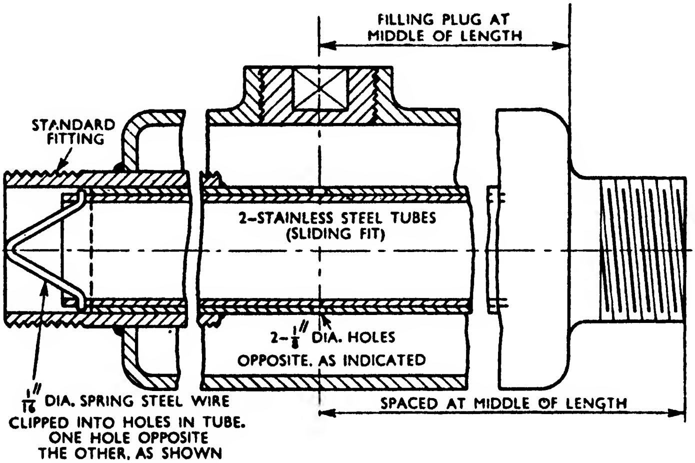

Fig. 1. Air-Line Lubricator

In order to investigate scientifically the performance of various rock drills for different working conditions, the Walker testing machine,* (Fig. 2) was constructed. The rock drill is placed on the plunger and is held in position by the two wooden clamps and a weight of sufficient magnitude is placed on top of the drill to keep it hard against the plunger of the testing machine when the drill is in operation and prevent it from "knocking" or "jumping". The blows of the drill strike the plunger, which in turn communicates the energy of the blows to the oil contained in the plunger cylinder of the testing machine. The oil discharge passes from the cylinder through a pipe in which is placed a spring-loaded ball valve (Fig. 3). The stiffness of this valve spring is made adjustable so that there is no discharge of oil when the drill is not in

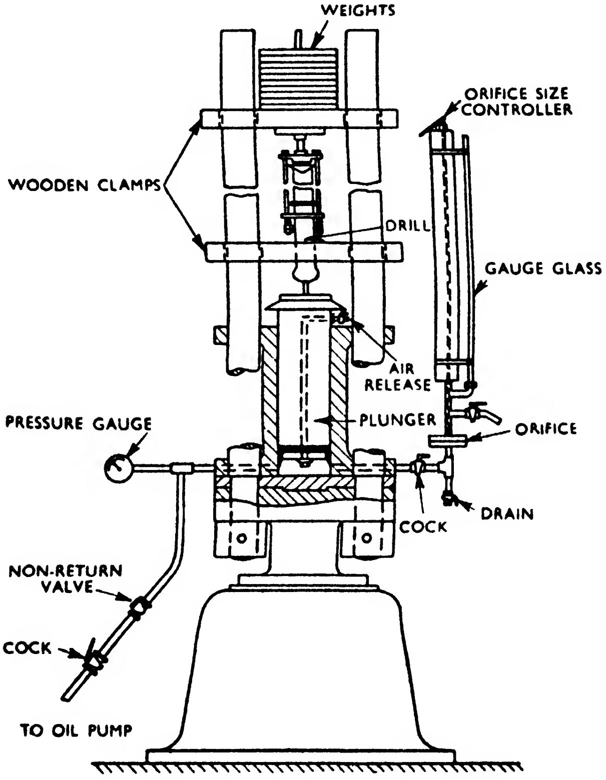

Fig. 2. Rock Drill Testing Machine

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* Jl. Chemical, Metallurgical and Mining Soc. of South Africa, 1931 Feb.

operation. When the drill is working, the quantity of oil discharged through the orifice is measured in the gauge glass (see Fig. 2).

A needle valve (Fig. 3) regulates the size of the discharge orifice and thereby can simulate the various degrees of hardness of rock by increasing or decreasing the resistance to the flow of oil through the orifice. In the particular machine constructed the diameter of the orifice is ½ inch and the pipe diameter is 3 inches. It was also found that excessive "knocking" or "jumping" of the pneumatic drill was often caused by air below the plunger. On opening the air release cock and allowing the air to escape, the rock drill will again run steadily. With the ½-inch diameter orifice the areas of opening for the various turns of needle valve were:—

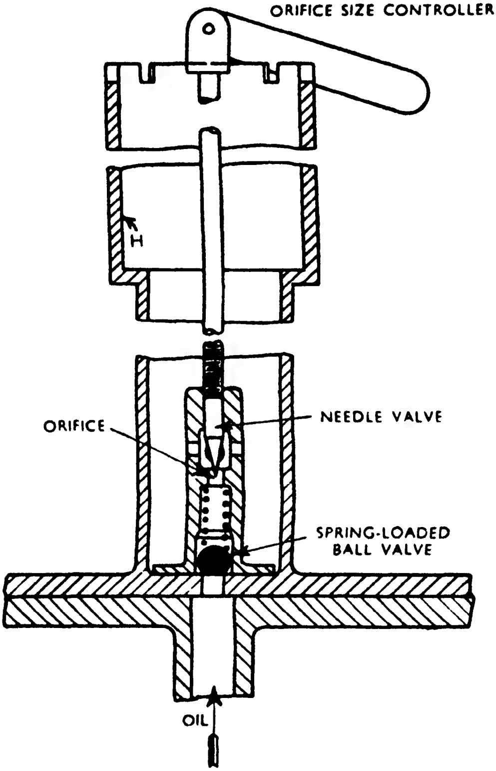

Fig. 3. Needle Valve for Rock Drill Testing Machine

Turns	½	¾	1	1½
Area, square inches .	0·0245	0·0309	0·0419	0·0664

Other observations required during a test are air pressure, oil and air temperature, oil consumption, air consumption (taken by means of an orifice meter), and number of blows per minute of rock drill. It was found that the last-named can be very conveniently observed by means of a frequency meter of the vibrating reed type connected to the rock drill merely by a piece of fairly stiff wire. The theory of the testing machine is as follows:—

Let h=height of oil column produced per second
W=weight of oil in pounds per cubic foot
a=area of orifice opening in square feet
A=area of oil column in square feet
V=velocity of oil passed through orifice in feet per second.

Then $V=Ah/a$.

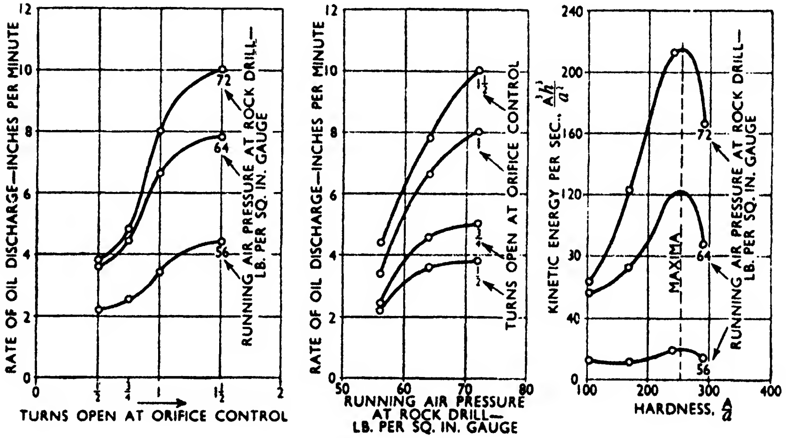

Fig. 4. Results of Tests on One Rock Drill

The kinetic energy per second of the column

$$=\frac{WaV(V^2)}{2g}=\frac{WA^3h^3}{2ga^2}=\frac{A^3h^3}{a^2}\times\text{constant}$$

The simulated "hardness" of rock can be written A/a.

In Fig. 4 are reproduced graphs obtained from a series of tests on one rock drill. It will be realized that the testing machine only serves to compare the behaviour of different rock drills under varying conditions as regards lubrication, air pressure, weight loading, temperatures, and other factors which influence the operation of such machines.

In the near future, in conjunction with tests such as are indicated above, it is hoped to obtain diagrams of pressure variations in the rock drill cylinder on a piston stroke base with the aid of a piezo-electric crystal and oscillograph. This apparatus will then be used to obtain the pressure variation, on a time base, in the oil cylinder of the testing machine. It may also be possible to apply this latter apparatus to tests on rock drills under actual working conditions.

LUBRICATION IN WIRE DRAWING

By R. Goodacre, B.Sc., Ph.D.*

There have been various attempts to make mathematical investigations of the wire drawing process, but the results obtained have only been very approximate. The earliest equation that the writer has been able to trace is due to Gewecke (1909)

$$p=k(\tan\theta+\mu)(a_1-a_2)$$

(where p is the power required to draw, θ is the angle of the die, μ the coefficient of friction, a_1 and a_2 the cross-sectional areas before and after drawing, and k a constant). Applying this to the drawing of copper wire it was found that μ was apparently constant for any one lubricant.

Another equation, due to Sachs (1927), is

$$p=k\left\{1+\frac{\tan\theta}{\mu}\right\}\left\{1+\left(\frac{d_2}{d_1}\right)^{\frac{2\mu}{\tan\theta}}\right\}$$

(p, k, and μ as above, θ is the semi-angle of the die and d_1 and d_2 the original and final diameters). Using this equation, Sachs (1927) calculated that the value for μ in drawing steel through steel dies is 0·21, a very high figure compared with the value of 0·66 for unlubricated surfaces. However, using the same formula, Francis and Thompson (1931) obtained more reasonable values for a series of non-ferrous metals.

A third useful equation is due to Lewis (1933)

$$p=\tfrac{1}{2}y(d_1-d_2)(1+\mu\cot\theta)$$

(p, d_1, d_2, θ, and μ as above, and y the mean yield point of the metal while passing through the die).

Lewis has found that the value for μ varies between 0·065 and 0·110 for dry drawing and 0·13 and 0·19 for wet drawing of steel wire, depending on the lubricant and with the usual die angles.

Seeing that even under the best conditions the energy lost due to friction in the die is very large (Giraud's (1928) experiments with mild steel suggest upwards of 50 per cent of the total), it is essential to keep the lubrication as efficient as possible.

There are two general conditions of lubrication: (*a*) fluid, and (*b*) boundary. Taking into consideration that the pressures in the

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* **Chief Chemist and Metallurgist, Bruntons (Musselburgh), Ltd.**

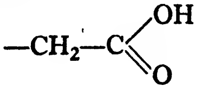
wire drawing die can range between 5 and over 70 tons per sq. in., according to the material and degree of reduction, that the friction is practically independent of the speed of drawing, and that the conditions make it very improbable that a continuous wedge of lubricant is produced, Francis (1932) concludes that the lubrication is of the boundary type. Lewis (1933) thinks that, with the exception of the unique conditions applying for dry drawing of iron and steel wire, there is no true lubrication, but that the wire practically comes in direct contact with the die, which leads to the burnished finish of most non-ferrous and wet-drawn ferrous materials.

The unique method for dry drawing of steel wire requires some explanation. After cleaning in acid, and washing, a wet rust or sull coat is formed on the surface, probably consisting of mixed hydroxides. The rod is then dipped in a cream of hydroxide of lime, usually hot, which partly neutralizes any remaining acid and partly helps in the subsequent drawing, which is carried out after the rod has been baked at about 100 deg. C. to dry it thoroughly and drive out the absorbed hydrogen. As a lubricant for drawing it is usual to use a good dry sodium soap which becomes bonded with the lime and the hydroxide during the process. Lewis suggests that in this method the frictional forces act between the hydroxide-lime coat and the die and not the steel. Certainly there is no burnishing effect on the metal surface, for if the soap-lime-hydroxide skin is removed, the metal underneath presents a matt surface similar to that which remains after the pickling of the rod.

The physical and chemical conditions of boundary lubrication have received considerable attention. It is well established that solids possess a degree of free surface energy dependent on the material and due to the partially unlinked electronic fields of the surface atoms. These can join up with similar partially linked fields of other materials which may come in contact with them, setting up a type of combination usually referred to as adsorption. Various workers (Hardy and Doubleday 1922, Hardy and Bircumshaw 1925, Woog 1926), have shown that an adsorbed film of lubricant exists on the surface of metals when in contact, while Trillat (1926), investigating by X-ray methods the surface conditions of metals in contact with fatty acids, has shown that some form of chemical combination had occurred.

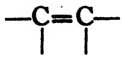
The efficiency of lubrication depends upon various active chemical groupings whose order (Bonzel 1934) is as follows:—

(1) The free carboxyl radicle



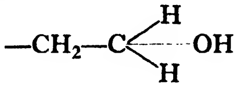
$$-\mathrm{CH_2}-\mathrm{C}\genfrac{}{}{0pt}{}{\diagup \mathrm{OH}}{\diagdown\!\!\diagdown \mathrm{O}}$$

(2) The double bond



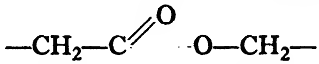
$$-\underset{|}{C}=\underset{|}{C}-$$

(3) Hydroxyl or alcohol radicle



$$-CH_2-\overset{H}{\underset{H}{C}}-OH$$

(4) The combined carboxyl with alcohol, i.e. glyceride



$$-CH_2-\overset{O}{\overset{\|}{C}}-O-CH_2-$$

The terminal ending of the ordinary mineral oils possesses very little free energy and cannot (Francis 1932) set up a boundary condition for lubrication. The efficiency is also connected with the molecular volume: the larger the value the better. Symmetrical molecules are better than asymmetric ones and straight chain compounds should be better than isomeric branch chain ones. The presence of two active radicles, e.g. double bond and carboxyl, can be disturbing, presumably because it gives a double effect and prevents the closest possible packing of the molecules.

The foregoing applies essentially to liquid lubricants but it probably also applies to solids, although (Francis 1932) conditions must be set up where the molecule can be so orientated that the active radicle comes into contact with the residual field on the surface of the metal. Francis thinks that this probably occurs in the die; under high pressure the solid becomes plastic and flows as a viscous liquid. This suggestion is confirmed in certain respects by Williams (1935), who suggests that the lubricant must adhere strongly to at least one of the surfaces, wire or die. It must be in a solid state before being subjected to the relative shearing of the surfaces to be lubricated and it must melt or be transformed into a fairly mobile liquid between the surfaces while shearing takes place. However, he further suggests that the coefficient of friction with a solid lubricant that meets the above conditions depends largely upon the viscosity and film thickness rather than upon the chemical nature of the lubricant or the surfaces and this indicates a "fluid" rather than a boundary state of lubrication. Williams did not obtain parallel results with "18–8" stainless steels and copper and explained this by differences in adhesion, the influence of the differences in temperature produced in the metal deformation, and by the specific effect of roughness or harshness of the surface. The author thinks that the important factor is the adsorbed oxygen or passive layer on the surface of the "18–8" stainless steel; as is well known, this type of passive layer is very difficult to break down, the stainless properties being attributed to its presence. Very little adsorption of lubricant could occur owing to the saturated condition of the surface and this may offer an explanation for some of the difficulties encountered in drawing stainless steels and other high chromium alloys.

The position of soap as a lubricant in wire drawing is unique, for, although theoretically it should possess only a little free energy and have very moderate lubricating properties (probably coming after the glycerides) it is nevertheless almost invariably used in dry drawing, and often as a basis for the more common wet drawing solutions, although probably its main action is as an emulsifier for other oils and greases present and its action as a lubricant is only incidental.

In dry drawing of steel wire the sodium soap of olive oil is commonly used. For efficient working it should be as dry as possible. The results from friction and dry drawing load tests (Thompson and Francis 1931, Francis 1932), show that sodium soaps head the list of lubricants for both ferrous and non-ferrous materials, but a considerable variation was also noted in the different soaps studied. These investigations confirm that the drier the soap the less the friction losses and also indicate that the presence of free glycerides is undesirable, while free glycerol is distinctly deleterious.

The mechanism of lubrication by means of soap is still obscure. It seems doubtful whether there is much adsorption effect, so for the time being it can only be assumed that the process is purely mechanical. The work of Francis (1932) on the effect of elevated temperatures upon lubrication tends to confirm this suggestion. Most lubricants have a temperature coefficient showing a lessening of friction as temperature rises and the same effect was found to exist in wire drawing, except for commercial soap, which gave a constant lubricating effect at all temperatures tried below 150 deg. C., at which decomposition commences. As an increase in temperature usually results in increased chemical activity there should also be an increase in the adsorption effect. The constancy noted for soap suggests that there is no increase in free energy and tends to confirm the hypothesis that its action in lubrication is chiefly mechanical. However, against this is the fact that both barium oleate and more particularly sodium stearate showed temperature coefficients. Chemically, these compounds are soaps and the second one is closely related to the usual commercial type of wire drawing soap. There can be little doubt that the free energy of such compounds is possessed by the metal radicle since it is combined with a weak fatty acid, and a tentative suggestion is that in the dry drawing of steel wire there is a partial linkage between the metal radicle and the hydroxyl groups in the hydroxide coat. Explanation of the mechanism for other metals is more difficult as no hydroxide coats are used in these cases, but there is no reason to believe that some form of adsorption does not occur.

The conditions of wet drawing do not appear to have been studied in much detail. One of the essentials in the wet drawing of steel wire appears to be the formation of a very thin coat of copper or tin on the surface of the metal, usually the former. Most baths used are acid, some containing as much as 2·5 per cent by weight of sulphuric acid. Most contain some form of lubricant of the glyceride type, e.g. tallows and waxes and the presence of an emulsifying agent, e.g. soap, is almost essential. The production of the metal coat on the steel will reduce the primary coefficient of friction for the metal to the die, possibly by about half, and the presence of the acid is essential in the forming of such coats, those of copper formed in neutral solutions being practically non-adherent. At the same time the usual concentrations are much higher than is necessary for this purpose alone; Bonzel (1934) suggests that the acid probably helps in the lubrication, stating that a 10 per cent solution of sulphuric acid can be used in drawing steel wire through both diamond and synthetic dies without the presence of fatty matter, oil, or metallic coat. The mechanism of lubrication by the oils and greases is probably the same as described above, but they are introduced to the interface in the form of an emulsion rather than a continuous film and consequently not so efficiently. Gulick (1926) states that the efficiency of lubrication depends upon the particle size, which in turn is bound up with the stability of the emulsion. The normal type of soap does not produce a very stable emulsion, even when used at the optimum concentration, which is about 0·5 per cent, so it is usual to add some form of stabilizer in the form of colloids, usually starch or gelatin. Sulphonated fatty acid compounds, either glycerides or salts, produce emulsions with a very fine particle size which are very stable and do not require the addition of protectors.

It might be thought that graphite would have proved to be a satisfactory lubricant for wire drawing, but on the whole it has been disappointing. Further, it is costly and its use is only warranted with wire for which the monetary return is adequate or where no other methods are possible. It has been used fairly satisfactorily in the drawing of tungsten and similar metals where the operation is carried out at 600 deg. C., at which temperature no other form of lubricant is possible. It is also fairly satisfactory for the Nichrome type of alloy, but is difficult to remove if the final product has to be in a bright condition. Graphite-bearing soaps are quite satisfactory, but it is doubtful whether they are any better than ordinary soaps, in fact, experimental results obtained by Francis (1932) suggest that they are no better.

Soaps containing metallic radicles other than sodium have been introduced, notably calcium and aluminium, and lead soaps have been tried in the United States. However, these do not appear to have any advantage over the usual sodium soaps and presumably the chief reason for their introduction is that they can be supplied in a practically anhydrous condition.

Metal coatings, other than copper and tin, have been tried with some success in the wet drawing of steel wire. Dudzeele (1932) states that a thin layer of lead of the order of 0·012–0·015 mm. (approximately 0·0005 inch) facilitates the drawing of metals, e.g. the Nichrome type of alloy. However, it can cause subsequent trouble such as the formation of small beads of lead during bright annealing processes. A recent note (Anon. 1936) mentions a new material called "Amaloy" as being suitable for drawing stainless steels; it apparently consists of 98 per cent lead together with a bond, but no mention is made of the method of application though claims are made for its efficiency.

The forecasting of possible future developments in lubrication during wire drawing is practically impossible. Taken as a whole the industry remains faithful to the old and tried methods which have been developed empirically. But the introduction of continuous machinery is changing the complexion of the industry and new methods usually demand better technique. Already the introduction of the synthetic die has reduced power costs because the primary coefficient of friction of metal to die is lower than with the older type of steel die; claims have been made that the saving is as much as one-third. Scientific investigation has given us the clue to the mechanism of lubrication in the die and has given suggestions as to why some methods are better than others, but there still remains a wide field for investigation. The possibility of the higher homologues in the fatty acid series could be explored and the question of the free energy and spatial configuration of active chemical groups with regard to the rest of the molecule requires further elucidation. One obvious development resulting from the work of Francis (1932) should be the use of lubricants at temperatures in the region of 100 to 120 deg. C. At these temperatures many lubricants are quite equal to the best of soap and, in the region of the boiling point, aqueous mixtures are nearly as efficient, while the same quality of surface is produced as with normal wet drawing. Adam (1936) believes that such methods will lead to a definite revival in the wet drawing method for steel wire.

Reduction of the frictional forces will increase the output per die because of increased die life, and repolishing and the need for opening out to large sizes would then be less frequent. Thompson and Francis (1931) state that an increase in μ from 0·01 to 0·20, when drawing 20 per cent reduction with a die of 5¾ deg. angle, results in a 126 per cent increase in the tension, and when it is considered that, under normal conditions, upwards of 50 per cent of the power used is needed to overcome frictional forces it is obvious that if by means of improved lubricants even small reductions could be obtained in the coefficients of friction which obtain at present in the wire drawing die, a considerable saving in power costs alone would be seen in even a medium sized wire mill.

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EXPERIMENTAL STUDY OF BALL AND ROLLER BEARINGS

By Professor Ch. Hanocq*

Tests were undertaken by the Société Générale Isothermos, of Paris, in order to find the coefficient of friction for roller bearings at the same loads and speeds as those usual in practice for smooth bearings.

It has been found that, provided lubrication was abundant enough, a coefficient of friction could be obtained under hydrodynamic conditions with smooth bearings which was of the order of 0·002 and even 0·0015. The problem was to find the corresponding value for the same journal under the same load and speed of rotation when a smooth bearing was replaced by a roller bearing. The roller bearing was of the common railway type, and had two rings placed side by side; its dimensions were as follows:—

Journal diameter	120 mm.
External diameter	260 mm.
Diameter of the rollers in their plane of symmetry	30 mm.
Width of the rollers	36 mm.
Number of rollers	64 in 4 rows.

Two sets of experiments were carried out: one with a diametral clearance on the bearing of 0·06 mm. (Group I) and the other with a diametral clearance of 0·11 mm. (Group II).

Table 1 (p. 75) gives the results obtained both with the roller bearing and an "Athermos" bearing. The latter is lubricated with oil throwers and the brass is of the type in which the oil is distributed by multiple jets, this type having been used in tests on bearings with partial brasses, the clearance being 3 mm. A helical domestic fan was used to obtain ventilation, the speed of circulation being some 16 kilometres per hour.

It was found that a clearance of 0·06 mm. was insufficient for proper working at the temperatures reached during the work, so a clearance of 0·11 mm. was employed. Lower values were then obtained and the temperature increased but slowly with increasing speed, while the coefficient of friction fell to below 0·002. For this reason, only the figures in Group II (Table 1, p. 75) should be considered, though the importance of the clearance in respect of the coefficient of friction is realized.

With a load of 12,000 kg. and a speed of 800 r.p.m., the temperature of the smooth Athermos brass remained at 79·5 deg. C., whereas with

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* University of Liége.

TABLE 1

	P	N	Rollers with fan			Athermos bearing			
			Air temp., deg. C.	Bearing temp., deg. C.	$f = \frac{1}{1000}$	Air temp., deg. C.	Brass temp., deg. C.	$f = \frac{1}{1000}$	
Group I. Clearance, 0·06 mm.	6,000	530	21	55·3	3·12	15	57	1·69	No fan used
	8,000	530	22·5	60·5	2·97	20	58·5	1·67	
	10,000	530	21	63·0	2·66	20·5	64·5	1·65	
	12,000	530	22	66·0	2·53	21·5	69·0	1·63	
	7,000	290	20·5	38·0	1·80	26·0	48·5	1·35	Observations after 4 hours' run
	7,000	580	20·0	60·5	2·37	—	—	—	
	7,000	700	22·0	83·5	3·82	24·0	66·0	1·23	
	12,000	700	20·0	95·0	4·55	—	—	—	Bearings not fitted with felts
	12,000	800	—	—	—	24·5	79·5	1·18	
			Grease, deg. C.	Bearing, deg. C.	$f = \frac{1}{1000}$	Oil, deg. C.	Brass, deg. C.	$f = \frac{1}{1000}$	
Group II. Clearance, 0·11 mm.	6,000	635	40·0	52·0	1·80	47·0	66·5	1·50*	Air temperature for the Group II tests was 22 deg. C.
	6,000	692	41·5	54·0	1·87	49·0	69·5	1·27	
	6,000	750	44·5	58·5	1·84	49·0	73·0	1·29	
	6,000	850	47·5	60·0	1·79	48·0	76·0	1·32	
	7,000	635	51·5	52·5	1·64	48·0	67·5	1·44	
	7,000	692	52·5	55·5	1·69	49·0	70·5	1·31	
	7,000	750	57·5	61·0	1·79	50·0	76·0	1·33	
	7,000	800	61·5	64·5	1·91	50·5	78·5	1·29	
	7,000	840	57·5	59·5	1·52	50·0	78·0	1·21	

* Rough-finished bearing with fairly hard whitemetal.

the roller bearing under the same load at the same speed and a clearance of 0·06 mm., the temperature rose to 92 deg. C. after 140 minutes and the test had to be stopped as the temperature continued to rise.

These apparently unexpected results are by no means in contradiction with results obtained in the writer's laboratory, using other methods.

Using the deceleration method (Fig. 1) the coefficient of friction of ordinary ball bearings was determined with a shaft 40 mm. in diameter. After starting up the combination of flywheel, shaft and bearing, the deceleration curve was obtained and the corresponding torque deduced.

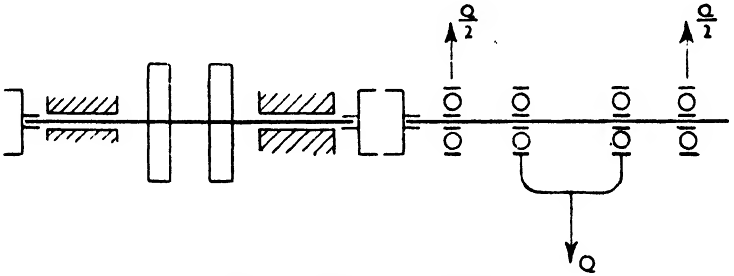

Fig. 1. Method of Testing

Then the torque M corresponding to the four ball-bearings could be obtained by difference, whence

$$f = \frac{M}{4\left(\frac{1}{2}Q\right)r}$$

The results, giving f in terms of N (revolutions per minute), are plotted in Fig. 2 and strongly suggest a linear expression for N. That this should be so can be proved, for it can be assumed that the power W absorbed by a ball bearing is made up of the following terms :—

(1) The loss by bearing friction can be taken as proportional to f_r, the coefficient of bearing friction, and to the load $\Sigma P'^{\frac{4}{3}}$, as was shown by Professor Dumas of Lausanne. Under these conditions we can write, for the whole

$$W_r = 2f_r k_1 wr \Sigma P'^{\frac{4}{3}} = 2f_r k_1 wr P^{\frac{4}{3}}\left[\left(\frac{P'}{P}\right)^{\frac{4}{3}} + \left(\frac{P''}{P}\right)^{\frac{4}{3}} + \ldots\right] \quad . \quad (1)$$

the coefficient k_1 representing the opposing force considered as applied to the periphery of the interior ball race, whereas wr represents the tangential speed of the shaft, all in relation to the radius r of the shaft.

(2) The loss due to the sliding friction of the balls in their race, the peripheral speed of the balls being strictly equal to the tangential speed of the race only in the plane of symmetry, whence

$$W_g=2f_gk_2wr\Sigma P' \quad . \; . \; . \; . \; . \; . \; . \quad (2)$$

k_2 being inserted because the sliding speed is only a very small fraction of the peripheral speed wr.

(3) The loss due to the slip of the layer of oil along the balls, the oil flowing back as the balls advance, and also to oil scraped off by the race itself. Applying Newton's law, the force applied to the periphery of the balls can be written as

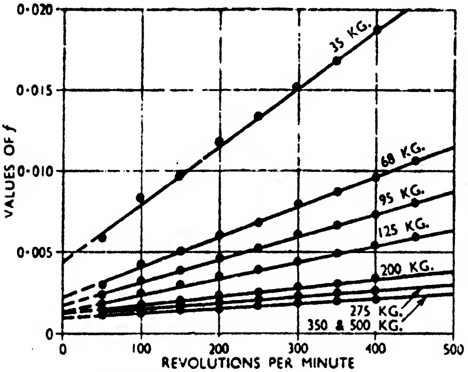

Fig. 2. The Value of *f* in Terms of N

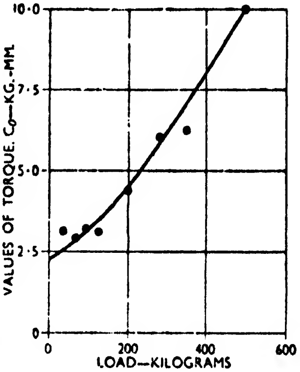

Fig. 3. Torque under Different Loads

$$R=\mu k'\rho^2\times\frac{k''wr}{e}\times n$$

where $k'\rho^2$ represents the surface of each ball affected by the removal of oil in the race and the reflux of oil on the race, $k''wr$ the speed of slip, e the thickness of the oil film, and n the number of balls. Movement requires a force applied to the periphery of the interior ball race equal to 2R and therefore the equation for this loss can be written as

$$w_\mu=2\mu(k'n\rho^2)\frac{k''wr}{e}\times wr_i \quad . \; . \; . \; . \; . \; . \; . \quad (3)$$

where $r_i=k''r$.

(4) The loss absorbed initially under no load owing to the positional tension of the balls and imperfections in manufacture, owing to which the torque is not strictly zero under no load

$$W_i=C_iw \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (4)$$

Summing these four terms and seeing that the total power absorbed can be expressed in terms of the coefficient of friction as Pfwr, we have

$$f = 2k_1' f_r P^{\frac{1}{3}} + 2k_2 f_g + k'k''^2 n \frac{\rho}{e} . \frac{\rho}{r} . \frac{\mu V}{\frac{P}{2r}} + \frac{C_i}{Pr} \quad . \quad . \quad . \quad (5)$$

where V represents the peripheral speed of the shaft, k_1' being distinguished from k_1 in order to take into account the bracketed term in equation (1).

From equation (5) it is seen that f should appear as a linear function of N in the experiments carried out under constant conditions of load and temperature. If the curves of Fig. 2 are extended to the ordinate axis, the value of Pfr can be found for N=0 and the curves Pfr=C_o (Fig. 3) can be drawn as a function of P. From this curve

$$C_i = 2{\cdot}3 \text{ kg.-mm.} \qquad 2k'_1 f_r r = 0{\cdot}0018 \qquad 2k_2 f_g r = 0{\cdot}00113.$$

To render equation (5) applicable to all geometrically similar bearings the specific pressure must be introduced in the first term instead of P; in other words the first term must be multiplied by the ratio of the squares of the radii of the balls to the $\frac{2}{3}$ power, or,

$$\frac{(0{\cdot}01)^{\frac{2}{3}}}{2\rho}$$

where 0·01 is the diameter, in metres, of the balls of the bearing actually tested, and 2ρ is the diameter of the bearing under consideration.

As regards the expression C_i, which depends essentially on the mechanical finish, it can be said that, provided the quality is the same, it certainly increases with $2r$, so that for the equation to be generally applicable, the expression must be multiplied by $2r/0{\cdot}04$.

The equation can now be written

$$f = 9 \times 10^{-5} P^{\frac{1}{3}} (0{\cdot}01/2\rho)^{\frac{2}{3}} + 5{\cdot}66 \times 10^{-5} + 2{\cdot}88 \frac{1}{\frac{P}{2r}} + 965 \frac{\mu V}{\frac{P}{2r}} \quad . \quad . \quad (6)$$

Let this equation be used to calculate the coefficient of friction of a ball bearing geometrically similar to that used in these experiments, but suitable for a shaft diameter $2r$=120 mm., i.e. three times the size of the experimental ball bearing (40 mm.). This bearing had practically the same dimensions as the roller bearings tested by the Isothermos Laboratory and had 30 mm. rollers arranged in two rows of 16 rollers each, as in the 40 mm. bearing. The load P/$2r$ corresponding to the total stress of 6,000 kg. must be taken as equal to 3,000/0·120=25,000 kg. since the bearing is double.

To calculate the value of the last term, a value must be assumed for the viscosity of an oil such as was used, the viscosity at 60 deg. C. being probably 0·005. With this value and N=800 r.p.m., then

$$f=10^{-5}(60{\cdot}7+5{\cdot}66+11{\cdot}5+95)=1{\cdot}73\times10^{-3}$$

which corresponds closely to the observed value ($1{\cdot}79\times10^{-3}$).

It would be of interest to work out the curve of μ in terms of the bearing temperature (which was close to that of the grease in contact with the ball race), as the experimental values could then be made to agree with the calculated values. Thus the Group II experiments under a load of 6,000 kg. yield the curve of Fig. 4, which certainly resembles the viscosity-temperature curve. (The circles refer to experiments with a load of 7,000 kg.) It will be seen that the last experiment carried out during running-in under that load gave a very low figure comparable to that obtained with a load of 6,000 kg.

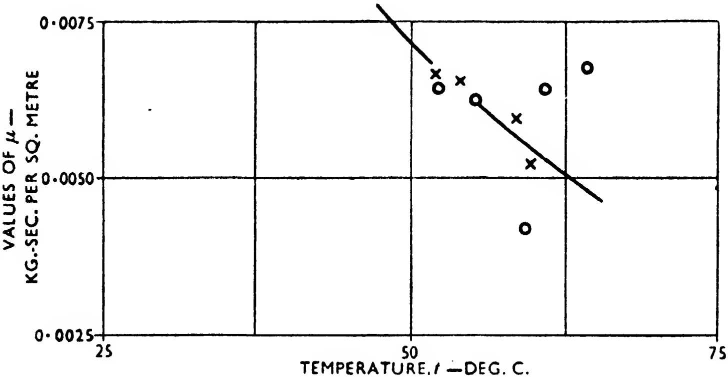

Fig. 4

There is thus sufficient ground for affirming that the proposed equation can be applied fairly widely as well as fairly strictly. Obviously, it must be remembered that a roller bearing is mechanically very complex and that the different coefficients in the equation may be modified when one type is replaced by another, according to the mechanical finish and even to the temperature when the clearances are not correctly adjusted for a particular working temperature.

THE LUBRICATION OF WIRE ROPES

By M. A. Hogan, D.Sc.*

Introduction. This paper is limited to the consideration of problems affecting flexible wire ropes moving over pulleys and drums. Standing ropes which are not bent or moved present somewhat different problems. Flexibility, the most characteristic property of a wire rope, is derived from the relative motion of the individual strands and wires. The bending of a rope over a pulley or drum is accompanied by a continuous process of readjustment of stress between the wires and deformation of the cross-section of the rope. The result of these actions is to reduce the maximum stress on the wires and, consequently, the stress on wires in a rope of given size is much less than on a straight bar of the same size bent to the same radius. Thus a wire rope may be regarded as a machine composed of a number of small units which continually slide upon each other and therefore require lubrication.

The lubricant applied to wire ropes serves: (*a*) to reduce internal friction between the wires and external friction against the drum and pulleys, and (*b*) to preserve the metal from corrosion. Formerly, doubt was thrown on the value of lubrication in relation to the first of these functions, and in some of the laboratory experiments carried out by Scoble (1935) it was found that dry ropes gave better results than ropes which had been lubricated. Subsequent experiments showed that, except under severe bending conditions, the differences between the two were not very great: sometimes lubricated ropes had longer life but occasionally dry ropes were to be preferred. The earlier experiments by Biggart (1890), and by Woernle (1934) showed that lubricated ropes were definitely better than dry ropes so that on the balance the experimental evidence is in favour of lubrication.

Practical experience definitely shows the desirability of keeping wire ropes effectively lubricated. With colliery winding ropes, for example (Dixon, Hogan, and Roberton, 1936), corrosion and corrosion-fatigue, due to the failure of the lubricant to protect the wires, account for most of the deterioration observed; with more efficient lubrication most of the deterioration would be avoided.

Two classes of rope must be distinguished, stranded and locked coil. In a stranded rope a number of strands, each containing six or more wires spun over a core wire, are spun over a main core, usually of fibre. In a locked-coil rope the wires are spun in concentric layers

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* Safety in Mines Research Board.

about a single core wire, the inner layers are of round wires, followed by trapezoidal or other specially shaped wires, and on the outside there is a special locking layer of Z-shaped wires. Internal wear is liable to occur at the contacts between the strands, between the different layers of wires in the strands, and between the different layers in a locked-coil rope. As a rule the length of lay of successive layers is different and consequently the contacts between the wires are oblique. In these oblique contacts any two wires have, in theory, only point contacts; but in practice there is some deformation of the metal which enlarges the contact area by plastic wear during the manufacture of the rope.

Internal Lubrication. Owing to the high contact pressures between the wires in a rope, seizing is liable to occur and wear may be very rapid unless a suitable material is present to lubricate the contacts. The conditions call for a material which can retain its lubricating qualities under high pressures and maintain an effective film on the surface of the wires in the presence of moisture and dirt. The lubricant must also resist being squeezed out of the rope in service.

In stranded ropes, lubricants may be introduced at various stages of manufacture, and where more than one type is used consideration must be given to the possible effects of interaction between the different lubricants. The fibre for the core is spun with the aid of a "batching" fluid which may be an emulsion of oil and water, or a thin mineral oil.* The core is next impregnated with a heavy mineral oil or grease by simple immersion, or by a vacuum and pressure process. The lubricant is usually heated to reduce its viscosity and assist penetration and after impregnation the core may contain upwards of 20 per cent of lubricant. It is to be noted that the thin batching oil already present in the fibre dilutes the heavier oil used for impregnation so that the viscosity of the lubricant in the fibre after impregnation will be less than that of the oil used for impregnation, and for this reason the fibre after batching should not contain more than about 5 per cent of batching oil. Experience shows that where a two-phase material—e.g. oil and wax—is used for impregnation the lighter phase is squeezed out in service leaving behind the wax, which has little lubricating or protective value (Mayne 1937). The strand interiors in ropes of this type are also filled with lubricant during the spinning of the wires and further lubricant is applied during the spinning of the strands on the core.

In locked-coil ropes, lubricant is applied as the rope is built up

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* In British Standard Specification No. 525, "Fibre Cores for Wire Ropes", the use of an acid-free pure mineral oil is specified.

layer by layer. Thorough lubrication of the interior of ropes during manufacture is of great importance because in practice it is seldom possible to replenish the internal lubricant should it be squeezed out or cease to be effective. Failures are due more commonly to the disappearance of the lubricant rather than to its decomposition.

External Lubrication. Lubricant is applied to the exterior of a rope to replenish the internal lubricant, or at least to seal up the surface and prevent the entry of moisture and dirt, and to preserve the exterior of the rope. The first of these functions is generally the more important because internal deterioration is, as a rule, more dangerous and more rapid than external. In some instances the presence of lubricant on the surface of a rope may give rise to troubles due to the reduction in the coefficient of friction—on endless haulages, for example, there may be slipping of the clips or of the rope on the driving pulley—and some other form of preservative dressing may have to be used.

The external lubrication of wire ropes is done at intervals usually of a week or more, the interval varying with the nature of service and the working conditions. Ropes working under dirty conditions should, if possible, be cleaned, and wet ropes should be dried before lubrication. The cleaning of long lengths of rope presents considerable difficulties. The simplest method of cleaning is by means of some form of scraper—a loop of wire for a locked-coil rope or a grooved wooden block for a stranded rope. Other methods involve the use of jets of compressed air, or wire brushes. Probably the best method of drying a wet rope is to keep it on the drum in a warm engine house and carry out the lubrication as the rope leaves the drum. The temperature of the lubricant should be regulated so that its viscosity is suitable for the method of application adopted.

Application of Lubricant. Lubricant may be applied to a rope by three methods; brushing, immersion, or spraying. If the lubricant is well worked on, a very satisfactory coating of lubricant can be obtained by brushing, but when long lengths of rope are being dealt with there is a tendency to omit the brushing and let the lubricant run on the surface of the rope. In the second method the rope is passed through a vessel containing the lubricant and the excess is scraped off as the rope leaves the vessel. This method is successfully applied to locked-coil ropes which can be effectively cleaned before lubrication so that the oil adheres without any further working. Stranded ropes cannot be cleaned so effectively and simple immersion does not usually give a very adherent coating of lubricant. In the third method a jet of compressed air causes a fine spray of lubricant to impinge on the surface of the rope at a high velocity. Under favourable conditions the speed of the oil particles enables them to penetrate through any thin films of dirt or moisture which may be on the surface of the wires and thus satisfactory adhesion is secured. In the past, jets of steam were also used, but the drawbacks of playing a jet of steam on a wire rope are obvious.

Experiments have recently been made on the continuous lubrication of winding and haulage ropes at collieries. A thin mineral oil—a penetrating oil—was applied to the rope passing over the pithead pulley by means of a pump. Ropes treated in this way keep very clean and the external condition is satisfactory, but the experience obtained is insufficient to decide whether the condition of the interior of the rope also remains satisfactory.

Types of Lubricant. It will be seen that wire rope lubricants have to work under difficult conditions. No agreement has yet been reached as to the best type of lubricant, but research and experiment on this matter are being actively pursued by manufacturers and users. The requirements are rather conflicting as materials having satisfactory adhesive powers and resistance to pressure are not, as a rule, good lubricants.

Petroleum oils either alone or admixed with other materials are the most generally used lubricants. Various grades are used ranging from the very thin oils for continuous application to the heavy greases used during the manufacture of ropes. The addition of wax to thicken an oil is undesirable because waxes are inert bodies having little or no lubricant or protective value, and the wax must be regarded as an adulterant. At ordinary temperatures the mixture tends to separate and, as already mentioned, in ropes having fibre cores the oil flows out leaving the wax behind. Another method of thickening is to add a soap to the oil, and properly made lubricants of this type have given good results in service. On the whole petroleum base oils are not sufficiently adhesive and much study has been given to the addition of materials to increase the adhesion.

In some instances a percentage of wood tar has been used, but the presence of organic acids in the tar introduces the danger of corrosion. Other adhesive materials which have been tried are sticky distillation residues or asphaltic compounds, but with these there is a risk that they may not retain their value in contact with the other oils in the rope. A few years ago an attempt was made to protect the inner faces of the strands by coating the fibre core with a heavy asphaltic dressing. Unfortunately when the lighter fractions of the petroleum grease with which the core had been impregnated were squeezed out they precipitated the asphalt so that its value was entirely lost.

Possibly the best results so far obtained in colliery winding have been with locked-coil ropes treated with internal lubricants containing graphite. In applying graphite to a rope during manufacture it is necessary to incorporate the graphite in a grease. Natural graphites of low ash content are used and as these do not remain suspended in oil, soap has to be added to give the mixture a proper consistency.

Formerly oils of animal origin were used on ropes but they have now been almost entirely abandoned. In a few instances in recent years trouble has been experienced due to the use of sperm oil for internal lubrication. The oil is fairly readily hydrolysed in the presence of water and the products of hydrolysis assist in the corrosion of the rope. Seal oil used as external lubricant on ropes working in very wet shafts appeared to give excellent results as the surface of the ropes remained bright and clean. However the oil also penetrated within the rope and leached out the mineral oil placed there during manufacture, leaving the wires unprotected and exposed to internal corrosion.

Besides the above materials mention must be made of the use of metallic zinc in the form of galvanized coatings. The zinc coating acts as a lubricant between the wire contacts as well as protecting the steel from corrosion.

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CUTTING FLUIDS AND COOLING OILS IN MACHINE TOOLS

By A. H. Lloyd, B.Sc., M.I.Mech.E., and H. H. Beeny, M.Met.*

Cutting fluids may be plain oils or they may be water-soluble oil emulsions or "suds".

The cooling qualities of oils are never outstanding, but if properly compounded they possess excellent lubricating properties so that friction between the chip and the tool and the tool and the work is reduced, thus facilitating machining by reducing the actual evolution of heat. Water-base liquids, on the other hand, possess extremely good cooling qualities, but have only in a minor degree the property of lubrication.

Metals, such as brass, cast phosphor bronze, gunmetal and grey cast iron, which break up easily to form short chips, are usually machined without any cooling or lubricating fluid. Similarly, aluminium and magnesium alloys are usually machined dry as the tensile strength is low and, except with certain aluminium-silicon alloys, there are no hard internal constituents which cause abrasion. Steels, Monel metal, and certain copper alloys, on the other hand, form a continuous chip which presses heavily upon the face of the tool. In such cases, when using high-speed steel tools, a cutting fluid is very desirable and the choice will naturally be governed by the predominant quality required. Thus, for rough machining or high-speed cutting with simple types of tool, soluble oil emulsions giving good cooling properties are usually selected. Tool wear need not be closely watched as regrinding is straightforward and the watery liquid is cheap and gives clean conditions of operation. On the other hand, in forming, hobbing, threading, and gear cutting, expensive form tools are frequently used so that it becomes important to preserve the edge of the cutting tool and therefore straight oils are chosen, the cost and quality of the oil increasing according to the demands of the operation and requirements for economical production.

The plain cutting oils may be of several types. Thus thin mineral oil may be used as it is cheap, stable, and causes no corrosion. On the other hand, it lacks a lubricating quality suitable for the extremely severe conditions obtaining in metal cutting and is not, therefore, to be advised for use alone. Such oil is good, however, in that it is practically immune from oxidation or gumming under the influence of cutting heat when exposed to the atmosphere. Fatty oils, either of

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* Alfred Herbert, Ltd., Coventry.

animal or vegetable origin, greatly improve the lubricating quality, but are expensive and tend to deteriorate rapidly in use. A compromise is reached by blending mineral oil with fatty oil. The fatty oil provides the principal lubricating quality while the mineral oil cheapens the mixture and helps to protect the expensive fatty ingredient from oxidation. The proportion of fatty oil, which in the best types is usually lard or neatsfoot oil, may vary in amount from 5 per cent up to about 50 per cent. It is noteworthy that the cost of straight cutting oils is closely proportional to the amount of fatty oil present.

In metal cutting the pressure between a tough continuous chip and the nose of the tool reaches very high values, and it is logical, therefore, that lubricants of the extreme pressure type should prove useful. Sulphurized or chlorinated fatty and mineral oils have been tried for this purpose and the sulphur-bearing oils are finding an increasing use, it being claimed that the sulphur both helps to prevent seizure between the chip and the tool and absorbs heat. In compounded mineral oils the sulphur is usually present in chemical combination with the fatty ingredient, it being possible to process a fatty oil so that it absorbs from 1 to 20 per cent of the element. Combined sulphur presents no danger of separation and, except that in damp weather there is some evolution of sulphurous gases, the sulphurized oil is quite stable. More recently a colloidal sulphur has been produced in the form of a concentrated liquid, which, when added to a cheap thin mineral oil gives a greater increase in life on hobbing operations than can be obtained with expensive sulphurized oil in the chemically combined state. The disadvantage is that after a period of use part of the sulphur tends to settle out, and it may be necessary to make provision to draw the oil supply from a suitable position in the sump.

The chief point in examining cutting oil is the amount and nature of the neutral fatty oil, as this mainly governs the quality and price of the material. The percentage of free fatty acid is also important as an indication of the degree of rancidity and probable bacterial content; the figure should be limited to about 3 per cent as greater amounts increase the risk of septic infection to those in contact with the oil and favour an etching or tarnishing action on exposed machine parts, or even an attack upon certain types of bearing material. A comparative heating test at 200 deg. C. gives an idea of the gumming propensity and probable loss by evaporation during machining. The closed flash point, which should not be less than about 350 deg. F., is useful as an indication of the temperature at which distillation, with the production of objectionable fumes, will commence, while it also determines the safety of the oil as regards firing. The viscosity at various temperatures indicates the ease with which the oil will flow over and cool the tool and the work during cutting. For a given fatty oil content the oil should be as fluid as possible, bearing in mind also the flash point and evaporation on heating. A comparatively cheap cutting oil, which is giving excellent service, combines a Redwood viscosity at 60 deg. F. of 261 sec. with the remarkably high closed flash point of 368 deg. F. This oil also has the advantage that it is unusually transparent, so that the condition of the work and the tool is readily seen.

In general, the use of a good quality blended cutting oil ensures a maximum tool life between grinds with accurate sizing of parts, but an exception will be found where much heat is being generated so that full cooling powers are required in the cutting fluid; here a soluble oil emulsion will most efficiently prevent the actual softening of the cutting edges and so will give the longest life between grinds.

Neat oils have advantages over soluble oil emulsions in that they lubricate the moving parts of machine tools, such as slides and turrets, with which they come into contact, and if there is any leakage into headstocks or gearboxes containing lubricating oil the contamination is unlikely to do harm; in addition, rusting does not occur and the appearance is preserved because the paintwork remains free from the softening or peeling which frequently occurs with soluble oil emulsions. Thus a good neat cutting oil prolongs not only the life of the tools, but also the life of the machine in which they are used. The more expensive the machine and the more intricate the design, the greater is the argument in favour of the neat oil.

So far the virtues of the neat cutting oils have been emphasized, but they also have disadvantages when compared with the aqueous emulsions. The cost per gallon of cutting fluid is from ten to thirty times greater than that of the emulsions, and there is a comparative lack of cooling quality which restricts cutting speeds; the tendency to fume produces a close oily atmosphere and the conditions of working generally are not so clean, so that there is always the danger of oil-dermatitis owing to the oil and dirt clogging the pores of the skin and the possibility of the oil becoming affected with *Staphylococcus aureus* and other disease-producing bacteria. On this account it is essential that plant be installed to clean the oil periodically. This can be done by heating the oil to 140 deg. F. to assist sedimentation and destruction of bacteria, and then by centrifugal separation while still warm to remove the remaining dirt and fine metal particles. It should be added that an effective disinfectant is now available which has a sterilizing action on oil and so prevents the conveyance of septic infection from one worker to another.

Water-base cutting fluids vary considerably in detail, but consist normally of a mineral oil, such as a pale spindle or machinery oil, along with a fatty oil which has been partly saponified with caustic soda to form a soap. Wood alcohol, among other substances, may be added to stabilize the emulsion and a phenolic disinfectant is also usually inçorporated. Upon adding water the soap emulsifies the inert mineral and neutral fatty oils which provide the lubricating qualities of the emulsion, minute droplets being formed which remain in suspension for an indefinite period and cause the characteristic milky appearance. In many of the soluble oils now on the market the soap is partly replaced as an emulsifier by sulphonated castor or other fatty oil, or even by a sulphonated mineral oil. These sulphonated soluble oils withstand greater dilution than was recommended for the earlier types and the emulsion, which may be made up with as much as thirty parts of water to one of oil, combines cheapness with satisfactory machining performance. Practically all these soluble oils, however, contain a considerable amount of free fatty acid which, in the presence of water, stimulates the tarnishing and rusting of iron and steel parts. It is desired to call particular attention to this matter as in many instances great damage has been done to the cast iron lathe beds and other parts of machine tools by the use of emulsions which are unsuitable owing to the quality of the neat oil, or which have been rendered unsuitable by dilution with an excess of water. It is advisable to limit the amount of free fatty acid to a maximum of 0·75 per cent in the neat oil, although no hard and fast rule can be drawn, as with varying types of soluble oil more than one factor can intervene.

About three years ago attention was drawn to the detrimental action of certain soluble oils on lathe beds and, as it was found that experimental grey cast irons, made both with and without the addition of nickel, chromium, and copper in proportions such as could be used commercially, were vulnerable to a similar degree, the problem was brought to the notice of an oil company. Investigations were begun in order to develop a new and improved quality of soluble oil, it being emphasized that existing emulsions in general lacked stability so that renewals of the cutting fluid had to be made fairly frequently, and that the water liberated by the separation was the main agent in producing rusting and pitting. A cutting fluid was produced which gave a comparatively clear and slightly alkaline liquid when mixed with water, and possessed greatly improved stability and a much decreased tendency to cause corrosion even at dilutions greater than those formerly used. The new type of cooling fluid has been used exclusively for soluble oil applications during the past fifteen months and, at a dilution of forty parts of water to one of oil, has shown a large economy. This saving has been achieved despite the higher initial cost of the new grade of oil and takes no account of the better condition of the machine tools themselves. It is due primarily to the high dilution and permanence of the liquid. The extraordinary stability of the emulsion is probably accounted for by the fact that the oil is in a condition more nearly corresponding to true solution than occurs in ordinary types of soluble emulsion. This stability is demonstrated by the strengthening of the solution which takes place during several months of use: water is lost, so that, notwithstanding additions at the original strength to make up the volume, a 40 to 1 mixture may ultimately show a ratio of only 15 parts of water to 1 of oil.

The neat oil mixes readily with cold water to form a non-corrosive liquid, but if the greatest freedom from rusting is desired under workshop conditions, it is beneficial to dissolve 0·5 per cent of common soda in a hard water, such as that found in the Coventry district, before adding the oil. This addition precipitates calcium and magnesium salts which would otherwise combine with some of the emulsifying soap in the oil to form compounds which rise to the surface as a scum.

Experiments with the use of emulsions on automatic and semi-automatic machines have been unsuccessful owing to the wear of tools, the danger of corrosion, and the sticking of moving parts, such as turrets and slides. The effort was renewed with the new and improved cooling fluid on several automatic screw-cutting machines, but was abandoned on account of the heavy wear of screwing dies, which lasted only approximately one-third of the time yielded with straight oils. There was no corrosion and any tendency to sticking was obviated by increasing the proportion of soluble oil in the cooling fluid, but owing to the wear it is now the definite policy of Messrs. Alfred Herbert, Ltd., to recommend a neat cutting oil for all automatic and semi-automatic machines. As already mentioned it is usually advisable to employ a "straight" oil for forming, hobbing, and gear cutting operations, or where an elaborate layout of tools is required.

The corrosion of cast iron lathe beds occurs mainly adjacent to the headstock where metal chips collect. These chips hold the cutting emulsion so that unless the machine is cleaned down the attack with an unsuitable fluid is continuous. The localized action can be accounted for by the absorption of oil on to the freshly cut metal particles so that the surrounding liquid is weakened; in addition electrochemical action probably occurs, since brass chips, for instance, cause corrosion on a cast iron surface much more readily than steel chips. A test which has been developed and standardized from practical workshop observations is used for checking the corrosive qualities of soluble oils. In this test small piles of steel and brass chips are placed in recesses milled into the top surface of a grey cast iron block. The metal is similar to that used for lathe beds and the cutting liquid is allowed to drip very slowly on to the prepared surface from a syphon. Six or more tests may be carried out side by side on the same slab of iron under identical conditions. The results obtained on any given emulsion are consistent and have proved a very valuable index of the behaviour of the cutting fluid in the machine shop.

An insidious form of corrosion has been noticed where a soluble oil emulsion has leaked into a housing containing steel gears and other mechanism. Though the fluid has not actually come into contact with these parts, evaporation and condensation has occurred so that water has been deposited above the level of the original liquid, bringing about mechanical failure due to severe corrosion. A similar action occurs where steam from the cut settles on to adjacent steel surfaces, producing red rust. These possibilities are, of course, unavoidable with any water-bearing cutting liquid and do not reflect upon the quality of any particular grade of soluble oil.

When using tools of the recently developed sintered hard carbide type on a modern lathe, cutting speeds are normally very high and any cutting fluid used is, therefore, scattered violently by the centrifugal action of the work in the chuck. It is the general practice, therefore, to cut dry under these circumstances, particularly as the remarkable resistance of the new tools to both heat and abrasion renders the use of cooling and lubricating fluids less necessary than with high-speed steels. In some cases, however, particularly where older types of machine tool are in use, lower speeds are necessary and soluble oil emulsions may then be used with advantage even with the new tools, as the liquid washes away the chips and prevents the work from heating.

THE LUBRICATION OF GEAR TEETH

By H. E. Merritt, D.Sc., M.I.Mech.E.*

A necessary preliminary to a discussion of gear lubrication is a description of the various typical forms of gear tooth failure which result from imperfect lubrication.

GEAR TOOTH FAILURES

Gear tooth failures may be classified mainly as (I) direct breakage of the tooth proper; and (II) tooth surface failures.

Group I comprises two classes: (*a*) those in which temperature, and therefore lubrication and cooling, play no part; they may be traced to faulty design, material, manufacture, or mounting, or to overload or abuse; and (*b*) those which are the result of excessive temperature, due either to imperfect surface lubrication and excessive friction, or to inadequate cooling (the one possibly reacting on, and aggravating the other).

Examples of this last-named class are:—

(1) The stripping of wormwheel teeth (worm underneath and normally immersed) through lack of oil (Fig. 1, Plate 1). Failure to conduct the frictional heat away from the teeth causes the temperature of the wormwheel to rise to a point at which the material will no longer withstand stresses which it will normally carry with safety. It does not follow that the gears have been allowed to run completely dry before this occurs: the oil level may have fallen to a point at which, whilst oil is still in contact with the worm threads when the gears are stationary, insufficient immersion occurs when running. The temperature of the gears may then rise to a point sufficient to rob the remaining oil of its lubricating value, and thus cause increased tooth friction, still higher temperature rise, and final failure.

(2) Excessive wear, scoring or cracking of wormwheel teeth or worm threads (worm overhead, wheel dipping in oil bath). If the viscosity of the oil is insufficient, enough oil will not be carried by the wheel up to the worm to cool it. The surface of the worm threads then becomes so hot that the film breaks down. The worm threads may then soften or crack, the bronze of the wormwheel seizes or "picks up" on to the worm, and the sliding bronze–bronze combination wears rapidly.

(3) Bronze gear rims shrunk on to a steel or cast iron centre will become slack on account of the difference in the coefficient of expansion

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* David Brown and Sons (Hudd.), Ltd.

Fig. 1. Wormwheel Stripped through Imperfect Lubrication and Cooling

Fig. 2. Case-Hardened Double-Helical Pinion Exhibiting Scuffing and Pitting

Fig. 3. Spur Gears (Driving and Driven) Exhibiting Ridging and Dragging

Fig. 4. Wormwheel showing Pitting (with Correct Lubrication)

Fig. 5. Tooth of Helical Pinion showing Pitting due to Initial Concentration of Loading, Scoring due to Foreign Particles, and Smooth Abrasion

if the temperature rise is excessive. Local overloading may then cause retaining screws or dowels to fail; the rim will creep sideways until the teeth jam and either seize or break.

The above types of failure are associated with worm gear, in which, on the one hand, the greater sliding velocities render the frictional losses more sensitive to incorrect lubrication, and on the other hand, the bronze of the wormwheel suffers a greater loss of physical properties with rise in temperature than do steels in general. The consequences of inadequate lubrication are thus more likely to be serious for worm gears than for spur, helical, or bevel gears. Proper selection and application of the oil are, therefore, of special importance.

The surface failure group II shows greater variety and the typical forms are as follows:—

(1) *"Smooth" Abrasion*. The teeth of spur, helical, and bevel gears (which all have a comparatively high "rolling" or contact-line velocity in relation to the sliding velocity, and are thus in contrast to worm gears), may show a form of wear in which the tooth surfaces remain smooth and sometimes quite highly polished. The rate of wear, although more rapid than is desirable, is still comparatively slow, and the tooth surfaces do not show a marked change of character at or on either side of the pitch line. The result is similar to what would be expected if the oil contained a small quantity of very fine abrasive; and whilst the particles of metal removed may later act as such, the wear might, at a guess, be initially ascribed to an extremely thin oil film which, stressed just below the point of rupture, occasionally breaks down and each time permits removal of a few minute particles by abrasion.

In several cases which have come to the author's notice, hardened gears carrying much less than normal loads have worn in this way in the presence of a light oil of unsatisfactory lubricating qualities (as measured by friction tests); the substitution of an oil of slightly higher viscosity but substantially lower friction cured the trouble before the gears were irreparably damaged. With traction gears, using the heaviest practicable lubricant, the same thing may occur, although much more slowly, and may be presumed to occur mainly under the heavy loads and low speeds of starting. It has also been suggested that wear of this kind is due to oxidation.

(2) *Scuffing*. This is the condition next in severity to smooth abrasion. It may occur with both soft and hardened gears, and consists in a tearing or scoring of the surfaces in the direction of sliding. Fig. 2 illustrates the scuffing of heavily-loaded case-hardened gears, pitting also being present. The oil film has apparently broken down under a pressure in excess of its capacity (which may be diminished by

high surface temperature) and the surfaces have locally welded and produced the characteristic effect. Although encountered with soft industrial gears lubricated by normal oils of too low a lubricating value, it has been a more serious problem with very highly stressed automobile gears (particularly spiral bevel and hypoid gears). Failure of the previously used heavy gear oils led to the development of extreme-pressure lubricants.

The precise action of extreme-pressure lubricants is somewhat controversial. It does not appear that they necessarily lower the coefficient of friction between the surfaces, or that there is any direct relation between "film-strength", so called, and surface friction; further, the fact that new gears, which show scuffing when fully loaded with ordinary gear oil, become smooth and polished when first run with extreme-pressure lubricant, and thereafter run satisfactorily with ordinary gear oil, lends itself to various explanations, none of which appears to have received practical confirmation. The effectiveness of such oils, however, is established, and many industrial applications await their development in the lighter range of viscosities.

(3) *Ridging or Dragging*. Associated with, and forming an extension of, scuffing, is an aggravated form of oil film failure (to which lack of surface hardness contributes) which leads to the formation of a groove along the pitch line of the driving tooth and the building up of a corresponding ridge along the driven tooth (Fig. 3, Plate 2).

With breakdown of the oil film, the friction between the surfaces, possibly increased by intermittent local seizure, produces in ductile materials a flow of the surface layers in the direction of sliding. Since the direction of sliding reverses at the pitch point, acting away from the pitch line of the driver and towards that of the driven gear, plastic flow alone would explain the ridging, but it may, of course, be accompanied by a re-welding of particles dragged from the surface at the pitch point, where sliding motion momentarily ceases.

(4) *Pitting*. This type of surface failure may occur with any gear or gear material (Figs. 4 and 5). Cracks develop in the surface, spread beneath it at a distance which bears a relation to the depth of penetration of stress, and emerge again at the surface, usually forming a closed area surrounding a flake of metal, which becomes detached. It is not uncommon for pitting to develop early in the life of a gear, reach a certain point, and then cease to spread. Sometimes it disappears with wear of the surface layers. Pitting is definitely associated with the stress conditions in the material, and can be reproduced by means of cylinders pressed into contact at their peripheries and rotating with various combinations of surface velocities. Arrested pitting is nearly always found in traction worm gears and turbine reduction gears, and is

usually attributable to local high spots (due to errors in either the cutting or the running position of the gears) which result in stresses sufficient to produce pitting before the irregularities have had time to be removed by smooth abrasion. Pitting and scuffing or dragging may sometimes be found together on the same gear, as in Figs. 2 and 5.

It has not usually been considered that pitting is related to lubrication, but Way * suggests that the propagation of a crack, once started at the surface, is effected by hydrostatic pressure developed in oil contained in the crack when the outer end is sealed by deformation of the surface layers under load. In support of this theory, his experiments show that pitting does not occur with pure rolling in the absence of lubrication. It may be added that although, under static loading, line contact stresses have no tendency to produce a crack starting at the surface, gear teeth in motion present two additional factors, namely, the tangential load at the surface due to sliding friction and the effect of the very high and rapidly changing temperature difference in the surface layers. These factors are both largely influenced by the intensity of loading, however, and whilst it is essential to provide a lubricant which will prevent the types of failure directly attributable to breakdown of the film, there is to the author's knowledge no conclusive evidence gained from practice that different lubricants will, other things being equal, either accelerate or retard pitting; provided that the film strength is high enough to prevent scuffing.

THE SELECTION OF GEAR LUBRICANTS

Gear lubricants should properly be selected having regard to their primary function of forming a protective friction-reducing film between the teeth under the prevailing conditions of tooth surface pressure, speed and gear materials; the choice of method by which lubricant is applied to the teeth will then be guided by considerations of gear disposition, speed, and viscosity. If the method of application imposes limitations on viscosity, it may be necessary to reduce tooth pressures by redesign.

The precise behaviour of the oil film which separates gear teeth is not clearly understood. It is probable, however, that a boundary film exists between the surfaces over the area which would make contact under static conditions, and that this is reinforced by a hydrodynamic film induced by surface motion in the wedge adjoining the static contact area. That such a film can afford protection to the surfaces under the

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* Trans. A.S.M.E., 1935, vol. 57, p. A49.

severe combinations of pressure and speed encountered appears to be due largely to the effect of the hydrodynamic film. Experiments on rolling-sliding disks have shown that with a clean lubricant, continuous motion of a kind which aids the hydrodynamic film, and pressures at which the surfaces remain polished, no wear measurable by instruments reading to 0·0001 inch can be detected after millions of revolutions; whereas if the motion gives no hydrodynamic effect (equal and opposite rolling) the film breaks down at negligibly small loads.

The limiting conditions of surface motion and pressure for the maintenance of an unbroken film for a comprehensive range of combinations of materials and lubricants are not, however, precisely known, and much work remains to be done on the subject.

The first step is to compare the conditions of surface pressures and speeds, and actual figures are somewhat startling in comparison with those encountered in other mechanisms. The calculation of tooth pressures and surface velocities is properly the province of the gear designer, but it is necessary to present in simple form a method of reducing gear tooth loading conditions to first principles in order that the design of laboratory apparatus for further investigation can be definitely related to actual conditions instead of giving, as happens with some lubricant testing machines, misleading or conflicting results.

Comparison of gear tooth pressures in terms of "load per inch of width" is misleading and obsolete. Surface curvature must be introduced, and this may be done without reference to gear tooth geometry by the use of the "surface stress factors" and "zone factors" introduced by the author, and since embodied in British Standard Specification No. 436 of 1932, No. 535 of 1934, and No. 721 of 1937. With the aid of these specifications, the intensity of loading on any pair of gears may be expressed by the surface stress factor, denoted by S_c; this means that the pressures and stresses in the region of the line of contact are comparable with those existing between a cylinder of 1 inch radius and 1 inch length, and a plane, making line contact under a force of S_c lb. (the cylinder and the plane being made of the same materials as the gears).

This method of stress comparison was adopted because allowable stress values found from experience cannot be directly related to the usual physical properties of materials and require the introduction of additional empirical factors. Moreover, the pressure distribution across the width of the band of contact is not accurately known, neither is the proportion of the load carried by the entering "wedge" film. The surface stress factor is, therefore, a convenient basis of comparison in the absence of more complete knowledge of what takes place.

The surface stress factor can, however, be reduced to maximum surface pressure, if static conditions are assumed, from the approximate relation

$$S_{\max}=0{\cdot}418\sqrt{\frac{S_c.2E_1E_2}{E_1+E_2}}$$

where E_1 and E_2 are the moduli of elasticity of the materials (units in pounds and inches). The calculation of the velocities of relative surface motion is somewhat more complicated, but Table 1 indicates the range of values of S_c, $S_{\max}$, and sliding velocity v_s encountered in various typical gear drives.

In order to put the selection of gear lubricants on a rational basis, what is needed is the establishment of a "film-destruction criterion" which will take into account the S_c value, the materials, and the relative surface-motion conditions. Test apparatus in the form of rolling-sliding cylinders or their equivalent might be applied to the determination of this criterion, and also serve for the rating of lubricants over the whole range of gear-tooth loading conditions. At the moment, such work appears to be restricted to the severest conditions requiring extreme-pressure lubricants. The author has been using apparatus of this kind for many years, but the results obtained have mainly related to worm gear materials and conditions, and the industrial gear field has been only partially explored.

With pitch-line speeds ranging from a few inches to upwards of 12,000 ft. per min., the range of types of lubricant must necessarily be wide, and in each individual case it is desirable to reduce viscosity to a minimum consistent with film strength, in order to minimize oil-churning losses. In the absence of a better criterion based on more exhaustive tests, it is necessary to regularize practice by a somewhat arbitrary classification of lubricants and of the conditions of application. "Recommended practice" must be suitable for commercial use; the necessity for calculations of a highly technical nature must be avoided.

As introduction to the classification of lubricants for gears of the "direct-acting" types, which include spur, helical, and bevel gears, it may be mentioned that the following broad generalizations have been drawn from the results of experiment and from experience of difficult cases:—

(*a*) For straight mineral oils of a given type (i.e. base) there is a rough relation between the viscosity and the coefficient of friction measured under the conditions of gear-tooth engagement, the higher viscosities giving the lower coefficient of friction.

TABLE 1. TYPICAL GEAR TOOTH PRESSURES

Type of drive	Materials	Speeds, r.p.m.	S_c	S_{max}, lb. per sq. in. (approx.)	v_s, ft. per min.	Remarks
Plate mill . . .	0·55 per cent C, forged	300–35	330	40,000	340	Normal
	0·40 per cent C, cast		2,000	90,000	340	Peak
Marine turbine reduction	Ni-Cr-Mo, forged	4,900–450	270	36,500	1,575	High pressure
	3·5 per cent nickel, rolled	4,000–450	390	44,000	1,325	Intermediate pressure
		2,600–450	480	48,500	900	Low pressure
Bevel gears (industrial)	Ni-Cr, air-hardened (80–100 tons per sq. in.)	300–460	740	60,000	370	Continuous
Automobile gearboxes (commercial)	Ni-Cr, case-hardened	At 1,000 r.p.m. (engine)	16,000	285,000	190	1st speed
			8,000	200,000	180	2nd speed
			4,500	150,000	250	3rd speed
Automobile double reduction rear axle (commercial)	Ni-Cr, case-hardened	At 1,000 r.p.m. (engine)	21,000	320,000	80	Bevels } 1st speed
			21,000	320,000	25	Helicals } 1st speed
Automobile worm drive rear axle	Nickel, case-hardened steel, phosphor bronze	At 1,500 r.p.m. (engine)	3,000	100,000	175	1st speed
High-speed worm drives (industrial)	Ditto	5,000–700	200	26,000	3,750	Continuous operation
Aircraft propeller reduction gears	Ni-Cr case-hardened steel	At 1,000 r.p.m. (engine)	7,000	190,000	600	—

(*b*) For the same viscosity, the coefficient of friction is appreciably lower with paraffin-base than with asphaltic-base or naphthene-base oils. Since the former are characterized by lower specific gravity, this property gives a useful indication of the quality of a gear lubricant.

(*c*) Excessive refinement, and the use of some solvent extraction processes, are liable to increase the coefficient of friction, and to involve the risk of more rapid tooth wear.

(*d*) Results obtained in practice, but not yet confirmed by laboratory tests representing industrial gear conditions, show that oils with a lower coefficient of friction (see (*b*) above) give greater protection against wear. It cannot yet be said, however, that this relation invariably holds good, or that it applies, in particular, to extreme-pressure lubricants.

Pending the accumulation of more exhaustive data, the above generalizations must be treated with some reserve; but they have been the basis of a range of specifications which fit in fairly well with the products of the leading oil suppliers and, in conjunction with Table 3, work out fairly well in practice.

TABLE 2. ABRIDGED SPECIFICATION,† GEAR LUBRICANTS

Grade No.	Approximate viscosity at 140 deg. F. (sec. Redwood)	Specific gravity at 60 deg. F. (max.)	Cold test, deg. F. (approx.)	Percentage compounding	Demulsibility test (max.)
1	90	0·890	25	—	2
2	110	0·890	25	—	4
3	140	0·890	25	—	6
4	200	0·895	25	—	13
5	275	0·895	25–30	—	—
6	400	0·895	25–30	—	—
7	600	0·895 *	30–35	5	—
7 T	800–1,000	0·900 *	25	5–10	—
8	1,000	0·910 *	35–40	5	—

* Before compounding.

† Practice of Messrs. David Brown and Sons (Hudd.), Ltd.

The viscosity at 140 deg. F. only has been given, since this is a safe operating temperature; the limitation of specific gravity usually ensures an oil of good quality, and a rigid limitation to other properties (flash point, viscosities at other temperatures, carbon residue, etc.) has not been found necessary for the majority of applications. An accepted and standardized test for surface protection and surface friction is, however, much to be desired.

Table 3 indicates the field of application of the various grades of oil given above, for spur, helical, and bevel gears.

Roughly speaking, the viscosity increases as the tooth pressures increase, for the allowable intensity of surface loading on the gear material increases with the hardness of the material and decreases with velocity. In modern practice, tooth loading is based on revolutions per minute rather than pitch line speed, but this is partly compensated by the fact that high-speed gears are of finer pitch, and have proportionately lower tooth-sliding velocities.

TABLE 3. SPUR, HELICAL, AND BEVEL GEARING

Totally enclosed units lubricated by splash or spray.

Material or ultimate tensile strength of wheel	Pitch line speed, ft. per min.						
	Slow, speed up to 100	100–200	200–500	500–1,000	1,000–2,500	2,500–5,000	Over 5,000
Fabric, cast iron, or bronze	6	5	4	3	2	1	—
Steel:—							
30–40 tons per sq. in.	7	6	5	4	3	2	1
40–50 tons per sq. in.	7	6	5	4	3	2	1
50–65 tons per sq. in. *	7	7	6	5	4	3	2
65–80 tons per sq. in.*	7	7	6	5	4	3	2
80–100 tons per sq. in.*	8	7	7	6	5	4	3
Case-hardened or surface hardened . . .	8	7	7	6	5	4	3

* When two nickel-chromium steel gears are running together specify an oil of the next heavier grade to that listed above.

The recommendations given above apply, of course, to gear drives of the highest quality, loaded up to their full capacity, and from which maximum performance and endurance are required. Many cases, in which the loading or the requirements are less severe, can be catered for by less expensive lubricants.

The lubrication of industrial worm drives calls for a rather different approach. In such drives the surface loading is usually well within the capacity of the gears on a surface-stress basis, and the limitation is one of temperature rise. The emphasis is therefore upon the coefficient of friction, to which, other things being equal, the efficiency loss by tooth friction and the temperature rise are proportional.

The difference between various types of lubricant in this respect is considerable, as the few typical curves given in Fig. 6 will show. These were measured on a "disk" machine, having disks of 8-inch (bronze) and 4-inch (case-hardened steel) diameter respectively, rotating in the same angular direction with an S_c value of 600, and with the rolling velocities two-thirds and one-third of the sliding velocity respectively. Particulars of the oils corresponding to curves A–G are given in Table 4.

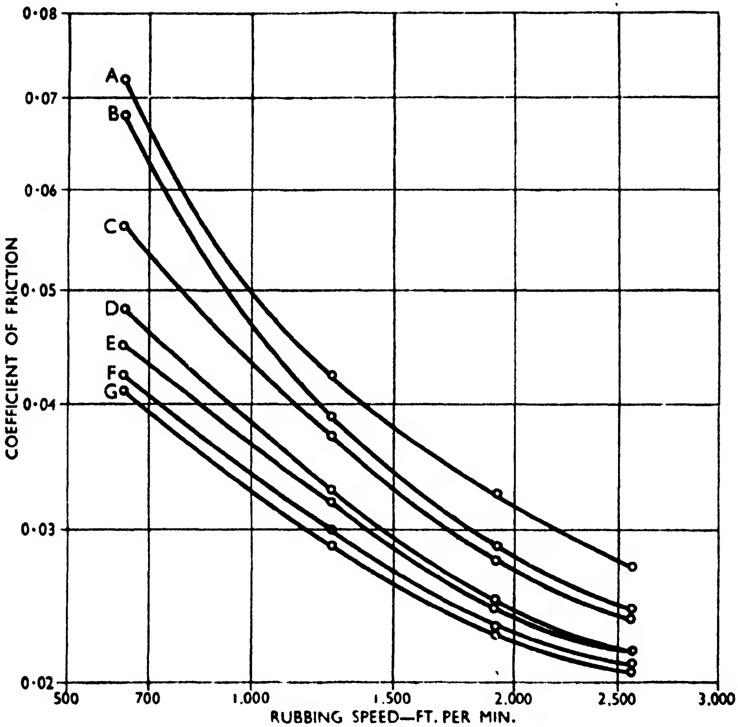

Fig. 6. Coefficient of Friction of Various Lubricants

The characteristics of lubricants A to G are given in Table 4.

In the lubrication of industrial worm gears, the necessity for minimizing oil churning losses arises, and since the intensity of surface loading diminishes with increase in speed, and heat dissipation at the higher speeds necessitates lower viscosity, the viscosity may be reduced as the sliding velocity increases. Table 5 gives values which have been found to give satisfactory results in practice.

Of all types of worm gears, vehicle rear-axle drives require the greatest care in the selection of oil. Surface pressures and running temperatures are both higher than in industrial work, and the worm is usually below the wheel, and completely immersed, thus increasing the agitation and tendency to oxidation and sludge formation. No mineral oil approaches castor oil in low tooth friction and low running temperature, but it is not foolproof, requiring careful watching and frequent changing, and it must not be mixed with mineral oil. For these reasons, mineral oils are usually preferred, a friction test on a disk machine being a good guide to suitability.

TABLE 4. OILS CORRESPONDING TO THE CURVES OF FIG. 6

Designation	Viscosity at 140 deg. F., sec.	Specific gravity at 60 deg. F.	Percentage compounding
A	85	0·883	—
B	198	0·904	—
C	101	0·905	—
D	390	0·906	—
E	107	0·875	—
F *	850	0·935	5·5
G	410	0·895	—

* Black oil.

TABLE 5. TOTALLY ENCLOSED WORM GEARS

Type of gear . .	Traction gears	Industrial gears						
Rubbing speed, ft. per min.	—	0–200, extra heavy duty	0–500, heavy duty	0–1,000, medium duty	1,000–2,000	2,000–3,000	3,000–5,000	Over 5,000
Grade of lubricant	7T	8	7	6	5	4	3	2
Method of application	Splash	Splash	Splash	Splash	Jet or splash	Jet, 10 lb. per sq. in.	Jet, 30 lb. per sq. in.	Jet, 50 lb. per sq. in.

Of the oils which the author has tested in this way, the addition of extreme-pressure base (with one exception) does not reduce the coefficient of friction; whether it reduces tooth wear of worm gears remains to be proved.

The foregoing outline of present-day practice, necessarily condensed, attempts no more than a bare outline of the problem, and of the way in which it has been approached. Its purpose is chiefly to draw attention to the need for further research by oil technologists on lines which definitely link up with gear design practice.

The author is indebted to Messrs. David Brown and Son (Hudd.), Ltd., for permission to include matter based on their experience and practice.

BALL BEARINGS FOR HOT SHAFTS

By Dr. A. Meyer*

In the application of uncooled-gas and exhaust-gas turbines as used by Messrs. Brown, Boveri and Company, Ltd., for their Velox boilers and turbo-superchargers, difficulties have been experienced on account of the high temperatures at which these machines are run. These difficulties and the way in which they have been overcome are described in this note.

The above-mentioned gas turbines drive blowers which supply the air for the combustion process from which the exhaust gases driving the turbine are obtained. A high overall efficiency of the turbo-blower set is essential here, for if the efficiency is too low, insufficient air will be available for the combustion process; too little exhaust gas is then delivered to the gas turbine, with the result that the output of the turbine and, accordingly, the delivery volume of the blower are still further reduced.

For this reason ball or roller bearings have to be used, at least for the smaller types of such turbo-blowers, comparative tests having shown that improvements in the efficiency up to 15 per cent can thus be obtained, as compared with journal bearings. Arduous requirements are placed on these bearings since the turbo-blower sets are run with very high gas temperatures and at speeds varying between 3,000 and 30,000 r.p.m.

The following difficulties had to be contended with at the gas turbine end when ball bearings were used for these sets. Owing to the high temperatures of the rotor (the temperature may be as high as 850 deg. C. in the guide blades of the turbine and 650 deg. C. in the moving blades) a considerable amount of heat was transmitted to the bearings through the shaft. This heat transmission is facilitated by the fact that the shafts must be very thick, so that they always run below the first critical speed, because the turbo-set may have to run at any speed and for any length of time between zero and the maximum speed, according to the load of the supercharged engine. On account of this transmission of heat the inner race of the ball or roller bearings heated up much more quickly than either the balls themselves or the outer race; the latter permanently transferrred heat to the casing. Consequently the diameter of the shaft and the inner race increased more than that of the outer race, so that the clearance first diminished and finally disappeared

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* Brown, Boveri and Company, Baden.

altogether. The ball bearing thus became a ball mill and very quickly ground itself to destruction.

Tests made with such bearings showed that the inner race could attain a temperature of 350 deg. C. Attempts to provide a remedy by more efficient cooling through more abundant lubrication did not always prove successful, since the influx of heat was still greater than its efflux. It was consequently necessary to attempt to reduce the influx of heat to such an extent that it could be removed by the available means of cooling.

This is achieved as follows. A thin sleeve is fitted on the part of the shaft carrying the inner race of the bearing. This sleeve rests on the shaft on small surfaces at each end only in order to diminish the heat transmission to the bearing by reducing the heat transmitting surfaces as well as the heat transmitting sections. In this way it is possible to cool the part of the sleeve supporting the bearings and the bearing itself so efficiently, that the diameter of the inner race is maintained at a value which does not impair the clearance appreciably. Consequently, the diameter of the sleeve carrying the bearing varies over its length according to the different temperatures of the various sections of the sleeve. At the points where the sleeve touches the shaft it will have a diameter corresponding to the diameter and temperature of the shaft, and at the point where the bearing rests on the sleeve it will have a diameter corresponding to the temperature of the bearing. The diameter thus varies continually between these two limits, so that the sleeve takes up a shape similar to that shown in Fig. 1, which, however, is exaggerated.

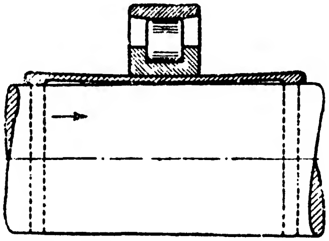

Fig. 1. Shape (exaggerated) Assumed by the Sleeve

It is possible to calculate, for given temperature conditions, the stresses set up in the sleeve by the variations of its diameter. These calculations as well as the absence of permanent deformation of the sleeves show these stresses to be well within the yield point of the material.

Fig. 2 shows the section of such a bearing. The influx of the heat comes through shaft *a* from the right. The heat can only flow to the bearing *b* through the surfaces c_1 and c_2 and section *d* of the sleeve *e*, which is slightly shrunk on the shaft, while the bearing itself is a tight fit on the sleeve.

Abundant cooling by oil or in special cases by air is, of course, essential and must be specially provided, as the normal amount of lubricating oil is insufficient for this purpose. On the other hand, special means must be applied so as not to flood the bearing with too much oil in order to avoid a pumping action of the bearing, with high bearing losses as a result. The shields *f* serve this purpose.

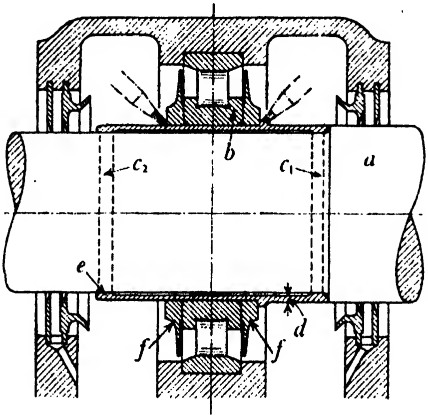

Fig. 2. Section of the Special Bearing

a Shaft. *b* Bearing. *e* Sleeve. *f* Shields.

This method of preventing undue heating and destruction of ball and roller bearings has been successfully applied during the last few years, not only for new machines but also in several cases where normally mounted ball and roller bearings failed completely.

EXTREME-PRESSURE LUBRICANTS AND LUBRICATION

By F. L. Miller*

Perhaps no recent single development in the field of lubrication has done more to revolutionize thought on lubrication than the introduction of the hypoid gear (Griswold 1936; Clayden 1937). Since its lubrication involves radical departures from previous practice, special lubricants have had to be developed. Some compare the action of these lubricants to that of an anti-weld agent. In any event, the fact remains that the gears can be operated successfully with these special lubricants, whereas scoring and failure is almost certain to occur with ordinary mineral gear oils. The hypoid gear is not the only one, however, which may require extreme-pressure lubricants for proper operation. The current trend in the automotive industry toward even higher power means that gear tooth loads will be higher in all differentials, even though perhaps not as high as in hypoid gear drives. Thus figures given by Wooler (1931) for a number of representative cars, show that the average gear tooth load rose from 1,180 lb. per sq. in. in 1924 to 1,450 lb. per sq. in. in 1930. Similar figures for 1936 or 1937 are not available, but it is probable that the increase in gear tooth loads was even greater from 1930 to 1936 than for the preceding six-year period. This is indicated by the fact that several manufacturers of cars equipped with spiral bevel gears specify the use of extreme-pressure lubricants. Worm gear differentials, in which one of the parts is bronze, may under certain conditions also benefit from the use of extreme-pressure lubricants. Further, the application of the proper extreme-pressure lubricants in industrial gears, often heavily overloaded, has in many cases eliminated gear lubrication troubles and resulted in much longer gear life.

Like any other gear lubricant, the different extreme-pressure lubricants must have certain well-defined properties. They should:—

(1) Possess high load-carrying ability.
(2) Not lose their load-carrying ability.
(3) Not cause wear owing to the presence of abrasive or corrosive compounds.
(4) Not cause corrosion of the operating or other parts.
(5) Not oxidize or thicken too rapidly in service.
(6) Not "channel" or foam.

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* Esso Laboratories, New Jersey, U.S.A.

In spite of abundant research on extreme-pressure lubricants during recent years, essentially only three elements—lead, sulphur, and chlorine—find commercial application in providing extreme-pressure properties, though in isolated cases certain phosphorus-containing derivatives have been mildly effective. The lead is usually employed in the form of soap which is wholly or partially oil-soluble, whereas the sulphur and chlorine are introduced in the form of a number of chemical compounds or sometimes directly. The lead, sulphur, and chlorine compounds are used singly, in combinations of any two, or occasionally, in combinations of all three, depending on individual ideas of how best to obtain the extreme-pressure properties desired. Further distinction is usually made between the sulphur-containing lubricants, depending on whether they are "active" (corrosive to a copper surface) or "inactive".

Of the possible extreme-pressure lubricants involving the use of various combinations of sulphur, chlorine, and lead, actually the number of promising types is limited largely to the following: (1) sulphur alone; (2) sulphur and chlorine; (3) sulphur, chlorine, and lead; (4) lead and sulphur (inactive); and (5) lead and sulphur (active). Seemingly, sulphur is a necessary component in every instance and the other elements are beneficial largely in supplementing its effectiveness.

Load-Carrying Capacity. The most important single property of an extreme-pressure lubricant is its load-carrying ability. Some gears require lubricants having only slight or mild extreme-pressure properties, while others, such as the hypoid gears, may require the strongest possible extreme-pressure characteristics. If properly made, the strong, or "powerful", extreme-pressure lubricants may, of course, be used with gears requiring only mild extreme-pressure lubricants or straight mineral gear oils. To measure load-carrying ability by actual gear or car tests is both tedious and costly, and is rarely feasible. Great impetus has thus been given to the development of simpler and more convenient test machines. Among the more familiar of these laboratory testing machines are the Mougey (Mougey and Almen 1931), Almen (Wolf and Mougey 1932), Timken (Wooler 1931; Maag 1932; Boden and Maag 1931; Frame and Graham 1933), Floyd (Automotive Industries 1931), Faville-Lavally, and S.A.E. (McKee and co-workers 1933; Neely 1936) machines. The last-named machine has been developed through a co-operative programme sponsored by a committee of the Society of Automotive Engineers (S.A.E.), and is now being critically tested by various laboratories.

Although all these machines purport to measure the same property of lubricants, hardly any two show even reasonable agreement as to the relative order in which they rate the load-carrying abilities of different types of lubricants. This is illustrated by the results, given in Table 1, of tests on several of the better-known testing machines of representative extreme-pressure lubricants.

It will be seen that no lubricant carries the highest possible load on any of the four machines listed. The active lead-sulphur lubricant carries the highest load on the S.A.E., Almen, and Mougey machines, but ranks only third when tested on the Timken machine. The inactive lead-sulphur carries the highest load on the Mougey machine and ranks first on the Timken, but does not perform at all well on either the S.A.E. or Almen machines.

TABLE 1. LOAD-CARRYING CAPACITIES

Type of lubricant	S.A.E. 1,000 r.p.m., lb.	Almen, weights*	Mougey 100 r.p.m., weights*	Timken, lb.
Active lead-sulphur . . .	550	15	25	50
Inactive lead-sulphur . .	50–200	2	25	65
Active sulphur with a saponifiable fatty oil	250–550	15	12–15	45–50
Inactive sulphur with a saponifiable fatty oil	100–350	5–10	10	45–50
Sulphur-chlorine . . .	50–200	15	15–25	65
Maximum load on the machines .	550	15	25	—

* Cf. Wolf and Mougey 1932*c*.

It is now fairly well agreed that the S.A.E. machine, when run under proper conditions of speed (750–1,000 r.p.m.), rate of loading (83·5 lb. per sec.), and roll ratio (14·6/1), generally shows the most consistent correlation with actual gear performance of any of the laboratory extreme-pressure testing machines yet developed. This is shown by service tests reported by Zwahl (1937). In proving-ground gear tests with Chevrolet cars and hypoid axles, in which 111 different lubricants were tested, a total of 70 lubricants performed satisfactorily without scoring the gears, and 41 resulted in scored gears and failed the test. The classification of these lubricants by chemical types is shown in Table 2.

Of the 70 lubricants which passed the gear-scoring tests at least the 62 active lead-sulphur, and presumably the five sulphur lubricants carried the top load on the S.A.E. machine at 750 r.p.m. Most probably these would also have performed equally well at 1,000 r.p.m., which has recently been accepted as the more desirable operating speed, although such data were not given. Both the active lead-sulphur and sulphur lubricants have come to be known as "powerful" extreme-pressure lubricants. However, despite their high load-carrying capacity on the S.A.E. machine, the lubricants containing sulphur alone have not always given satisfactory service performance and the five oils which passed the proving-ground tests were not approved for this reason. The three lead-sulphur-chlorine blends were the only oils which did not carry the top loads on the S.A.E. machine, but still performed satisfactorily in the rear axle. Because of this consistently excellent performance of the leaded lubricants, at least two American car manufacturers now specify that the "powerful" extreme-pressure lubricants, containing lead, shall be used in their cars.

The 41 lubricants which failed in the proving-ground tests included the inactive lead-sulphur, sulphur, chlorine, sulphur-chlorine, and lead-sulphur-chlorine types. Most, if not all, of these failed to carry the top load at 750 r.p.m. on the S.A.E. machine, and they would have performed even more badly at the higher speed of 1,000 r.p.m. Lubricants of these types, some of which show load-carrying abilities under these severe conditions only slightly greater than mineral oils themselves, are generally termed "mild" extreme-pressure lubricants and are suitable only where service requirements are not as severe as in the hypoid gear.

TABLE 2. CLASSIFICATION BY CHEMICAL TYPES

	Lead soap, active sulphur	Lead soap, sulphur-chlorine	Lead soap, little or no sulphur	Sulphur lubricants	Chlorine	Sulphur-chlorine	Miscellaneous	Total
Passing .	62	3	0	5	0	0	0	41
Failing .	0	2	15	7	13	2	2	70

Maintenance of Extreme-Pressure Characteristics. As extreme-pressure lubricants depend on chemical reaction for their effectiveness, they all tend to lose their extreme-pressure properties during use or under heating. The loss may be due either to chemical rearrangements within the lubricants themselves, or to continuous reaction with the gear teeth surfaces, with consequent depletion of chemical ingredients—probably both factors are involved. The extent of the loss of extreme-pressure characteristics may be great enough to be serious, or so slight as to be undetectable by any available methods. It is safe to say, however, that it is now possible to produce lubricants of the "powerful" type which will retain their extreme-pressure properties almost indefinitely. An illustration of the loss of extreme-pressure ingredients which may occur is given in Table 3, which sets out results of hypoid gear tests run under relatively mild conditions (60 m.p.h. and road load). The lubricant was a commercial sulphur-chlorine extreme-pressure lubricant of the mild type. Within a run of 3,600 miles the chlorine content was reduced to almost one-half of its original value, indicating that the lubricant would have a limited life, depending only on the length of time required to consume the remainder of the chlorine. Almost the same behaviour can also be demonstrated by heating the lubricant (at 200 deg. F.) in the presence of metal, such as steel wool (Table 4).

TABLE 3. LOSS OF EXTREME-PRESSURE PROPERTIES AND INGREDIENTS BY SULPHUR-CHLORINE LUBRICANTS

Packard car tests			Almen machine tests	
Miles run (at 60 m.p.h.)	Chlorine, per cent	Sulphur, per cent	Load	Condition of test pin after test
0	0·62	1·00	15	Smooth
1,400	0·48	1·03	15	Smooth
3,600	0·32	1·04	15	Noticeably scratched

TABLE 4. HEATING TESTS (IN PRESENCE OF STEEL WOOL)

Hours heating at 200 deg. F.	Chlorine, per cent	Sulphur, per cent	Almen machine tests	
			Load	Condition of test pin after test
0	0·62	1·0	15	Smooth
1	0·38	Not determined	15	Slightly scratched
2	0·21	,, ,,	10	Seized and sheared
5	0·18	,, ,,	7	,, ,, ,,

Other types of lubricants may also lose their extreme-pressure properties. Active lead-sulphur products, if improperly compounded, may show loss in load-carrying ability on use or from heating, but when properly made with suitable base stocks, correct incorporation of the optimum quantity of sulphur in the oil, and proper blending of lead soap, these lubricants do not lose their properties even after long use (Table 5).

Wear. If extreme-pressure lubricants are to enjoy the fullest development, they should cause no greater wear of the various parts of the rear-axle assembly than ordinary mineral gear oils. In many cases, however, they have been accused unjustifiably of causing greater wear, particularly of ball bearings. Leading automotive engineers (Zwahl 1937) state that extreme-pressure lubricants of the proper type and of sufficiently high load-carrying capacity will not cause wear of gears. Rather, gear wear is found to occur when the lubricant is not sufficiently powerful to provide adequate lubrication, thus allowing scoring or scuffing. This, therefore, leaves the ball or roller bearings as essentially the only parts which might possibly be adversely affected by these lubricants. Studies of ball bearing wear, however, have shown that extreme-pressure lubricants of proper quality will not cause undue wear. Further, with the active lead-sulphur lubricants, wear occurs largely only when harmful abrasives are present, either in the original base oil or derived from the manufacturing procedure. Oils of this type which are entirely free from sediment show the same exceedingly low wear tendencies that are observed with well-refined mineral oils. The instrument employed in making the wear measurements was a modification of the Timken extreme-pressure lubricant testing machine, in which a ball bearing assembly (in one test, a roller bearing assembly) was mounted on the main shaft in lieu of the usual roller. The load was applied by the normal lever arrangement through a supporting collar designed to fit the outer surface of the bearing assembly. Runs were 150 hours in length, during which a load of

TABLE 5. ACCELERATED HEATING TESTS WITH ACTIVE LEAD-SULPHUR LUBRICANTS

Lubricant	Hours heating at 200 deg. F.*	S.A.E. machine load at 1,000 r.p.m., lb.
Active lead-sulphur (weak)	0	550
	48	350
	64	200
Active lead-sulphur (satisfactory)	0	550
	64	550
	150	550

* Procedure specified by the Packard Motor Company.

1,000 lb. and a temperature of 165–175 deg. F. were maintained. Wear was determined by the loss of weight of the bearings (Table 6).

The data in Table 6 merely illustrate the trend of wear with these different lubricants under normal conditions and no attempt has been made to introduce other complicating factors such as moisture or contamination from outside sources which might be encountered in certain types of service. It will be observed that with the sulphur-lead lubricants the extent of wear is related to the benzol-insoluble sediment content or to the non-combustible sediment content. For some unexplained reason, wear was somewhat higher with sulphurized and sulphur-chlorine lubricants. In no case, however, was wear excessive, considering the severity of the test and the load to which the bearings were subjected.

TABLE 6. BALL BEARING WEAR IN A MODIFIED TIMKEN MACHINE

Lubricant	Loss in weight, grammes	Sediment content,† grammes	Non-combustible sediment (abrasives), grammes
Pennsylvania bright stock	0·008	None	None
Active lead-sulphur A	0·004	Trace	Trace
,, ,, B	0·007	Trace	0·003
,, ,, C	0·018	0·15	0·009
,, ,, D	0·027	0·30	0·015
,, ,, E	0·013	0·30	0·008
Sulphurized (inactive)	0·012	Trace	—
Sulphur-chlorine	0·028	Trace	—
Active lead-sulphur E*	0·007	0·30	0·008

* This run was on a Timken roller bearing (designation ♯362).
† Determined by adding 9 parts benzol and centrifuging.

Attempts have been made to measure wear on extreme-pressure testing machines by determining the loss of weight of the rolls, bearings, pins, etc., under arbitrarily selected operating conditions. Results obtained in this manner, however, have shown almost complete disagreement among themselves, and so far no evidence has been offered to indicate that any one of the tests of this nature correlates with service.

Resistance to Oxidation and Thickening. Extreme-pressure lubricants, like other gear lubricants, should show a minimum tendency to thicken during use, usually as the result of oxidation. While, in general, the various extreme-pressure agents tend to decrease resistance to oxidation, this effect is slight and it is possible to make products which are practically as stable as the base mineral oils themselves. The most important consideration is the oxidation properties of the base stocks used, since their stability is reflected in the characteristics of the finished extreme-pressure lubricants. This is shown by the results (Table 7) of laboratory accelerated heating tests (100 hours at 300 deg. F.*) on several representative base oils and on the extreme-pressure blends made from them.

For most applications exceptional stability against oxidation and thickening is probably not necessary. Transmission and rear axle temperature under normal passenger car operating conditions seldom exceeds 200 deg. F., a temperature which is not particularly severe. In more trying service, however, such as often occurs in bus and truck operations, temperatures are higher, occasionally reaching or exceeding 300 deg. F., and under these conditions resistance to oxidation is extremely important. If a single product is to be recommended for all types of operation it is essential that the maximum possible stability be provided, using base stocks of the highest quality and purity.

TABLE 7. EFFECT OF HEATING ON VISCOSITY

Lubricant	Saybolt viscosity at 210 deg. F.		
	Before heating	After heating	Increase, per cent
Mineral oil base A: S.A.E. 90 .	82·0	84·9	3·5
Active sulphur-lead lubricant made with base A: S.A.E. 90	84·4	88·5	4·8
Mineral oil base B: S.A.E. 90 .	85	101·9	20
Active sulphur-lead lubricant made with base B: S.A.E. 90	88	111·7	27
Active sulphur-lead lubricant made with base C: S.A.E. 90	86	94·5	9·8

Corrosion. The word "corrosive" as applied to certain of the sulphur-containing extreme-pressure lubricants has been a much abused term. It indicates that the lubricant will stain or blacken a polished copper surface under the conditions of the "copper-strip" corrosion test. The mere fact that the sulphur is present in a sufficiently chemically active form to discolour copper does not mean that it will react similarly with the surfaces which are encountered in the usual differential assembly. Even the most active of the various sulphur lubricants will usually not stain steel and some will only slightly stain normal bronze. On the

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* This test was suggested by General Motors specifications.

other hand, certain types of lubricants will not discolour copper in the copper strip test although they actually corrode or dissolve the metal as determined by measuring the loss of weight of the strip during the test. Neither when copper is blackened with sulphur lubricants nor when there is corrosion without discoloration is any information given regarding the actual service performance of the lubricants except that the ingredients are in a chemically active and immediately available form.

Of greater significance are the reactions which may occur in operation and the effect of the final reaction products. There is ample evidence to show that sulphur and chlorine in these lubricants react with the gear tooth surfaces in service. Under ordinary conditions it is probable that the films which are formed are stable and are partially renewed as the rubbing action under high pressures removes the old surface. In the presence of moisture, however, which is very often found in the differential as a result of condensation, the behaviour may be entirely different. Sulphide films, particularly with the lead-containing lubricants, show little or no tendency to react with the moisture. Chloride films, however, generally react readily with moisture, forming "rust" which is quite abrasive in nature and may be injurious if sufficient quantities accumulate. Wolf and Mougey (1932) observed this behaviour in car rear-axle tests, failure occurring with sulphur-chlorine lubricants following corrosion and wear in the presence of moisture. At the present time there are no adequate laboratory tests which will predict the extent or degree of corrosion which can be expected in service.

Channelling and Foaming. Channelling and foaming characteristics are as important for extreme-pressure lubricants as they are for ordinary mineral gear oils. Channelling at low temperatures is probably even more serious with extreme-pressure lubricants, since any lack of lubricant is apt to result in gear failure. Such lubricants must, therefore, flow freely at lower temperatures than do normal gear oils. The term channelling is applied rather loosely since no method for measuring this property is generally accepted. The Society of Automotive Engineers and the American Society for Testing Materials are now studying this problem.

The need for freedom from foaming is well recognized, but further work is desirable. The General Motors and Packard laboratories in their purchase and approval specifications suggest the use of a simple beater type of test. This test is widely employed at present, but is of doubtful merit, since it is not necessarily run under conditions representative of those encountered by the oil in service.

Future Trends. Rapid progress has been made in recent years and extreme-pressure lubricants of greatly improved quality are now available. With proper choice of ingredients and care in manufacture, the petroleum industry is able to provide the "powerful" lubricants required by the newer cars. These lubricants are not only able to carry extreme loads but are exceptionally stable against deterioration, thickening, etc., and free from objections of increased wear, channelling, and foaming. Unfortunately, because the problem has not yet been sufficiently clarified, it has been necessary for certain of the automotive concerns to draw up requirements for extreme-pressure lubricants, covering not only the load-carrying capacity and other usual properties, but also the chemical type. This is regrettable, but, presumably, this situation is a temporary one which will be corrected as soon as field experience permits.

More research in this important field is still needed. Petroleum chemists and engineers are still limited in their choice of ingredients for the manufacture of satisfactory lubricants. It seems probable that eventually the list of materials which will provide satisfactory extreme-pressure properties will be considerably extended. Further work, too, is urgently needed in developing tests for load-carrying capacity, wear, and corrosion. In recognition of this need, the Society of Automotive Engineers is continuing its excellent work and has committees studying all of these items. It is hoped that within a reasonably short period many of the questions connected with the problem will be eliminated and that soon the properties of these lubricants will become so generally understood that they will be used as widely in gear lubrication as straight mineral oils have in the past.

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THE APPLICATION OF CUTTING FLUIDS

By John F. Miller *

This paper is primarily concerned with the application, from the chemist's point of view, of cutting fluids in the machine shop and the troubles which may arise from the influence of extraneous machining conditions.

The cutting fluids generally obtainable fall within two broad classes, the so-called "water-soluble" oils and the straight or neat oils. The water-soluble or emulsifying oils consist essentially of a sulphonated oil or rosin soap which emulsifies the other constituents, namely mineral oil, fatty oil, and combined alkali in the form of soap. Another type of soluble oil usually contains more soda and, when added to water, gives an emulsion which is comparatively transparent. The most usual straight oils are the viscous, dark-coloured sulphurized oils which consist of mineral oil and fatty oil with which sulphur is intimately amalgamated during manufacture. These are offered in concentrated form and can be diluted with mineral blending oil according to the service needed or are sold ready blended for use at various concentrations. A pale sulphurized oil, which is relatively translucent, is also available. Other straight oils include medium heavy mineral oil, with or without the addition of fatty matter, and lard oil, either employed neat or blended with mineral oil.

Essential Functions of Cutting Fluid. No matter what the medium is, the prime functions of a cutting fluid are:—

(1) To cool the tool and work piece effectively.
(2) To provide adequate lubrication so that tool resistance during cutting is lessened.
(3) To establish a thorough flushing action so that swarf may be carried away from the tooling area as quickly as possible.
(4) To "wet" the metal quickly.

(1) In all metal cutting operations, the rapid dispersion of frictional heat is vital. The wedging action of the tool creates a cavity bounded by the parent metal, the chip, and the tool, and it is the ability of the cutting fluid to occupy that cavity continuously which enables it to effect cooling, in that the small quantity trapped is quickly replaced by other and cooler fluid.

(2) The cutting fluid must also possess a degree of oiliness. It is

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* Fletcher Miller, Ltd., Dukinfield.

suggested that oil fills the space created by the chip parting from the metal and as this cavity is formed immediately in front of the tool track, a very considerable lubricating action is produced. Furthermore, as the chip cleavage leaves a rough, fractured surface to be cleaned up by the tool point, it is suggested that lubricant occupies the minute "hills" and "valleys" momentarily to provide a lubricating layer, if only for a fraction of a second. It is believed that sulphur increases the oiliness of mineral or fixed oils so that when applied to metal cutting, friction is reduced and also the amount of heat released. Whether sulphur additions tend directly to increase the cooling capacity of cutting fluids is a moot point; it is more probable that decreased heat generation is the relevant factor. Sulphur does, however, increase the "bite" of cutting tools and also improves the adherence of oils with which it is incorporated.

(3) To accomplish the rapid removal of swarf from the cutting area so that tool keenness and component finish are not endangered, the cutting medium must flow freely and be fed in sufficient volume for the scouring action to be automatic and complete. Ready separation of the swarf from the cutting fluid, either by physical or mechanical means, should be a further characteristic.

(4) The "wetting" propensities of a cutting fluid have a profound influence on its efficiency. If a straight oil or aqueous emulsion fails to "wet" the metal surface being machined, it will bead off and provide very doubtful cooling or lubrication. Conversely, high adherence combined with quick "wetting" properties ensures that the oil clings to the metal, even at high speeds and flows rapidly over the "wetted" surface to establish a layer of cutting fluid just where it is needed. This point may be illustrated from practice. When machining an 8-inch diameter phosphor-bronze wormwheel with special tipped tools, taking a cut ranging from $\frac{1}{16}$ inch to $\frac{1}{8}$ inch in depth, the operation is carried out at the amazing speed of 2,100 surface ft. per min. As a 1 in 25 emulsion of soluble oil is used with complete success, there is obviously much more in this "wetting" property of cutting oils than is generally admitted.

Failures with Soluble Oil. The soluble oils can cause difficulties, mainly owing to the fact that their emulsions establish a balance between oil and water and when any outside influence disturbs that balance, inferior cutting control or complete breakdown of the emulsion may result. Many troubles are caused because the user does not carry out the maker's mixing instructions to the letter. If the maker says "Add the oil to the water" it is fairly certain that the reverse process will not work. Similarly, if the instructions say "Stir thoroughly whilst adding the oil", it is probable that the oil will not emulsify properly if all the soluble oil required for a particular mixing is dumped into the tank and then stirred. Again, when the maker says the oil must not be used at greater concentration than 1 in 10 parts water, a 1 in 5 mixing will probably separate out. The explanation is that the ratio of oil to water increases to an extent where there is not sufficient water present to hold all the molecules of oil as a stable emulsion. The mixture is oversaturated and some of the oil content separates out. So if the 1 in 10 emulsion will not suit the operation in hand, a straight oil should be employed, provided of course that failure is not really due to inadequate flow. Unless a tooling operation is really flooded with cutting fluid, one cannot say that machining conditions are correct. Furthermore, that flow must be at low pressure, otherwise splash-back sends the essential fluid away from its objective. Insufficient flow of oil is a common cause of unsatisfactory machining. Generous flow is also advisable because the cutting fluid has then a longer effective life when delivered in abundance at low pressure. Each molecule of oil makes contact with the heated tool and work for a shorter period and is then replaced. This implies that the oil remains cool and is better able to maintain the tool and work piece at a relatively even temperature.

Hard water is the most common cause of failure of soluble oil due to separation. If the water supply is hard, it is politic either to install a water-softening plant; add a softening agent to the water before mixing the soluble oil; or obtain an alternative supply of soft water. The importance of this matter is shown by the fact that water of about 16 deg. hardness destroys nearly 12 oz. out of every gallon of soluble oil concentrate when circulating a 1 in 20 emulsion. It would therefore repay any considerable user of soluble oil to adopt water softening, whether or not he is troubled with separation.

Many soluble oil emulsions break down when machining cast iron. An explanation is that cast iron is chemically very active and readily combines with the fatty acid in the soluble oil to cause separation. Even if separation does not immediately follow, there is always the risk of rusting the work or machine parts owing to the emulsion strength falling. It is therefore advisable to employ one of the soluble oils specially blended for machining cast iron.

Emulsion breakdown encountered when machining acid-pickled metals can be averted by adding a small proportion of soda to the mixing water to neutralize the acid which causes the trouble.

When machining brass with high lead or zinc content, there is risk of emulsion failure due to slight attack between the fatty oil, lead, and zinc. The oleates of zinc and lead which are formed, increase as more brass is machined until a point is reached where they settle out as a curd in the emulsion and disturb its homogeneity. This difficulty is best overcome by renewing the cooling fluid more often than when machining steel, or by using one of the special blends of soluble oil marketed for machining cast iron or by the employment of a light duty straight oil instead of a soluble oil emulsion.

Peculiarities of Aqueous Emulsions. Amongst troubles associated with the use of soluble oil comes the discoloration of the emulsion. This may arise from machining rusty metal stock, from rust in the circulating system, or in the collecting ducts of a centralized system of cooling fluid supply. The emulsion becomes brown, but so long as rust particles are excluded from recirculation, little harm will result. Cooling fluid used in this state should, however, be renewed more frequently to avoid the possibility of ultimate separation caused by high concentration of iron oleate. The other principal cause of discoloration is due to copper oleate, identified by the familiar green emulsion formed when machining any copper-rich alloy. Copper oleate quickly colours a large volume of emulsion but normally is harmless. On copper for example, the attack is so slight that no etching effect is discernible at 45 magnifications after 36 hours' immersion in a 1 in 20 emulsion of good quality soluble oil. Shop foremen often incorrectly diagnose this coloration as verdigris (copper acetate).

Gummy deposits on machine parts may occur with soluble oils containing a large amount of rosin and, when slides and guides are affected, free movement may be restricted. A soluble oil of good quality does not cause gumming. Rusting of the work, machine, or tools may arise from incorrect mixing, the use of hard water, or the practice of leaving machined parts lying in the shop for extended periods. Excessive dilution may also be the cause.

There seems to be some nervousness about the use of soluble oil on automatic lathes on account of possible interference with the bearings. The author knows many engineers who use soluble oil on automatic lathes at dilutions up to 1 in 35 parts water and has never heard that the bearings gave trouble because an aqueous emulsion was present. It must be emphasized that many of the difficulties arising from the application of soluble oil are infrequent and that the author still considers a first-class soluble oil to be the most adaptable and economical cutting fluid available to engineers to-day.

Machining Troubles with Straight Oil. Smoking, perhaps the most frequent trouble, may be due to exceptional surface speeds, heavy cuts on tough materials, and inadequate flow of oil, either separately or in combination. The last-mentioned cause often recurs, but if the delivery of oil is increased to give a real flood of well-diffused oil at low pressure, the smoking often disappears. Should it persist, however, a slight reduction in surface speeds will probably lessen or cure the smoking. Smoking may also be due to the use of an oil of too high viscosity in relation to the cutting speed. In such cases, the oil fails to spread quickly enough for it fully to remove the heat generated. Tearing of the metal often accompanies smoking and may therefore be prevented in the same manner.

Tarnishing of copper-rich alloys machined with a sulphurized straight oil is also encountered. This stain is due to a sulphur-copper reaction which only occurs in the presence of moisture. It is very difficult to exclude moisture from the oil in circulation, but sulphurized oil should not be stored in a damp place in an open container. Care in the stores here will prevent trouble in the machine shop.

Regarding dermatitis and skin complaints, the author believes that isolated outbreaks in engineering shops can be more often traced to the operative than to the cutting oil. It is, however, incumbent upon the management to instruct the worker how to protect himself against skin troubles as he is generally quite uninformed on such matters. If a machine tool worker does not wash his hands and forearms thoroughly after each shift, he invites skin troubles, that is, if he is susceptible, and not everybody is. Lack of attention to open sores and the delayed removal of metal particles embedded in the skin, further predispose to infection. There is generally very little risk of dermatitis or oil cancer when using either straight or soluble oils, provided that the operatives look after themselves. The author is not convinced that anti-dermatitis creams are either necessary or desirable, for their use encourages a lower degree of vigilance in personal hygiene. Incidentally, most cutting oils contain an antiseptic, such as carbolic acid, which affords a protection against possible bacterial growth within the oil. The sulphur incorporated in certain straight oils also acts as an antiseptic.

Wet versus Dry Cutting. With the advent of special tipped cutting tools, much machining can now be safely undertaken in the dry state; and on certain metals, magnesium alloy for example, cutting fluid is inadvisable. Apart from this notable exception, some kind of cutting fluid should always be used for the very good reason that tool life and component finish can be greatly improved when an oil is present. If finish is unimportant, then tool life is the deciding factor.

The author believes that the cutting fluids of to-morrow will differ greatly in composition from those of to-day. There is a tendency even now to employ strongly polar compounds, such as alcohol and paraffin, when machining at exceptional speeds and it is probable that a proportion of these materials will be introduced into cutting fluids for extremely severe service. Sulphur incorporation to increase tool "bite", improve oil adherence and increase oiliness, is a common practice to-day and it is likely that more work in this direction will be carried out, either by the addition of certain metallic oleates or other chemical substances. One development would be the stabilization of soluble oils when emulsified. Soluble oil is the most adaptable cutting fluid in use to-day, so the criticism that its emulsion is too susceptible to outside influences is doubly unfortunate. Cutting fluids perform an onerous duty in the machining of metals, and from all indications their function will become increasingly important with the years.

LUBRICATION OF WIRE ROPES FOR HOISTING IN MINES ON THE WITWATERSRAND GOLD FIELDS

By W. G. C. Nixon, B.Sc.,* and W. G. Jackson, B.Sc.†

Lubrication of wire ropes for winding purposes presents a somewhat different problem from ordinary lubrication. The problem would appear to be entirely different from that generally conceived under this heading, in that the possibility of adsorbed films and even of separation of the solid surfaces by a viscous film, is remote. The conditions under which the winding rope operates, i.e. bending, reversed bending, torsion and tension, introduce bearing conditions between wire and wire, and between strand and strand which would in all probability break down a film-forming tendency. This would result in sliding of bare surfaces not necessarily clean when considering the rope from the outer wires to the core.

For the protection of the crown wires of the rope from friction wear and the rope as a whole from corrosion, it becomes necessary to employ greasy "lubrication".

Object of the Investigation. The object was to determine :—

(1) The value of internal lubrication.
(2) The lubricant giving the most consistent results.
(3) The best method of application for particular conditions.
(4) The effect of speed.
(5) Effect of winding in dry or wet shafts on the choice of lubricant.
(6) Effect of upcast or downcast shaft winding.

For the purposes listed above data sheets were submitted to all the thirty-nine mines on the Witwatersrand with shaft depths varying from 720 to 6,306 feet and winding speeds from 1,000 to 4,000 ft. per min., calling for particulars under the following heads : name of mine ; name of shaft (vertical or incline) ; upcast or downcast ; wet or dry ; size and construction of rope ; maximum winding speed ; brand of dressing used ; frequency of application of dressing and approximate quantity applied per 1,000 feet of rope ; method of application and if hot or cold ; and reasons for using the brand adopted, e.g. quantity used for given foot tonnage, etc. Further data regarding the approximate

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* Consulting Mechanical Engineer, Johannesburg Consolidated Investment Company, Ltd.

† Technical Assistant, Consulting Mechanical Engineer's Dept., Johannesburg Consolidated Investment Company, Ltd.

analysis of the various dressings used was supplied by the several manufacturers.

(1). In order to make a reasonable survey of the whole position, it is necessary to review the manufacture of the rope and the conditions of its working life. In practically every instance, the main core and strand cores are found to be lubricated prior to laying up in the rope or strand respectively, and in some cases lubricant is applied during the laying up.

There would appear to be no uniformity of opinion as to the most effective type of lubricant. It is taken that the lubricant as a whole must not be acid or strongly alkaline, but beyond this there is no evidence to show that the essential internal lubrication is studied in conjunction with the particular conditions of winding.

The bedding down of the rope strands on the soft core, in ropes with a main hemp core, causes the core lubricant to be forced through the whole rope. In certain cases where the main core is purposely left large in the laying up, the amount of lubricant squeezed out is such that no external lubricant need be applied as such for some months. Thus at the first and initial bedding down of the rope on the core, good internal and external lubrication is assured. The loss of internal lubrication can probably be attributed to two main factors:—

(*a*) Loss by pressure during the working life of a rope owing to its passage over drum and sheave, and to rope elongation during loading and winding.

(*b*) Loss owing to the breaking down of the lubricating qualities of the dressing, or probably complete chemical change of the lubricant through the action of the corrosive elements encountered under winding conditions.

The first cause can be guarded against in manufacture as shown above.

Under (*b*) tests are required to find the type of lubricant which offers the greatest resistance and which will maintain its body and lubricating qualities when acted on by the known corrosive elements in the shaft. In this connexion it is not suggested that a particular type of lubricant will be required for each given set of winding conditions.

The classification could embrace the following points:—

(*a*) Material mined, taking cognizance of the corrosive nature of the various substances found.

(*b*) Point (*a*) in conjunction with the shaft conditions, i.e. wet or dry, upcast or downcast.

In most winding equipments, the normal operation is that of raising a heavy load with the conveyances in balance. The squeezing out of the lubricant will therefore practically always occur on the ascending rope and admission of corrosive elements to the centre of the rope will occur on the descending rope, except when lowering heavy loads of men or material.

External Lubrication. The function of the external lubricant is primarily to protect the rope against external corrosion and abrasive wear, with the possibility of penetration into the rope itself. The lubricants used may be roughly divided into two classes:—

(*a*) Those which aim at the total exclusion of corrosive elements, by including constituents which make the dressings as a whole repellent to water and vapours.

(*b*) Those which allow for absorption of moisture or vapours and include in the constituents means for neutralization of acid or alkali.

TABLE 1. ANALYSES OF VARIOUS DRESSINGS FOR WIRE ROPES

		A	B	C	D	E	F	G
Water	per cent	1·1	Trace	4·6	—	Trace	6·5	6·0
Lime soap	,,	1·8	2·5	10·2	—	—	—	—
Saponifiable oil	,,	3·1	13·6	—	—	—	9·5	5·0
Vaseline	,,	79·2	—	—	—	—	—	—
Zinc oxide	,,	15·4	—	—	—	—	—	—
Lime	,,	0·5	—	—	—	—	6·5	14·2
Free fatty acid	,,	0·2	—	—	—	—	—	—
Mineral oil	,,	—	73·4	68·9	60	68·1	47·5	54·8
Mineral ash	,,	—	13·6	11·8	—	Trace	—	—
Carbonaceous matter	,,	—	6·6	—	—	—	—	—
Resin oil	,,	—	4·1	—	—	—	—	—
Bitumen	,,	—	—	4·5	40	20·0	30·0	20·0
Acidity	,,	—	—	—	—	0·3	0·15	0·4
Fixed oils	,,	—	—	—	—	11·9	—	—
Total ash	,,	18·45	—	—	—	—	—	—

In class (*a*) occur the dressings which have a bitumen base, the lubricating quality being obtained by blending with mineral oils.

Class (*b*) covers a wider range of mixtures and the neutralizing medium may take the form of lime soap.

(2) The data sheets show that some sixteen brands of external dressings are used on the Witwatersrand. Approximate analyses of seven brands of dressings are given in Table 1.

It is interesting to note the wide ranges over which the essential components of a wire rope dressing can be varied, and still give the desired results.

Analysis of the duty with regard to shaft depth does not allow any definite statement to be made about the type of lubricant best suited to any particular set of winding conditions.

(3) The most favoured method of application is to run the rope through a box containing the lubricant hot or cold. In inclined shafts, the general method is to apply the dressing cold by hand.

(4) The effect of speed on the lubricant is most marked. The adhesive qualities must necessarily be greater, the higher the shaft speed. The tendency for the lubricant to be thrown off during the passage of the rope to and from the sheave and drum indicates that dressings which give good results at relatively slow speeds, do not remain on the rope to give the necessary protection when the rope is run at higher speeds.

(5) Under this heading the effect of varying temperature on the viscosity of the lubricant must be considered. No discrimination appears to be made in the choice of lubricant according to whether the ropes are running in wet or dry shafts.

It has been found necessary to change the grade of dressing in deeper shafts and to apply a more viscous compound, owing to the tendency to "run off" when subjected to temperature changes.

(6) When used in upcast or downcast shafts the same remarks apply as under (5).

Conclusion. The analysis of the data submitted gives evidence of such contradictory nature that no particular point of importance can be defined. The basis of the choice of wire rope lubricant would appear to be one of economy consistent with maintenance of adequate protection of the rope from corrosion. It is apparent that good internal lubrication is essential, and this is a matter for the rope manufacturer.

It would be a step in the right direction if each wire rope manufacturer gave a choice of external lubricants to be used so that external dressings which are deleterious to the internal lubricant, would not be applied.

The question of external lubrication should be investigated.

No data are to be found regarding:—

(1) The efficacy of the externally applied lubricant as a preservative of, or even to enhance the effect of, the initial internal lubricant.

(2) The amount and quality of lubricant required to ensure minimum slip between headgear sheave, and rope.

(3) The value of the lubricant as such to the exposed wires of the rope.

(4) The value of varying percentages of known lubricating constituents in the composition of the dressing.

While few consistent indications regarding the duty of the lubricant appear from the analysis, it can be stated that the desired results are achieved and the ropes are well protected from corrosion during their working life.

It is a rare occurrence on the Witwatersrand fields for a discarded rope to show signs of internal corrosion, in spite of the rather haphazard methods so far employed in lubrication, but it seems more than possible that a scientific investigation of the whole subject might lead to results which would give better rope lives.

It would also appear that manufacturers of ropes should be as much concerned with such an investigation as users.

LUBRICATION OF PIVOTS OF LIGHT MECHANISMS

By R. J. Parker *

The lubrication of light mechanisms of the clockwork type has received very little attention in the past, and practically no consideration has been given to the correct solution or maintenance of a lubricant at the frictional surfaces. This no doubt was because the quantities of lubricant involved are small and there was insufficient recompense for the cost of the close investigation required. During the last decade, however, considerable advance has been made in lubricants for light mechanisms, particularly in mineral oils.

For the proper lubrication of pivots the following points should be given consideration:—

(1) Design of the frictional elements.
(2) Replenishment of lubricant.
(3) Service conditions.
(4) Requirements of the lubricant.
(5) Earlier types of lubricant.
(6) Modern types of lubricant.
(7) Investigational data.

The term "light mechanism" includes apparatus of a delicate nature which needs very fine adjustment after final assembly, and in which the complicated moving parts are small. Consequently, very little room is available in which to apply and hold the lubricant. It is often impossible to provide oilways, and in most cases true film lubrication cannot be obtained. Frequently only boundary lubrication is achieved, so that the film of lubricant produced is only of temporary duration.

The possible combinations of moving elements are numerous, and the following are some of the prominent classes. The rotating spindle may pivot in a small cup bearing, or it may rotate in a cylindrical bearing, solid at one end to take thrust. The spindle may be stationary with the moving member rotating upon it. The bearing may be a hole in the main component member, or a surface, groove, or "V" upon which a knife-edge operates. In nearly all these cases the actual frictional surfaces are exceedingly small in area and frequently of smaller length than the diameter of the shaft carried.

Owing to the relative minuteness of the wearing surfaces to be lubricated and the general inability of the surrounding surfaces to

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* George Kent, Ltd., Luton.

provide a reservoir, the main oiling must be performed during the assembly of the apparatus. It is therefore of vital importance to select the correct lubricant to be used during initial assembly. Also it is most important that during this operation, complete lubrication be effected and excess oil removed. The opportunity should always be taken to oil the apparatus again when it is dismantled for repair or cleaning. Intermediate replenishment is necessary and usually can only be performed with a brush or feather, percolation being assisted by imparting motion to the elements in order to ensure the introduction of a film of lubricant between them.

The lubricant selected must meet the service exposure conditions under which the apparatus may have to work, in addition to being suitable for the working requirements of the apparatus itself. These service conditions will frequently affect the physical characteristics of the oil through exposure to atmospheric temperatures and dust polluted atmospheres.

The lubricant must be chemically inert and without corrosive action on the parts to which it is applied, or with which it is likely to make contact inadvertently. The physical nature of the lubricant must be such that it can give a continuous adherent and tenacious film strong enough to resist the pressures encountered without rupture. Mobility must be such that in conformity with the other properties the oil readily flows into position and "stays put" under operating pressures and temperatures. Obviously it is at times impossible to find a lubricant which will fulfil all the requirements under the working conditions, and a compromise has often to be resorted to.

In order to ascertain the most suitable lubricant for given conditions it is frequently advisable to carry out chemical and physical tests, as far as possible under conditions approaching the site conditions. Formerly the lubrication of light mechanisms was practically confined to the use of a few fixed oils of marine, animal, and vegetable origin. These have the great disadvantage that they combine with atmospheric oxygen and tend to dry up or become gummy. They also tend to become rancid and their characteristics change, with adverse effect on the oiliness and also corrosive action on the bearing metals. The conditions promoting gumming and rancidity are nearly always present, since the oil is rarely insulated from light, oxygen, and moisture.

The chief fixed oils used for this class of lubrication are sperm, neatsfoot, porpoise and dolphin jaw, hazel nut, and olive oil. Dolphin and porpoise jaw oils are the most suitable for light mechanisms, particularly as they possess the property of "oiliness" to a greater degree than mineral oils. It must, however, be remembered that dolphin and porpoise jaw oils tend to gum and oxidize.

During the last decade great progress has been made with the production of mineral oils for lubricating light mechanisms. A considerable number of these oils are available for light mechanism lubrication, and as they have widely varying characteristics, it is essential to know the properties of individual oils before applying them to any piece of apparatus. Practically the first mineral oil adopted for oiling clocks and similar mechanisms was solar oil produced from the residue of Baku petroleum, known as "Ostatki". Until quite recently mineral oils for light lubrication purposes were produced with low viscosity, and consequently did not prove successful where it was necessary for the lubricant to be retained in the bearings. Now, however, mineral oils are available with properties suitable for any light lubrication problem which may arise. Mineral oils are far superior to fixed oils in their resistance to oxidation and the development of organic acidity, stability, and non-corrosive action on the bearing parts. They are almost colourless and so do not present a visible film on bright metal parts and bearings which might be objectionable in appearance. Having low cold test values they are suitable for oiling mechanisms exposed to low temperatures, and the increase in fluidity with increase in temperature is not sufficiently great to cause spreading or flowing off from the bearing surfaces.

While all mineral oils are very much cheaper than the fixed oils, too much stress cannot be laid upon the point that it is essential to use only the best of the mineral oils. The following table gives the results of some tests carried out on two similar light mechanisms, No. 1 being lubricated with a cheap mineral oil and No. 2 with high-grade mineral oil.

Test temperature, deg. F.	Mechanism	Completed revolutions	Sticking faults	Number of times lubricant is replenished
65–85	No. 1	1,000,000	5	10
65–85	No. 2	1,000,000	0	1
135	No. 1	2,000,000	8	13
135	No. 2	2,000,000	3	3

Many gels which are employed as filtering media for purifying these mineral oils, are slightly soluble in or microscopically miscible with the warm oil. It is vitally important that these gels be completely removed from the oil before use, for their accumulation not only causes cloudiness when the oil chills in cold weather, but they deposit themselves in the holes and journals in which the spindles work and cause, at least, retardation of the movement if not final clogging.

In conclusion the writer would like to express his thanks to the technical staff of Messrs. Fletcher Miller, Ltd., of Dukinfield, for their co-operation in the preparation of the paper.

SOME PRACTICAL CONSIDERATIONS IN THE LUBRICATION OF ROLLER AND NEEDLE ROLLER BEARINGS

By C. G. H. Richardson, B.Sc., M.I.Mech.E. *

Theoretically, only line contact occurs between the rollers and the tracks in a roller bearing, but in actual practice there is a definite area of contact due to deformation of the material and in consequence a small amount of sliding friction is unavoidably present, and heat is generated in the bearing. Further, in most designs of cage type roller bearings the cage has a running clearance of only a few thousandths of an inch over the outside diameter of the inner ring. This feature ensures concentricity of the cage and eliminates vibration, but it introduces a source of sliding friction. There is a similar action between the rollers and the cage pockets, and also rubbing between the ends of the rollers and the shoulders of the raceways. In full type roller bearings (i.e. those in which the rollers are not separated from each other by a cage) and needle roller bearings, adjacent rollers touch and at the points of contact are revolving in opposite directions, thus again causing sliding friction.

The requirements of a roller bearing lubricant may be summarized as follows:—

(1) To minimize internal sliding friction in the bearing.
(2) To protect the highly polished surfaces from corrosion.
(3) To dissipate the heat generated by deformation due to the loads carried by the rollers and tracks.
(4) To act as a seal against the ingress of moisture or foreign matter of any kind.

It will be seen therefore that the principles involved in roller bearing lubrication differ fundamentally from those appertaining to plain journal bearings where the load is carried by an oil film, the maintenance of which under varying conditions of service is the main factor in the selection of the type of lubricant as well as the method by which it is to be applied. In a roller bearing, the high unit pressures obtained definitely preclude the maintenance of an unbroken oil film between the loaded portions of the bearing.

Either an oil or a grease may be used for the lubrication of roller bearings, the selection depending on the service conditions and the

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* General Manager, Ransome and Marles Bearing Company, Ltd., Newark on Trent.

design of the bearing housing. Oil is the ideal form of lubricant owing to the certainty with which it can be maintained in the working parts of the bearing, but, owing to the simpler retention designs required, grease is employed whenever possible. For needle roller bearings oil should always be used, except where the loads are negligible or the bearings are used for oscillating motion only. Needle roller bearings are always "crowded" with rollers and there is therefore very little space for grease to remain inside the bearings. Further, the rollers are long in comparison with their diameter and it is difficult to ensure that grease is always present in the middle of the bearing.

Grease Lubrication. Greases are, broadly speaking, homogeneous mixtures of soap and mineral lubricating oil, which vary considerably in consistency. By far the most common soaps used in grease manufacture are those having lime and soda bases, although aluminium soap is slowly finding a more general use and, to a lesser extent for special applications, lead soap; for the subject under review only lime- and soda-base greases are considered.

Lime-base greases are usually of a smooth, buttery consistency and have melting points in the region of 160–210 deg. F. They are not satisfactory lubricants where high temperatures and pressures are to be expected, since a water content of approximately 1–3 per cent is necessary to ensure a stable product, and a temperature sufficiently high to cause evaporation of the inherent moisture causes separation of oil and soap. Their use, therefore, is limited to comparatively light duties, particularly where extreme moisture conditions prevail, as the insoluble lime soap provides an effective seal against the ingress of water.

Soda soap greases are usually of a fibrous nature and may be divided into two classes: (1) those of high soap content (18–25 per cent) and (2) those of low soap content (10–20 per cent). They have a much higher melting point than lime-base greases (275–310 deg. F.), and are therefore suitable for high-temperature work. They are quite as good as lime-base greases for lighter duty and are therefore preferable where considerable variation in operating conditions is likely to be encountered, or where it is desirable to have only one grade of grease which will be suitable for applications of varying severity. In the presence of water they emulsify and therefore where water cannot be excluded they should be used only when any other form of lubricant is unsuitable. With many soda greases the emulsion formed by water acts as quite an efficient lubricant, but there is danger of rusting if the bearings are left standing for long periods. Soda soap greases having a low soap content give a comparatively high initial peak temperature due to the excessive churning caused by their tenacious nature, but they will withstand a fair amount of oil separation without hardening sufficiently to cause any trouble. Owing to their semi-fluid nature, they tend to flow back to the bottom of the housing and the bearing when the machine is stopped and thereby avoid permanent channelling such as is liable to occur with some of the stiffer forms of soda grease. Soda greases having a high soap content give a lower initial peak temperature, but only slight oil separation can be permitted or the grease will become too hard and will lose its lubricating properties.

Greases of different makes should never be mixed unless an assurance has been obtained from the manufacturers to the effect that no detrimental effect is likely to be encountered. It may be taken for granted, however, that lime- and soda-base greases should not be mixed, as the presence of lime in a soda-base grease not only reduces the melting point considerably but often causes total disintegration of the grease. The mixing of two lime-base greases or two soda-base greases is not usually so serious but, nevertheless, it is a practice which should not be adopted. Wherever possible the old grease should be thoroughly cleansed from the bearing before a different one is applied.

When designing roller bearing housings for grease lubrication it must be borne in mind that there is a tendency for the grease to be thrown away from the rotating parts of the bearing, and therefore the housings should be so designed that the lubricant will be held as far as possible against the rollers. At the same time, however, the space in the housing must be sufficient to allow packing with an adequate quantity of grease, otherwise evaporation of the oil content may reduce the volume of grease to such an extent that an adequate volume for efficient lubrication is not maintained.

In initially packing a bearing and housing with grease for a normal horizontal job, as much grease as possible should be applied to the interior of the roller bearing itself, and the lower third of the housing space of each side of the bearing should be filled with grease. In this way a mass of grease will normally be kept in contact with the rollers and small quantities will be continuously picked up and carried into the bearing itself, thus providing constant lubrication.

The requirements for a vertical application are somewhat different, as there is a constant tendency for the grease to fall to the bottom of the housing and a danger of the upper surfaces of the roller bearing becoming devoid of lubricant. For this reason, it is advisable to pack the top housing space practically full of grease initially, but the space below the bearing should be only one-third filled.

When starting up with new grease a considerable amount of churning and aeration takes place in the grease which must therefore occupy considerably more space than normal. If the housing has originally been packed too full of grease, an excessive peak temperature is caused and in extreme cases total separation of the oil content results, leaving a hard soap deposit which is detrimental to the free running of the bearing. With greases of low soap content there is nearly always a slight leakage of grease through the housing sealing device during this initial period of running, but this usually ceases after a few hours, when the grease has settled down to normal working consistency.

The frequency with which renewals or replenishments of grease are required depends entirely upon the operating conditions of the bearing and no hard and fast rule can be laid down. As a rough guide it may be taken that under normal conditions of working, additions should not be required at less than 500-hour intervals. It should, however, be remembered that it is preferable to add comparatively small quantities of grease at fairly frequent intervals rather than large quantities at widely spaced intervals, and this applies particularly to vertical applications on which provision should always be made for adding grease *above* the bearing. In some greases a shrinkage occurs almost immediately after starting and it is necessary to add grease very early, after which more normal replenishment at longer intervals is satisfactory.

The failure of grease lubrication, supposing a suitable grease has been selected, is generally due to one of the following conditions:—

(1) Escape of grease due to inefficient sealing of the housing.
(2) Evaporation of oil from the lubricant due to high operating temperatures.
(3) Too large a space in the housing, allowing the grease to be thrown clear of the rotating parts.
(4) The initial application of too much grease in a housing, resulting in oil separation.

Oil Lubrication. As mentioned above, oil is the ideal lubricant for roller and especially needle roller bearings, owing to the certainty with which an adequate supply can be maintained in the revolving parts of the bearing. For normal work, only pure mineral oils should be used, but in most applications of needle roller and roller bearings where oil lubrication is adopted, the oil is not provided solely for bearing lubrication but has also to lubricate other parts of the mechanism such as gears, etc., and in these instances the type of oil to be used is often dictated by the requirements of the other parts of the mechanism. It frequently happens that a compounded oil which may be eminently suitable for the other parts, cannot be adopted owing to its corrosive action on the bearings; hence the selection of the most suitable grade of oil is sometimes more difficult than the choice of a grease.

The desirability of consulting the bearing manufacturer or the oil supplier cannot be too strongly emphasized, as one of the most frequent causes of bearing failure is the use of an oil which has either oxidized or carbonized owing to the unsuitable duties it has been called upon to withstand.

For roller bearings only three grades of oil are required to cover all normal variations of load, speed, and temperature, and the lightest bodied oil which will cope with the conditions imposed should always be chosen, as this avoids unnecessary frictional losses. For speeds up to roughly 4,000 r.p.m. and temperatures not exceeding 140 deg. F. an oil having a Redwood viscosity of about 250 to 400 sec. at 140 deg. F. should be satisfactory. For slow speeds, but at temperatures from 140 to 250 deg. F. a cylinder oil having a viscosity of 150–170 sec. Redwood at 200 deg. F. is recommended. For high-speed applications such as textile spindles a light spindle oil is, of course, essential.

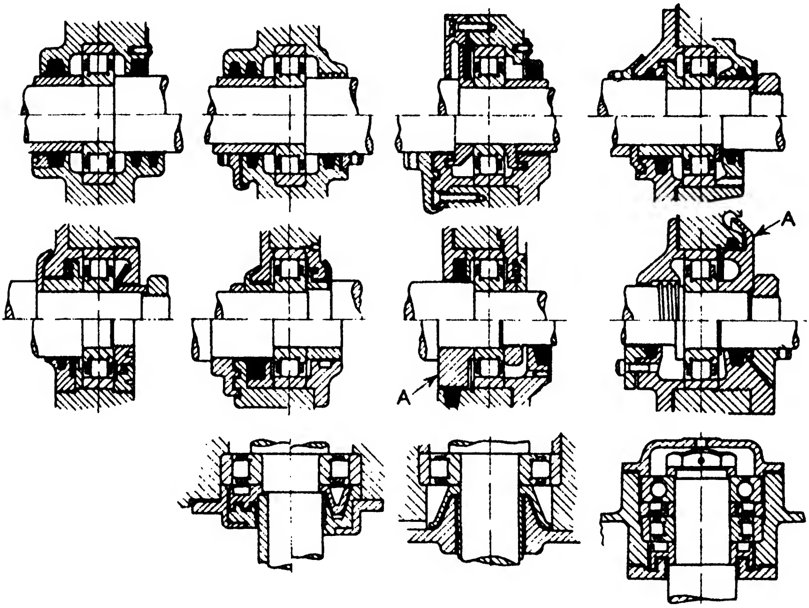

Fig. 1. Designs of Housings for Horizontal and Vertical Applications of Roller Bearings

A. This scheme should only be used when it is impossible to fit an end cap with felt pad.

There are many methods of oil lubrication, but the splash system is probably still the most common. In this system the bearing is lubricated by oil splashed into it by the other revolving parts of the mechanism, but this cannot be adopted for high-speed work because of the excessive churning which develops, leading to the production of high temperatures.

Where it is possible to isolate the bearing, an oil bath system is the most satisfactory, particularly if the oil is kept constantly as high as the centre of the bottom roller in the bearing. This may be achieved by fitting a drain at the requisite height and supplying new oil from a sight-feed or wick lubricator. The wick feed has the additional advantage of filtering the oil before passing it to the bearing.

On vertical applications a circulating oil system is common practice, as it enables the mechanism to run for fairly long periods without attention, but in such cases particular care must be exercised in choosing a suitable oil, as the churning which is inevitable with such a system imposes a very severe test on the stability of the oil, especially at high working temperatures. An unsuitable grade of oil will rapidly oxidize, leaving a gummy abrasive deposit on the bearing which eventually causes seizure or excessive wear. Circulatory oil systems are usually adopted where the temperatures encountered are only moderate, and relatively light oils may therefore be used. Heavy oils are unsuitable as they give rise to excessive internal fluid friction in the oil itself and so generate high operating temperatures.

Mention may be made of the extreme-pressure lubricants which are now being more widely used in automobile practice. Experience with these is not yet very extensive, but some of them cause corrosion of brass and bronze cages and care must be taken in their selection. There is an indication that they may have a beneficial effect on the life of rather heavily loaded bearings.

For very high-temperature work where needle roller bearings are used the most suitable lubricant is colloidal graphite. This is best applied to the bearing in admixture with a volatile oil, which is burnt off when subjected to the heat present and leaves a film of graphite deposited evenly over the rollers and tracks of the bearing. As a rule only the colloidal form of graphite is permissible for roller bearing lubrication as flake graphite almost invariably contains a slight amount of siliceous material which, being highly abrasive, is fatal to the bearing.

In special applications where roller bearings are called upon to operate submerged in water and where efficient glands are impracticable, soluble oils have been found to give satisfactory results even when as much as 60 per cent water is present in the oil, but instances of this type are comparatively rare and it is advisable in such a case to consult the bearing manufacturer because some alternative will usually be found possible (e.g. the use of stainless steel bearings).

The danger of using a soluble oil and water emulsion lies in the possibility that the emulsion may become unstable if the machine is left idle for lengthy periods, so that corrosion of the bearing parts is liable to take place.

Owing to the varied factors influencing the selection and application of a suitable oil, no arbitrary rules can be laid down while, further, no simple laboratory tests have yet been devised which will accurately forecast the behaviour of an oil or grease under actual working conditions.

Since the design of the bearing housing plays a very vital part in the successful lubrication of roller bearings a few designs for both horizontal and vertical applications are reproduced (Fig. 1).

LUBRICATION IN RELATION TO PIVOTS AND JEWELS IN ELECTRICITY METERS

By G. F. Shotter *

The study of the correct lubrication of the lower or footstep bearing of an electricity meter presents many problems which are rarely encountered in normal lubrication practice. Usually, this lower bearing consists of a hardened ball-ended steel pivot which rotates on a highly polished sapphire jewel cup. The pivot radius varies from about 0·015 to 0·035 inch and the jewel radius from about 0·04 to 0·075 inch.

Other arrangements and materials are used, such as a steel ball running between two sapphire cups. Usually, where the weight upon the bearing is high, diamond is used instead of sapphire for the cup. Various pivot materials have also been tried and in some cases used.

Most meters have bearings of the steel-sapphire type, but the fundamental problem facing the designer remains the same whatever the arrangement or materials used. This problem concerns the extremely high pressures which exist in bearings of this type, despite the fact that the weights imposed upon them are very small. These weights range between 12 and 60 grammes approximately. As an illustration of the actual value of the stress, the initial pressure exerted on a steel pivot, having a radius of 0·025 inch and with a weight of 13 grammes imposed on it, is in the neighbourhood of 75 tons per sq. in. The difficulties encountered in finding an ideal form of lubrication under such conditions can now be more readily appreciated.

The first point to be considered is the necessity for lubrication. The major factor which has led meter engineers to seek a suitable lubricant has been the production of ferric oxide in the jewel cup, a phenomenon directly associated with the above-mentioned high pressures. This oxide, being highly abrasive, produces an increase in friction which slows up the meter, particularly at low loads, and thus may cause a considerable loss in revenue to the supply undertaking. Generally speaking, the amount of the oxide produced is a function of the weight, number of revolutions, and the optical axis of the jewel. Detailed surveys of the production of this oxide have been published.†

Experience over many years has proved that the length of life obtained

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* North Metropolitan Electric Power Supply Company.

† Collected Researches of the National Physical Laboratory, 1931, vol. 24, p 1; Shotter, G. F., Jl. I.E.E., 1934, vol. 75, p. 755.

when this type of bearing is lubricated is far longer than that obtained when the bearing is dry, the improvement produced by the use of oil over the dry condition having been found to give a ratio as high as 400 to 1 for the same increase in friction.

An important fact which emerges from an examination of dry and lubricated bearings is that a totally different type of wear is produced in each case. When the debris consisting of ferric oxide and sapphire particles is removed from the dry bearing, the surfaces of the pivot and jewel cup are found to be rough, the friction still being fairly high. In a lubricated bearing, while rust or ferric oxide is still found, it is dispersed throughout the oil providing that this is sufficiently viscous. Experience has shown that a petroleum base oil which has been widely used in the past assumes the characteristics of vaseline after a few years. The grease holding the debris in suspension is found just off the running circle of the pivot around the wall of the jewel cup, which then has a rusty greasy centre. As might be expected, the surfaces of the pivot and jewel are polished and after the unit has been cleaned the friction is very small, approximating to that of a new unit. Even before cleaning the friction is very low, particularly where a large quantity of oil or grease is present.

The above-mentioned oil has a viscosity of 1,000 sec. Redwood at 60 deg. F. and one of the objections to its use is that under certain conditions, in which wear is rapid, its viscosity is not sufficient to hold the debris formed in suspension. The debris thus collects in the bottom of the cup, causing an excessive increase in friction, particularly at low speeds, a condition similar to that of the dry bearing.

Some controversy has taken place about the formation of a film between the bearing surfaces. Doubt has been expressed whether a film could form under such high pressures. The question is further complicated by the fact that in friction tests the total friction includes a fluid drag which tends to mask any decrease in friction that might be expected to occur due to the oil film. The author has always held the opinion that filming, or at least partial filming, does take place, particularly during rotation.

Very briefly the foregoing sums up the situation as it existed at the time when the British Electrical and Allied Industries Research Association began general research on the problem of these bearings and invited the author through his company to carry out the investigations. One of the sections of this research concerned lubrication. Bearing in mind the experience already outlined, the author's object was to find an oil which had the characteristics that he felt would satisfactorily fulfil the conditions. The following specification was drawn up for such a lubricant, taking into consideration the fact that a bearing of this type usually has to remain untouched in service for periods ranging between five and ten years:—

(1) The oil should have a high film-forming factor.
(2) It should not be prone to oxidation at normal temperatures.
(3) It should be stable.
(4) It should not solidify at freezing point.
(5) It should have such a viscosity as to hold any debris formed in suspension after dispersion by the movement of the pivot.

From the outline of the problem previously given, the reason for the first condition is obvious. In regard to the second point, it is again obvious that an oil prone to oxidation could cause corrosion in the mounting metals, the degree of corrosion depending upon the type of metal used for the mount.

An important point is that highly refined oils frequently develop organic acids by oxidation more rapidly than other oils of a lower degree of refinement. It follows that the best results are not necessarily obtained from the use of white oils such as medicinal paraffin. The author's experience has proved that the use of a grease of high viscosity is not perfectly satisfactory as it may tend to trap air and moisture and cause oxidation by the normal process, quite apart from the fact that the fluid drag may become excessive. The objection to a lubricant of low viscosity has already been explained.

Stability, the third point in the specification, is related to the first in that the oil must retain its viscosity and its other characteristics. Again, it is also necessary that the viscosity of the oil should not increase beyond a certain limit at low temperature. The reason for the last point in the specification is self-evident in view of the previous description of the phenomena associated with wear.

The author held a consultation with a certain lubricant manufacturer, and after a discussion on the above characteristics, a Pennsylvanian oil was put forward as being a suitable lubricant.

The characteristics of the oil were given as follows:—

(1) It has the highest film-forming factor of any oil.
(2) It is extremely difficult to oxidize the oil at normal temperatures.
(3) It may be obtained with various viscosities.
(4) With its highest viscosity, it does not solidify at freezing point.

This oil is refined by the "Alchlor" process, involving the use of aluminium chloride instead of sulphuric acid or solvents; and it was stated that this special process removes all the unstable substances without impairing the lubricating properties of the oil. Tests were then carried out to find the initial friction on new pivots and jewels lubricated with two grades of this oil, and another meter oil known as "Sangamoil," comparing the results obtained with those obtained with the dry bearing.

"Sangamoil" is a completely refined petroleum hydrocarbon oil, colourless, odourless, and tasteless, corresponding in all essential characteristics to the normal B.P. medicinal paraffin.

Space does not permit a detailed description of the apparatus used, but it is sufficient to state that an electrical method was adopted and it was possible to measure the friction down to 0·25 dyne-cm. This is equivalent to one-thousandth part of the torque of an electricity meter at its lowest tested load, i.e. one-twentieth load. The method permitted both the static (or starting) friction and the running friction to be measured. Table 1 gives the mean results obtained in the tests. The pivot radius was 0·025 inch and the jewel radius 0·0725 inch in every case. Four different weights were used and the bearings were tested with the jewel cups in both smeared and flooded conditions.

TABLE 1. MEAN FRICTION VALUES OBTAINED WITH A STEEL PIVOT, A SAPPHIRE JEWEL, AND VARIOUS LUBRICANTS

	Mean friction, dyne-cm.						
Lubricant	—	Sangamoil; viscosity approx. 600 sec. Redwood		Pennsylvania Meter Oil No. 1; viscosity, approx. 1,400 sec. Redwood		Pennsylvania Meter Oil No. 2; viscosity, approx. 6,600 sec. Redwood	
Lubrication	Dry	Smeared	Flooded	Smeared	Flooded	Smeared	Flooded
Weights used, grammes	*a*	*b*	*c*	*d*	*e*	*f*	*g*
17	1·33	1·4	1·55	1·2	1·2	1·25	1·45
32	2·85	3·1	3·6	2·0	2·7	2·45	2·6
47	4·3	4·3	5·0	3·3	4·0	3·85	4·0
62	6·15	6·5	7·0	5·3	5·2	5·3	5·9

The chief point revealed by these tests is that the Pennsylvanian oil, except in one case, decreases the total friction of the bearing compared with the dry condition, in spite of the fact that added "friction" in the form of a fluid drag has been included. Therefore there appears very little doubt but that an oil film exists between the bearing surfaces despite the high pressure between them. It also seems quite possible that this film exists even with oils of the paraffin type, although not to the same extent as with the special Pennsylvanian oil.

The case with the 17-gramme weight, col. *g*, is the only exception to the frictional decrease as against the dry condition with Pennsylvanian oil. This might be expected, as the fluid drag, which is constant for any given viscosity, flooding, and pivot and jewel dimensions, forms a larger proportion of the total friction when a light weight is used.

Life tests carried out between similar units have shown the marked superiority of the Pennsylvanian oil over the paraffin type. Work on the problem is continuing and though it has not yet been completely solved, it can be said that satisfactory progress has been made towards finding a lubricant which will give the minimum change in friction during the life of an electricity meter bearing.

SOME EXPERIMENTS ON THE LUBRICATION OF PIVOT AND JEWEL BEARINGS

By Verney Stott, B.A.*

Whilst no comprehensive investigation on the lubrication of pivot and jewel bearings has been carried out at the National Physical Laboratory, some results obtained in the course of a more general investigation have a direct bearing on the problem. In the course of this investigation an apparatus † was made for carrying out wear tests on pivots and jewels. During a wear test the pivot rotates in a vertical spindle and the jewel, mounted at the centre of a disk which it supports on the pivot, remains stationary. The wear test can be interrupted at intervals and the torque due to friction between the pivot and jewel measured. The jewels used were of the cupped type employed in electricity meters and compasses.

Some years ago there was considerable divergence of opinion as to whether it was or was not advantageous to lubricate the bottom bearing of electricity meters. In view of this it was decided to carry out comparative wear tests with lubricated and non-lubricated steel pivots and sapphire bearings. Two pivots were used, both made from hardened Stubs's steel wire, 2 mm. in diameter, finished to a 60 deg. cone terminating at the apex in a portion of a sphere of 0·8 mm. diameter. Two jewels of synthetic sapphire, one (No. 11) cut so that the axis of the jewel made an angle of 71 deg. with the optical axis of the sapphire and the other (No. 12) having this angle equal to 39 deg., were used. Subsequent work has shown that this difference in orientation is not likely to have influenced seriously the results obtained. The radius of curvature of the jewel cups was 1·7 mm. The load on the pivots was 22 grammes during the wear test and 20 grammes when the frictional torque was measured; the speed of rotation of the pivots during the wear tests was approximately 150 r.p.m.

One pivot, No. 108, was run in contact with the jewel No. 11 without any lubrication. The torque due to friction was measured about every 50,000 revolutions until the pivot had made 1,000,000 revolutions, when the test was stopped as the friction had become excessive.

The second pivot, No. 112, was run in contact with the second jewel, No. 12. The friction was first measured with both pivot and jewel dry and then after filling the jewel cup with "superfine clock oil" and replacing it in its inverted position over the pivot. When both were

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* National Physical Laboratory.

† National Physical Laboratory, Collected Researches, vol. 24, paper 1.

dry the frictional torque was 1·4 dyne-cm. and when oiled 2·2 dyne-cm. The pivot was then allowed to make 50,000 revolutions, when the jewel cup was again filled with oil and the pivot rotated for a second 50,000 revolutions. This procedure was repeated up to a total of 250,000 revolutions, after which the pivot was allowed to make 100,000 revolutions between oilings up to a total of 1,050,000 revolutions. Throughout all this period the torque due to friction varied by only a few tenths of a dyne-centimetre. The test was continued beyond 1,050,000 revolutions but with the difference that, although the friction was measured after every 100,000 revolutions, no further oil was placed on the bearing. After the first million revolutions the pivot and jewel simply ran on their reserve of oil.

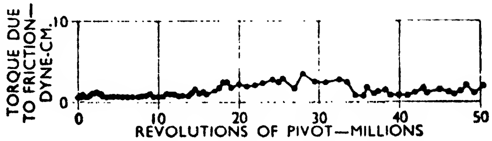
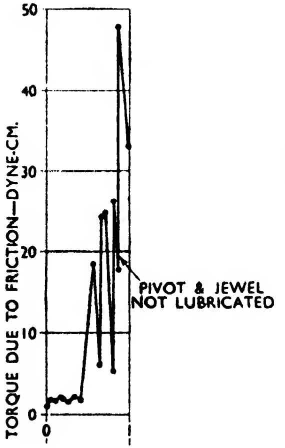


Fig. 2. Friction between a Steel Pivot and a Diamond Jewel during a Wear Test lasting 3½ Years



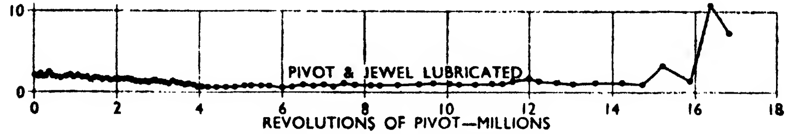


Fig. 1. Results of Friction Measurements during Wear Tests on Steel Pivots and Sapphire Jewels

The results of the friction measurements are shown in Fig. 1. For the pivot and jewel which were run without lubrication, the friction remained low (about 2 dyne-cm.) for the first half-million revolutions and then began to increase rapidly, rising to nearly 50 dyne-cm. before 1,000,000 revolutions had been completed.

For the pivot and jewel which were lubricated the initial friction was practically the same as for the dry pivot and jewel. The friction, however, maintained this low value, not only during the first million revolutions, when the oil was replenished at intervals, but also for a long time after the final oiling at 1,050,000 revolutions. The friction remained at about 2 dyne-cm. over a period of six months, during which time the pivot made a total of 15,000,000 revolutions. The friction then began to rise, and by the time 16,500,000 revolutions had been completed had reached 10 dyne-cm. The rate of increase when the friction finally began to rise was much less than the rate of increase with the dry pivot and jewel. The contrast between the two sets of results is very marked. The presence of oil, though it did not materially alter the initial friction, gave a useful life for the second pivot and jewel thirty times as great as that for the pivot and jewel which were run dry.

The pivots and jewels were examined at intervals under the microscope during the test. With the pivot and jewel run dry copious rust was formed, as invariably occurred in all experiments when steel pivots and sapphire jewels were run without lubrication. The sequence of events is first that a small patch showing brilliant interference colours appears at the centre of the pivot. This patch increases in area and at its edge a ring of small rounded nodules, bright brownish-red in colour, appears. As wear proceeds these nodules coalesce into patches of irregular shape interspersed with isolated nodules of varying size and isolated particles are scattered over the surface of the pivot well away from the area of contact. The centre of the jewel cup remains comparatively free from rust throughout but, as wear proceeds, this central area is bounded by a ring of heavy rust deposit and beyond this, rust particles are scattered over the whole surface of the jewel.

At the end of 1,000,000 revolutions a very pronounced flat had been worn on the pivot run dry, but with the lubricated pivot and jewel no change in the profile of the pivot could be detected at the end of 1,000,000 revolutions. Though rust finally developed on the pivot and jewel which were lubricated, it made its appearance much later. After 3,500,000 revolutions, for example, there was very little evidence of rust.

To summarize, the presence of oil did not materially influence the magnitude of the initial friction, though it did enable the initial low friction to be maintained very much longer than in the absence of lubrication. A possible explanation of this beneficial effect of lubrication is that the presence of oil may reduce the amount of rust formed, and may also enable any rust or debris formed to become distributed throughout the oil film, thus preventing the rapid increase in friction which occurs in a dry bearing when the rust formed remains near the area of contact of pivot and jewel.

Corroboration of the effects described was obtained by experiments on electricity meters carried out in conjunction with the electricity department. Two meters initially running on the fast side, were running slow after 5,000,000 revolutions at full-load speed without lubrication. The meters had steel pivots and sapphire jewels, and rust developed precisely as already described. The pivot and jewel bearings of the meters were then thoroughly cleaned and oiled and their performance again tested. After making 5,000,000 revolutions at full-load speed they were again tested and their performance was practically unchanged.*

Another experiment was carried out with sapphire pivots and sapphire jewels. A sapphire pivot and a sapphire jewel were subjected to a short wear test without lubrication under the same conditions as those previously described for the wear tests on steel pivots and sapphire jewels. At the end of this test, the jewel being still in good condition, it was used in a second short wear test with another similar sapphire pivot under the same conditions as before, except that a drop of clock oil was placed on the pivot and the jewel cup filled with oil before starting the run. The sapphire pivots used had a radius of curvature of 0·3 mm. The jewel had a radius of curvature of 1·24 mm. and its axis made an angle of 65 deg. with the optical axis. The orientation of the pivots was not determined. The results obtained in the wear tests are given in Table 1.

A comparison of the second and fourth columns of Table 1 shows that the increase in friction during the first 2,000,000 revolutions was slightly less in the presence of oil. The difference between the dry and oiled sapphire pivot and sapphire jewel combinations is, however, of an entirely different order from the large differences shown in Fig. 1, for steel pivots and sapphire jewels when dry and when oiled.

The last experiment of which details will be given was one which extended over 3½ years. A diamond jewel (No. 88) was used, having a radius of curvature of 2·75 mm. A Hoffmann hardened steel ball, $\frac{1}{32}$ inch (0·8 mm.) in diameter, mounted on a brass wire, was used as a pivot (No. 446). During the wear tests, which were carried out without lubrication, the pivot rotated at approximately 100 r.p.m. and the load on the pivot was 25 grammes. The torque due to friction was measured at intervals, the load on the pivot during the friction measurements being 20 grammes. The results obtained in the friction measurements are plotted in Fig. 2. Throughout the whole period of 3½ years during which the wear test continued, the pivot remaining stationary for considerable periods during this time, but making a total of just over

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* Detailed results are given in The National Physical Laboratory Collected Researches, vol. 24, paper 1, p. 43.

TABLE 1. RESULTS OF FRICTION MEASUREMENTS DURING WEAR TESTS ON A SAPPHIRE JEWEL AND SAPPHIRE PIVOTS

Sapphire jewel No. 62; sapphire pivot No. 93; no lubrication		Sapphire jewel No. 62; sapphire pivot No. 94; lubricated	
Number of revolutions	Torque due to friction, dyne-cm.	Number of revolutions	Torque due to friction, dyne cm.
0	1·1	0 without oil	$0{\cdot}8_5$
286,173	1·3	0 with oil	$0{\cdot}9_5$
500,001	1·2	250,000	0·9
750,000	1·5	382,000	0·9
1,012,000	1·8	605,258	1·0
1,262,000	1·4	887,258	0·9
1,412,000	1·4	1,137,258	1·1
1,715,000	2·0	1,382,000	1·1
2,000,000	1·9	1,787,000	1·3
2,000,000 (after cleaning)	1·3	2,000,000	1·3

50,000,000 revolutions, the torque due to friction remained very small, i.e. within the range approximately of 2 dyne-cm.±1 dyne-cm.

The profile of the pivot was practically unchanged at the end of the run of 50,000,000 revolutions as may be seen in Fig. 3, in which the small projections are debris on the pivot. For comparison, the profile of an initially identical steel pivot (No. 355) with a heavy deposit of rust, formed after only 2,000,000 revolutions without lubrication in

Fig. 3. Profile of Steel Pivot (No. 446) after 50,000,000 Revolutions in Contact with Diamond Jewel (No. 88)

contact with a badly orientated sapphire jewel (No. 37), is shown in Fig. 4.

At the end of the run with the diamond jewel there was a ring of debris, of about 0·33 mm. external diameter and 0·25 mm. internal diameter, on the pivot. The debris consisted mainly of isolated nodules less than 0·01 mm. in diameter and of a pale fawn colour. Inside this ring the pivot was free from debris but exhibited interference colours. There was very little debris outside the ring. The debris was entirely different in appearance from the copious rust deposit obtained with a steel pivot and sapphire jewel when run dry.

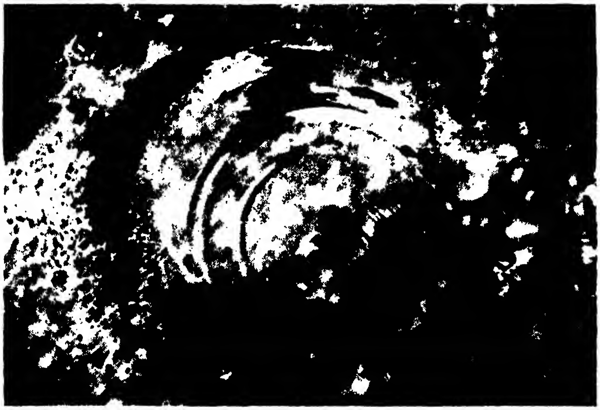
Fig. 4. Profile of Steel Pivot (No. 355) after 2,000,000 Revolutions in Contact with a Badly Orientated Sapphire Jewel (No. 37)

Fig. 5. Surface of Diamond Jewel (No. 88) after 50,000,000 Revolutions of Steel Pivot (No. 446) in Contact with Jewel

On the surface of the diamond jewel at the end of the run there was a central area about 0·3 mm. in diameter practically free from debris.

Outside this area and extending to a diameter of about 0·5 mm. was an area of grey deposit interspersed with clear jewel surface and exhibiting interference colours in places. A photograph of the jewel surface is reproduced in Fig. 5. There is no sign of damage to the jewel surface. For comparison, the surface of the badly orientated sapphire jewel (No. 37) after only 2,000,000 revolutions with the steel pivot (No. 355) is shown in Fig. 6. Here the dark areas are heavy deposits of rust and there is obvious damage to the jewel surface.

No experiments with lubricated steel pivots and diamond jewels were carried out, the excellent performance without lubrication indicating this to be unnecessary.



× 100 dia.

Fig. 6. Surface of Badly Orientated Sapphire Jewel (No. 37) after 2,000,000 Revolutions of Steel Pivot (No. 355) in Contact with Jewel

Conclusions. Wear tests carried out at the National Physical Laboratory have shown that:—

(1) With a steel pivot and sapphire jewel, lubrication had markedly beneficial effects.

(2) With a sapphire pivot and sapphire jewel, over a short run of only 2,000,000 revolutions, there was little difference between the increase in friction when the combination was lubricated and when it was not.

(3) With a steel pivot and diamond jewel, friction remained small over a long period (3½ years) and for a large number of revolutions (50,000,000) with no lubrication.

The results show clearly that lubrication must be studied separately for each combination of materials used for pivots and jewels. In a pivot and jewel bearing of the type dealt with, the function of lubrication is not to reduce the initial friction but to maintain a low value for the friction over a long period. Lubrication is only one factor, though for some combinations of materials an important one, in obtaining efficient service from pivot and jewel bearings. The problem as a whole involves many other factors, e.g. choice of materials and shapes for pivots and jewels, the kind of debris produced and the mechanism of its production, the direction of cutting jewels relative to the crystal structure, the structure of the material used for the pivot, the physical properties of the polished surfaces of pivots and jewels, and the effect of the atmosphere.

LUBRICATION IN WIRE DRAWING

By Professor F. C. Thompson, D.Met., M.Sc. *

Although the calculation of the exact distribution of stress on the various portions of a wire-drawing die is most difficult, there can be no doubt that this stress may attain very high values, probably of the order of 100 tons per sq. in. in heavy reductions of high tensile steels. The necessity for adequate lubrication is, therefore, at once apparent. Unsatisfactory lubrication leads to wastage of power, to excessive heating of the die, with consequent risk of fracture in carbide dies owing to the thermal stresses set up, and probably to unsatisfactory wire.

With regard to the nature of the film of lubricant between the metal being drawn down and the die, the experimentally established facts are as follows:—

(1) The pull required (Fig. 1†) to effect a given reduction under standard conditions is independent of the speed of drawing, at any rate over the range from about 20 to 600 ft. per min., which latter is the highest speed the author has employed. (It should be pointed out, however, that at very low speeds the same conditions do not obtain and the pull required increases as the speed of drawing is raised. It is not, however, intended to deal with these conditions here.) The independence of the pull and, therefore, presumably the friction, and the speed of drawing is a result which is inconsistent with the idea that the lubrication is of the "fluid" type, but would be expected with a "boundary" film.

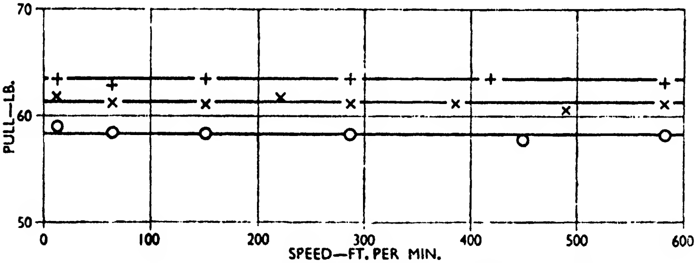


Fig. 1. Pull Required with Three Different Soaps

Mild steel, 5 deg. die, 17 per cent reduction of area.

+ Brown soap. × Special hard soda soap. ○ Soap number 3,535.

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* Department of Metallurgy, University of Manchester.

† Figs. 1, 3, 5 are reproduced by permission of the Iron and Steel Institute from Carnegie Scholarship Memoirs, 1932, pp. 23, 16, and 26 respectively.

(2) The addition of small percentages of fatty acids to mineral oils markedly reduces the coefficient of friction under boundary conditions. That the same holds under the high pressures which obtain in wire drawing will be seen from the following results. In tests on the pull required to reduce a mild steel wire by 20 per cent through a tungsten carbide die of 5 deg. angle, the pull with a certain mineral oil amounted to 89 lb. The addition of a small amount of stearic acid brought about a reduction in the pull by about 8 per cent. This result is consistent with the view that the lubrication is of the boundary type.

(3) When the tension required to draw a given wire under standard conditions is plotted against the reduction of area effected the curve obtained (Fig. 2*) is a straight line. For ferrous metals this appears to extend up to about 30 per cent reduction, after which the pull necessary increases less rapidly, but for non-ferrous materials the curve remains strictly linear at any rate up to a reduction of 50 or 60 per cent. When the conditions of lubrication are altered, all other factors remaining constant, a similar straight line relationship is obtained. The new curve is strictly parallel to the former but is displaced vertically according as the lubricant employed is better or worse. It thus appears that the intercept made by such curves on the stress axis is directly related to that part of the pull required to overcome the friction between the metal and the die.

It can readily be shown that the area of the surface of contact between the metal and the die is proportional to the reduction of area. If it be assumed that, as is probable, the total pressure normal to the surface of the die is proportional to the tension imposed on the wire, which latter in turn has been shown to be directly proportional to the reduction of area, it follows that the total pressure itself is also proportional to the latter value. According to Amonton's formula for boundary lubrication, the friction is proportional to the product of the coefficient of friction and the total pressure. If the total pressure increases with the reduction of area, so, therefore, should the friction, and the fact that the curves in Fig. 2 are parallel throughout their length is difficult to explain. The same difficulty, however, would still obtain if the friction were of the fluid type, when the frictional loss would again be directly proportional to the area of contact. Comparatively little is known about lubrication under the extremely high pressures which may obtain in the drawing of wire, and in particular the manner in which the coefficients of viscosity and friction vary with the pressure, and it is uncertain how far the relationships which exist at "ordinary" pressures will hold under pressures such as are now being considered.

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* Reproduced by permission of the Institute of Metals (Jl. Inst. Metals, 1931, vol. 46, No. 2, p. 321).

(4) Under conditions of boundary lubrication, there is very strong evidence that an adsorbed film of the lubricant is anchored to the metallic surface. The film of soap left on a wire, after use as a lubricant in the drawing, is so adherent that prolonged rubbing with a linen rag will not remove it.

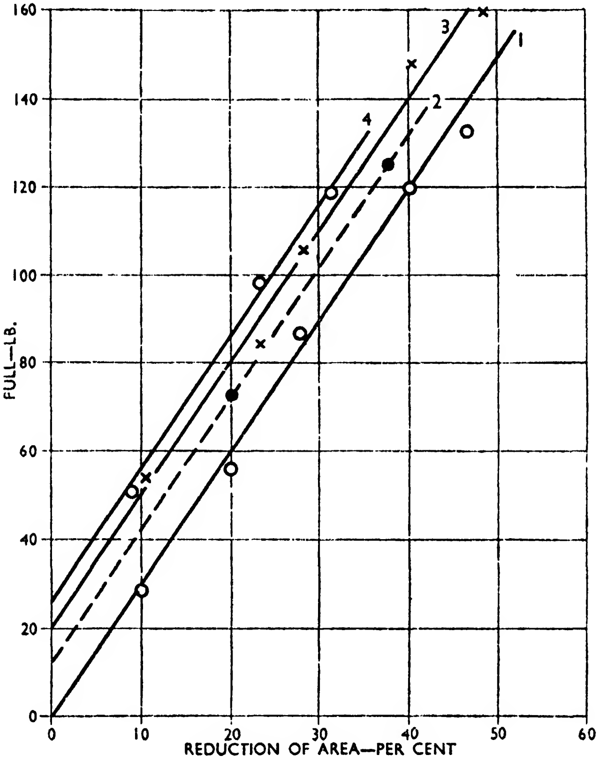

Fig. 2. Tension Required to Draw a Wire (7 per cent Nickel-Silver) compared with the Reduction of Area Obtained

1. Annealed: soap.
2. Annealed: radial taper die.
3. Annealed: "Germ" oil.
4. Cold-drawn: "Germ" oil.

On the whole, then, although the evidence is not completely satisfactory, it would appear strongly that only on the assumption that the lubrication is of the boundary type can these results be explained, and this would be expected in view of the probable difficulty of thick films of lubricant withstanding the extremely high pressures encountered.

It is possible, however, to go some way further in demonstrating that this conclusion is probably correct. In Fig. 3 are collected some results of tests made on a friction testing machine* loaned by Mr. J. E. Southcombe. Against the actual pull required to reduce a mild steel by some 20 per cent when various lubricants were employed, are plotted the figures obtained for the same lubricants on the friction testing machine. The latter values are proportional to the coefficients of boundary friction, and the general proportionality of the two sets of determinations is evident. One may, therefore, conclude that the lubrication in wire drawing is probably of the boundary type.

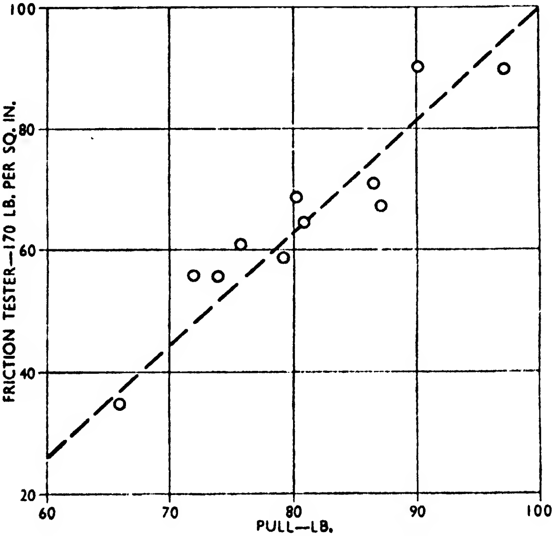

Fig. 3. Pull Required to Reduce a Mild Steel by 20 per cent Using Various Lubricants, Compared with the Coefficient of Friction of those Lubricants

Since considerable heat is produced in the die, some consideration of the effect of temperature upon lubrication is desirable. It appears to be the general opinion that any appreciable rise of temperature is harmful, though on what grounds this view has been reached, and whether it is the wire or the die which is supposed to suffer, is by no means clear. Elaborate precautions are taken, however, especially in continuous drawing, to ensure that the heat is dissipated as rapidly as possible, either by cooling the circulating lubricant itself or by means of

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* Designed by J. H. Wells.

water-cooled dies. At any rate with carbide dies, which are used for the drawing of tungsten wire at 700 deg. C., there is no obvious reason why the relatively small rise of temperature to be expected in ordinary drawing should exert any serious effect. It must be borne in mind, however, that marked *fluctuations* of temperature will set up thermal stresses in such dies which may result in fracture.

With regard to the actual temperatures attained in wire drawing much uncertainty still persists. It is well known, however, that temperatures may at times be attained in the drawing of steel wire sufficiently high to yield definite oxidation tints on the surface. With really efficient lubrication and normal reductions of area, the temperature probably does not appreciably exceed about 200 deg. C. Apart from the influence of this rise of temperature upon the life of the die, the two factors which have to be considered are: (1) the influence on the lubricant itself, and (2) any possible effect produced on the mechanical properties of the wire. Dealing with the latter first, it has been shown that drawing brass or steel wire at temperatures of the order of 170 deg. C. gives a material the mechanical properties of which are essentially identical with those obtained when the drawing is done at room temperatures. In Table 1 two sets of such determinations with castor oil as the lubricant are recorded.

TABLE 1. EFFECT OF TEMPERATURE IN DRAWING ON THE PROPERTIES OF BRASS AND STEEL WIRE

Material	Temperature, deg. C.	Maximum stress, tons per sq. in.	Elongation on 2 inches, per cent	Reduction of area at fracture, per cent	Torsions on 100 × diam.
70/30 brass .	18	35·2	16·0	74·7	73
	165	35·3	15·5	74·2	75
0·5 per cent carbon steel	20	60·7	15·0	55·0	33
	175	60·8	14·5	56·0	32

Passing on to consider the effect of the temperature rise upon the lubricant, Fig. 4 shows that when soap is used breakdown does not occur until a temperature of about 175 deg. C. is reached. Up to that temperature the drawing had proceeded smoothly and with progressively lower loads as the temperature was raised. Above 175 deg. C., however, the tension required suddenly increased to an appreciably higher value and the drawing proceeded irregularly in a series of violent jerks, the wire appearing to stick periodically in the die.

Experiments with liquid lubricants show that as the temperature is increased the pull required is progressively diminished. Some of the curves (Fig. 5) show a curious and so far unexplained dip in the neighbourhood of 60 deg. C. One curve of special interest is that for water, which is, at a temperature just below 100 deg. C., a far from inefficient lubricant. A comparison of the relative efficiencies of the soap, a particularly good one, and the liquid lubricants shows that, at room temperatures, the former is much the more satisfactory. As the temperature is raised, however, the position may be reversed, and at temperatures from 80 deg. C., upwards to 140 deg. C., arachis oil, for instance, is a distinctly superior lubricant.

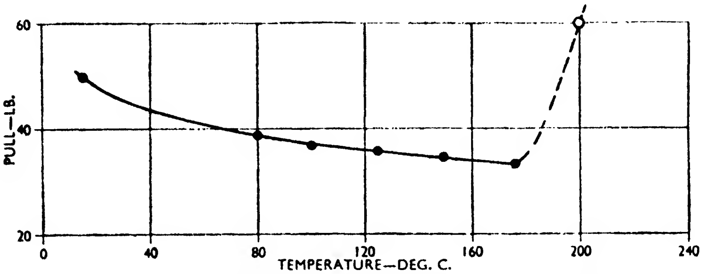

Fig. 4. Effect of Temperature on Soap Used as Lubricant

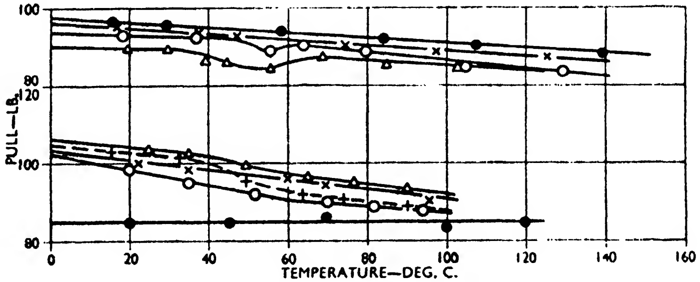

Fig. 5. Effect of Temperature on Tension when Liquid Lubricants are employed

Upper curves:
- ● Castor oil
- × Rape oil.
- ○ Arachis oil.
- △ Stearic acid.

Lower curves:
- △ Sodium oleate (15 per cent solution).
- + Water.
- ○ "Germ R.O.D." paste (15 per cent solution).
- ● Dried soap.
- × "Aquadag."

Mild steel, 5 deg. die, 20·3 per cent reduction in area.

Several other metallic salts of organic acids have been investigated, such as barium oleate. At a temperature of 120 deg. C. this substance is the most efficient lubricant which the author has employed. There is a wide, and almost completely unexplored field in the investigation of other solid lubricating materials.

One aspect of the use of soap as a lubricant in wire drawing is the dermatitis experienced by some workmen when using certain kinds of soap. This complaint is apparently experienced only where soaps, as distinct from oils or greases, are employed. Regarding the relationship, if any, of the disease and any special constituent of the soap, it appears that dermatitis has never been known to follow the use of a true olive-oil soap, though there are fairly strong grounds for the belief that soaps made from fat charges, including a large proportion of coconut- or palm-kernel oil, may lead to dermatitis in those susceptible to this complaint. It has been suggested that lime on the surface of the wire may be an important contributory factor, as it is the wire drawers and not the men working in the soap store who are the actual sufferers. A fact, however, which appears to put out of court the possibility of lime being the direct cause is that when a true olive-oil soap is employed, even with a lime-coated wire, dermatitis does not appear to result. The author is informed that the trouble may be eliminated if the hands are well rubbed prior to working with a mixture of unsalted lard and 5 per cent boric acid.

Numerous experiments have been carried out using lubricants, both solid and liquid, to which graphite has been added. The inclusion of graphite in wire-drawing soaps has been the subject of much research, the results of the author's experiments on such compositions being distinctly satisfactory. In most cases the graphitic liquids have proved quite satisfactory, "Castordag" in particular being one of the best lubricants that the author has tried in the drawing of mild steel wire at 130 deg. C. The satisfactory behaviour of many of these liquid lubricants depends, however, in large measure on the nature of the surface of the wire, and small traces of oily matter, by preventing the metal being wetted by the lubricant, may lead to very uneven drawing.

It is now generally agreed that the "lime coat" on steel wire serves to assist in the formation and retention of a stable film of the soap lubricant employed. When drawing high-tensile non-ferrous wires there would appear to be a fruitful field of investigation in the production of similar coats. Such coatings have been used with some success in the drawing of high-strength nickel-chromium alloys.

Soft metals have been employed as surface coatings in the drawing of steel wire. The use of copper in the wet drawing of steel is well known, and its more recent application to such materials as the nickel-chromium alloys is of interest. Lead has been employed in the Dudzelle process for the drawing of wire and tubes of austenitic stainless steels, and lead coatings enable much more work to be done on the material before annealing is required than is otherwise possible. It would appear most probable, though not yet certain, that the metallic coat acts mainly in assisting in the formation of a stable, adherent film of the soap lubricant which is also employed. The pull required for the lead-coated wire is very slightly greater than that for the untreated material, but the mechanical properties after a given reduction appear to be independent of the existence of the coat up to a reduction of area of about 70 per cent. For higher reductions, lead-coated wire is slightly softer and more ductile than is the uncoated steel. The evidence would suggest that the soap film anchors itself on to the lead coat with the result that it is able to withstand higher pressures without breaking down. Further, a chemical reaction may take place between the lead and the soap.

TABLE 2. COMPARATIVE EFFICIENCY OF LUBRICANTS IN THE DRAWING OF FERROUS AND NON-FERROUS METALS

Lubricant	Steel die, 6½ deg. semi-angle, 19·3 per cent reduction of area, minimum drawing load, lb.			
	Copper	Brass	Mild steel	7 per cent nickel-silver
A	13	43	61	53
B	16	44	62	58
C	19	46	63	54
D	21	48	64	54
E	21	47	64	60
F	25	49	66	60
G	25	58	70	80

Until comparatively recently the type of lubricant employed in the drawing of steel and non-ferrous wires respectively was essentially different, soaps being used for the former and grease lubricants for the latter materials. Without venturing to suggest the idea that a different type of lubricant for the two kinds of wire was unjustified, it may be pointed out that in the author's experience the most efficient lubricants for steel have been equally efficient with brass, copper, and nickel-silver and that, at any rate roughly, the order of a series of lubricants for ferrous metals was essentially the same when non-ferrous ones were being drawn. In Table 2 a few results of such tests are recorded.

Although efficient lubrication is clearly desirable, there is at any rate one instance where the use of a less satisfactory lubricant is preferable. It has already been pointed out that the lubricating film tends to adhere to the drawn wire, giving it a greasy surface. Where a subsequent treatment of the wire, as for example, in enamelling, is to be employed a bright-surfaced wire is at times preferable and this, so far as the author's experience goes, can only be obtained where the lubricant is not of an efficient type. Water at a temperature just below the boiling point, which, as we have seen, is a moderately efficient lubricant in wire drawing, has given in the author's experience a bright-surfaced wire which has proved satisfactory when coats of other materials are subsequently to be employed.

This short paper has concerned itself solely with results obtained in the author's laboratories, since it appeared that this would prove more useful than a cursory survey of so large a field.

In conclusion the author would like to acknowledge the great assistance rendered to him by Mr. E. L. Francis, M.Sc., who was working under his direction while most of this work was being done.

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GEAR LUBRICATION: THE APPLICATION OF LUBRICANT

By W. A. Tuplin, M.Sc.*

Having selected a gear lubricant which will give the necessary protection to the tooth surfaces, it must be applied so as to (*a*) conduct frictional heat away from the teeth, and (*b*) avoid excessive oil churning losses. The method of application will vary with the viscosity of the lubricant and the speed of the gears, and the following broad classification may be made.

(1) *Hand Application.* Grease or extremely viscous lubricant may be applied intermittently by hand to comparatively slow-running open gears, adhesion of the lubricant being depended upon to overcome the effect of centrifugal force. Open gears are not now favourably regarded; amongst other things, atmospheric dust and worn particles remain in the lubricant, which becomes more and more abrasive in character. Grease is not a good lubricant for heavily loaded gears, and the very viscous black oils now available are preferable. In a comparative test made to select a lubricant for a very highly stressed worm drive required to run for two minutes, greases good of their kind broke down in ten seconds, whilst a proprietary heavy black lubricant was running well after three minutes.

(2) *Dip Lubrication.* This may be of two kinds: (*a*) the teeth of one or both gears may dip in the oil, or (*b*) an oil-carrying member such as a flinger or annular trough may convey oil up to a point at which it will strike, or drain down on to the gears.

Dip lubrication is commonly employed in industrial gears up to peripheral speeds of 2,500 or 3,000 ft. per min., but much higher speeds are used for special purposes where the running time is short and some increase in churning loss is less troublesome than a forced lubrication system.

Different combinations of speed and viscosity can produce very different effects. For a given viscosity, the amount of oil carried round by the teeth will increase with the peripheral speed of the gear up to the point at which it begins to be thrown off by centrifugal force; this quantity will also increase with the depth of immersion and with the viscosity. As the speed increases, the quantity carried round diminishes, and the continuous stream of oil gradually gives place to a "Catherine wheel" type of spray. Ultimately, only the tips of the leading tooth faces will be wetted by the oil, the trailing faces remaining

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* David Brown and Sons (Hudd.), Ltd.

quite dry. Lubrication then depends upon some of this spray striking the mating teeth, rebounding back and forth until the point of mesh is reached. (The same thing may occur with a low-pressure oil jet).

If a "solid" stream of oil is carried up by the gear, it will be expelled with considerable velocity from the ends of the tooth spaces at the point of mesh, in the form of an intermittent jet. This may build up pressure in adjacent ball or roller bearings and cause leakage along the shafts unless intercepted by baffles; or it may strike the case wall and form an unsuspected cause of noise.

In order to minimize oil churning losses, the depth of immersion must be reduced to a minimum, bearing in mind that, on the one hand, the oil level will fall, when the gears commence to run, by an amount corresponding to the volume of oil "in circuit" round the gears and flowing down the walls of the casing, and, on the other, frothing may occur which virtually raises the level as regards both gear immersion and oil leakage along shafts.

Gears between 3 and 5 inches in diameter, and running at speeds between 7,000 and 10,000 r.p.m. have been successfully lubricated by splash. Although oil churning losses are greater than with spray lubrication, they are somewhat reduced because practically the whole of the oil above the lowest point of the dipping gear is converted to froth, which fills the casing. The froth is low in viscosity and creates less drag on the gears, although the degree of agitation is effective in removing heat. For industrial gears, it is rarely necessary for a gear to dip more than about three times the depth of the tooth spaces.

Care is necessary in designing the form of dip-lubricated gear casings. Close-fitting cases should be avoided, not merely because the quantity of oil may be undesirably small, but chiefly because of the excessive oil drag which will result.

A dipping gear produces rapid surface motion in a direction at right-angles to the teeth. Since the displaced oil must circulate back, this causes a variation in oil level between different points, and the oil level, as indicated by a gauge inserted in the casing wall, may either rise or fall according to its position. Helical gears or worms in a close-fitting gear casing may act as viscosity pumps, "pile up" the oil at one end, and cause trouble with oil leakage along shafts. In circulating, the oil gives up its heat to the walls. Width of oil sump is more valuable than depth.

Double helical gears with continuous teeth sometimes produce excessive noise due to oil trapping. If the gears run "apex trailing" and too much oil reaches the point of mesh, oil filling the tooth spaces then becomes trapped as successive apexes of driving and driven gears intermesh. This oil can escape only through the comparatively small tooth clearances, and very high pressures, suddenly applied, produce violent shocks, the sound of which resembles that of a machine gun.

Viscosity implies resistance to motion, and for this reason, is, in general, a nuisance. It can, however, often be turned to good account in effecting or preventing, as the case may be, the conveyance of oil from one place to another. Not only does the dip-lubrication of gear teeth rely upon viscosity; overhead oil scrapers feeding bearings, oil return screws, and screw pumps also depend upon it.

The proper use of viscosity in gear lubrication finds greatest scope in worm drives. It is sometimes said that overhead worm gears (wheel dipping) are inadequately lubricated and that oil drains away on stopping, leaving the teeth dry when starting up. With oils suitable for worm gear lubrication, this is not so; sufficient oil remains on the surfaces to provide the thin lubricating film required; the extra oil required for cooling arrives with the first half-revolution. Only if the viscosity is too low in relation to the speed of the wheel, and enough oil is not carried up to the worm at normal speed, will trouble arise; such an oil will, for all normal work, be too light to be a satisfactory lubricant, even if the volume were adequate.

(3) *Spray Lubrication.* This is employed either when the peripheral velocity exceeds the usual dip-lubrication limit, or when additional cooling is required and an external oil tank or oil cooler is employed.

Helical gears running at speeds of from 2,500 to 12,000 ft. per min. or over may be lubricated by a spray operating at 8–10 lb. per sq. in. Such a spray will have a low velocity in comparison with that of the gears, but if directed on to the gears on the approach side of the region of engagement, the oil will be carried between the teeth.

If the quantity or viscosity of the oil is excessive, application on the ingoing side may cause excessive power loss in oil churning, and for this reason the sprayers are sometimes divided, part of the oil being fed at the point of approach for lubrication, and the remainder on the leaving side for cooling. If, however, the quantity, viscosity and form of spray are correctly proportioned, no useful purpose is served in dividing the sprayers. The following figures relate to a test made to verify this conclusion:—

Gear centres, 24 inches.
Gear ratio, 4·5/1.
Wheel speed, 910 r.p.m. (direction "over and inwards").
Oil pressure, 10 lb. per sq. in.
Oil viscosity, 90 sec. Redwood.
No-load loss, no sprays, 22·6 h.p.

No-load loss, sprays at point of mesh, 25·0 h.p.
No-load loss, sprays at top of wheel, 23·6 h.p.
Quantity of oil through sprays, 9·6 gal. per min.

The total no-load loss represents 1·0 per cent of the rated capacity of the gears, and the difference in power loss when changing the direction of the sprays from the top of the wheel to the point of mesh therefore makes a difference of only 0·055 per cent in the overall efficiency.

High-speed worm gears require jet lubrication, and it is important to remember that oil flung off the gears is mostly flung away, and does not rebound back to the point of mesh as it does with helical gears. Lubrication is best effected by two "solid" jets (as distinct from sprays), one striking the wheel teeth in a direction parallel to the axis of the worm, and the other striking the worm threads parallel to the wheel axis, in both cases just before the point of mesh. The jet velocity should be somewhat higher than the peripheral speed of the wheel.

Worm drives for turbine governor gears are nearly always badly lubricated, which is probably the reason why they are more troublesome than any other worm drive. Not only is turbine oil lower in viscosity than is desirable for worm gears, it is too hot on the high-pressure side, and at too low a pressure on the cool side, whilst oil jets, if used, are seldom properly disposed.

Oil sprayers are designed to project a flat, fan-shaped film of oil, which in still air can reach a surprising width without breaking up. The most usual type is a nozzle giving a single circular jet, which strikes an inclined deflector plate. The form of the resulting spray is rather sensitive to oil viscosity, contracting rapidly in width at viscosities above 400 sec. Redwood, and pressures below 10 lb. per sq. in. Twin intersecting jets, however, produce a form of spray which shows less change in width and thickness with varying pressure and viscosity.

The total quantity of oil required for cooling will be determined by the power losses in the gears and bearings, and the allowed temperature rise of the oil between ingoing and outgoing. Overall efficiency varies greatly with the type of gear and bearings; high-speed helical gears mounted on plain bearings usually have a total power loss of about 1·5 per cent, with ball or roller bearings, 1 per cent or less; of this, about 0·5 per cent represents tooth losses. The losses in spur, bevel, and worm drives increase in that order, and must be calculated individually. An oil temperature difference (in and out) of between 10 and 15 deg. can usually be obtained without excessive rate of oil circulation. Gear running temperatures under continuous full load, range from 120 deg. to 160 deg. F., the former being regarded as cool. Higher temperatures (200–220 deg. F.) may be used in hot surroundings, if the oil is suitably selected.

Oil vapour from high-speed gears can be troublesome if allowed to escape in the neighbourhood of electrical machinery on account of its effect on insulating materials. Once formed, it seems impossible to condense it, and it must be either isolated or led away. There are two main causes of the vapour: (*a*) air leaks into the suction side of the pumps, (*b*) the impact of drops of oil, moving at high speed, on gear-case walls. High-speed air currents exist in high-speed gearcases, due both to the peripheral velocity of the gears and the air displacement due to the intermeshing teeth; these result in zones of high and low pressure in the case, and vents should be disposed with these in mind. Case contours should be arranged where possible, so that oil striking them does so obliquely rather than perpendicularly.

Complete force-feed gear lubricating systems may be more, or less, elaborate, according to the conditions which have to be met. Strainers are always desirable, and should be capable of being cleaned without shutting down or interrupting the oil flow. If oil must be circulated for cooling but considerations of speed do not require sprayers, an open jet may be used; but sprays require the provision of a pressure gauge accompanied in important cases by an alarm to give warning if pressure should fail or fall. Cooling may be effected by cooling by coils inside the gear case, by an external tank, or by a special cooler.

REPORT ON PAPERS IN GROUP III

INDUSTRIAL APPLICATIONS

By Lt.-Colonel S. J. M. Auld, O.B.E., M.C., and E. A. Evans

If the papers in Group III are regarded as a narrative, and the individual papers merely as chapters written by several authors, it is possible to obtain a comprehensive idea of the subject as a whole. In this way the conclusion may be drawn that the discussion should be on "Industrial Applications" as a whole, and not necessarily on individual communications. This point of view was kept in mind when preparing this report, but one exception had to be made. Hanocq's paper on "The Experimental Study of Ball and Roller Bearings" does not seem quite to fit into the narrative. It might have received even greater consideration in Group I, because it is so closely connected with his paper in that Group. It does, however, form a very useful link between Group I and Group III.

Grease Lubrication. A decade ago it seemed hardly likely that the use of grease in modern lubrication practice would persist. To-day it is being used in ever-increasing quantities. The reasons are that, by virtue of its solidity, grease for many purposes has a big advantage over oil and that up-to-date methods of design and manufacture can ensure efficient and standard products having little relationship to the haphazard greases of the past. Catterall and Maitland's survey is important because it shows this trend clearly, and will convey to many people the need for according full recognition to a form of lubrication of which they may have been only somewhat vaguely aware. In various parts of the world, for example, large rolling mills are being erected in which very elaborate systems of grease lubrication are being incorporated. Grease may be delivered continuously and regularly by means of mechanical lubricators to points 1,000 feet distant.

Anti-friction bearings also are calling more and more for various and suitable greases. Such products must be developed to meet particular requirements, for it would be inadvisable, of course, to use for ball or roller bearings a grease which had not been prepared specially for the purpose.

Theoretically ball, roller, and needle bearings require little lubrication; indeed, to use Cooper's words "the ideal is an oil mist". In practice, however, an oily atmosphere is insufficient, so that oils of various types are used, as in other forms of metal bearings. It must be realized that there is always slight deformation of the metal due to the loads carried, and also a small amount of sliding friction. In needle bearings, the needles are revolving in opposite directions, thus causing sliding friction. Such forms of friction are revealed in heat, which if not removed would cause damage. If oil is applied by one of the recognized methods, the friction is reduced and the heat minimized.

But, if oil, as expressed also by Richardson, is the ideal lubricant, why use grease? The answer is, to get simpler retention designs and maximum protection. Anti-friction bearings are made very accurately, and whatever upsets their accuracy and finish militates against their advantages. Rusting must be prevented and anything which may cause abrasion kept out. The simplest way to effect this exclusion is to fill the bearing and a suitable part of the housing with grease. Bearings when in service inhale and exhale. If the air space is large much dust and moisture can be inhaled, but if the air space is limited and the metal surfaces are covered with grease little damage can result.

When using grease the most careful selection is necessary; it is a fact, of course, that all makers of anti-friction bearings exercise great vigilance in this direction. It is little wonder, therefore, that they are difficult people to satisfy, both in the choice of lubricants and the manner in which they are applied. The ease with which these bearings operate tempts us, sometimes to our cost, to leave them unattended. Grease must be added from time to time, but it is important that the housings be not packed too full.

Although there is no paper dealing with miniature ball bearings, it should be pointed out that much interest is being taken in the bearings with an overall dimension of 1·5 mm. which are on exhibition at the Science Museum.

Viscous Emulsions. Allied to grease lubrication is the application of viscous emulsions. These emulsions resemble greases in some respects, but differ essentially from them in their larger water content. In the cold drawing of steel tubes, where it is necessary to lubricate the die and the mandrel, neither grease nor a soluble oil is suitable, but a soda-base type of grease emulsified with water is proving satisfactory.

The cold pressing of sheet steel is a somewhat similar operation. The metal is drawn out by pressure in the presence of a lubricant. The omission of a lubricant between the sheet and the dies in the cold pressing of steel is usually immediately evident in resulting damage to either the pressing or the dies, or to both. Whilst the use of simple lubricants such as mineral or fatty oils has in the past generally been sufficient, there have always been cases where such lubricants have proved inadequate under conditions of extreme pressure. The development of improved pressing steels and the increase in the speed of press operations have increased the necessity for improved drawing lubricants. The condition of lubrication during cold pressing is one in which an extremely thin film of lubricant is operating. It is possible that the work of Bowden at Cambridge, and the research upon which Swift is now engaged at Sheffield, may help to solve this problem.

Arrowsmith believes that film strength is the most important property of a drawing compound. To achieve this, resort has been made to solid lubricants, such as talc. More recently, and perhaps more logically, chlorinated and sulphurized oils are being tried. Oiliness, too, is a desirable quality, but curiously enough it is a property which must be kept within the limits prescribed from practical observations. There are also other essentials, but in practice a compromise is frequently made by using fatty oil emulsions containing chalk powder or similar material.

Cutting Compounds. Viscous emulsions found favour for a number of years under the designation of cutting compounds. Whilst they were an improvement upon the old compound oil-soap-water mixtures, they were not good enough and their place was taken by soluble oils which are readily mixed with water to give stable emulsions.

Water-soluble oil emulsions or "suds" are used for rough cutting where tool wear is not important. For hobbing, threading, and gear cutting, however, expensive tools are used, and the preservation of their edge becomes an important factor. Oils are consequently chosen according to the demands of the operation and the requirements of economical production. There is no golden rule to guide the choice of cutting fluids, though much has recently been accomplished in correlating machinability, through the physical properties of the metal, with empirically chosen oils and solutions. It is common practice, however, to machine dry those metals which form short chips, while those which form continuous chips which press heavily upon the face of the tool are cut wet. Boston has studied the performance of cutting fluids, and their particular applications to specific metals, tools, and cutting operations. He summarizes under thirteen headings those properties which he considers essential. He then examines the influence of cutting fluids on the torque of a twist drill in various metals. His conclusion is that the performance of one cutting fluid on a given metal is not a criterion of its comparable performance on another material. When the thrust curves for the same materials are plotted, no similarity is seen between the curves of any two metals. Those graphs are likely to prove of considerable assistance in large shops where power costs have to be considered.

Lloyd and Beeny state that aluminium alloys are usually machined dry, but Boston's results seem to indicate that some advantages are obtained by wet machining. J. F. Miller states that except for magnesium alloys, some kind of cutting fluid should always be used. Perhaps somebody will be able to explain this apparent divergence of view.

Possibly both views are wrong, because Miller believes that the cutting fluids of to-morrow will differ greatly in composition from those of to-day. He instances a mixture of alcohol and paraffin as a useful fluid when machining at exceptional speeds. If such a low flash point fluid were used extensively, it would be interesting to observe the reactions of the fire insurance companies !

Sulphurized Oils. In metal cutting the pressure between the chip and the nose of the tool reaches very high values. Engineers have long known the practical value of adding flowers of sulphur to oil, but they had no idea whether the sulphur acted as a solid lubricant or whether it acted in some other way. It was not until the introduction of the hypoid gear that a serious study of sulphurized oils was undertaken. Since that time sulphurized oils for high-duty work have become a commonplace of the machine shop.

Sulphurized oils form part of that group of lubricants known as extreme-pressure ("E.P.") lubricants. The term "E.P. lubricants" may be a little unfortunate because a pressure which is considered extreme at one moment may be commonplace at the next.

Evans has presented a survey of the relative film rupture strengths of a number of oils and greases and has recorded some of his investigations upon pure organic compounds. F. L. Miller goes so far as to state, "Perhaps no recent single development in the field of lubrication has done more to revolutionize thought on lubrication than the introduction of the hypoid gear". Its lubrication certainly involves radical departures from previous practice. The hypoid gear, however, is not the only form of transmission which may require extreme-pressure lubricants. Many motor vehicles are using them to enable greater loads to be carried by a mere change of the lubricant. The possibilities are really enormous; so great in fact that no engineer can afford to be unacquainted with them. Obviously, initial troubles have been encountered, as in every other field of development. During the pioneer work, experimenters were severely handicapped by the dearth of laboratory testing apparatus, and by the expense of making tests under running conditions. This applies even to-day. It is not possible to rate extreme-pressure lubricants in the same order of load-carrying capacity by any two machines. In Miller's opinion, and in that of others, it is becoming recognized however, that the tentative S.A.E. machine shows most consistent correlation with gear performance. Clayton has studied seizure characteristics under conditions of high pressure between hard steel balls, using the well-known four-ball apparatus. The apparatus is not intended for direct tests of gear lubricants, but Clayton's view, with which the reporters are in agreement, is that it has great advantages for research purposes. Anyone studying this subject should consider Brownsdon's paper in Group IV.

It is significant of the intense interest taken in extreme-pressure lubricants that more than half of the papers in Group III are devoted directly or indirectly to their study.

Light Mechanisms. The lubrication of light mechanisms of the clockwork type, or the footstep bearing of an electricity meter, or the bearing of an electric clock seems to be so insignificant as to pass unnoticed, until attention is suddenly arrested by the information that the bearing pressure may be 75 tons per sq. in. If the main bearings of a petrol engine were loaded to 75 tons per sq. in. there would be some consternation. How little does the householder, who uses electric light, realize that in his house is one of the most heavily stressed bearings in existence, and that if it is not lubricated properly he may receive more electrical units than he pays for.

The bearing of the electricity meter consists of a hardened ball-ended steel pivot which rotates in a highly polished sapphire. As the steel pivot has a radius of only 0·025 inch the difficulties of finding the ideal form of lubrication can be imagined. First of all, is lubrication necessary? Both Stott and Shotter have come to the same conclusion, that the bearing runs perfectly happily for a time without oil, but if it is to operate for a prolonged period it must be lubricated. If, however, a diamond is used there is much less need for lubrication. The major factor which has led meter engineers to seek a suitable lubricant has been the production of ferric oxide from the steel pivot in the jewel cup. This oxide is highly abrasive, so the lubricant used must be capable of holding it in suspension. It would be interesting to know whether electron diffraction technique has been applied to this very complex problem, and whether an oxidation inhibitor possessing extreme-pressure properties has been found to be of value.

Wire Drawing. By virtue of the pressures involved, the problem can perhaps be visualized by comparing it with wire drawing, where the pressure in the wire-drawing die can range from 5 to over 70 tons per sq. in. Goodacre quotes the theory that there is no true lubrication, but that the wire practically comes in direct contact with the die. The so-called dry drawing of steel wire requires some explanation. After cleaning in acid, and washing, a wet rust coat is formed on the surface. The rod is then dipped in a cream of calcium hydroxide which partly neutralizes any remaining acid and partly helps in the subsequent drawing. As a lubricant for drawing it is usual to use a dry soap, generally sodium soap, which becomes bonded with the lime. For efficient working the soap should be as dry as possible. The use of soap as a lubricant is regarded by Goodacre as almost unique, because theoretically it has little free energy. It gives a constant lubricating effect up to a temperature of 150 deg. C. Sodium stearate behaves rather differently.

The conditions of wet drawing do not seem to have been studied in much detail. One of the essentials in the wet drawing of steel wire appears to be the formation of a very thin coat of copper or tin on the surface of the metal. Most baths are acid. They usually contain a fatty lubricant, and the presence of an emulsifying agent, e.g. soap, is almost essential. The sulphuric acid is essential; in fact a 10 per cent solution of sulphuric acid can be used in drawing steel wire through a die without the presence of fatty matter. When it is considered that 50 per cent of the power used in wire drawing is needed to overcome frictional forces, it is obvious that if, by means of improved lubricants, even small reductions could be obtained in the coefficients of friction, a considerable saving in power costs alone would be effected.

Gear Teeth. The paper by Thompson on wire drawing provides an excuse for discussing boundary lubrication if desired, although Merritt refers to it in his description of the lubrication of gear teeth. Merritt's paper is a storehouse of information, and must be read in its entirety to appreciate its value. Reference must be made to his gear lubricant testing machine. It has some points in common with the S.A.E. machine already referred to. Any comparative results obtained in the two machines would prove of considerable interest. Tuplin continues the subject of gear lubrication by telling us how the lubricants should be applied—most necessary advice.

Surface temperature measurements on gear teeth under extreme-pressure lubricating conditions are helpful to those who are interested in gears for industrial applications, and reference may be made to Blok's paper on this subject in Group IV. It forms a useful association with Bowden's paper in that group on "The Friction of Sliding Metals".

Chains. Passing from power transmission by gears, we come to chains. Chains may be divided broadly into two classes: (1) chains for conveying materials; (2) chains for the transmission of power. Coulson restricts his remarks to the latter class. The pressures under which the joints operate may be 3,000 lb. per sq. in. calculated on the projected area of the bearing. Where the service required is continuous, pressures of 1,250 lb. per sq. in. are aimed at.

The conditions under which the joints of a chain operate are not so severe as might be anticipated. During the transit of the chain the motion of the bearings is intermittent, so that there is ample opportunity for the fluid lubricant to enter the clearances between the roller and the inner plates. So long as the fluid lubricant does not form sticky deposits any oil having a viscosity of 600–900 sec. Redwood at 70 deg. F. can be used, but mineral oils are preferred. Grease is not recommended, because of the difficulty of inserting it between the moving parts. On the rear transmission chains of motor cycles and similar exposed drives, however, where it is impracticable to arrange for continuous lubrication by means of oil, grease is used. It must be of the mineral jelly type, so that it can be melted to penetrate. Mention might have been made in this connexion of the use of penetrating oil containing colloidal graphite.

There are so many chains in use on conveyer work that even in the absence of a special paper it may be prudent to invite discussion upon them. Chains for this type of work are used under all sorts of conditions, from the automobile industry to the bakery trade for transporting bread through the ovens. Generally speaking the older method of applying a viscous oil by hand is being replaced by mechanical lubrication.

Applications in Mining Engineering. To mining engineers the papers on rock drill lubrication and wire ropes will be welcome. The rapid progress in mining on the Witwatersrand, and elsewhere, owes much to the improved efficiency and construction of pneumatic hammer drills. The co-operation, and co-ordination of research between the various mining groups and research organizations are leading to greater safety in mines and considerable easing of the lot of the miner. Any improvement, for example, which will reduce the risks of silicosis is particularly pleasing. There is little doubt that an efficient rock drill does much in that direction.

A modern pneumatic hammer drill strikes from 1,700 to 2,000 blows per minute. Such a high-speed machine presents certain lubrication difficulties which are recorded in Gimkey's paper. These difficulties have been surmounted by introducing the lubricant into the machine together with the compressed air, by means of an air line lubricator which should atomize the oil. Since only one kind of oil can be used for lubricating all the parts, careful thought must be given to its selection. Gimkey describes an air line lubricator which was designed at the University of Witwatersrand. Doubtless discussion will bring to light the existence of other types of air line lubricators, and the types of oils which are in use.

Nixon and Jackson observe that the lubrication of wire ropes for winding purposes presents a somewhat different problem from ordinary lubrication. They therefore undertook to ascertain the various conditions under which the ropes were operated, also on the Witwatersrand. The conclusion was that the evidence, though interesting, was so contradictory as to make any definition difficult.

At the same time Hogan, of the Safety in Mines Research Board, gives a comprehensive survey of the subject and points out that wire rope lubricants have to work under difficult conditions. The lubricant has two functions. First it must reduce internal friction between the wires and external friction against the drum and pulleys; second, it must preserve the metal from corrosion. Owing to the high contact pressures between the wires in a rope, seizure is liable to occur and wear may be very rapid. This being so, it may be a little difficult to appreciate how a thin mineral oil, or indeed a penetrating oil, such as he is experimenting with, could be used with any degree of satisfaction. If a penetrating oil should prove to be satisfactory, it would be necessary to consider its composition, or alternatively to ascertain whether the contact pressure between the wires is as great as is imagined. If the pressure is really high it might be that an extreme-pressure lubricant, possessing suitable characteristics, could be used with advantage.

Hogan exposes many practices which deserve close attention, and which require more detailed definition. What are the data for the statement that waxes are inert bodies and undesirable? If a wax is a monohydric alcohol combined with a fatty acid, then we should have to review the position of sperm oil and wool wax. It may be a little difficult to accept the theory that sperm oil is more readily hydrolysed than seal oil, but the practical results, of course, are not challenged. Whatever may be the necessities for research it is comforting to learn from Nixon and Jackson that on the Witwatersrand field it is a rare occurrence for a discarded rope to show signs of internal corrosion. That, we fear, is not always the case with our home colliery ropes.

Textile Machinery Spindles. Of the three classes of textile spindles, namely (*a*) mule, (*b*) flyer or speed, (*c*) ring, according to King only the ring need be considered, because the speeds at which the others run do not in themselves lead to any lubricating problem. King, the author of the paper on this subject, protects himself with the words "in themselves". Perhaps someone will enlarge upon this interesting phase of the subject by telling us whether the coefficient of friction is considered an important characteristic of an oil for flyer spindles. Ring spindles rotate at 10,000–12,000 r.p.m. At this speed they can easily be thrown out of balance and caused to vibrate, unless proper precautions are taken. As fluid drag at these high speeds would be most unwelcome, an oil of the lowest viscosity that will maintain a complete lubricating film with minimum friction and lowest operating temperatures, is used. Oils having a viscosity of 65–70 sec. Redwood at 70 deg. F. are recommended.

The important subject of wear has been examined. The result of this investigation seems to indicate that the best conditions are obtained with cast iron containing 0·50 per cent of combined carbon with a network of phosphide eutectic.

Anti-friction bearings at first sight may appear attractive for these spindles, but the initial cost renders the saving in power consumption and lubricant somewhat doubtful.

DISCUSSION ON PAPERS IN GROUP III

14th October 1937

The **Chairman**, Sir H. Nigel Gresley, opened the meeting on 14th October 1937, at 2.30 p.m., and called upon Lt.-Colonel S. J. M. Auld and Mr. E. A. Evans to present their report on the papers.

Mr. E. A. **Evans** (Messrs. C. C. Wakefield and Company) suggested that Colonel Auld, as President of the Institution of Petroleum Technologists, should present the summary of the papers in Group III.

Colonel S. J. M. **Auld** (Vacuum Oil Company), in presenting the report, said he desired first of all to pay a tribute to Mr. Evans as his co-reporter, because during his own absence abroad, the work in preparing the report had fallen almost entirely on Mr. Evans. In addition, Mr. Evans had organized the very fine exhibition at the Science Museum, and it was fitting that those present should appreciate the burden that Mr. Evans had been carrying, and his present self-effacement. Colonel Auld then read the report (p. 167).

Discussion

Mr. W. E. **Gooday** (Brettenham House, W.C.2) remarked that the so-called extreme-pressure lubricants were emerging from the experimental stage, so that it would be very dangerous to be dogmatic at present. Nevertheless, the period was one of transition between the use of mineral oils (straight mineral oils or those compounded with fatty oils) and that of built-up lubricants. There was every likelihood that synthetic products of the future would not only withstand extreme pressures, but would tend to alleviate cylinder liner wear. A colleague had suggested that the refiner removed substances from a lubricating oil which the engineer immediately put back into the oil; but the substances which the engineer now added were not derived from the oil. Where conditions were more favourably fulfilled by the use of lighter oils, the employment of extreme-pressure bases would assist in the use of lighter oils where formerly only medium or heavier oils had been within suitable viscosity range.

A speaker that morning had referred to the premises on which the conclusions were based. Was one always certain, when comparing results, that they were derived from comparable samples? He was inclined very seriously to doubt that, since the further away from the refinery one might be, the more unlikely it was that samples for investigation would be identical.

Mr. Guy had referred (vol. 1, p. 367) to air, dust, and surface condition as the factors affecting performance. The factors affecting the life of a lubricant were air as a source of oxidation; temperature as a means of accelerating that oxidation; turbulence as a mechanical means of presenting more oxygen to the oil in bulk; water as a means of emulsification, or of introducing impurities; impurities as dust from various sources; and treatment as the way in which the oil was kept clean during use (e.g. by using a centrifuge), particularly in a circulatory system. Mr. Guy must not throw the dust only in their eyes. The impurities might be lead, zinc, copper, iron oxides, bronze or silica, or dust containing cement, flour, coal, or acid fumes. He thought Mr. Guy was not referring to substances found in the oil, but rather to materials which gained access to the oil during use. If, however, Mr. Guy did refer to any traces of mineral matter in the oil, he trusted that, if the oil technologist was accused of those motes, then the engineer-user would remove his beams!

Dermatitis was mentioned in the reporters' summary. The various forms of, and the prevention of, dermatitis had been dealt with by the Home Office Medical Officers, but an up-to-date account would be very useful to users of petroleum products, particularly oil, in machine-shop operations.

Mr. G. H. Thornley (Messrs. C. C. Wakefield and Company) remarked that Mr. Gimkey's paper gave an excellent analysis of the lubrication of rock drills, but it seemed to neglect the quality of the oil to be so applied.

The great extent of rock-drill operation and the amount of oil and replacement parts consumed thereby, were perhaps not generally realized. At one mine alone on the Rand, the average monthly cost of rock-drill spare parts was £1,500, which was reduced to £700 per month by changing over to a more suitable lubricant.

The air line lubricator described by Mr. Gimkey had the advantage of simplicity, but apparently it would still feed oil by the annular capillary, even when the drill was stopped, although at a much slower rate than when air was passing. Another type of air line lubricator depended upon felt plugs fitted with leather washers to close the oil feed holes. By making the oil feed holes of suitable size, the fit of the felt plugs and washers effectively controlled wastage of oil either by the unbalanced pressure or by gravity when the machine was idle.

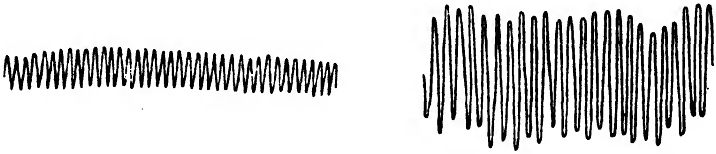
The lubricant used in wet-drilling suffered very arduous conditions. The water passed through at high pressure; it was frequently extremely "hard" and, containing alkali and alkaline earth metal salts in solution, it had strong corrosive properties. The water flowed down the central *stationary* tube into the hollow drill, or jumper, which had both reciprocating and rotational movement, so that an effective water joint could not be made and the water escaped along the inside and outside of the chuck, carrying the lubricant with it away to waste. The result was to set up serious corrosion of the unlubricated surfaces every time the machine was allowed to stand idle for a few hours.

Numerous laboratory experiments suggested the emulsification of the hard water with the lubricating oil, and that theory had been amply proved in full scale research with actual rock-drills and in industrial operation. However, it was not sufficient merely to provide a lubricant which possessed emulsifying properties; a water-in-oil or mayonnaise type of emulsion was required as distinct from the oil-in-water or soluble oil type, and the water-in-oil emulsion must be stable so that it did not break down on standing. In addition, it must have a high degree of adhesiveness to metals in order to resist the washing action of the stream of water. The kernel of rock-drill lubrication for wet-drilling undoubtedly was to prevent the corrosive water from making contact with the metal, and that could be achieved by absorbing it into the lubricant to form an emulsion in which each particle of water was surrounded by a tenacious film of oil which itself maintained contact with the metallic surfaces.

Mr. R. POOLE (Metropolitan-Vickers Electrical Company, Ltd.) said that whilst considerable trouble was taken to reduce friction, it was very often of equal importance to reduce vibration. In attempting to reduce friction it must not be forgotten that, if it were desired to reduce wear, the vibration of the moving parts must be minimized. Several authors had stated that at certain speeds, there was severe rotary vibration on hydro-generator sets. The problem of vibration in spinning machines had also been mentioned. The difficulty of reducing vibration arose in other instances, particularly in reciprocating engines.

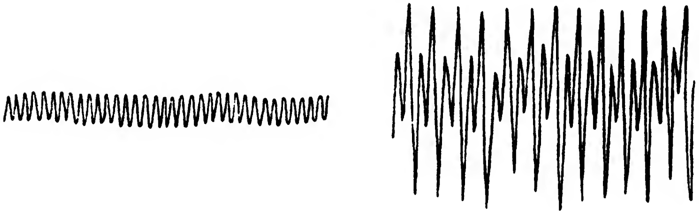
When the vibration on the pedestal or housing of a machine was recorded, results were obtained showing that the vibration was almost sinusoidal under widely varying conditions; but on taking measurements on the shaft or journal itself, vastly different results were found. For instance, with a large turbo-alternator running at 3,000 r.p.m., the type of vibration diagram obtained was as in Fig. 1.

The vibration of the pedestal followed quite a smooth curve, but the vibration of the journal just at the nose of the pedestal was going through the form of oscillation illustrated in Fig. 1 *b*. There was a very pronounced double frequency. Readings were then taken at various speeds, and it was found that whereas the vibration of the pedestal changed very little, the form of vibration of the journal itself was completely different and at certain speeds would be seven or eight times that of the pedestal vibration. That implied that there was a considerable amount of damping between the journal and the pedestal, and, in order to check the effect of a change in oil (both quality and quantity) the supply of the same type of oil was reduced to something like one-third. Strangely enough, there was a reduction in the pedestal vibration to roughly 70 per cent but, at the same time, there was an increase in the journal vibration of about 50 per cent (Fig. 1 *c* and *d*).



a Pedestal vibration *b* Journal vibration

Normal oil supply



c Pedestal vibration *d* Journal vibration

Reduced oil supply

Fig. 1. Comparison of Pedestal Vibration and Journal Vibration, showing the Effect of Reducing the Quantity of Oil

Rotor running at twice the first critical speed.

Reference was made in one paper to distortion of crankshafts. The question of damping was particularly important in that branch of lubrication. Dr. Shannon * had shown that by applying the correct quantity of oil to an engine, the vibration could be reduced to something like 30 per cent of the normal vibration. Moreover, Dr. Shannon had

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* Shannon, J. F. Proc. I.Mech.E., 1935, vol. 131, p. 387.

shown that it was possible for the engine to run safely at its critical speeds. The speaker's firm had investigated the axial vibration of engines, and had found again that by correct lubrication, it was possible to reduce the vibration to 25 per cent of the normal; and in particular, since most high-speed engines showed a pronounced critical speed of axial vibration, the lubrication of the thrust block and the quantity of oil supplied were of primary importance.

The problem facing petroleum experts was to produce an oil which not only reduced friction, but had very high damping properties. That such a combination of properties was possible was revealed in the papers dealing with "oiliness" and, or alternatively, the variation of viscosity with pressure.

With a heavily loaded journal bearing having forced lubrication the loaded side of the bearing would fall in the high-pressure, low-viscosity region so that the coefficient of friction would be small. The low viscosity of the oil in the oil film would result in a relatively low vibration damping. On the other hand the oil in the unloaded part of the bearing would have a higher viscosity and would therefore be more effective in damping out vibration.

The most suitable oil for machines subjected to vibration would therefore appear to be one which showed a big reduction in viscosity with increase in pressure.

The paper by Bradford and Vandegrift (vol. 1, p. 23) and also that by Burwell and Camelford (vol. 2, p. 261) provided valuable information for the study of the problem of vibration damping in bearings.

Mr. J. E. SOUTHCOMBE (Germ Lubricants, Ltd.) remarked that Mr. Clayton had been investigating extreme-pressure lubrication by the use of the four-ball machine of Boerlage. His (Mr. Southcombe's) laboratory had been doing similar work with a Timken machine. Mr. Clayton found a break in the curve of wear against dead load at fairly moderate loading. The break was then followed by a kind of seizure effect, and a stable state of lubrication was arrived at later on. Mr. Clayton called the break in the curve the "recovery period." That recovery appeared to him (Mr. Southcombe) to represent simply the change from an unstable state of lubrication (if there was any lubrication at all) to a condition where the surface had worn itself to a bed and so reduced the specific pressure to a relatively low value where boundary or semi-fluid lubrication could exist. His laboratory had found that a boundary or semi-fluid state existed in the period after recovery, as was instanced by the fact that the friction was influenced in that part of the curve by the presence of a fatty acid.

Comparatively low values for specific pressure were found. For a stable state of lubrication under conditions of extreme pressure, it seemed wrong to state that some oils had very high film strength, as even so-called extreme-pressure oils did not withstand specific pressures of 20,000 lb. per sq. in. He protested most strongly against the use of such an expression as "high film strength." Oils did not possess high film strength. In the literature, the expression "high film strength" was even coupled with "oiliness", which was quite erroneous. Wherever there was line contact or point contact, wear would immediately take place to relieve the load, and all that the extreme-pressure lubricant did was to permit wear to take place smoothly, and therefore relieve the load. An interesting corollary was observable in wire drawing. The pressure on the die when drawing copper wire was estimated at about 5,000 lb. per sq. in., and under such pressures, the copper wire-drawing die could be satisfactorily lubricated with a fatty oil or with a material containing fatty acid. But in drawing hard steel, the pressure on the die was nearer 100,000 lb. per sq. in., and there all liquids generally failed as lubricants and the wire had to be lubricated by a ferroso-ferric hydroxide coating which was bound together by a solid material. In other words, in all those conditions of very high pressure where a hydrodynamic film could not form, liquid lubricants would fail to carry a load much in excess of 20,000 lb. per sq. in.

Mr. H. H. **Beeny** (Messrs. Alfred Herbert, Ltd., Coventry) said that he had been interested in the remark made by the Reporters in quoting Professor Boston to the effect that aluminium alloys could, with advantage, be cut on machine tools with cutting fluids. Great caution was required in making general statements on metal cutting. Nowadays, tungsten-carbide tipped tools were commonly used to machine a metal such as aluminium which did not set up much cutting pressure, and therefore could be cut with more safety to the expensive tools. Therefore, there was less necessity for any cutting fluid with tungsten-carbide tipped tools than there was with high-speed steel, because their wear resistance was so great. Also, such high cutting speeds could be used with them that, if any cooling fluid was used, it tended to be flung from the machine. The modern practice was to machine metals of the aluminium class dry under most conditions. Where a particularly good surface finish was required, turpentine or paraffin was often used; but cutting was then done at comparatively slow speeds, and was usually a tool-room job.

Concerning the use of a cutting fluid (particularly an aqueous cutting fluid) with a tungsten-carbide tipped tool, if the cutting fluid was allowed suddenly to impinge upon it, it was quite possible that the expensive metal alloy tip might be cracked. He wished to stress the importance of a careful choice of cutting emulsions for machine tools. Great damage was frequently done to machine tools by the choice of an unsuitable type of aqueous emulsion. As good types were available, there was no excuse for using types which caused damage.

Dr. F. P. **Bowden** (Laboratory of Physical Chemistry, Cambridge University) remarked that it was important to know what was the real area of contact between metal surfaces. It was customary, for example, to calculate the pressure between surfaces as being so many pounds per square inch, but before the real value of the pressure could be ascertained the real area of contact must be known. Even if the surfaces were carefully polished, hills and valleys would still be present, and the surfaces would be held apart by those irregularities. If a piece of polished steel, say one foot square, was laid on top of another, how much of the surfaces really touched ? The area of intimate contact was very much less than a square foot. He and his collaborators had made some rather primitive attempts to estimate that area. One method was to measure the electrical resistance between the surfaces and attempt to calculate how much metal was touching. The theoretical treatment of those experiments was necessarily approximate, but they indicated that the real area was extremely small, something of the order perhaps of 0·0001 sq. cm. for, say, a foot of steel, even for a highly polished surface. *That meant that the intrinsic pressure was very high because all the load was carried on that very small area.*

The classical work on the area of contact between surfaces was due to Hertz, who took the case of a sphere resting on a flat surface. He assumed that the sphere was *elastically* deformed by the pressure and then calculated the area of contact. That could be applied to steel, and a measurement of the indentation made on the sphere showed that the area was in approximate agreement with the theory. Was it right to assume, however, that the steel behaved like an elastic body ? Was it not possible that the flow of the steel was *plastic?* That is to say, was it not more like a ball of putty pressed against the surface rather than a ball of rubber ?

A calculation could be made showing how the electrical resistance between the surfaces should vary for elastic and for plastic deformation. In each case there was a different relationship between the electrical resistance and the load. Recent measurements made by Mr. Tabor indicated that the deformation was *plastic*, i.e. the steel at the points of contact flowed like putty. That did not mean that the deformation of the whole ball was plastic because the intense pressure only occurred locally at the surface irregularities and it was there that plastic flow occurred. Over the rest of the ball the pressure would be much less, and its deformation would be elastic. As an analogy, it behaved as a rubber ball with knobs of putty on it. But at the points of contact—and they were the things that mattered for wear and friction—the pressure, even on "flat" surfaces, was great enough to cause plastic flow of the steel. Even for lightly loaded surfaces, the local conditions at the points of contact might be those of "extreme pressure".

Mr. W. H. **Oldacre** (Chicago, Illinois, U.S.A.) said that he had been asked by Professor Boston to bring to the meeting his greetings and to convey his regrets that he was unable to be present. In the paper by Messrs. Lloyd and Beeny, reference was made to oil dermatitis, and Mr. Gooday had asked for comments on that subject, while Messrs. Lloyd and Beeny implied that there was some connexion in the neat oils, as they termed them, between the acidity of the oil and oil dermatitis. American experience indicated that the only type of dermatitis that could be truly attributed to the oil was oil acne, which manifested itself as a small red pustule appearing on the parts (generally the forearm) which were in contact continuously with the oil. It disappeared rapidly when the part affected was removed from contact with the oil, and was very difficult to eliminate so long as the part remained in contact with the oil. American experience indicated that the cause was physiological rather than bacteriological. In America they had had experience of cases of oil dermatitis with all types of oil—straight petroleum oils, straight fatty oils, and mixtures of the same, with and without the addition of antiseptic or germicidal materials. They had been able to control the trouble to a certain extent by increasing the viscosity of the oil. Kerosene, which was used sometimes as a diluent in cutting oil, tended to aggravate oil acne.

Mr. J. C. **Geniesse** (Atlantic Refining Company, Philadelphia, Pennsylvania, U.S.A.) agreed with the remarks made by Mr. Gooday. The lubricating oil of the future would be a synthetic lubricant. In America they were making rapid strides in that direction. Increased speeds, higher operating temperatures, and the desire of the automobile manufacturer to make smaller gears and to reduce the weight and cost of the car had thrown many problems on the oil manufacturer. Higher temperatures had led to the introduction of new types of bearing alloys, which were corroded by some lubricants under service conditions. At the high temperatures in the engine the lubricating oil formed acids which corroded the bearings. Later experience had shown that reduction of the crankcase temperature eliminated that corrosion problem. In some cases the oil refineries had developed inhibitors and methods of correcting corrosion. Other problems were sludge or oxidation. High temperatures meant more oxidation and more sludge. The oil refineries had partly met that by solvent refining, and other refining methods, which removed some of the sludge-forming materials. Another problem was that of viscosity index. In Great Britain that was not so great a problem as in the United States where the temperature might vary from −30 deg. F. to +50–60 deg. F. within a week's time. To obtain an oil which had a proper fluidity over that temperature range had necessitated the introduction of viscosity index improvers, or solvent refining, which also increased the viscosity index of the oil. Another problem was that of the flow of the oil at low temperatures, and for that purpose inhibitors had been introduced. Still another problem was that of oiliness. In America they believed they knew enough about it to begin to add oiliness agents to the lubricating oils. Further, the desire to manufacture cars with very small gears, to reduce unsprung weight, and to reduce the cost of the vehicle, had meant larger loads, which resulted in the introduction of extreme-pressure ingredients. Most of those addition agents were applied to automotive lubricants, but in the future many "dopes" would be used in industrial oils.

Dr. B. KJERRMAN (S.K.F., Göteborg, Sweden) remarked that Mr. Cooper in his paper said that the consistency of a grease must not be so low as to permit escape from the housings and not so high as to cause channelling under working conditions. The speaker pointed out, however, that where grease was put in continuously in small quantities it certainly would not channel, but where it could not be continually renewed it often did channel. He did not agree with Mr. Cooper's statement that a good ball-bearing grease should not channel. If it did not channel it would develop an extremely high temperature at no very high speed. Because of that a lubricating system which worked continuously had to be very carefully regulated, especially if the speed was high. Mr. Cooper also stated that "for low speeds and temperatures the consistency should be a little greater than that of vaseline, increasing as the conditions became more arduous." He would like to know what Mr. Cooper meant by "vaseline." Mr. Cooper further stated with reference to the high-speed grades of soda-soap greases: "They also have an increased tendency for oil separation to occur under working conditions. This separation must only be slight, but an amount sufficient to provide a film of oil over the working surfaces of the bearing is obviously beneficial for high-speed applications." What Mr. Cooper said was quite true, but personally he had never seen such a grease. Some oil could work out of the grease under capillary action, but he had never come across any instance of good ball-bearing greases leaking.

Professor G. B. **Karelitz** (Columbia University, New York) said that as soon as the speakers that morning began to talk about extreme-pressure lubrication, they entered into some unknown factors of interconnexion between surfaces and the lubricant itself. Quite unexpected phenomena were found which had to be understood before one could talk intelligently about the application of special lubricants. He understood from Mr. Geniesse that the oil manufacturer had not time to wait for the physicist to tell him what to do because he had to sell his products; but it was necessary now to start to develop that physical foundation.

Mr. W. N. **Bray** (British United Shoe Machinery Company, Leicester) said that of recent years there had been a growing tendency to introduce machinery which was controlled by hydraulic pressure; and therein lay certain new problems which required investigation. His own company used very largely small sleeve valves for oil control. Those valves did not move at high speeds, but they had to fit extremely accurately. His company could not afford to have any wear taking place if they wanted to obtain a reasonable life for the valve. Therefore, amongst other things, the oil they employed had to be a good lubricant. In addition, there was the question of viscosity because they had to get a very quick response in fairly narrow channels in the system; and obviously they could not have a sticky oil. Furthermore, they wanted an oil the viscosity of which did not increase a great deal with a drop in temperature, because they had long channels which were often full of oil, and when they started the machine up from cold, they wanted to get an active control of blast; they did not want to wait until the machine warmed up.

There was a third problem. If oil in a pipe was flowing through a valve, and then the valve was suddenly shut, a compression wave was formed which ran back along the pipe. If there was a rapid series of turning on and off of the valve there was a possibility of a synchronous vibration building up that might interfere with the working of other parts. To make the problem calculable it would be necessary to know both the density and bulk modulus of the oil, and he believed that figures for the latter were not generally recorded by the manufacturers. He would be grateful for any information about special oils which had been produced for pressure control.

Professor Karelitz had referred to the particles laying themselves down in the direction of motion. He himself had found, in special cases of pulleys running on pins at a high speed, that seizing was avoided by grinding the pins so that the direction of the travel round the pin lay in the same direction as that in which the pulley was going around the pin. That presumably gave the same laying-over effect of the particles when running in the bearing as was pointed out by Professor Karelitz.

The CHAIRMAN, referring to the remarks of Mr. Bray, said there was also the question of the class of oil used in the remote control of the steering gear of ships. No reference had been made to that, and information on the point would be welcome.

Mr. B. PRINGLE (B.T.H. Company, Rugby) observed that, although it was fairly easy to define what a grease should be for ball bearings, it was a little difficult to say what precise greases fulfilled that function. He believed that Mr. Cooper, when he referred to greases of the consistency of vaseline, was thinking possibly of the lime-base greases. The use of grease, particularly in the ball bearings of electric motors, had been very extensive for reasons relating to its retention in housings; but the problem of effective lubrication by grease became more difficult the larger the bearing. In the larger sizes there were very high peripheral speeds between the cage and the race, and lubrication at that point was very difficult. The lime-base greases did come in very intimate contact with all working parts because they were usually softer and more pliable than the soda-soap greases. Whilst tracking rarely occurred with a lime-base grease, it was more likely to occur with a soda-soap base grease. Other problems became more acute as the quantity of lubricant increased in the housing. That was why efforts had been made to load soda-soap greases with oil so that whilst the relative stiffness of the grease retained it in the housing, it exuded thinner lubricant to get into the working parts.

Mr. S. J. AYRES (Hoyland Common, Barnsley, Yorks) said that his company had a number of aerial ropeways working on the bi-cable system, the carrying ropes being locked coil wire ropes and the hauling ropes being of Lang's lay construction. The carrying ropes worked under rather severe conditions, being subjected to heavy strains and exposed to the weather. The rope lives had increased from 300,000 tons net carried, to 500,000 and even to 700,000 tons. That was no doubt due in the main to improved lubrication. At the start, the ropes had been lubricated infrequently, but in the past few years an oiling car travelled over the line daily. Failure had been due in almost all cases to corrosion and fatigue on the inner surface of the outer locked coils. That was obviously due to weather penetration when under load and was in agreement with Dr. Hogan's results. He agreed with Dr. Hogan that it was difficult to ensure that the oil penetrated the outer locked layer, but it should be possible to do so as the moisture did penetrate. Graphited greases were being used in the manufacture of ropes, and his company had adopted a graphited oil. No results were yet available, though they had fewer broken strands in the past year than in any previous year.

Metallic coatings, such as zinc, appeared attractive so long as the coating did not crack. The splicing of the rope might be a source of weakness, as it was difficult to carry out a galvanizing process in the field. The findings of Messrs. Nixon and Jackson, that little internal corrosion had been found in ropes used on the Witwatersrand, were remarkable considering the various lubricants. It certainly showed that the conditions met with were very favourable and not the same as in Great Britain. If that was not so, then the use of a bitumen-base dressing was a solution of the problem.

Mr. J. CARTER (Glacier Metal Company) gave an account of an experiment which had been carried out with a view to finding the abrasion point. Whether the particular material was lubricated or not, it yielded the same characteristic curve ; it mattered not whether it was a hard material or a soft one. Some material had been run for twenty-four hours without lubrication at all, and it was found that, at a pressure of between 610–810 lb. per sq. in., there was a vibrational period. The loading was variable by a matter of 30 lb. each five minutes of running. One could continue from the starting of the vibration until the vibration stopped, and abrasion did not take place in some materials until there was more than double the maximum vibrational loading to which the bearing was subjected. Beyond 610 lb. per sq. in. there was hysteresis due to the small asperities becoming flattened out, or to some other phenomenon. Lubricating oil was supplied at the rate of 7 drops to the minute and distributed over the bearing surface by means of a felt pad. The same characteristic curve had been obtained on a dry piece of metal without lubrication.

Mr. A. E. HOPE (Shell-Mex and B.P. Company) remarked that the Reporters mentioned that there had been a revival in the use of grease as a lubricant. Four or five years ago, one plant of which he knew was making about 2,000 tons of grease a year. Since the revival due to specialized lubrication—particularly of the automobile—the output had gone up to 5,000 tons a year.

Mr. Richardson in his paper seemed to consider lime-base greases to be unsuitable for anti-friction bearings, and stated that they were not satisfactory lubricants, since they must contain water, and when that water evaporated in use they became unstable and the oil separated from the soap. That was not so. There were various lubricants on the market which, when the water had been removed, were perfectly stable. In fact, one of the tests applied to such a grease, was to heat it, not only to just above its melting point, but up to the flash point of the oil (over 400 deg. F.) and then to let it stand for twenty-four hours. After that time—in fact, after many hours—there was no separation of oil, though the grease was anhydrous.

A previous speaker had said that a lime-base grease was a better lubricant in ball bearings because of its intimate contact. It did not track. On the other hand, soda-base greases, which some people preferred, had a long fibrous structure, and probably it was on that account that tracking took place. He had known of a case where a roller wheel bearing had fused, although the hub had had some soda-base grease in it. The grease did not melt and prevent fusion even at that temperature. A lime-base grease which did not separate when the water had gone was a better lubricant than a fibrous soda-base grease; it was only a question whether the bearing closure would hold the grease. An earlier speaker—an engineer—had thrown down a gauntlet to the oil men, but he himself would like to retaliate and ask engineers to provide better bearing closures which would hold the grease even though softened under heat, because, when it cooled again, it would still be a grease even though the water had gone. He believed that certain manufacturers of anti-friction bearings did not guarantee their products if run at temperatures over 180 deg. F. The lime-base greases which he had in mind would not melt at that temperature, so that if the bearing was used properly and had an effective closure, lime-base grease was the proper lubricant.

Lt.-Colonel S. J. M. **Auld**, in reply, said he had been disappointed at the discussion on extreme-pressure lubrication, which was so important that he would particularly invite written contributions on the subject. He had been especially struck with a remark made by Mr. Southcombe in connexion with the behaviour of extreme-pressure lubricants in use, because it showed how little one visualized what was actually happening. Mr. Southcombe had pointed out, very rightly, that people were beginning to use terms of film strength in connexion with extreme-pressure lubrication, whereas that was not correct. All that extreme-pressure lubrication did was to allow of gradual wear. There would always be wear, but there was increased reduction of the points of contact which made for real trouble. The element *par excellence* which produced the extreme-pressure effect was sulphur, either alone or in combination with chlorine, phosphorus, etc. He did not say that was the end of it all, but, in the meantime, sulphur seemed to be the basis. Sulphur was the typical anti-welding material, but that only begged the question as to what was meant by anti-welding. Apparently, it was simply the ability of the extreme-pressure material not to form a skin but rather to give a chemical action whereby small particles were torn from those asperities which otherwise would cause seizure. That picture could easily be translated into practice. It was a helpful hypothesis to use in searching for materials, because, although it was stretching it rather far to be fully in accord with one motor manufacturer who refused to consider extreme-pressure materials unless they were corrosive, it was fairly certain that, at the local temperatures under high-pressure conditions, some sort of chemical reaction of the type in question did occur. What, then, must take place was that the sulphur, or other element, had to be provided in such a form that it was present at the right moment and in the right manner; and with sulphur, it was probable that one would always ultimately get some form of polysulphide (he was now talking about free sulphur), but where sulphur itself was used in cutting oils, it was very likely that one must have a polysulphide effect. A most interesting observation was that the sulphurization of an oil depended on the crude oil which was used, and that very frequently it was the oils which already contained sulphur which reacted most effectively with the new dissolved or loosely combined sulphur; and something of the kind which he had described would take place.

Mr. E. A. Evans said that he was very glad that Colonel Auld had emphasized the importance of extreme-pressure lubricants, because when Colonel Auld and himself had prepared their report, they were hoping that this subject would almost take priority in the discussion, because of its importance. How the extreme-pressure lubricants reacted no one knew, but it was quite clear that no sulphide was produced on the surface. One paper did state that sulphides were produced. He had tried to ascertain what did really happen on the surface, and he had sought assistance from Professor Finch, who had utilized electron diffraction. Professor Finch was not able to be present and though it was not within his province to give the results of that work, he could state that sulphides were not produced. Some other action took place. Perhaps he might be allowed to quote the last sentence from his own paper on "Extreme-Pressure Lubricants":—

"The numerous references to extreme-pressure lubricants in the patent literature testify to the growing interest and importance of the subject. The vast array and diversity of compounds which have been patented, gives a clear indication that little is known about the chemical behaviour in extreme-pressure conditions, and how the additive agent attacks the metal surface."

In the patent literature alone, quite apart from any chemical literature, quite 200 entirely different types of compounds were mentioned. Of what use were all those products ? One could only conclude that some other compound was being produced on the surface and acted as a film.

A word of warning might be given with regard to extreme-pressure lubricants. People said "These things are all very well in theory, but have they come to stay ?" The answer was "Yes", provided that the temperature range in which they were safe to use was stated. The limit of temperature with copper or copper alloys was 85 deg. C., but in his own paper mention was made of the temperature limit as about 100 deg. C. for copper and copper alloys and about 130 deg. C. for steel (that was the running oil temperature). Extreme-pressure lubricants ought not to be criticized if they failed under conditions for which they were not intended.

In the discussion, reference had been made to skin troubles arising from the use of oils. There had been a scare, particularly in Lancashire, but the trouble was not due to oil, but to dirt. If people would only wash themselves properly there would be no need for those troubles to arise. One should use plenty of soap, rubbing the hands together in order to form an emulsion, and then that should be washed off gently with water. If that were done, no more skin troubles would occur.

One speaker had referred to water separation from grease. He understood the speaker to say that if one heated the grease to its flash point, water would be driven off. He did not think that that was so. Emulsions did not break up necessarily when heated, and he thought the remark by Mr. Cooper, that the water was essential for the stability of the grease, was justified.

The temperature at which ball bearings ran with lime soap greases had been referred to. The important factor was not so much the stability of the grease as the softening point of the grease.

With regard to vibration, he did not connect it with the quality of the oil so much as with its viscosity.

For aerial rope lubrication, they would have to wait for more work to be done on the subject to find an answer.

Dr. B. **Kjerrman** did not agree with Mr. Evans that high-pressure lubricants did not form sulphides. Personally he did not know about the high-pressure lubricants, but he knew about cutting oil used in turning. His own firm put sulphur in that oil and, during turning, sulphides were formed on the chips, which increased 100 per cent and more in sulphur content. That was a danger, because his firm used the scrap in their steelworks.

A vote of thanks, moved by the CHAIRMAN, to the two reporters, Mr. Evans and Colonel Auld; to the authors of the various papers; and to all those who had taken part in the discussion, was carried with acclamation, and the meeting then terminated.

Communications

Mr. H. H. BEENY (Messrs. Alfred Herbert, Ltd.) wrote that Mr. Lloyd and himself fully agreed with Mr. Oldacre that clean habits on the part of the worker were of first importance in avoiding oil-dermatitis, but with a contaminated oil, there must always be an increased danger of carrying infection from one worker to another and sepsis might be set up if the skin was even slightly cut or damaged. It was the practice of their firm to treat all new cutting oil entering the works with 1 per cent of an oil disinfectant, which had been proved by test to kill *Staphylococcus aureus* and other pathological organisms; in addition, used oils were first heated to 140 deg. F. and then centrifuged to remove dirt and sharp metal particles, before being returned to the machines. Those safeguards had kept the works remarkably free from sepsis and oil infections and were felt to be of real importance.

Mr. N. E. F. HITCHCOCK (Messrs. C. C. Wakefield and Company, Ltd.) wrote that in his description of internal lubrication, Dr. Hogan stated that it was seldom possible in practice to replenish the internal lubricant of a wire rope, and in the next paragraph he said that the primary function of the externally applied lubricant was to replenish the interior. The conclusion evidently was that lubrication was seldom of the first order. Furthermore, he stated that excessive lubricant was a disadvantage. Dilution of the external wire rope dressing with a fairly volatile penetrating agent might conceivably cure both those evils.

Regarding types of lubricant, whilst it was agreed that wax was of relatively small value in wire rope lubrication, primarily because of its poor protective value, the statement that it had little or no lubricating quality, should not pass unchallenged. One of the most effective lubricants for miniature wire ropes of the Bowden type was wax or a mixture of wax and graphite.

Reference was made to the conflicting properties of the ideal lubricant for wire ropes, namely, satisfactory adhesive power and resistance to pressure. It might be possible to overcome that difficulty by incorporating one of the extreme-pressure agents mentioned in other papers (e.g. by Mr. Evans) in a material possessing the necessary adhesive powers. Messrs. Auld and Evans, in their summary of the papers in Group III, asked for information concerning the rates of hydrolysis of sperm and seal oils. Experiments undertaken by the writer appeared to show that the rates of hydrolysis of sperm, rape, and seal oils were in that order, the first named being the greatest.

Dr. Hogan might care to elaborate his statement: "However, the oil (seal oil) also penetrated within the rope and leached out the mineral oil placed there during manufacture, leaving the wires unprotected and exposed to internal corrosion."

Another possible line of attack which was not suggested in the paper, was the use of a material capable of forming water-in-oil emulsions. Such a material could be confined to the wire ropes used in wet shafts.

Professor H. LOUIS (Newcastle upon Tyne) wrote that when concerned with gold mining in West Africa in about the year 1878, he found that the usual greasy lubrication interfered with the amalgamation on the copper tables. By substituting soft soap containing about one-third of its weight of graphite for the semi-solid grease used previously, the trouble was overcome. Later, when investigating an explosion in an air compressor, it was concluded that part of the lubricating oil was being cracked in the long and tortuous passages of the compressor, giving rise to an explosive gas. The difficulty was surmounted by the use of soft soap and graphite, which answered admirably.

Mr. D. F. PILKINGTON (Transport Officer, Lancashire Associated Collieries) wrote to ask whether any testing had been done or results had been published on the lubrication of leaf springs; there was a question of grease, ordinary oil, penetrating oil, graphite, or any other kind of dope and inter-leaves. The last item might not be called a lubricant, but it acted like one and, therefore, presumably should come under discussion.

Mr. C. G. H. RICHARDSON (Messrs. Ransome and Marles Bearing Company, Ltd.) wrote that Mr. A. E. Hope was mistaken in stating that he (the writer) had represented that lime-base greases were unsuitable for anti-friction bearings. That was not so; but it was stated that they were considered to be suitable for only comparatively light duties and were not suitable for high temperatures and pressures. Those observations were based on long experience and extensive tests, and appeared to be borne out by Mr. Cooper (vol. 2, p. 43) and Dr. Kjerrman (vol. 2, p. 341). A lime-base grease having characteristics approaching those mentioned by Mr. Hope had not been met with by the writer and, in any case, a test such as that described by Mr. Hope was valueless unless it was conducted in conjunction with actual running tests.

Mr. H. A. STURGESS (Menno Compressed Air Greasecup Company), in a written communication, said that in the paper by Messrs. Catterall and Maitland, a conclusion to be drawn was the difficulty of securing a proper and efficient method of application of grease lubricant to bearings. Various methods were mentioned, but no reference was made to the method of feeding by compressed air greasecups. With a cup of that description, automatic feed of a sensitive nature was provided. The latest design of Menno cup ensured absolute reliability of action under the control of compressed air. A very important feature of that cup was that its action—unlike that of spring-operated cups referred to in the authors' paper—had no deleterious effect on the quality or consistency of the grease lubricant.

Professor F. C. THOMPSON (University of Manchester) wrote that he was interested in Mr. Southcombe's suggestion that in the drawing of steel wire with a "water-coat" the rust itself acted as a lubricant. His own impression was that the rust was merely the sponge which held together the soap or other lubricating material which was used. At the moment, however, it was not possible to differentiate between the two points of view.

Mr. W. A. TUPLIN (Messrs. David Brown and Sons (Hudd.), Ltd.) wrote that from the remarks made by more than one speaker at the Discussion, it appeared that the hypoid gear held the doubtful distinction of having led to the development of extreme-pressure lubricants because, as an automobile rear axle drive, it would not work satisfactorily without them. Actually, the surface pressure on the teeth of a hypoid gear was lower than that on the teeth of a spiral bevel gear of comparable dimensions. On the other hand, the sliding velocity was higher in the hypoid gear. It seemed, therefore, that it was the combination of surface pressure and sliding velocity of steel on steel that demanded a special type of lubricant. That was in accordance with the opinions expressed to the effect that "extreme pressure" lubricants did not possess any particularly high "film strength" but that their virtue lay in permitting initial bedding-in of the surfaces to take place by relatively rapid wear without risk of seizure. The conclusion appeared to be that extreme-pressure lubricants were misnamed, but the description was now so well-known that it was probably too late to attempt the substitution of a more accurate one.

GROUP IV. PROPERTIES AND TESTING

MOLECULAR FORCES IN FRICTION AND BOUNDARY LUBRICATION

By Professor N. K. Adam, Sc.D., F.R.S.*

This paper contains only theoretical, and to some extent speculative, considerations on the molecular mechanism of friction and lubrication, based partly on experimental work by the author and by others on the properties of molecules, especially at liquid surfaces, and partly on a study of other people's work on lubrication. It may, however, be of some service as indicating possible mechanisms for the very important operations of the lubricant and for the friction and seizure which takes place if the lubricant fails.

Every solid surface has, extending outwards from its surface, a field of attractive force due to the atoms present in the surface. The attraction is very powerful, and also of very short range; it extends probably only from 1 to 3 Ångstrom units (Å)† away from the surface, and is imperceptibly small at greater distances. The attraction is due to the "affinities" of the surface atoms, the same forces which produce cohesion within the solid; these forces are due, of course, to the complex interactions of the electrically charged particles (protons and electrons) which compose the atoms, and as a rule, though not invariably, they decrease in intensity approximately according to an inverse sixth power, as the distance between atoms is increased. This accounts for the intensity of attraction at extremely short range and its disappearance at slightly greater distances.

Sir W. B. Hardy's work showed that when two solids really touch, i.e. any parts of their surfaces approach each other within the exceedingly small range of these attractive forces, they "seize" or weld together at the points where they really touch; and when the solids are made to slide over each other, the points where the solids had welded together are forcibly torn apart; breakage occurs usually at a depth different from that which constituted the original surface of the solids, and the surfaces are badly torn and scored. It is probable that this happens, in engineering practice, whenever solid surfaces really touch, in the sense that any parts of their surfaces approach within a distance of an Ångstrom unit or two of any other solid surface.

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* Professor of Chemistry, University College, Southampton.

† 1 Å, or Ångstrom unit, is one ten-millionth of a millimetre, 10^{-8} cm.

Real contact in this sense is, fortunately, a rare occurrence, at least over any considerable area of a surface. There are two reasons for this. Firstly, no known solid surface can be produced plane to anything like this degree of accuracy; if a surface is optically plane, even to one-tenth of a wavelength of sodium light, there will still be irregularities of the order 200 times the distance to which the attraction of the surface extends. This means that, no matter how carefully trued a surface may be, it will not touch another surface, except at a few very highly localized points, sufficiently closely for this spontaneous welding to take place. Nevertheless, if the surfaces are clean, such real contact will occur at a very few points, which may each have an area possibly of the order a square tenth of a millimetre in size. Second, most solids have on their surfaces invisible films of contamination, which prevent the real contact of the substances of which the solids are composed. These films are not easy to remove, except by scraping off and exposing an entirely fresh surface of the metal (or other solid). The films are held by the same attractive forces which would cause seizing on simple contact; they may consist of "adsorbed" oxygen, or of oily matter or moisture; and to some extent, but probably not completely, these films, whose presence is accidental, mitigate the intensity of seizure, because they satisfy to some extent the attractive forces of the surface atoms which would bring about seizure.

When two surfaces, well-trued and clean according to the best engineering standards, are caused to slide over each other under moderate pressure, the minute elevations of each surface touch the other surface, at very sparsely distributed points. Welding (seizure) occurs, practically instantaneously, with the formation of minute bridges of strength equal to the solids themselves, between the surfaces. As sliding occurs, these bridges are being continually broken and re-formed, and as they may break anywhere, not usually at the point which constituted the original surfaces, the effects appear as scoring and disintegration of the surfaces. The continual formation and breakage of the bridges produces much heat, which may melt the surfaces locally, and if the temperature rises high enough, the surfaces may flow sufficiently for real contact to occur over large areas, and the bearing may "run" or "seize" completely. The speed of formation of these locally welded bridges is surprising; suppose the surfaces of a bearing move at a speed of 100 cm. per sec. relatively to each other, and the bridges are 0·1 mm. across * ; then the time of contact at any point must be of the order 10^{-4} seconds, during which the bridge is formed and broken. Since kinetic friction differs but little in amount

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* The *area* of real contact at any point is a matter of speculation at present, so that these figures refer to probable order of magnitude only.

from static friction, the formation of the bridges must be almost as perfect at this, or even higher speeds, as when the surfaces are stationary.

The problem before the lubricating engineer is to prevent the sliding surfaces from ever coming within each other's range of molecular attraction. If possible, this is done by "complete", or "film" lubrication, the surfaces being separated by a film of liquid many thousands of times the depth of the field of attractive force. The difficulties in the way of so completely separating the solid surfaces increase, as is well known, the greater the pressure between the surfaces and the smaller their rate of motions past each other. "Oiliness", or the power of producing "boundary" lubrication, consists in covering the surfaces with a closely adherent film of one, or at most a few, molecules in thickness, which will tend to prevent real contact between the metals, even at the minute areas elevated above the surface generally.

It has been shown by Langmuir and his colleagues that a layer of a long-chain fatty acid, such as oleic acid, only one molecule thick, will reduce the coefficient of friction of glass from unity to about 0·13, and layers of three molecules thick will reduce it still further. It can easily be seen how such a layer protects the metal from destruction by seizure. The molecule of oleic acid (or of other fatty acids) consists of an acidic end group which, by reason of its chemical constitution, adheres strongly to the metal surface (this is shown by the increased heat developed when a metal surface is wetted by an oil, if fatty acids are added to the oil); and also of a long, and flexible, hydrocarbon chain containing carbon atoms each with hydrogen atoms attached. This part of the molecule is some 20 to 30 Å long, and is similar to the paraffins in structure and properties. If the surface is covered by a film, even only one molecule thick, of such molecules, the contact between the elevated parts of the surface is between the hydrocarbon parts of the molecules of this adhering "boundary" film, instead of between the metals. It is known that the adhesion between two hydrocarbon molecules is much smaller than that between metals; in other words the tensile strength of the bridge formed between the surfaces where they really touch is very much less between the outsides of the film molecules adhering to the surfaces than elsewhere. Hence, normally, the bridges formed between the surfaces will break without disturbing the adhering films, and no damage will be done. Thus one function of the molecules in the boundary film is to provide a weak link in the bridges necessarily formed between the surfaces on contact, thereby protecting the underlying metal.

Again, though the evidence is not yet complete, there is some probability that the flexibility of the molecules in the boundary film is important. This can be regarded in the following way. The molecules of boundary lubricant are probably standing upright, or at any rate at a steep angle, to the solid surface. If they are flexible, they can yield to the pressure of an elevated portion of the other surface moving across them, provided that this elevation is not greater than the total length of the molecules in the boundary films on both surfaces. They may thus function as a yielding cushion, preventing the projections on one surface from hitting those on another, or at least diminishing the chance of this taking place.

The permanent preservation of a boundary film is of at least equal importance with its properties when it is present. Here probably three points are desirable. First, there should be present in the lubricant a reserve supply of the molecules of the substance forming the boundary film, so that repairs can be made quickly at any point where the film is accidentally rubbed away. As a rule, the slowest stage in the repair of a boundary film will be the diffusion of molecules to the place where they are needed; a problem of transport rather than of fitting, for the adhesive field of force of the solid will fix molecules in place practically instantaneously if there is a gap in the film and if the molecules are there to fill the gap. Next, the chemical characteristics of the molecules chosen for boundary lubrication should be such that, once fixed in the boundary layer, they should not easily be displaced from it. This means a high adhesion to the solid surface, in other words a powerful chemical affinity between the end group of the molecule forming the anchorage to the surface, and the metal. Also a high boiling point, generally involving a heavy molecule, is desirable, so that the heat generated in a moving bearing shall not easily displace the film. The last two properties in the film molecules are closely interconnected; but what is probably required is not so much a high boiling point in the boundary lubricating substance in bulk, as a resistance to being *driven or boiled off the surface;* and this may not be exactly the same as the resistance to evaporation from a mass of liquid. Further, of course, chemical stability must not be forgotten, for if the boundary lubricant is decomposed it is useless, perhaps worse than useless; but the importance of chemical stability in lubricants of all classes is so well recognized that there is little danger of this being overlooked.

Something along similar lines may also be said about graphite as a lubricant. Graphite consists of plane sheets of carbon atoms, of indefinite and large extent, exceedingly tightly linked together in the sheets; each sheet is, however, rather loosely attached to the next sheet parallel to it, and sliding or separation can easily take place. It is known, from studies of electron diffraction, that graphite is easily attached to metal surfaces, these sheets being parallel to the surface; the sheets afford protection against seizure between the metals. When real contact occurs, it will be between the outside of the sheets of graphite and not between the metals. When separation occurs, the graphite is the weakest link in the bridge, and breakage occurs in the graphite and not in the metal. This tends to flake the graphite into tiny, flat particles, which will remain in the oil and be available for repairing any gaps accidentally formed in the layer of graphite covering the surface. Therefore, while graphite alone is valuable as a lubricant, the combination of graphite and oil is far better, for even apart from the lubricating power of the oil, which is of course considerable, the oil acts as a carrier of flakes of graphite to wherever they may be required, and thus tends to preserve the protecting covering.

To summarize, it may be said that the main function of "boundary" lubricating films is probably to provide weak links in the junctions or bridges formed between the elevations on the surfaces, at the points where contact is close enough for the molecular attractions to operate between the surfaces. Subsidiary requirements are that the molecules forming the films must adhere tightly to the surfaces, so as not to be easily displaced by heat, or mechanical means; and that a reserve supply of film-forming material, dissolved or otherwise distributed in the oil, should be available for repairs to the films.

SLUDGE AND DEPOSITS IN TURBINE OILS

By W. O. Andrews*

One of the major troubles in the lubrication of turbines is the formation of sludge and deposits in the oil pipes and coolers. The following notes have been prepared from the results of an investigation into the ways and means of preventing these deposits, with special reference to the quality of the oil used. Small-scale tests on various mineral oils have been followed up under practical conditions by observations on 48 steam-driven turbo-alternators and air compressors.

The main cause of deposits, apart from foreign substances such as dust, rust, and loose paint (from new plant), is the partial oxidation of the oil. This is hastened by the high temperatures to which the oil is subjected and by the rapid circulation and aeration of the oil, which may be in continual use day and night for months at a time.

There are two kinds of deposit. One, usually dark brown in colour, varies in consistency from a thick slime to a gummy varnish. It settles on the bearing housings, in the claw-couplings, and in the oil pipes. More particularly it is deposited on the tubes of the oil cooler, resulting in poor heat transfer and a general increase in the oil temperatures. This, in turn, increases the rate of oxidation of the oil and with it the rate of formation of further deposit. The conditions grow steadily worse until anxiety is felt about the high bearing temperatures. At this point it may become necessary to shut down the plant, to enable the oil coolers to be cleaned—often a difficult and unsatisfactory process.

The other kind of deposit is a spongy sludge which is formed when the oil is in bad condition and when, as often happens, small quantities of water or steam find their way into the lubricating system. It consists essentially of an emulsion of water and oil and frequently causes considerable trouble by clogging the passages and coolers. It may even clog the screens, in which case there is danger of starving the oil pump. Some of it usually settles in the dead spaces, including the reservoir tank, where it may be drained off, a wasteful process, as oil is lost.

The property of an oil of combining with water to form an emulsion is not constant, but gradually changes with the age of the oil. A new well-refined turbine oil will separate readily from an emulsion with water, but the readiness with which this emulsion will break up or separate again into its constituents gradually diminishes with use.

This property of separation constitutes a delicate criterion of the

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* Victoria Falls and Transvaal Power Company, Ltd.

condition of an oil, and has been employed by the author's firm for some time as a routine method of gauging the condition of turbine oils in service. Samples of oil are taken periodically from each machine and amongst other tests are subjected to the following emulsion test, known as the "R.L. emulsion test".

R.L. Emulsion Test. A sample of the oil is shaken up with water in a bottle and then allowed to stand. The rate of separation of the mixture into its constituents is noted and recorded for reference. A high rate of separation indicates that the oil is in good condition and a low rate of separation shows that changes have taken place in the oil, due to oxidation, which may render it unfit for further use. Details are as follows:—

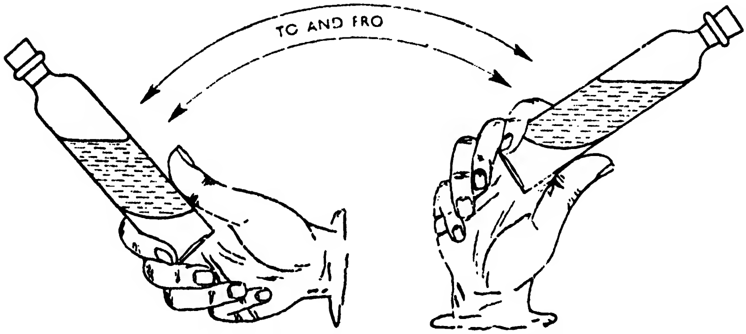

Fig. 1. Correct Method of Shaking to form a Water-in-Oil Emulsion

Distilled water is poured into a clean 2 oz. bottle (1 inch in internal diameter) up to the level of 40 mm. from the bottom (measured inside the bottle). The oil to be tested is poured on the water to a depth of 30 mm., thus making a total height of fluid in the bottle of 70 mm. The bottle containing water and oil is then shaken up to form a water-in-oil emulsion. Two different kinds of emulsion can be produced, namely water-in-oil and oil-in-water. These two types can be formed at will according to the method of shaking the bottle. For the purpose of the test it is important to obtain the correct type (water-in-oil) or the results are useless.

The quickest method of producing this emulsion is to shake the bottle, not violently, with a backward and forward motion perhaps better described as a jerk. The bottle is held near the bottom and gently jerked to and fro as shown in the sketch (Fig. 1). Each time the bottle is moved thus, a small amount of the water is projected into the oil and is suspended there for the time being, the continual motion preventing it from settling again. After about 5 minutes' agitation all the water is incorporated or distributed in the oil and the bottle may be shaken quite violently in a vertical direction to obtain the finest division of the water globules. The mixture generally takes on the appearance and consistency of cream which clings to the side of the bottle.

Another, but more tedious, method of obtaining the desired emulsion may be used. The correct amount to give 30 mm. of oil is put into the bottle. Five drops of water are then added and the bottle corked and violently shaken in a vertical direction. Five more drops are then added and the violent shaking is repeated. This process of adding water and shaking is continued until the full quantity of the water (40 mm. in height) is used.

Meanwhile a small water bath is heated on a hot plate to 130 deg. F. (chosen as a normal operating temperature for turbine oils) and maintained at that temperature to within 2–3 deg. F. The bottle containing the emulsion obtained by one or other of the methods described above is placed in the water bath for ten minutes in order that it may attain a temperature of 130 deg. F. During this interval the emulsion is prevented from breaking by shaking occasionally with a violent motion.

The bottle is given a final shake, and stood upright in the water bath. The emulsion gradually breaks into three layers, oil on the top, water underneath, and a layer of unbroken emulsion in between. The depth of oil separated on top is measured at intervals of one minute (or five minutes if the separation is slow). From the observations the average rate of separation for each period counting from the time the bottle was allowed to remain stationary is calculated in millimetres per hour and the highest number in this series is the figure required.

As the two kinds of emulsion have entirely different properties, it is important to distinguish the one from the other in order to confirm that the correct type is used for the purpose of the test. Fig. 2 illustrates the behaviour of the two emulsions. A shows a bottle containing the water-in-oil emulsion immediately after formation. B shows how, on standing, the mixture separates into the three layers of oil, emulsion, and water. C shows an oil-in-water emulsion immediately after formation, while D shows the rapid separation of this type into its constituents. Bottles B and D have been allowed to stand and separate for the same period of time. The second or incorrect type was obtained by placing the full quantities of oil and water in the bottle and shaking violently in a vertical direction.

In cases of doubt the type of emulsion can always be checked by examination under a hand lens in order to determine the nature of the suspended globules. If the emulsion consists of water globules suspended in oil (correct type) the globules will gradually sink, but if the emulsion consists of oil globules surrounded by water (incorrect type) they will gradually rise.

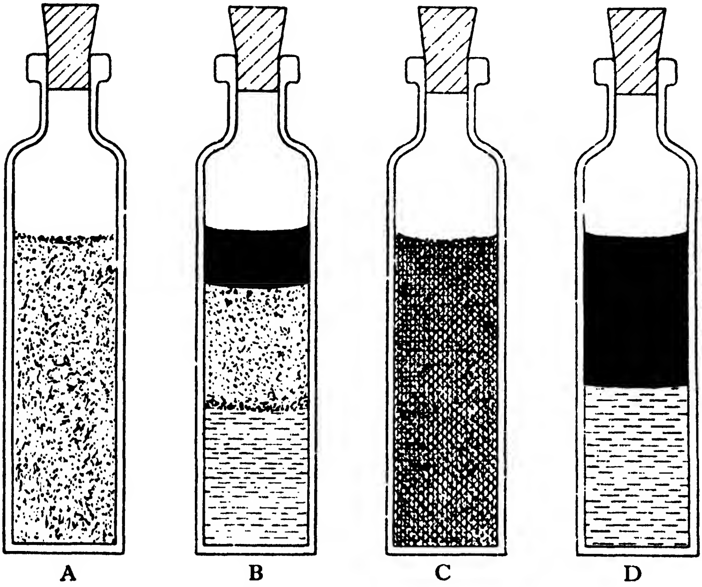

Fig. 2. Types of Emulsion

A Water-in-oil emulsion on formation.
B The same emulsion after separation.
C Oil-in-water emulsion on formation.
D The same emulsion after separation.

Typical results showing the effect of age on the separation according to the R.L. emulsion test are given in Table 1, which refers to turbine oils which have been in actual service.

TABLE 1. RATE OF SEPARATION OF OIL FROM EMULSION

Oil	1	2	3	4
Separation: new oil, mm. per hr.	400	1,400	780	700
Separation: used oil (after 1 year), mm. per hr.	5	150	130	0
Viscosity: sec. Redwood, at 140 deg. F.	112	80	70	78

It will be seen that in all cases the separation of the used oil was only a fraction of that of the new oil.

No. 1 oil tested after a year's run in a turbine gave a figure of 5 mm.

per hr. showing that it had practically lost its power of separation from water. Soon after making this test, water (as steam) gained access to the lubricating system of one of the turbines in which the oil was used. A practically permanent emulsion was formed from which very little oil could be reclaimed. The operating experience thus gained showed that the emulsion test figures were a good guide as to what might be expected if water accidently gained access to the system.

Oil No. 2 is of medium viscosity, and gave excellent results in practice, while oil No. 3, of light viscosity, also gave satisfactory results in service. No. 4 is an example of a good oil which has been subjected to very exacting conditions as regards heat and aeration. It was used in a hydraulic coupling where temperatures were high and aeration excessive. It will be seen that after this severe punishment, which extended over 12 months, it had completely lost its power to separate from water.

From observations carried out over a number of years it has been found that a used oil may be considered in satisfactory condition if it contains no deposits and separates from a water emulsion at a rate of not less than 50 mm. per hr. when submitted to the R.L. emulsion test.

Accelerated Ageing Test for New Oils. Although the above emulsification test gives a very good indication of the condition of a used oil it does not indicate how a new oil will behave after being in service for some time. In order to cover this point a further test has been devised which has proved a valuable guide in selecting a new oil for turbine use. The method of test consists of heating the oil in contact with air for a fixed period. The resulting oxidation causes an artificial ageing of the oil, and the quality of the oil is then tested by means of the emulsion test already described. Details are as follows.

Fifty cu. cm. of the unused oil are placed in a glass beaker of 3 inches internal diameter. A copper plate about $2\frac{3}{4}$ inches × $1\frac{1}{2}$ inches × $\frac{1}{16}$ inch is polished and placed lengthwise, standing on edge, in the bottom of the beaker so that part of the strip is in oil and part exposed to the air. The copper strip is introduced in order to accelerate the oxidizing action of the air and thus shorten the test. The bulb of a thermometer is then placed in the oil and the beaker, loosely covered by a clock glass, is heated by means of a hot plate to 302 deg. F. (150 deg. C.). The temperature is maintained as closely as possible by moving the beaker nearer to or away from the source of heat. With practice the temperature can be maintained remarkably steady (within 3 deg. F.) by this simple means. A number of these tests can be carried on simultaneously. Two tests are made on each oil, one heated for one hour and one for six hours, both at 302 deg. F.

After heating, the samples are allowed to cool and are set aside over night. Both samples are then examined and a note made of any brown deposit. It is found that most turbine oils will withstand heating for one hour without giving a deposit, whereas many fail after six hours' heating. The oils are filtered if necessary to remove deposit and the clear oil is then subjected to the emulsion test already described.

In Table 2 are given typical results obtained on turbine oils of different brands.

TABLE 2. RATE OF SEPARATION FROM EMULSION OF ARTIFICIALLY AGED OILS

Oil	1	2	3	5
Oil (new), mm. per hr.	400	1,400	780	400
Heated 1 hour, mm. per hr.	36	150	180	130
Heated 6 hours, mm. per hr.	2	120	110	120
Deposit	Nil	Nil	Nil	Heavy
Viscosity (Redwood) at 140 deg. F. (new oil)	112	80	70	75

There is thus a considerable difference between the behaviour of new oils when subjected to the ageing test, and it is interesting to compare the above tests with practical experience of these oils in turbines over a number of years.

The ageing tests on oil No. 1 show no deposit, but a marked falling off in the ability to separate from water. These results were confirmed by practical experience. Very little brown deposit was noticed on the oil coolers, but the oil itself was very sensitive to the presence of water, when it gave considerable trouble due to emulsion and sludge. It was only possible to use this oil in those turbines where there was no leakage of water or steam into the lubricating system. The tests on oil No. 2 after ageing show a fairly ready separation from emulsion with no deposit, thus indicating a satisfactory quality. This finding was substantiated by observation of the behaviour of the oil in use. Sample No. 3 is another oil which, judging from the tests, is suitable for use in turbines. It was found in practice to give little trouble with deposits in coolers or with separation from extraneous water. Oil No. 5 gave a heavy deposit in the ageing test, but separated fairly well from water. In service this oil deposited a brown slime on the coolers which made it necessary to clean them periodically. No difficulty was experienced on those occasions when water gained access to the lubricating system. Here again the practical observations on the plant confirmed the small-scale artificial ageing tests.

In selecting new oils experience has shown that the most satisfactory results are obtained with those oils which, in addition to the usual requirements as regards viscosity, etc., will also pass the following tests: After artificial ageing by heating to 302 deg. F. for 6 hours the oil shall give no deposit and shall separate at a rate of not less than 50 mm. per hr. when tested by the R.L. emulsion method.

In conclusion the author desires to thank the management of the Victoria Falls and Transvaal Power Company, Ltd., for permission to publish the information contained in this paper.

THE FLOW CHARACTERISTICS OF PETROLEUM LUBRICANTS

By D. P. Barnard *

Viscosity is usually the most important characteristic associated with lubricants and provides the principal basis for their selection and use. This in itself would constitute sufficient reason for leaving no stone unturned in the search for information upon the property of lubricant viscosity. The following discussion presents results obtained in a preliminary study of viscosity when observed under other conditions and in other apparatus than those highly standardized forms which are generally used.

Arveson (1932) has discussed several of the more widely used commercial viscometers as well as the apparatus and work of several other investigators. Arveson was led to design and construct a "constant shear viscometer" in which any preselected rate of shear could be maintained at a fixed value regardless of lubricant viscosity. In this instrument (Fig. 1) a synchronous motor A drives an eight-speed transmission B, which in turn drives a pair of worm gears C, functioning as nuts drawing forward the screws F, attached to the head G, riding on the rails supported by plates D and E. The piston H, constructed on the Bridgman (1931) principle, is attached to a head G, and in operation is forced into the cylinder I, forcing the oil out of the cylinder through duct J, through pressure block K into the viscometer L, forcing the disk M downward. The mercury from the bed about the capillary N moves through a duct O, then through the pressure block P, to the manometer at R until equilibrium is reached. The sample between the disk M and the capillary support N1 is forced out through the glass capillary N3 (see diagram of small details), which is ground to a shoulder as shown at 2, and cemented into place at 4 in the support 1. The bottom of the support is threaded so that the whole acts as a bolt to be fastened into position. The top opening in the viscometer is closed by a plug X, consisting of two parts, 1 and 2, separated on the flat face by a rubber washer. The viscometer is in a thermostat V, controlled by a mercury regulator T. The bath liquid was thoroughly agitated. Oil was first used as the liquid, but was found to be unsatisfactory. A 15 per cent soluble-oil emulsion was used with excellent results. The heat transfer was satisfactory, and no corrosion of the steel occurred.

The pressure blocks K and P are constructed so that gauges S and

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* Assistant Director of Research, Standard Oil Company (Ind.), Indiana.

manometers both function with the Bridgman plug in the back position, while only the gauges function when the plug is in the forward position. The plugs are operated by two valve wheels Q.

A detail W of the Bridgman tube connexions is shown. The block has a duct 3, a plain portion 1, into which a head 4 is fitted; and a threaded section 2, to hold the plug 8. Between head 4 and plug 8, two copper washers 6, and a rubber washer 7, are placed. Tension on the rubber seal is furnished by plug 8. When pressure is built up in the system, it applies to the full end surface of 4, producing a force which is transmitted to the rubber washer 7, and, causing pressures higher than the internal pressure, thus maintains a seal. For further comments on Bridgman's developments in high-pressure construction, his original paper should be consulted.

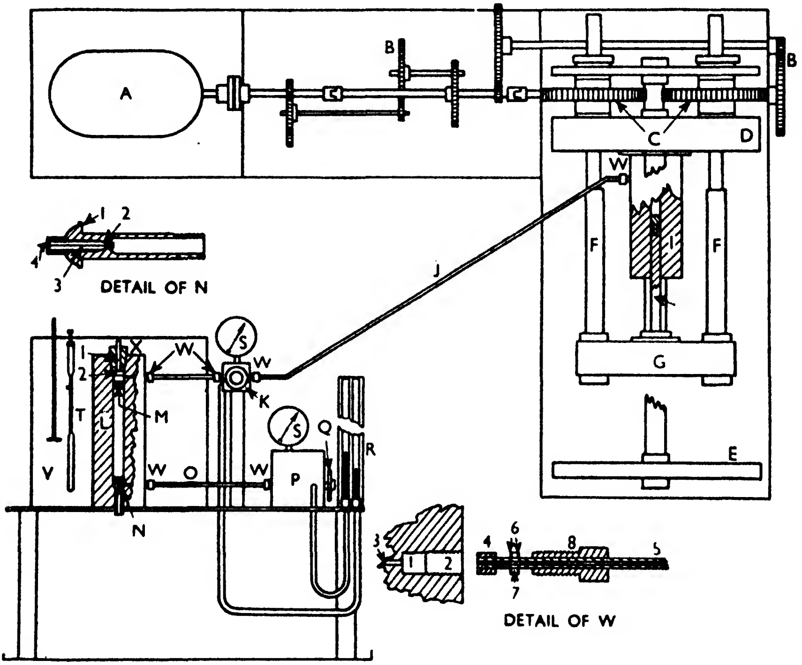

Fig. 1. Constant-Shear Viscometer

The method used by Arveson for relating apparent viscosity to rate of shear (in the purely viscous flow region) is given as follows: The apparent viscosity is the value calculated from Poiseuille's equation, which, when analysed as shown, gives the stress at the wall and the rate of shear:—

$$\eta_a = \frac{PR^4}{8LV/t} = \frac{PR/2L}{4V/tR^3} = \frac{F}{S}$$

where η_a is the apparent viscosity in poises, P the pressure in dynes per square centimetre; R the radius of the capillary in centimetres, L the length of the capillary in centimetres, V/t the rate of efflux in cubic centimetres per second, F the stress in dynes per square centimetre, and S the rate of shear in reciprocal seconds. The symbols are those conventional in the science of rheology. The dimensions required were determined in the conventional manner.

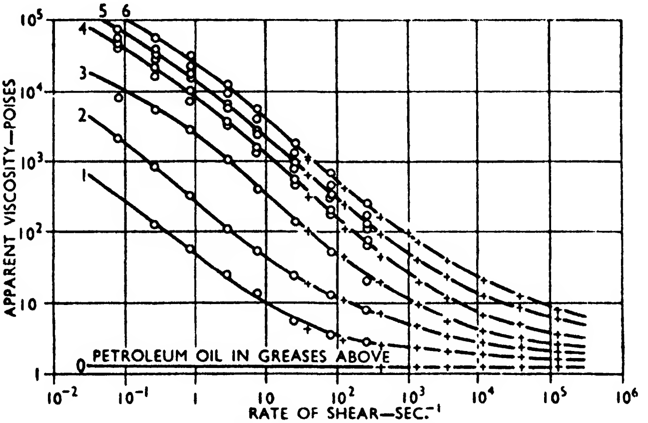

Fig. 2. Variation in Apparent Viscosity with Changes in Rate of Shear

Curve	0	1	2	3	4	5	6
Percentage of soap	0	3·0	6·1	10·1	13·6	18·8	22·5

○ No. 3 Cap. + No. 1 Cap.

Variation in apparent viscosity with changes in rate of shear is illustrated in Fig. 2 which is reproduced from the paper by Arveson. This diagram shows several important points, chief of which is the tendency (in so far as conventional greases are concerned) for the apparent viscosity of the thickened, soap-containing product to approach that of the "base" oil at the higher rates of shear. As will be seen later, this tendency is exhibited to a greater or less degree by all thickened products and is characteristic of "tenderness" or "false" viscosity. Fig. 3, also reproduced from Arveson's paper, compares two types of thickened products with their base oil. In each case the base oil showed no change in apparent viscosity, although the rate of shear was varied from 0·1 to over 100,000 reciprocal seconds. The cup grease (containing a calcium soap) showed a marked degree of "tenderness", while the pulp oil (containing an aluminum soap) exhibited a less pronounced effect. It may be regarded as having less "false" viscosity than the cup grease.

Later, Arveson (1934) extended his work to include the temperatures 10, −4, and −17·7 deg. C. (50, 25, and 0 deg. F.) in addition to 25 deg. C. (77 deg. F.), the temperature previously studied. It was found that for greases the "false" viscosity effect was similar over the entire temperature range and that the various effects could be expressed by a "soap factor" which varied consistently with temperature and concentration and which was characteristic of the soap used but independent of the base oil. The basic effects were, however, unpredictable and in no case was it possible to reconcile the data with existing equations for plastic or viscous flow.

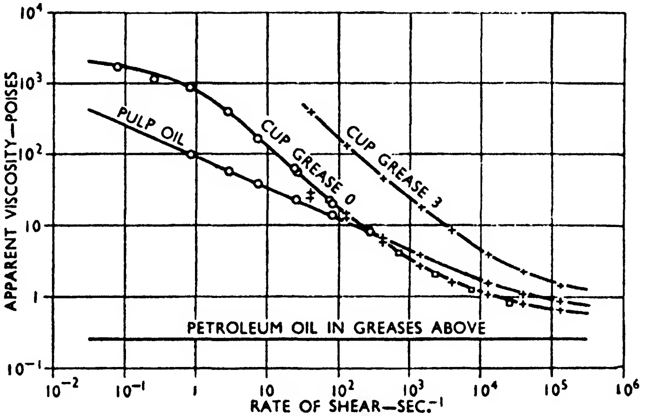

Fig. 3. Comparison of Thickened Greases with their Base Oil

		R	L
+	Capillary No. 1	$0{\cdot}0238_1$	$3{\cdot}73_5$
□	,, No. 2	$0{\cdot}0417_6$	$3{\cdot}81_5$
○	,, No. 3	$0{\cdot}187_1$	$9{\cdot}60_1$

The instances of viscosity varying with rate of shear cited above will be recognized, of course, as rather extreme. The greases containing calcium soap, for example, show infinite viscosities at extremely low shear rates and possess many of the characteristics of definite yield points. The pulp oil, however, behaves under all conditions as a liquid whose viscosity diminishes with increasing rate of shear. A small amount of rubber (0·1 per cent) will show a somewhat similar behaviour when added to an oil. Such materials have the further effect of increasing low shear viscosities even at higher temperatures, with an apparent decrease in the slope of the temperature-viscosity curve or, in other words, an improvement in viscosity index. To what extent such an "improvement" may be reflected in actual performance is problematic. An insight as to what may be expected, however, may be gained from a study of the shear-viscosity characteristics of the product. In general, viscosity index improvements effected by the addition of very small amounts of such substances as rubber can be expected to be apparent rather than real, as shown in Fig. 5.

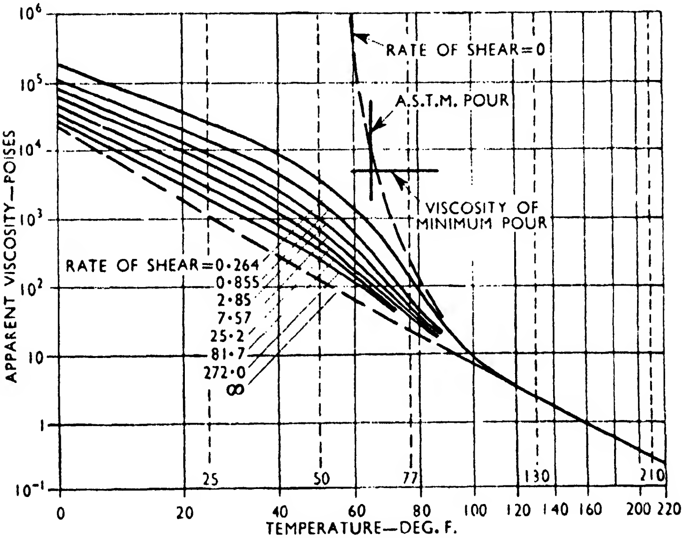

Fig. 4. Viscosity-Temperature-Shear Diagram of a Wax-Bearing Oil

An observation by Arveson, hitherto unpublished, concerns the effect of wax upon the viscosity-shear relations of a petroleum lubricating oil. A sample of filtered steam-refined cylinder stock was run in the constant shear viscometer and the results plotted upon the viscosity-temperature chart due to Sloane and Winnig (1931). The results, in the form of a series of lines of constant shearing rate, are shown in Fig. 4. It is recognized that oils may exhibit viscosity peculiarities at temperatures well above their pour points. Such effects are clearly illustrated by the data plotted in Fig. 4. The solid lines have been plotted from actual data while the dotted lines represent the investigator's estimate for infinite and zero rates of shear. The line for infinite rate of shear is drawn with a slightly increasing slope at the lower temperatures because, while it is true that a wax-free oil should give a straight line on this diagram, an oil from which wax is crystallizing actually becomes higher in viscosity as each fraction of wax changes from the liquid to the solid state. The zero rate of shear line must, of course, approach infinite viscosity. The line shown actually agrees fairly well with the results of independent observations upon the viscosities corresponding to "no flow" in the pour point test of the American Society for Testing Materials (A.S.T.M.). These data, it is believed, give for the first time the relation between viscosity and its two determining factors—temperature and rate of shear—for a wax-bearing oil. Correlation with the A.S.T.M. pour point is also shown.

In all of the foregoing examples the effects on viscosity of changes in rate of shear have been quite pronounced. These effects, however, may be exhibited to a much smaller degree by other materials. Fig. 6 gives the results of an examination of two oils containing added materials which improved their apparent viscosity indexes by about 20 units, the actual data observed upon the Saybolt universal viscometer being given in Table 1.

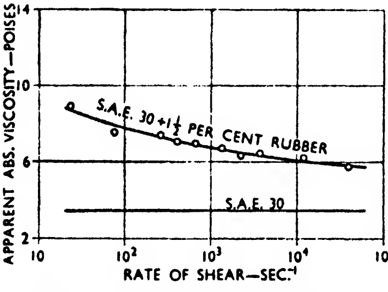

Fig. 5. Variation of Viscosity with Rate of Shear

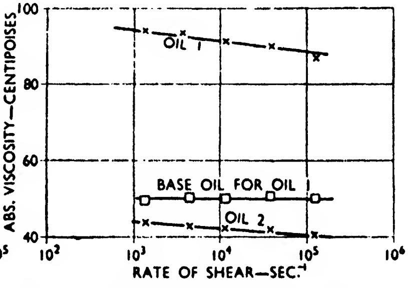

Fig. 6. Effect of Varying Rate of Shear on Viscosity at 100 deg. F.

It is obvious that the reality of this improvement in viscosity-temperature characteristics must be judged in light of the prevailing rate of shear. If the lubricant is to be employed in apparatus where flow rates are low and there are passages of generous proportions then it may be expected to behave as a normal liquid whose viscosity characteristics are as indicated by the orthodox laboratory examination. If, however, rates of shear are generally high, as in the clearance spaces of automotive engines, some loss in apparent viscosity index may be anticipated. Fig. 6 indicates that at rates of shear of 10^5 reciprocal seconds, the apparent absolute viscosities of the lubricants in question may be about 10 per cent less than those indicated by the Saybolt data. Data of this type throw light upon the functional mechanism of added substances which, when present in small proportions, exert pronounced effects upon viscosity.

As an increasing amount of interest is being shown in such substances for lubricating oils (including materials for improving viscosity index), it appears that a thorough study of the viscosity–rate of shear characteristics of oils containing such agents is desirable. As mentioned above, these effects are by no means clearly understood and may not be important in many cases. However, the foregoing discussion is intended to indicate that they do exist, that undoubtedly they will continue to play an increasingly prominent part in lubrication, and that a constant and controllable shear viscometer, such as that used by Arveson, serves as an effective tool for studying the relation between variations in rate of shear and liquid viscosity. It is felt that work of this type should yield profitable results in many instances and should serve to correct what appears to be an unjustified looseness in regarding viscosity as a rigidly fixed property rather than one which may depend to a marked degree on the conditions under which it is determined.

TABLE 1. VISCOSITY DATA FOR OILS SHOWN IN FIG. 6

Oil No.	Base oil for No. 1	Oil No. 1	Oil No 2.
Viscosity at 37·8 deg. C. (100 deg. F.) :—			
Saybolt universal	261	493	228
Centipoises	48·1	92·2	41·8
Viscosity at 98·8 deg. C. (210 deg. F.) :—			
Saybolt universal	49·9	70·6	51
Centipoises	5·9	10·8	6·1
Viscosity index :—			
(Dean and Davis)	98	121	125

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THE DETERMINATION OF THE VISCOSITY OF OILS AT HIGH TEMPERATURES

By Guy Barr, B.A., D.Sc.*

In determinations of viscosity by means of U-tube viscometers the conventional glass-fronted thermostat bath frequently gives trouble either from leakage at the glass-metal joints or from cracking of the glass at temperatures above about 80 deg. C. It is rarely necessary to control the temperature with the accuracy that is possible at 25 deg. C. or so and a tall 5-litre beaker has been found more satisfactory for occasional measurements: with water in the beaker, regulation to within 0·1 deg. C. is possible up to 100 deg. C. when streams of air bubbles are used for stirring. Replacement of the water by glycerin solutions allows temperatures well above 100 deg. C. to be obtained conveniently with the same arrangement: for higher temperatures than 120 deg. C. an oil bath becomes preferable. Even if the capacity is reduced to 1 litre, with a corresponding decrease in constancy of temperature, the fire risk soon becomes rather great as the temperature is raised, and above 180 deg. C. most oils begin to emit some smell.

In some preliminary work, in which the use of temperatures up to 300 deg. C. was contemplated, recourse was had to an air bath made up from a hollow cylinder of aluminium with walls about 1 cm. thick: the walls were slotted to allow observation of the viscometer marks, the slots being provided with double mica windows. The bottom of the cylinder was deeply covered with sand and the top was closed by a metal plug bored to suit the viscometer and thermometer. With gas heating it was not possible to regulate the temperature to much closer than 1 deg. C., and observation and manipulation were by no means convenient.

Vapour baths have been found to be much more easily operated. Fortsch and Wilson (1924) illustrate a form which they found suitable for use with a Bingham viscometer up to 258 deg. C. This consisted of a straight wide tube to which vapour was supplied from a bulb of about the same diameter through a short length of narrower tubing. The author has omitted the narrower tubing: a jacket 15 cm. long and of 4 cm. outside diameter accommodates the essential parts of a No. 2 British Standard viscometer and ends below in a bulb of 7 cm. diameter, while a 10 cm. continuation of 5·5 cm. diameter tubing at the upper end serves as a condenser. The arms of the viscometer are lengthened

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* Senior Scientific Officer, Metallurgy Department, National Physical Laboratory.

so as to project 5 cm. above the tube, and a metal disk is bored to pass over them and rest on the mouth of the tube. The top of the tube was made wider than the main jacket in order to allow for the introduction of additional condensing surface, e.g. of water-cooled channels in front of and behind the plane of the viscometer tubes. This addition has not been found necessary when using the vapour of tetrachloroethane (boiling point 147 deg. C.), but would probably be desirable at lower temperatures; further, the metal cover might well be slightly dished to ensure that any condensate on it drained inwards. A thermometer is supported with its bulb near the middle of the capillary; emergent-stem corrections may be large (and inaccurate) unless the thermometer is well chosen.

The bulb at the bottom of the jacket is charged to a depth of about 2 cm. with the selected liquid, and a few grains of broken silica are added to reduce bumping. The jacket stands vertically, on a ring above a gas burner, the top being guided rather than clamped. The viscometer is filled before insertion and adjusted to be vertical in the jacket. The bath liquid is then brought to the boil and the level of condensing vapour is raised, very slowly if the boiling point is high, until it reaches to about 2 or 3 cm. above the upper graduation mark of the viscometer. The volume of liquid in the viscometer is adjusted by means of a long drawn-out capillary and the determination of flow time may be begun within a few minutes.

The choice of bath liquid will usually be governed by convenience. Fortsch and Wilson (1924) used xylene, aniline, naphthalene, and diphenyl ether. The author has used tetrachloroethane and naphthalene in the bath above described, which is made of "Pyrex" glass: for regular work with very high-boiling inflammable liquids a fused silica bath would be preferred.

Relation between Viscosity and Temperature of Oils. If measurements of viscosity are made at temperatures provided by the vapours of liquids boiling under atmospheric pressure, it becomes essential, for purposes of comparison, that interpolation between results obtained at two temperatures should be made by a suitable method. At least a score of formulæ have been proposed to relate viscosity to temperature: in a recent paper by Erk and Eck (1936), which gives some references, three of these formulæ have been tested against accurate determinations made on oils of very different viscosities between 20 and 80 deg. C. Vogel's formula, which may be written

$$\eta = Ae^{b/(T-c)} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

where A, *b*, and *c* are constants for each oil and T is the absolute temperature, gave the most accurate results. Walther's formula, reproduced as

$$(\nu+a)=Ae^{b/T^c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $a=0{\cdot}8$, $A=1$ (with ν in centistokes), and b and c are constants for each oil, led to values that were normally too high by about 1 per cent when the viscosity at 40 deg. C. was interpolated from results obtained at 20 deg. C. and 60 deg. C., while extrapolation from 60 deg. C. to 80 deg. C. tended to produce larger errors of opposite sign. This formula would apparently introduce no appreciable error if it were used to convert viscosities at odd high temperatures to those at the nearest multiple of 10 deg. C. when a second determination at one of the standard lower temperatures was available. The third formula tested by Erk and Eck is

$$\eta=Ae^{b/T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

A and b being constants for each oil; this they find totally unsuitable for oils. This formula is quoted as being due to Andrade, but Andrade (1934) has disclaimed priority for the deduction.

The method of plotting viscosity-temperature relationships that is suggested by Walther's formula, namely, with log log $(\nu+a)$ as ordinate and log T as abscissa, has received approval from the American Society for Testing Materials; and some valuable practical applications have been described by Ubbelohde (1936). The magnitudes of the constants b and c do not, however, convey directly any idea of the relative viscosities and temperature coefficients of different oils. Thus two mineral oils studied by Erk and Eck (1936) gave the following values:—

	η_{20} centipoises	η_{80} centipoises	b	c
Oil 1935/2 . .	14·58	2·48	10·471	4·207
Oil 1935/1 . .	1,774	54·26	7·959	3·217

and castor oil gives values for b and c that lie between these limits. The constant c might have been expected to bear some relation to the extent to which association occurs.

A formula proposed by Cragoe (1933) has the merit of securing a linear scale for T. The formula may be expressed thus:—

$$\nu=Ae^{b/(T-c)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where $A=0{\cdot}05$ (with ν in centistokes) and b and c are constants for each oil. A straight line should result when reciprocals of $\log(\nu/\mathrm{A})$ are plotted against T, but when the formula was tested against some of the data provided by Erk and Eck it was found to be less satisfactory than that due to Walther.

It appears probable that Walther's formula might become more accurate if the value $\mathrm{A}=1$ were modified: a value $\mathrm{A}=1/0{\cdot}8$ was found to give better results in two cases where it was tried. The most satisfactory concordance with experimental data has been obtained, however, by means of a formula which is believed to be new, namely,

$$\nu+a=10^{f}$$

where $f=(\mathrm{A}+b/\mathrm{T})^{c}$, while $a=0{\cdot}8$ (with ν in centistokes), $c=3{\cdot}333$, and A and b are constants for each oil. In the logarithmic form this becomes

$$[\log_{10}(\nu+a)]^{0\cdot3}=\mathrm{A}+b/\mathrm{T} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The addition of the term a is made, as in equation (2), to avoid the occurrence of negative logarithms and to increase accuracy when ν is small. The viscosity scale defined by $[\log(\nu+a)]^{0\cdot3}$ is relatively more open at high viscosities than that defined by $\log\log(\nu+a)$. To balance this, the reciprocal temperature scale is relatively more open than the logarithmic one at low temperatures.

The new formula is purely empirical, and it is not more easy to suggest a physical basis for its form than for that of Walther. It has been tested against the data for oils between 20 and 80 deg. C. provided by Erk and Eck (1936, Tables 1, 7, 9, 12), where the viscosities at 20 deg. range from 16·4 to 12,240 centistokes, and also against data given by Fortsch and Wilson (1924) for various oil fractions from kerosene to heavy motor oil up to temperatures of 255 deg. C. In every case the agreement with experiment was better than when Walther's formula was used: when the viscosity of a "light motor oil" at 170 deg. F., for example, was deduced by interpolation, between the values at 75 deg. F. and 496 deg. F. in Fortsch and Wilson's paper, the result was 1 per cent low by formula (5) and 6·5 per cent low by formula (2). More extensive tests will be necessary before the new formula can be accorded preference on the grounds of higher accuracy in general work. If its superiority can be established it will be desirable to construct a table for the conversion of ν to $[\log(\nu+a)]^{0\cdot3}$ as has been done by Ubbelohde for the conversion of ν to $\log\log(\nu+a)$, but the table should give five places of decimals to avoid errors in manipulation. A second table giving values of 1,000/T for every 1 deg. C. would facilitate calculations.

It may be expected that the new function should be roughly additive for mixtures, as is claimed for the Walther function and for that derived from equation (4); no calculations have yet been made to test this point.

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THEORETICAL STUDY OF TEMPERATURE RISE AT SURFACES OF ACTUAL CONTACT UNDER OILINESS LUBRICATING CONDITIONS

By H. Blok*

Following up a previous paper (Blok 1937), the temperature rise at the surfaces of actual contact on surfaces lubricated under oiliness conditions have been investigated. By "oiliness conditions" are understood those boundary lubricating conditions under which the *mean* pressure and the *mean* temperature in the region of nominal contact are relatively low.

In considering two bodies, bearing upon each other, the following definitions will be used:—

(1) Surface of actual contact ("SAC"): any single surface on one of the bodies on which all points are points of actual contact with the other body.

(2) Region of nominal contact: the smallest region containing all SAC's of that body at a definite moment (in the particular case in which there is only one SAC, this SAC is therefore also the region of nominal contact).

(3) Total actual bearing area of a region of nominal contact: the sum of the areas of all SAC's in that region.

(4) Permanent SAC: any SAC that remains in contact during a sufficiently long time to reach a steady temperature field due to the frictional heat developed on it.

(5) Temporary SAC: all other SAC's.

Holm (1929) has shown that the region of nominal contact under low mean pressure is not one coherent SAC, but consists of small SAC's of, for instance, a few microns in diameter, scattered in the region of nominal contact. The total actual bearing area will be only a small fraction of this region and so the SAC's will be relatively far apart from each other; for example, if this fraction amounts to 0·001, the mean distance between the SAC's will be approximately $\sqrt{1,000}$ times their mean diameter. Under these conditions the mean pressure in the SAC's may sometimes attain values of 10,000 kg. per sq. cm. and higher.

If two surfaces lubricated under oiliness conditions slide over each other at a sufficiently high velocity, it is to be expected that a high temperature rise will occur in the SAC's due to frictional heat. Indeed,

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* Delft Laboratory, Royal Dutch Shell Group.

Bowden and Ridler (1936) have found this to be true. Still the ***mean*** temperature rise in the region of nominal contact may be small, as the total actual bearing area is small. Further particulars about these phenomena will be deduced hereafter by a calculation which will be illustrated by some numerical examples. Though this calculation will be applied to oiliness conditions, it appears to be applicable in principle to those cases where extreme *mean* pressures (10,000 kg. per sq. cm. and higher) occur in a region of nominal contact stationary with respect to one of the bodies bearing upon each other. Examples of these cases are the Timken machine, the four-ball apparatus, and cutting tools; for these cases the region of nominal contact may be treated as one single SAC.

We shall now consider the temperature rise due to a source of heat showing dimensions that are small with regard to the body to the surface of which it is applied. The following problems will be investigated first:—

I. Round stationary source of heat
  (A) Heat supply evenly distributed.
  (B) Heat supply paraboloidally distributed.
II. Square stationary source of heat; heat supply evenly distributed.
III. Round stationary source of heat between two large contacting bodies.
IV. Square source of heat, moving on a large body.

In the above problems the character of the source of heat has not yet been defined. In the further problems V and VI, which are applicable to actual lubricating conditions, this source of heat will be considered to be the frictional heat developed in one single SAC; then the source of heat coincides with the SAC. The difference between permanent and temporary SAC's will be investigated.

In all these problems the bodies to which the heat is supplied are considered to be so large compared to the dimensions of the source of heat, that they may be assumed to have an infinitely large heat capacity and that the surface of the bodies may be assumed to be plane, at least from a macroscopical point of view.

Problem IA (Fig. 1). Let heat be supplied continuously and distributed evenly with q kg.-cm. per sq. cm. per sec.$=q/42{\cdot}7$ cal. per sq. cm. per sec. over a round surface (radius R cm.), situated on the plane surface of a large body, it being assumed that no other heat is supplied or withdrawn from the body in such a place that it would affect the temperature of the round surface. The temperature rise

θ_T in the centre C of the round surface will be calculated as a function of the time T which has elapsed since starting to supply heat. The surface temperature field will apparently be rotationally symmetrical with respect to C. Let the amount of heat dq applied at the time t to a small area $rd\phi \, . \, dr$ be

$$dq = rd\phi \, . \, dr \, . \, qdt \text{ kg.-cm.} \quad \ldots \ldots \quad (1)$$

Then the temperature rise $d\theta$ at C will be, according to Riemann-Weber (1927),

$$d\theta = \frac{2rd\phi \, . \, dr \, . \, qdt}{\pi\sqrt{\pi} \, . \, \gamma c \, . \, 4at\sqrt{4at}} \, . \, e^{-\frac{r^2}{4at}} \quad \ldots \ldots \quad (2)$$

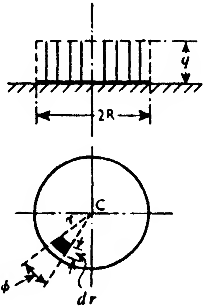

Fig. 1. Stationary Heat Supply, Evenly Distributed over a Round Surface

This formula incorporates the following values, considered to be constant for the material of the body:—

γ=specific gravity, in kilogrammes per cubic centimetre.

c=specific heat, in kilogramme-centimetres per kilogramme per deg. C.

$a = \lambda/\gamma c$ in square centimetres per second, in which:

λ=heat conductivity in kilogramme-centimetres per centimetre per deg. C. per second.

Integrating as to radius, angle, and time (limits respectively 0 and R, 0 and 2π, and 0 and T): and putting $R^2/4aT = \chi$,

$$\theta_T = \frac{1}{\sqrt{\pi}} \, . \, \frac{qR}{\lambda} \left\{ \frac{1}{\sqrt{\chi}} \, . \, (1 - e^{-\chi}) + 2\int_{\sqrt{\chi}}^{\infty} e^{-x^2} dx \right\} \quad \ldots \ldots \quad (3)$$

For T=∞ , i.e. $\chi=0$, a final temperature rise will be found:

$$\theta_{\mathrm{T}=\infty}=\frac{q\mathrm{R}}{\lambda}=\frac{1}{\pi}\cdot\frac{\mathrm{Q}}{\lambda\mathrm{R}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

The total heat supply per second, $\mathrm{Q}=\pi\mathrm{R}^2.q$.

It is seen from curve *a*, Fig. 2, that for $\chi=0{\cdot}1$, i.e. at a time $\mathrm{T}=10\mathrm{R}^2/4a$, 82 per cent of the final temperature rise will have been reached (e.g. $\mathrm{R}=10^{-3}$ cm., $a=0{\cdot}25$ sq. cm. per sec., then T is only 10^{-5} seconds).

*Problem I*B (*Fig. 3*). Similarly it may be shown that for a paraboloidal distribution of the heat supply over the round surface, the temperature rise θ_{T} at C, T seconds after starting the heat supply will be

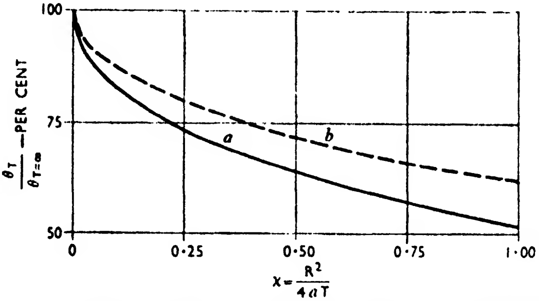

Fig. 2. The Ratio $\theta_{\mathrm{T}}/\theta_{\mathrm{T}=\infty}$ Plotted Against $\chi=\mathrm{R}^2/4a\mathrm{T}$

a Uniformly distributed heat supply (cf. Fig. 1).
b Paraboloidally distributed heat supply (cf. Fig. 3).

$$\theta_{\mathrm{T}}=\frac{2}{\pi\sqrt{\pi}}\cdot\frac{\mathrm{Q}}{\lambda\mathrm{R}}\left\{\frac{1}{\sqrt{\chi}}\cdot\left(1-\tfrac{2}{3}c^{-\chi}\right)+\tfrac{4}{3}\int_{\sqrt{\chi}}^{\infty}e^{-x^2}dx\right\} \quad . \quad . \qquad (5)$$

For T=∞ , i.e. $\chi=0$, and representing the mean heat supply by $q_m=\frac{1}{2}q_p$ (cf. Fig. 3),

$$\theta_{\mathrm{T}=\infty}=\tfrac{2}{3}\cdot\frac{q_p\mathrm{R}}{\lambda}=\tfrac{4}{3}\cdot\frac{q_m\mathrm{R}}{\lambda}=\frac{4}{3\pi}\cdot\frac{\mathrm{Q}}{\lambda\mathrm{R}} \quad . \quad . \quad . \quad . \qquad (6)$$

Now $\theta_{\mathrm{T}=\infty}$ is 4/3 times as high as in problem IA, provided that Q, the total heat supply per second, is the same. The ratio $\theta_{\mathrm{T}}/\theta_{\mathrm{T}=8}$ is given by curve *b*, Fig. 2.

Problem II (*Fig. 4*). In the case of a heat supply evenly distributed over a square surface with sides 2L, the temperature rise θ_T at C will be (putting $L^2/4aT=\chi$)

$$\theta_T=\frac{1}{\pi\sqrt{\pi}}\cdot\frac{q}{\lambda L}\int_{\sqrt{\chi}}^{\infty}\left\{\int_{-x}^{x}e^{-z^2}dz\right\}^2 dx \quad . \quad . \quad . \quad . \quad (7)$$

By numerical integration for $T=\infty$

$$\theta_{T=\infty}=200/\sqrt{\pi}\,.\,qL/\lambda \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Problem III (*Fig. 5*). In the case of two large bodies 1 and 2, pressed against each other, the surface of 1 has a protuberance, while the surface of 2 is plane, the round top (radius R) of the protuberance forming a SAC. Taking an evenly distributed heat supply, Q kg.-cm. per sec. in the SAC, let the temperature rise θ_T reached in the steady state be calculated for the centre C. Part of the heat Q, namely, $Q_1=A_1Q$, will penetrate into body 1 and the remaining part, $Q_2=A_2Q$, into body 2; obviously

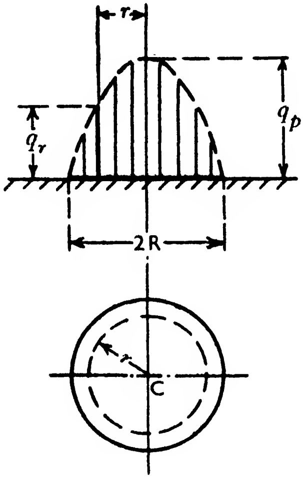

Fig. 3. Stationary Heat Supply, Paraboloidally Distributed over a Round Surface

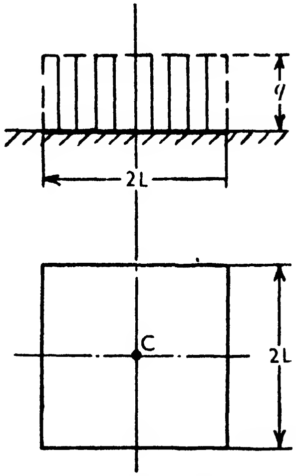

Fig. 4. Stationary Heat Supply, Evenly Distributed over a Square Surface

$$A_1+A_2=1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The coefficients A_1 and A_2 can be determined by assuming that nowhere in the SAC is there a temperature "jump" between any point on body 1 and the point of contact on body 2; by applying equation (4),

$$\theta=\theta_1=\frac{A_1}{\pi}\cdot\frac{Q}{\lambda_1 R}=\theta_2=\frac{A_2}{\pi}\cdot\frac{Q}{\lambda_2 R} \quad . \quad . \quad . \quad (10)$$

Therefore

$$A_1 = \lambda_1/(\lambda_1+\lambda_2) \quad . \; . \; . \; . \; . \; . \; . \quad (11a)$$

and

$$A_2 = \lambda_2/(\lambda_1+\lambda_2) \quad . \; . \; . \; . \; . \; . \; . \quad (11b)$$

According to equations (10) and (11)

$$\theta = 1/\pi \,.\, Q/(\lambda_1+\lambda_2)R = qR/(\lambda_1+\lambda_2) \quad . \; . \; . \quad (12)$$

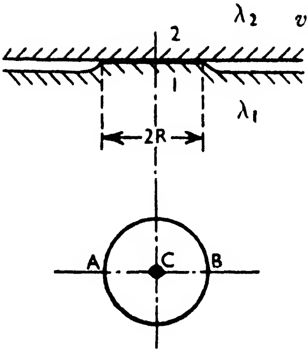

Fig. 5. Plane Surface of Body 2 Pressed against Round Protuberance on Surface of Body 1

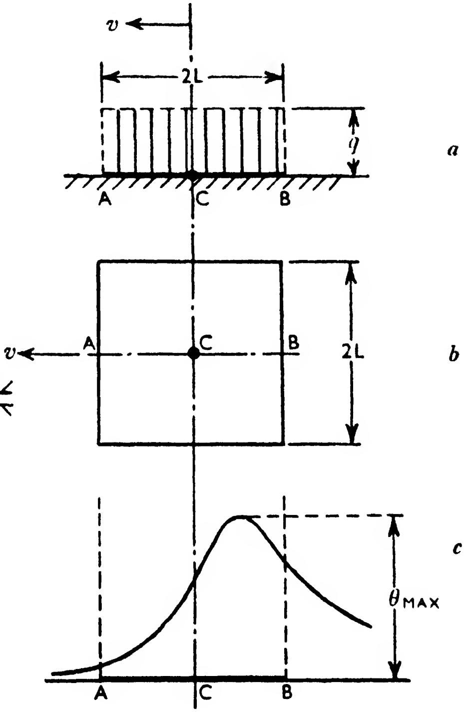

Fig. 6. Heat Source Moving with Velocity *v* cm. per sec. over a Surface

c Temperature distribution for section AB

Compare, for instance, the two particular cases: (1) Two bodies of the same material are pressed against each other; $\lambda_1=\lambda_2=\lambda$. (2) A body of the same material as under 1 is pressed against a perfect heat insulator; $\lambda_1=\lambda$ and $\lambda_2=0$. It follows from equation (12) that the temperature rise at the centre C (i.e. the maximum temperature) in case 2 will not be many times higher than in case 1, but only twice as high.

Problem IV (*Fig. 6*). Take the case of an evenly distributed and continuously applied source of heat in the shape of a *square* (sides 2L) moving over the plane surface of a large body at a uniform velocity v cm. per sec. in a direction parallel to one of the sides; it is assumed that the heat supply is started simultaneously with the motion. The temperature field (moving with the source of heat) will be symmetrical with respect to section AB and the maximum temperature will occur in section AB between C and B, or at a sufficiently high velocity even in B, as will be shown hereafter. Before evaluating θ_T at the centre C, at a time T after starting the motion, consider Fig. 7. After a time τ or T respectively, the centre of the source of heat will be in the position indicated by C_τ or C_T respectively. Now the temperature rise $d\vartheta$ at the time T at the point C_T of the surface, due to the amount of heat $d_q = q d\tau \, . \, dx \, . \, dy$ which is developed at the time τ during $d\tau$ seconds in a small square with sides dx and dy at a distance r from C_T, may be deduced in the same manner as equation (2) and will be

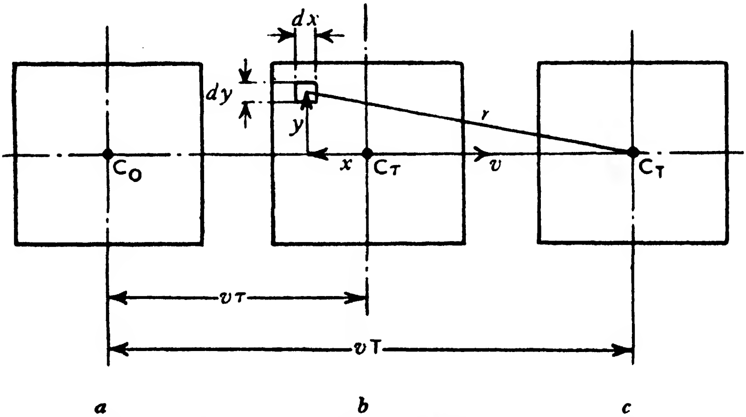

Fig. 7. Different Positions of Moving Heat Source

a Initial position; *b* position after τ seconds; *c* position after T seconds

$$d\theta = \frac{2q d\tau \, . \, dx \, . \, dy}{\pi\sqrt{\pi} \, . \, \gamma c \, . \, 4at\sqrt{4at}} \, . \, e^{-\frac{r^2}{4at}} \quad . \; . \; . \quad (13)$$

in which

$$r^2 = \{x + v(T - \tau)\}^2 + y^2 \quad \text{and} \quad T - \tau = t \quad . \; . \; . \quad (14)$$

The temperature θ_T may be found by integrating equation (13)

$$\theta_T = \frac{2q}{\pi\sqrt{\pi} \, . \, \gamma c} \int_0^T \left[\int_{-L}^{L} dx \int_{-L}^{L} e^{-\frac{r^2}{4at}} \, dy \right] . \frac{d\tau}{(4at)^{\frac{3}{2}}} \quad . \quad . \quad (15)$$

For T=∞ and putting $\sqrt{4a/v\mathrm{L}} = \psi$ and $\psi \, . \, \mathrm{L}/\sqrt{4at} = \epsilon$

$$\theta_{T=\infty} = \frac{\psi}{\pi\sqrt{\pi}} \cdot \frac{q\mathrm{L}}{\lambda} \cdot \int_0^\infty \left[\int_{\frac{-\epsilon}{\psi}}^{\frac{\epsilon}{\psi}} e^{-z^2} dz \, . \int_{\frac{\epsilon^2-1}{\epsilon\psi}}^{\frac{\epsilon^2+1}{\epsilon\psi}} e^{-z^2} dz \right] \frac{d\epsilon}{\epsilon^2} \quad . \quad (16)$$

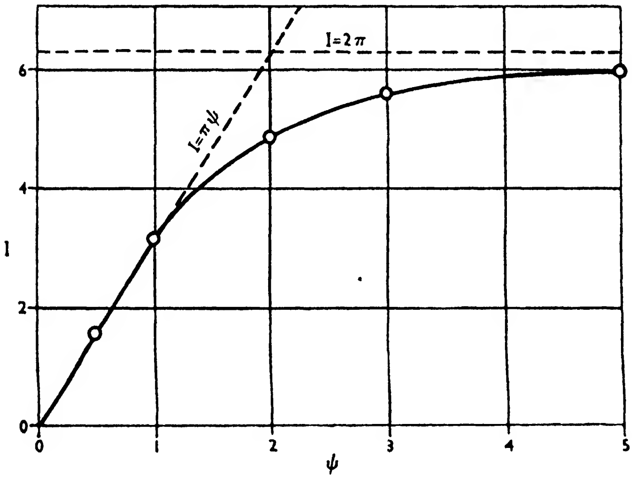

Fig. 8. Relation of I to ψ, when $\mathrm{I} = \psi \int_0^\infty \left[\int_{-\epsilon/\psi}^{\epsilon/\psi} e^{-z^2} dz \, . \int_{\frac{\epsilon^2-1}{\epsilon\psi}}^{\frac{\epsilon^2+1}{\epsilon\psi}} e^{-z^2} dz \, . \right] \frac{d\epsilon}{\epsilon^2}$

○ Points found by calculation.

The relation of the value, denoted by I in Fig. 8 (found by graphical integration) to ψ is given by a curve. It will be seen from Fig. 8 that:—

(1) For $\psi < 1$, i.e. for sufficiently high velocity ($v > 4a/\mathrm{L}$)

$$\mathrm{I} \approx \pi\psi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17a)$$

or

$$\theta_{T=\infty} \approx \psi/\sqrt{\pi} \, . \, q\mathrm{L}/\lambda \quad . \quad . \quad . \quad . \quad . \quad . \quad (17b)$$

It will be clear that at increasing velocities the "side flow" of heat (i.e. in a direction at right-angles to the direction of motion) into the body will be increasingly reduced, so that in so far as the section AB (Fig. 6) is considered the case of the square source of heat may be approximately compared with that of an infinitely long band-shaped source of heat (width 2L), moving with the same high velocity v at right-angles to its longitudinal axis, as in the latter case there will not be any side flow of heat. Indeed, it has been found (Blok 1937) that equation (17*b*) applies also to this last case, for which the maximum temperature rise occurs on the "rear" side of the source of heat and is $\sqrt{2}$ times as high as the temperature rise in the centre, if the velocity $v \geqslant 5.4a/\mathrm{L}$. This leads to the conclusion that the maximum temperature rise occurring in the square source of heat will also be approximately $\sqrt{2}$ times the temperature rise at the centre, if $v \geqslant 5.4a/\mathrm{L}$.

The above conclusions also apply approximately to a round source of heat, as the reasoning of the last paragraph remains in the main the same (L should be replaced by radius R).

(2) It will also be seen from Fig. 8 that for $\psi \geqslant 5$, i.e. for sufficiently low velocity $v \leqslant (1/25.4a/\mathrm{L})$

$$\mathrm{I} \approx 2\pi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18a)$$

or

$$\theta_{\mathrm{T}=\propto} \approx 2/\sqrt{\pi} \,.\, 9\mathrm{L}/\lambda \quad . \quad . \quad . \quad . \quad . \quad . \quad (18b)$$

Thus, $\theta_{\mathrm{T}=\infty}$ is independent of ψ, i.e. for a given heat supply q is also independent of the velocity. Equation (18*b*) corresponds approximately to equation (8), so that the source of heat may now be treated as a stationary one (compare case II). The maximum temperature rise, however, still occurs at some small distance "after" the centre of the source of heat; this distance will be the smaller and the temperature field will be the more symmetrical with regard to the centre the lower the velocity. As in this case the maximum temperature rise will not be very much larger than the temperature rise at the centre, the latter may be calculated approximately from equation (18*b*) if $v \leqslant 1/25.4a/\mathrm{L}$. It will also be evident from the foregoing that the temperature rise at the centre of a moving *round* source of heat with an evenly distributed heat supply may be calculated approximately from equation (4), provided that $v \leqslant 1/25.4a/\mathrm{L}$.

In problems I–IV the character of the source of heat has not yet been defined. In problems V and VI, which are applicable to actual lubricating conditions, this source of heat will be considered to be the frictional heat developed in one single SAC; then the source of heat coincides with the SAC. The difference between permanent and temporary SAC's will be investigated.

Problem V. Consider the case of two bodies 1 and 2, shaped as in Fig. 5, but with body 2 moving with a velocity v over the stationary body 1 such that $v \leqslant 1/25 . 4a_2/R$. ($a_2 = \lambda_2/\gamma_2 c_2$ sq. cm. per sec.). As the surface of body 2 is very large, the SAC is permanent. If the load on the SAC is evenly distributed (p kg. per sq. cm.), the coefficient of friction being f, then an evenly distributed heat supply q kg.-cm. per sq. cm. per sec. will be generated in the SAC so that

$$q = fpv \text{ kg.-cm. per sq. cm. per sec.} \quad . \quad . \quad . \quad (19)$$

It has been shown under problem IV (2) that the moving body 2 may be treated as a stationary one, so that equation (4) can be applied. Thus problem V is reduced to problem III, so that the temperature rise in the centre of the SAC may be approximated according to equation (12)

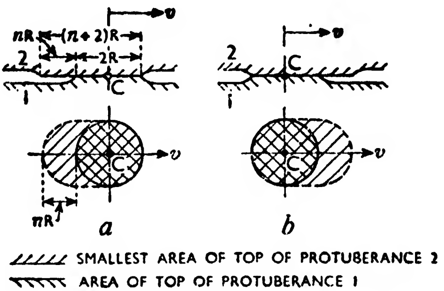

Fig. 9. Protuberance on Body 2 Moving over a Protuberance on Body 1 (Temporary SAC)

(*a*) Initial moment considered; (*b*) body shifted R cm. further on.

$$\theta \approx \frac{qR}{\lambda_1 + \lambda_2} = \frac{4fp}{\gamma_2 c_2} \cdot \frac{1}{1 + \dfrac{\lambda_1}{\lambda_2}} \cdot \frac{vR}{4a_2} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

It is seen that θ varies proportionately to the velocity.

So far the conclusions obtained for this problem refer to permanent SAC's; it will now be shown that under many circumstances they apply also approximately to temporary SAC's. In Fig. 9 the moving body 2 now also shows a protuberance; let its top have a dimension of $(n+2)R$ in the direction of motion (R being the radius of the round top of the contacting protuberance on the stationary body 1) and let it have a dimension of at least 2R in the direction perpendicular thereto. Then heat will be supplied to all points of the SAC during a time $T \geqslant nR/v$. If n is so large that $R^2/4aT = \chi_1$ and $R^2/4a_2T = \chi_2$

are both smaller than $1/25=0{\cdot}04$, the temperature rise at the centre will have reached at least 82 per cent of the final temperature rise indicated by equation (20) (cf. Fig. 2, curve a); this will be so if $T \geqslant 25 . R^2/4a_1$ and also if $T \geqslant 25R^2/4a_2$, i.e. n ought to be $\geqslant 25vR/4a_1$ (or $n \geqslant 25(vR/4a_2)(a_2/a_1)$) and also $n \geqslant 25vR/4a_2$. As it is supposed that $vR/4a_2 \leqslant 1/25$, then n must be such that $n \geqslant 1$ and also $n \geqslant a_2/a_1$, in order that a temperature rise of the order of magnitude as indicated by equation (20) will be reached on a temporary SAC; this may frequently be so. The reasoning and the conclusions remain the same if it is the stationary body which shows the largest protuberance.

It can be shown from equation (20) that for a velocity $v \leqslant 1/25 . 4a_2/R$ (i.e. $vR/4a_2 \leqslant 1/25$) the maximum temperature rise on the SAC's will always be low. If $f=0{\cdot}20$ and the specific heat per unit of volume $\gamma_2 c_2$ of body 2 is equal to 50 kg.-cm. per cu. cm. per deg. C., then according to equation (20), even at a mean pressure in the SAC's of $p=50{,}000$ kg. per sq. cm. which, even for hardened steel, is very high, θ will be lower than 32 deg. C. as $1/(1+\lambda_1/\lambda_2)$ is always smaller than unity.

Problem VI. Consider the case represented by Fig. 5, assuming, however, that body 2 moves with a high velocity v such that $v \geqslant 5 . 4a_2/R$ (i.e. $vR/4a_2 \geqslant 5$) over the stationary body 1. The total heat supply per second, $Q=\pi R^2 . q=\pi R^2 . fpv$ kg.-cm. per sec., will be divided between the two bodies in such a way that body 2 tends to absorb the greater part A_2Q of the heat Q, as during a certain time the moving body 2 represents more cooling material than the stationary protuberance. The remaining smaller part A_1Q will penetrate into the stationary body ($A_1+A_2=1$). The assumption that nowhere in the SAC will there be a temperature jump between contacting points of the bodies 1 and 2 leads to the conclusion that the heat supplies A_1Q and A_2Q respectively will not be evenly distributed over the SAC. That this is true may be shown by considering separately the respective temperature fields on body 1 and 2 which are established if A_1Q is evenly distributed with A_1q kg.-cm. per sq. cm. per sec. over the SAC on the stationary body 1, or $A_2Q=(1-A_1)Q$ also evenly distributed with $A_2q=(1-A_1)q$ over the SAC on the moving body 2, and if A_1 and A_2 are chosen in such a way that the respective temperature fields show the same maximum temperature ($\theta_{\max_1}$ and $\theta_{\max_2}$ respectively). The shape of these fields is roughly shown by curves 1 and 2 in Fig. 10 for a section AB through the centre C of the SAC, taken in the direction of motion. The shape of curve 2 relating to the moving body 2 has already been discussed under case IV (1) (cf. Blok 1937). The actual common surface temperature field—common as no temperature jump has been assumed—may be approximated by the average (cf. curve 3, Fig. 10) of the two temperature fields represented by curves 1 and 2. To obtain this temperature field it is necessary that, from the "rear" side A to about a point G given by the intersection of curves 1 and 2, more heat penetrates into the stationary body 1 and consequently less heat into the moving body 2 than corresponds to A_1q or A_2q respectively. As the reverse is true for the part of section AB between G and the "foreside" B, it is seen that the total heat supplies A_1Q and A_2Q cannot be evenly distributed over the SAC.

The exact calculation of the distribution of A_1Q and A_2Q and of the actual common surface temperature field presents considerable difficulties and therefore an approximate method has been adopted in order to obtain an idea of the maximum temperature rise reached. The average of the two temperature rises $\theta_{c_1}\approx\theta_{\max_1}$ and θ_{c_2} in the centre C (Fig. 10), respectively corresponding to the temperature fields originating from the heat supplies A_1Q and A_2Q which are considered to be evenly distributed, will give a first approximation of the actual maximum temperature rise $\theta_{\max}$; hence

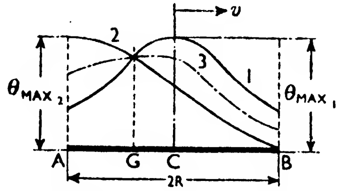

Fig. 10. Determination of Temperature Field in Section AB of the SAC

Curve 1: Temperature field on stationary body 1 for evenly distributed heat supply A_1Q.
Curve 2: Temperature field on moving body 2 for evenly distributed heat supply A_2Q.
Curve 3: Average of curves 1 and 2.

$$\theta_{\max}\approx(\theta_{c_1}+\theta_{c_2})/2\approx(\theta_{\max_1}+\theta_{c_2})/2 \quad . \quad . \quad . \quad (21)$$

According to Blok (1937), if $vR/4a_2>5$

$$\theta_{c_2}\approx 1/\sqrt{2}\,.\,\theta_{\max_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

and

$$\theta_{\max_2}\approx 4\sqrt{\frac{2}{\pi}}\,.\,\frac{A_2\,.\,q}{\gamma_2 c_2}\,.\,\sqrt{\frac{R}{4a_2 v}} \quad . \quad . \quad . \quad . \quad (23)$$

According to equation (4):

$$\theta_{c_1}\approx\theta_{\max_1}\approx A_1\,.\,qR/\lambda_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

By equalizing equations (23) and (24), putting $vR/4a_2=\phi_2$ and remembering that

$$\theta_{\max_1}=\theta_{\max_2} \quad \ldots\ldots\ldots \quad (25)$$

and

$$A_1+A_2=1 \quad \ldots\ldots\ldots \quad (26)$$

it will be found from equations (21) to (26) that

$$\theta_{\max}\approx\frac{2fp}{\gamma_2 c_2}\cdot\frac{\left(1+\frac{1}{\sqrt{2}}\right)\cdot\phi_2}{\frac{\lambda_1}{\lambda_2}+\sqrt{\frac{\pi}{2}}\cdot\phi_2} \quad \ldots\ldots \quad (27)$$

and

$$A_2\approx\frac{\frac{1}{2}\left(1-\frac{1}{\sqrt{2}}\right)+\frac{\lambda_2}{\lambda_1}\sqrt{\frac{\pi}{2}}\cdot\phi_2}{1+\frac{\lambda_2}{\lambda_1}\cdot\sqrt{\frac{\pi}{2}}\cdot\phi_2} \quad \ldots\ldots \quad (28)$$

It can be seen from equation (27) that $\theta_{\max}$ tends to vary proportionately to $\sqrt{\phi_2}$, hence to $\sqrt{v}$, if the velocity v is sufficiently increased, and from equation (28) that A_2 approaches to 1, which indicates that practically all frictional heat will then be absorbed by the moving body 2.

According to Blok (1937) the temperature field relating to θ_{c_2} in equation (22) will have been established if the evenly distributed heat A_2Q has been supplied during at least $2R/v$ secs. to all points of the moving SAC, as $v\geqslant 5 \,.\, 4a_2/R$. This condition is fulfilled if one of the two contacting protuberances (cf. Fig. 9) has dimensions of at least 4R (4R corresponds to $(n+2)R$ in Fig. 9, hence $n\geqslant 2$) in the direction of motion and of at least 2R perpendicular thereto; this may frequently be the case.

However, there is another restriction to these dimensions which may be deduced by considering the temperature field relating to θ_{c_1} in equation (24), i.e. the temperature field originating from the evenly distributed heat supply A_1Q. The time T during which heat will be supplied to all points in the SAC on the stationary body 1 will be $T=nR/v$ (cf. Fig. 9). It can be seen from curve *a* of Fig. 2 that for $\chi=R^2/4a_1T\leqslant 0{\cdot}10$, i.e. only for $T=nR/v\geqslant 10R^2/4a_1$ seconds, the temperature rise in the centre of the stationary protuberance will reach 82 per cent of θ_{c_1} in equation (24). From this it follows that only for $n\geqslant 10.vR/4a_1=10vR/4a_2\,.\,a_2/a_1$, i.e. $n\geqslant 50a_2/a_1$ (as it has been assumed that $vR/4a_2\geqslant 5$), a temperature of the order of magnitude indicated by θ_{c_1} in equation (24) will be reached; this, however, will not frequently be the case for temporary SAC's and so equation (27) will not frequently apply, even approximately, to temporary SAC's. That for $\phi_2 = vR/4a_2 \geqslant 5$ high temperature rises may indeed occur on permanent SAC's may be shown by the following example:—

In equation (27) let $f=0{\cdot}20$; $p=20{,}000$ kg. per sq. cm. (as may occur on hardened steel); $\gamma_2 c_2=50$ kg.-cm. per cu. cm. per deg. C., $\lambda_1/\lambda_2=1$, and $\phi_2=vR/4a_2=5$ (e.g. $v=1{,}000$ cm. per sec., $R=0{\cdot}005$ cm., and $a_2=0{\cdot}25$ sq. cm. per sec.), then the maximum temperature rise on the permanent SAC considered will be about 360 deg. C.

In conclusion the author wishes to express his gratitude to Professor W. J. D. van Dijck for affording him some of the fundamental conceptions incorporated in this study.

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THE FRICTION OF SLIDING METALS

By F. P. Bowden, D.Sc. *

Part I. The High Temperature of Lubricated Surfaces

The Surface Temperature of Sliding Metals. Elementary considerations of the amount of frictional heat liberated on sliding and of its possible rate of escape, lead us to expect that the *surface* temperature of sliding metals would reach a high value. Some years ago experiments were described which showed that this is indeed the case (Bowden and Ridler 1935, 1936). This surface temperature was measured by using the rubbing contact of two different metals as a thermocouple, and determining the electromotive force generated on sliding. The results confirmed the theory: the surface temperature increased with the load or the speed and could easily reach the melting point of the metal.

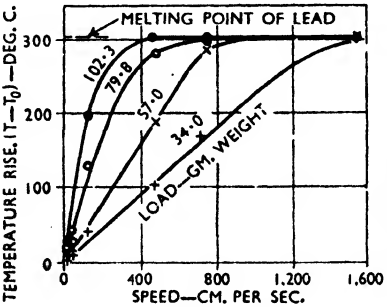

Fig. 1. Surface Temperatures produced by Lead Sliding on Mild Steel

Initial temperature T_0, 17 deg. C.

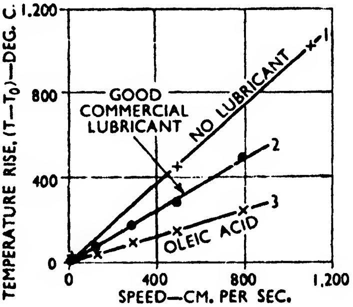

Fig. 2. Surface Temperatures produced by Constantan Sliding on Mild Steel, with and without Lubricant

Load, 102 grammes; initial temperature T_0, 17 deg. C.

Some typical results for lead sliding on mild steel are shown in Fig. 1. The observed rise in surface temperature is plotted against the speed of sliding. The different curves are for different loads. It will be seen that the surface temperature rises as the speed is increased until it reaches the melting point of lead. If a heavier load is used the melting point is reached at a correspondingly lower speed.

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* Laboratory of Physical Chemistry, Cambridge.

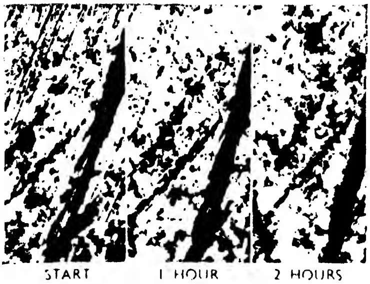

Fig. 3. Effect of Rubbing Wood's Alloy on Camphor : Surface Flow

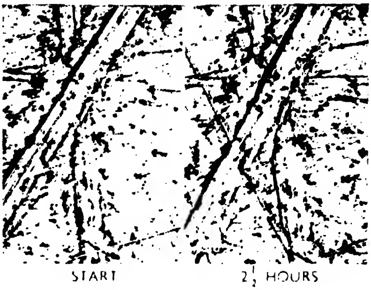

Fig. 4. Effect of Rubbing Tin on Camphor: No Surface Flow

Fig. 5. Fluctuating Friction: Steel on Steel

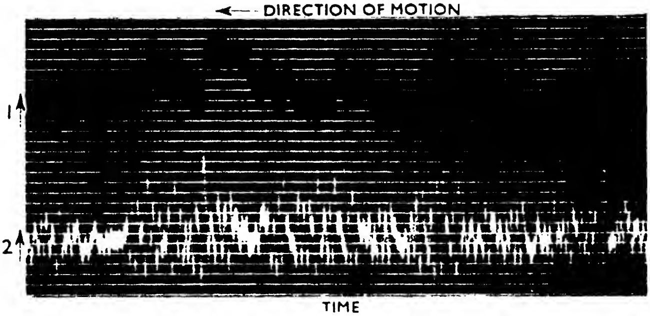

Fig. 6. The Fluctuating Friction (1) and the Fluctuating Surface Temperature (2) between Steel and Constantan Surfaces

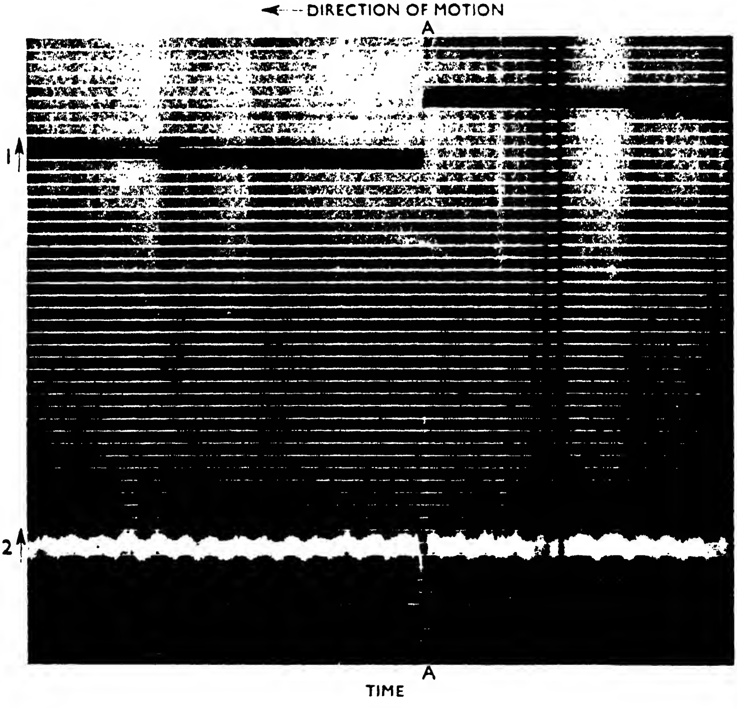

Fig. 7. Analysis of the Effect shown in Fig. 6

The loads used in this particular experiment were comparatively light. If the load is very heavy (the so-called "extreme-pressure" condition), high local temperatures may be reached at very low sliding speeds.

The highest temperature recorded in Fig. 1 is 327 deg. C. because the temperature cannot rise above the melting point of lead. With metals of higher melting point, however, local surface temperatures of over 1,000 deg. C. were reached. Although the surfaces may be very carefully prepared and optically flat, contact does not occur over the whole surface. The heat is liberated and the temperature rise occurs *locally* at the surface of the rubbing points of contact. The mass of the metal may be cold and the areas on the surface which are not actually rubbing may be comparatively cool. Even all the points that are rubbing may not be at the same temperature. The electromotive force measured is the integrated value of a number of thermocouples in parallel so that many of the points of contact may be at a higher temperature than the recorded value.

The Temperature of Lubricated Surfaces. Even if the surfaces are lubricated or are flooded with water these local high temperatures still occur. In Fig. 2, curve 1 shows the temperature reached by a surface of Constantan sliding on steel. Curves 2 and 3 show the corresponding temperatures when the surfaces are lubricated, under boundary conditions, with a good commercial lubricant and with pure oleic acid respectively. It will be seen that the temperature of the lubricated surfaces may be very high (500 deg. C. or more). This will cause volatilization and decomposition of most lubricants, and is an important cause of the breakdown of the lubricant film. If either the load or the speed is too great, a welding together of the "hot spots" may occur on a large scale with consequent damage or seizure of the surfaces.

The fact that an electromotive force is developed shows that some metal-to-metal contact is occurring through the lubricant film. This is in harmony with some earlier work (Bowden and Beare 1935) which showed that the boundary film of lubricant was unable to afford complete protection to the sliding surfaces. Some wear and abrasion of the surfaces always occurred.*

The Flow and Seizure of Metal Surfaces: Bearing Metals. Further experiments on the polishing and surface flow of metals have shown that the process is greatly influenced by the *relative melting point* of the polisher and the solid (Bowden and Hughes 1937). Surface flow

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* Quite recently Blok (World Petroleum Congress 1937) has applied this method to rubbing gear teeth of two different steels, and has been able to confirm these observations that local high temperatures occur. See also vol. 2, p. 14.

and the formation of the Beilby layer readily occur, *provided the melting or softening point of the metal is lower than that of the polisher.* The relative hardness, as normally measured at room temperature, is comparatively unimportant. For example, Fig. 3 shows the effect produced when Wood's alloy (melting point, 69 deg. C.; Vickers hardness, 25) is rubbed with a block of camphor (melting point, 178 deg. C.; very soft). Fig. 4 shows the result when tin (melting point, 232 deg. C.; Vickers hardness, 4) is rubbed, under the same conditions.

Surface flow and the formation of the Beilby layer have taken place on Wood's alloy because its melting point is lower than that of camphor. The surface of the tin is untouched because its melting point is higher than that of camphor. It is clear that hardness is unimportant. Although camphor is almost as soft as butter it readily causes the flow of the very hard Wood's alloy, and does not affect the softer tin. If a polisher of higher melting point is used it will readily cause the flow of tin, but not of a metal which melts at a still higher temperature, and so on.

Similar results were obtained with whitemetal and various bearing metals, and the experiments suggest that this high temperature softening and flow plays an important part in the practical working of bearings. The performance of a bearing metal may depend, not so much on its properties at room temperature as on its properties *at the high temperature of sliding*. It will depend particularly on the *relative* mechanical properties of the two surfaces at this high temperature. It may be very important that the melting or softening point of the bearing metal should be much *lower* than that of the steel journal or other sliding surface. In this case the local high temperature at the surface of the points of contact will cause the bearing metal to melt *first*. This melting will prevent the temperature from rising any higher. The journal will be undamaged since its surface is still hard at this temperature. If the amount of melting is not excessive, surface flow and polish of the bearing metal will occur, and no harm will be done. If, however, both metals possess a similar melting point it will be possible for both of them to soften and weld together at the "hot spots" and metal bridges will be formed. These will be torn away, *both surfaces* will be damaged and seizure and breakdown may occur.

It is also possible that surface tension effects may play some part. It is important for example that the molten or softened bearing metal should not "wet" the other surface or it will be wiped off. The surface tension will be influenced by the nature of the metals, by the lubricant and by oxide and other films.

These high temperatures will of course accelerate the oxidation and attack of the metals. The influence of this on metallic wear has been discussed elsewhere (Bowden and Hughes 1937). If the bearing alloy is non-homogeneous, parts of the surface may be melted or attacked before the others and pitting and damage may result.

PART II. THE NATURE OF SLIDING: THE MEASUREMENT OF FLUCTUATING FRICTION AND FLUCTUATING TEMPERATURE

By F. P. Bowden, D.Sc., and L. Leben *

The Measurement of Fluctuating Friction. Most methods of measuring kinetic friction give the average force necessary to maintain sliding. It was considered probable on theoretical grounds that this force might not be constant and some earlier experiments supported the view. More recently an apparatus has been constructed which has a period of a few thousandths of a second, and is capable of recording any rapid fluctuations in friction if they occur. Some of the results for steel sliding on steel at slow speeds are shown in Fig. 5. It will be seen that the frictional force is not constant but is undergoing the most violent fluctuations. The process of sliding is not a uniform one. The top surface "sticks" to the lower one and moves with it, until it is no longer able to withstand the increasing pull of the friction measuring device. A sudden and very rapid "slip" then occurs. This process is then repeated indefinitely.†

The Simultaneous Measurement of Fluctuating Friction and Fluctuating Temperature. Fig. 6 shows the simultaneous measurement of the fluctuating friction (the dark trace) and the fluctuating temperature (the white trace) between steel and Constantan surfaces. The sliding speed is again low (an apparent sliding speed of 0·5 cm. per sec.) and it is clear that the friction is not constant but is again fluctuating violently. Sliding is not continuous; "stick, slip" is occurring. The surface temperature also is fluctuating. A close examination will show that there is a marked correlation between the two. *The sudden rise in surface temperature occurs at the same instant as the slip.* Whenever a particularly big "slip" occurs the rise in temperature is correspondingly greater.

The effect is analysed in detail in Fig. 7 and can be seen very clearly. The sliding speed was in this case very low (apparent speed 0·003 cm. per sec.) and the camera was driven faster so that only one slip occurs on the photograph. The dark trace again represents the friction. The top surface sticks to the bottom one and moves with it until at the

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* Laboratory of Physical Chemistry, Cambridge.

† A more complete account of this investigation will be published elsewhere.

point A a sudden and very rapid slip occurs. The surface temperature (white trace) remains nearly constant for some time and then shows a sudden very rapid rise and fall. This rise again occurs at the same instant as the slip. These photographs are for unlubricated surfaces, but with many lubricants, the behaviour both of the friction and of the temperature is essentially the same.

Discussion. These results show clearly that the process of sliding may not be continuous, but may proceed by a process of "stick, slip". The surfaces jerk along and the frictional force and the surface temperature both undergo violent fluctuations. These fluctuations in the frictional force may be very rapid indeed. Since most of the research and standard methods of measuring friction record only the average force over a relatively long period of time, they are not capable of detecting these very rapid fluctuations. The temperature changes are also extremely rapid, and the whole rise and fall may be over in less than a thousandth of a second.

The bearing of these results on the theory of friction will not be discussed here, but it is clear that they have important theoretical and practical implications. It is at the instant of slip, that tearing of the metal bridges, wear, and abrasion of the surfaces may occur. It is true that in most machines the relatively moving parts are not able to halt, but are driven remorselessly on. Nevertheless the intermittent clutching and breaking away of the surfaces is taking place, and the area of contact, the surface temperature and the friction are undergoing violent fluctuations.

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THE INFLUENCE OF INCREASE OF TEMPERATURE, OF OILINESS, AND OF SURFACE CONDITIONS ON VISCOSITY

By Henri Brillié*

I. THE INFLUENCE OF TEMPERATURE

The elementary theory of viscosity is generally laid down† without taking into consideration either the influence of increase of temperature of the lubricant, or of its "oiliness," or the influence of the irregularities of the bearing surfaces. These three factors have a considerable influence, and it is necessary to take them into consideration.

Increase of Temperature. Pressure develops in the oil film only on account of the drive and drag acting on the various "layers" of the lubricant, under the effect of viscous friction. This friction results from the interactions of the fields of force of the molecules.

For a speed of 1 metre per second of the moving bearing surface, for instance, and for a film 0·01 mm. thick, whose speed curve can be approximately replaced by its chord, the relative speed of the molecules of two adjacent layers is equal to 100,000 times their height, that is to say that during a second, a field of force of one layer is successively penetrated by 100,000 fields of force of the next layer.

This penetration is the cause of the restraint affecting the layer adjacent to the moving surface, and also the drive communicated to the next layer. Further, this interpenetration increases the vibratory motion of the fields of force concerned and leads to an increase in temperature of the corresponding molecules. There is no reason why these increases in temperature should not be considered to occur in every region; they may be extremely variable according to the points considered, and according to the shape of the speed curve.

Regarding the increase of temperature resulting from viscous friction, two fundamental principles can be deduced from the foregoing.

The increase in temperature occurs in the mass of the lubricant itself; it occurs ***instantaneously*** and accompanies the passive resistances in every point, ***without any delay in time.***

It is not obvious that the transmission of heat between the molecules of various layers, then from layers next to the bearing surfaces to those surfaces, and finally from those surfaces to the mass of metal, will occur

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* Formerly Engineer-in-Chief, Compagnie Générale Transatlantique.

† Cf. Group I, Brillié, H., vol. 1, p. 38.

with the same rapidity. It is therefore of interest to examine the problem of increases of temperature in respect of the limiting case in which all the molecules of the lubricant retain the heat resulting from the drive and drag imposed upon them. The increases of temperature in every point of the film can then be calculated. In fact, at every point there is a passive resistance corresponding to the slope of the tangent to the speed curve, a relative speed corresponding to that same slope, and a flow corresponding to the abscissa of the curve. All the elements are thus available for the calculation of the temperature rise since the flow is known and all the heat is supposed to remain in the oil layer. These calculations have been made for the case where the speed curve is a straight line, and for the case of a parabola having its apex on the fixed surface (Brillié 1935).

When the apex of the speed curve is on the fixed surface, the heat produced at a point is proportional to the square of the slope of the tangent to the speed curve, and proportional to the square of the distance from the fixed surface (property of the parabola). This also applies to the flow: the temperature will therefore be the same at any point of the same section.

When the speed curve is a straight line, the heat produced is equal for every point of the section; the flow tends towards zero in the neighbourhood of the fixed surface; the temperature tends towards infinity at the points of the section next to this surface.

These general considerations indicate that it is possible that, under certain running conditions, there might arise, in certain points of the oil layer, large local increases in temperature, having no connexion with the temperatures that may be found in the pipework, in a sump, or even in the bearings. These increases of temperature will occur near the fixed surface; they will appear especially when the relative speeds of the layers are considerable, i.e. for a speed of the moving surface, when, for some reason, the film is very thin.

Sometimes blackish deposits are found on the bearing surfaces. These sediments come from the lubricant. The decomposition of the lubricant may be due to the considerable increases of temperature mentioned above, combined with the pressures resulting from the running of the bearing. The cause of the deposit is insufficient thickness of the film, that is to say, too high pressures or insufficient oil supply.

It thus seems advantageous to adopt devices corresponding to speed parabolas having their apex on the fixed surface: this would be the case with films of uniform thickness, with an upstream nozzle of suitable section for priming, and a downstream nozzle designed to avoid the dispersion of the film. With a complete film and slight eccentricity, the speed curves approach a straight line: therefore very high temperatures could arise at the surface of the bearing. These considerations reveal new advantages in favour of the partial film as against the complete film.

Increases of Temperature in Relation to the Length of the Film. When the speed of the moving surface is small, and the thickness of the film is great (light loads), little heat is produced, and the oil supply may be very small and wick lubrication should suffice. The influence of the supply is then negligible: a stable condition is attained and the heat produced is completely transmitted by conduction or lost through radiation.

With high speeds and a thin film (heavy load or small clearance) wick lubrication becomes insufficient, and oil circulation must be employed, circulation being the best way to remove the heat evolved. Usually the oil flow will be greater than the "film flow," i.e. the quantity of oil which flows in each transverse section and which would be equal to the quantity of oil supplied to the bearing, were the oil of the film entirely replaced at each revolution. Oil losses at the edges of the bearing are neglected. The pressure of the oil supply is usually much lower than the pressures in the film; therefore the excess supply of oil and the oil of the film will not be mixed. The oil supply produces a kind of a dam in the bearing, excess oil being lost at the edges. The film itself arises downstream below the dam, being dispersed at the edges when the molecules, after a complete circuit, reach the upstream area above the dam. There will therefore be a *progressive increase* of the temperature of the molecules, a maximum being reached if the heat remains entirely in the oil layer during the time corresponding roughly to a complete revolution, a time which will be very short for the molecules close to the journal if the speed is great ($\frac{1}{60}$ second for a speed of 3,600 r.p.m.).

If it is assumed that the shape of the speed curve approaches that of the parabola having its apex on the fixed surface, the temperature will be uniform in each transverse section, according to the height of the film. If, however, the variations in thickness of the film are small, it can be assumed that the heat produced per unit length is the same going from upstream in a downstream direction, and the variations in temperature of the film according to its length can be calculated. But the variations of the coefficient of viscosity with the temperature must follow some law. It is approximately true to state that, between 20 and 120 deg. C., the coefficient of viscosity of most oils varies inversely as the 2·6 power of the temperature ($\mu \times t^{2 \cdot 6} = \text{constant}$). The problem is therefore completely defined. It is possible to make the calculations once for all (Brillié 1935b); the results obtained by the author are summarized in another publication (Brillié 1935a). The principle employed is as follows: the running conditions of the film being defined, it is possible to calculate the temperature of the film τ for the last downstream section, if the coefficient of viscosity has the same value

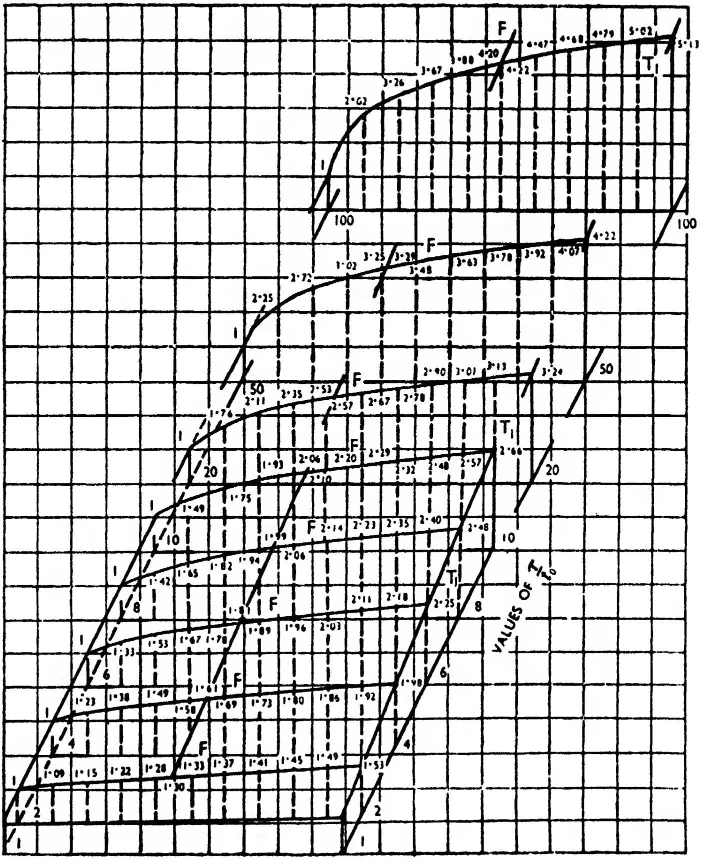

Fig. 1

as for the first upstream section at temperature t_0. This temperature τ is absolutely fictitious, and may be very high (1,000 deg. C. and more), without affecting the utility of the calculations, with the sole reserve that the downstream temperatures arrived at for the film remain below about 120 deg. C., a temperature at which the hypothesis of the variation of the viscosity coefficient would cease to be sufficiently approximate.

For various values of the ratio $\tau/t_0=1, 2, 4, 6, 8, 10, 20, 50$, and 100, the author has calculated the progressive increases in temperature of the film from the upstream section to the downstream section, and curves

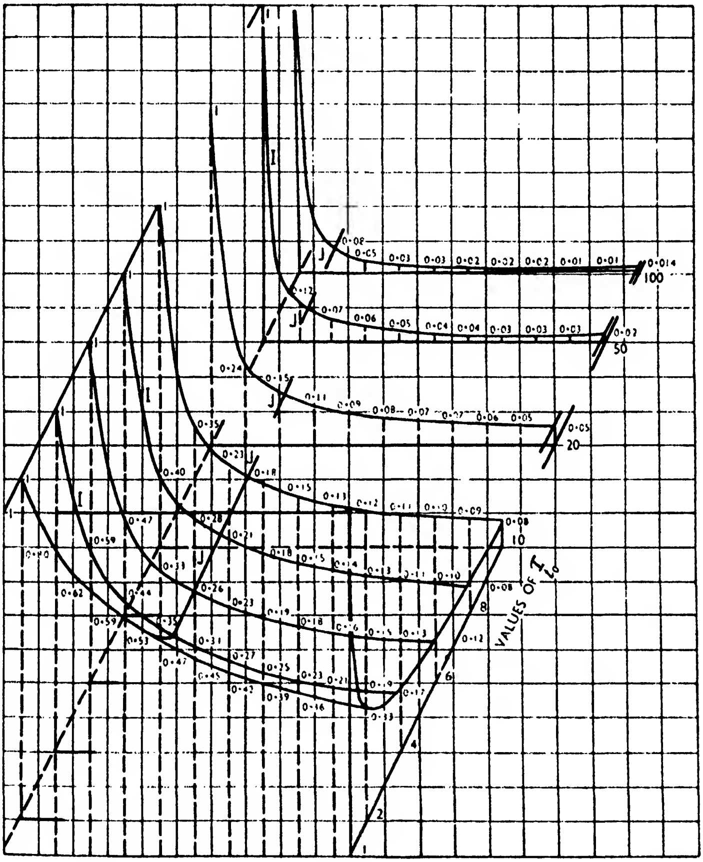

Fig. 2

have been plotted, giving values of the ratio of the temperature t at various points of the film to the initial temperature t_0, and also the values of the ratio of the coefficients of viscosity at various points of the film to the initial coefficient of viscosity (Figs. 1 and 2). The influence of the increase in temperature of the film is allowed for by adopting the corrective coefficient of temperature μ_m/μ_0 for the expression μ_0 of the coefficient of viscosity.

If the heat lost by conduction and radiation is neglected, and it is assumed that, in the region next to the bearing surface, the bush has an average temperature t_m corresponding to the temperature of a section M of the film, then, under stable conditions, there will be, downstream from M, transmission of heat from the film to the bearing surface; and, upstream from M, transmission of heat from the bearing surface to the film. For this reason the variations in temperature shown by the curves will be slightly lower.

When oil is supplied in excess it may be assumed that the excess oil and the oil of the film do not mix. The excess oil will carry away some of the heat, thus lowering the temperature t_m of the bush, so that the final temperature of the film will decrease. The same conclusion is arrived at if the heat is lost by conduction and radiation.

The results of calculations for a normal bearing of a high-speed engine may now be given. The appropriate data were as follows: $D=40$ mm.; $r=0{\cdot}01$ mm.; $N=3{,}000$ r.p.m.; $V=6{\cdot}28$ metres per sec.

With the speed of 3,000 r.p.m., the time necessary for a molecule to make the circuit from upstream to downstream in the layers adjacent to the journal is $\frac{1}{50}$ second. It is during this very short time that the increases of temperature mentioned in Table 1 occur. These increases of temperature are calculated for oil temperatures in the upstream section of 20, 40, 60, and 80 deg. C. respectively.

The figures in Table 1 have been calculated on the supposition (in

TABLE 1. INCREASE IN TEMPERATURE ACCORDING TO THE LENGTH OF THE FILM

1 Tempera- of the upstream section of the film, deg. C.	2 Coefficients of viscosity corresponding to temperatures (1), poises	3 Theoretical temperatures of the downstream section of the film, deg. C.	4 Downstream increases of temperature of the film from upstream in 0·02 sec. (3–1), deg. C.	5 Average temperature t_m (approximate temperature of the bush), deg. C.	6 Ratio of coefficients of viscosity for the upstream and downstream sections
20	2	96	76	74	0·08
40	0·7	116	76	90	0·17
60	0·26	120	60	98	0·33
80	0·12	128	48	104	0·52

agreement with the National Physical Laboratory experiments) that passive resistances are formed on the entire circumference of the bearing.

The Elementary Film. With the elementary film the fact must be considered that the restraint on the journal passes from $(V/e)\times\mu$, corresponding to the linear form of the speed curve, to $(2V/e)\times\mu$, corresponding to the parabolic form of the speed curve (with the apex on the fixed surface). On the other hand, the average speed changes from V/2 to V/3. For the same length of film, the average rise in temperature would be increased in the ratio 1/3. But if the length of the film is reduced, for instance, in the ratio of the total circumference to a 60 deg. arc, i.e. by 6 to 1, the increase in temperature is finally reduced by about one-half.

Values of τ/t_0 and of the Corrective Temperature Coefficient. The fictitious temperature τ will be the higher the greater the speed V and the less the clearance. With V=6 metres per sec., a film 0·01 mm. thick and 12 cm. long (automobile engine)

$$\tau-t_0=672\times\mu_0,$$

which leads, with a viscosity μ_0 of 2 poises, to

$$\tau-t_0=1{,}344 \text{ deg. C. (fictitious temperature)}$$

or, for $t_0=20$,

$$\tau/t_0=68.$$

The curves show that, for such values of this ratio, the oil becomes hot immediately; with $t_0=20$ deg., the temperature t is 80 deg., or an increase of temperature of 60 deg. at two-tenths of the film length, that is to say, in 1/250 second. The value of the corrective coefficient of temperature is less than 0·10; hence the great importance of the corresponding corrections and the necessity for taking them into consideration for the study of the phenomena of viscosity.

Experimental Verification. It would, of course, be difficult to verify experimentally the above theoretical results in detail; but they have easily been verified as a whole.

For each temperature of the oil in the sump and for a given speed, there is an average value of the coefficient of viscosity, a value of the passive resistance per second, and a value for the heat produced and removed by the circulating oil to the sump per second.

The rapidity of the increase in temperature of the oil per second in the sump was calculated, and by graphic integration a temperature-time curve for the sump was obtained. The curves were in very satisfactory agreement, considering the complexity of the question, with the curves obtained experimentally with an automobile engine (Brillié 1935b).

Conclusions. Viscous friction is considered to be the result of the reciprocal action of the fields of force of the molecules constituting the superimposed layers of lubricant comprised between the bearing surfaces. For every molecule, there is a sudden increase of the vibratory energy, and, consequently, a sudden increase in the temperature. Hence, for any point of the film there is an increase in temperature which can be calculated in accordance with the theory of viscosity, subject to the reserves mentioned above. These sudden increases in the temperature of the oil, which cause smaller increases in temperature of the bearing surfaces, are very important with great speeds and small clearances. They increase rapidly with the diameter, the number of revolutions, and the accuracy of fitting.

For high-speed engines (turbines, electric motors, Diesel engines) the layouts generally adopted, which correspond to the passive resistances of the complete film, usually give high increases in temperature. These high increases in temperature may be avoided by supplying excess of oil, and by special arrangements so that successive elementary films (films of limited length) replace complete films and are fed with appropriate nozzles. Devices to prevent the dispersion of the film, and the reduction of the bearing surfaces (central grooves) and their suppression where they only cause passive resistances and do not lead to the development of pressure also help.

These conclusions are in agreement with those results which yield the most advantageous coefficients of friction.

II. Surface Conditions

Oiliness and surface irregularities correspond to special running conditions of the bearings which the author describes as "surface conditions". Surface conditions may have an important influence on the passive resistances, the relative thickness of the film, the increases in temperature, and the coefficient of friction.

Taking the case of a perfectly smooth bearing surface and a lubricant without oiliness, the bearing surface will produce a field of force which corresponds, for friction occurring between the surface and the lubricant, to a very high increase of the ordinary coefficient of fluid friction corresponding to two adjacent layers of the lubricant. To this local increase of the coefficient of fluid friction, corresponds an "adherence" of the first layer of the lubricant. This adherence is complete, which means to say, that the extreme layers of the lubricant, in contact with the bearing surfaces, make one with those surfaces.

If the lubricant possesses "oiliness", then the active molecules settle on the bearing surfaces, forming a covering ("epilamen"); the field of force emanating from the metallic surfaces is neutralized, and for the first layer of the lubricant in contact with the covering, the friction is much less than that corresponding to the coefficient of viscosity. The partial neutralization of the field of force of the bearing surface is not confined to the first layer, but extends to a certain number of adjacent layers, so that three zones may be considered to be present in the oil bed (Fig. 3), namely, a middle zone where the coefficient of fluid friction corresponds to the coefficient of viscosity; and two extreme zones or oiliness zones, where the coefficient of friction decreases progressively in value, from the neighbourhood of the middle zone to the neighbourhood of the surface.

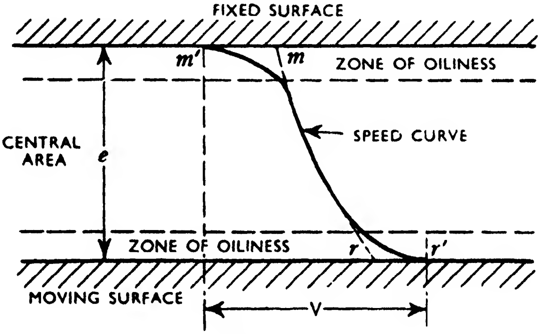

Fig. 3

An experiment due to Hirn suggests that the ratio between the coefficient of fluid friction along the covering μ_c and the coefficient of viscosity μ_e is very small, of the order of 10^{-3}. Various considerations, confirmed by laboratory experiments (Woog, Trillat, etc.) show that the height c of the zone of oiliness is very small, of the order of 0·000,010 mm. Under these conditions, theoretical considerations (Brillié 1935a) suggest the existence of an "oily sliding" along the bearing surfaces. This "sliding" is expressed by a "corrective coefficient" which influences the speed of the moving surface in the formulæ of the elementary theory of viscosity. The approximate value of this coefficient can be taken as $1-(g/e)$, where e is the thickness of the film (formula of viscosity), g is of the order $(\mu_0/\mu_e)\times c=10^{-2}$ mm.

The value of the coefficient of friction obtained in this way is in agreement with experimental results. Thus:—

(1) The coefficient of friction decreases progressively as oiliness increases (increase of coefficient g), but only to a certain extent.

(2) For a given percentage of active molecules (for instance, a known percentage of colloidal graphite), the decrease of the coefficient of friction is greater the smaller the thickness of the film e.

(3) When the thickness of the film e is of the order of a few hundredths of a millimetre, there is a considerable reduction of the coefficient of friction, and the correction coefficient may reach values such as 0·3, which shows that g may have a value of the order of a hundredth of a millimetre.

The decrease of the speed V, in the central zone of viscous friction, a decrease which is due to the "oily sliding", has the obvious consequence that the thickness of the film diminishes. When this thickness becomes of the same order as the surface irregularities, the influence of these irregularities prevails over the influence of oiliness. This is confirmed by experiment, which shows that the coefficient of friction increases with both grease and oil when the percentage of colloidal graphite exceeds a certain value.

Influence of Surface Irregularities. The height of the zone of oiliness is at the utmost of the order of 1/100,000 mm. The polish of the surfaces is far from being of the same order of size. Therefore the study of oiliness cannot be separated from the study of the influence of the surface irregularities.

Interesting information on the nature and extent of these irregularities is given by Shaw (1934). According to Shaw, the surface irregularities on the turned surface of a crankpin are of the order of about 0·003 mm. Turning a bearing on the lathe gives irregularities ten times greater in height, and it is with this in view that Fig. 4 has been drawn, enlarged 1,000 times in height and length, to show the effect of the irregularities of the bore of a bearing. From the author's previous paper (Brillié 1929) it is concluded that, across the hollows "oil rollers" will occur on which the true film will run. Across these oil rollers, extreme direct and return currents, with, in the centre, turbulence or eddies, will probably occur (Brillié 1937). The effect of the secondary irregularities of surface will be to moderate the movement of the oil rollers and thus produce additional passive resistances. The restraint of the oil roller also affects the film itself, so that the combined effect of the chief and secondary irregularities on the film is contrary to that of oily sliding. The extent of this effect can be represented by a coefficient of correction such as $\{1+(h'/e)\}$, which will influence the speed V in the formulæ for viscosity, just like the corrective coefficient for oiliness. On the other hand, from the author's study of oil pockets, it appears that the hollows correspond to regions where the pressures are considerably reduced, or even suppressed. This leads to a new corrective factor for the expression p_s, which will be $1-(h''/e)$, heights h' and h'' being connected with the height of the irregularities of surface.

Total Corrective Coefficient. It thus appears that, taking into account the combined influence of oiliness and of surface irregularities, the

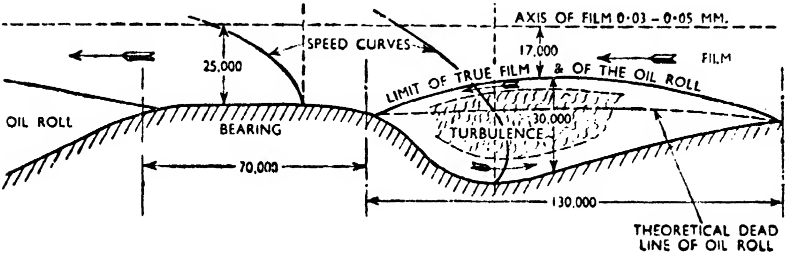

Fig. 4

coefficient of friction which was obtained in the theoretical study of viscosity (case of partial films) becomes

$$\mu = 10^{-3} \times 0{\cdot}46 \times \mathrm{K} \times \sqrt{(\mathrm{N}/p_s)\mathrm{Z}} \times \{1-(1/2)\}(g/e)\frac{1+(\frac{1}{2})(h'/e)}{1-(\frac{1}{2})(h''/e)}$$

The variations of the values of the coefficient of friction with speed will naturally depend upon the relative values of g, h', and h'', as well as of the values of these three factors in relation to the thickness e of the film.

Experimental Verification. The formula for the coefficient of friction obtained in this way agrees with experimental results. The correction coefficient for oiliness has already been shown to agree with the experimental results. Further, agreement has also been found to obtain with regard to well known peculiarities of the coefficient of friction for small values of the speed coefficient.

Imperfect Lubrication. It is well known that the friction-speed curve falls progressively with the speed to a critical point, afterwards suddenly rising hyperbolically for lower speeds. It is often stated that this hyperbolic portion of the curve corresponds to "mixed lubrication", "oily lubrication", or "imperfect lubrication", etc. The expression "oily lubrication" appears to have been very badly chosen for lubrication corresponding to less advantageous values of the coefficient of friction, whereas oiliness is a quality which is required in lubricants. The expression "mixed lubrication" suggests that oiliness intervenes, whereas, were it so, the coefficients of friction would show better values. The author prefers the expression "imperfect lubrication".

Consider a bearing under proper conditions of viscous friction, high values of the speed coefficient, and, consequently, high values of the film thickness *e*; the three coefficients of correction will have but a slight influence because of the relatively small values of *g*, *h'*, *h"*, with regard to *e*.

Let us suppose that the thickness of the film decreases progressively: after this period (1), then:—

(2) The lubricant being supposed to have great oiliness, the coefficient of correction for *g* will take effect and the values of the coefficient of friction will become lower than those resulting from the ordinary formula of the theory of viscosity, the difference increasing progressively.

(3) For a normal bearing, corresponding to the conditions set out in Fig. 4 for the valleys and hills corresponding to the irregularities of surface, when the thickness of the film becomes of the order of 0·05–0·06 mm., the influence of surface irregularities will be felt, especially that corresponding to the denominator of the coefficient of correction: the curve of the coefficients of friction will become hyperbolic when the thickness of the film is of the order of *h"*.

In fact, what is known as "oily lubrication" or "mixed lubrication" corresponds to purely viscous phenomena, in which the influence of surface irregularities intervenes.

(4) For a film thickness of an order less than 0·0001 mm., effects resulting from the reciprocal actions of both "epilamens" may occur. These effects correspond to reductions in the values of the coefficients of friction.

(5) For a certain value of the relative speed of two bearing surfaces, the relative speed of any one of these surfaces and of the adjacent layers increases as the film thickness decreases. There comes a moment when, under the influence of this speed, the molecules of the bearing surfaces are snatched away: this is the phenomenon of "wear", which will increase as the thickness of the film decreases.

(6) If the thickness of the film continues to decrease, a moment comes when but a few more or less regular layers of fluid molecules

remain between the bearing surfaces, or more exactly, between the tops of the irregularities of these surfaces: the slightest disturbance causes contact or direct action of solid molecules on solid molecules, so leading to seizure.

Running-In. These general considerations provide an explanation for the phenomena occurring during running-in. Further discussion would lead beyond the limits of this paper, and mention may merely be made of a publication by the author (Brillié 1937b) of some interesting results obtained in the Isothermos Laboratory (running-in phenomena with surface speeds of about 3·50 metres per sec. and pressures in the oil layer of 800 kg. per sq. cm., which ceased with a pressure in the oil layer of 300 kg. per sq. cm.), and of the experimental results set forth by Shaw (1934).

These different observations lead to the conclusion that wear is not due to the hydrodynamic effect of the lubricant acting on the molecules, but rather to viscous friction which produces a slow erosion of the metal. Oiliness increases the relative speed on the bearing surfaces, but considerably lessens the intensity of the fields of force corresponding to fluid friction; there may be an increase in the intensity of the running-in phenomena, but with relatively smaller increases of temperature (combined influence of the increase of relative speed and of the reduction of the intensity of the fields of force). Alternatively there may be a reduction of the extent of running-in, as appears from Shaw's experiments, under the influence of the reduction in value of the coefficient of fluid friction ("oily sliding").

The combined influences of temperature rise, oiliness, and surface irregularities bring about the complexity of the phenomena of friction, and confirm the author's opinion* that neither theory alone nor experiment alone can settle the basis of bearing technique.

The author extends his thanks to Messrs. Boswall, Clayton and Jakeman, Thomson, and Shaw for having furnished detailed information about their investigations.

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* cf. Brillié H., vol. 1, p. 38.

SOME OBSERVATIONS ON THE EFFECT OF A NUMBER OF ADDED SUBSTANCES ON THE LUBRICATING PROPERTIES OF OILS

By H. W. Brownsdon, M.Sc., Ph.D.*

This contribution is confined to the consideration of the effect of additions to a mineral oil in so far as they influence the wear of " 70–30 " brass when in frictional contact with a revolving hardened steel wheel. In an attempt to assess the relative value of such additions many comparative experiments have been carried out on a machine designed by the author.† In this machine a small hardened steel wheel 1 inch in diameter, 0·1 inch thick, and with a periphery of 0·05 inch radius, is rotated at a speed of 500 r.p.m., and is brought into frictional contact with a flat sample of the metal to be tested for wear under a predetermined load, the lubricant being dropped on to the revolving periphery of the wheel. An oval wear impression is made on the metal sample, the length of which can be accurately measured and taken as a criterion of the amount of wear that has taken place during a specified time, a suitable time for running being 15 minutes. When using a straight mineral oil having a Redwood viscosity of 40 seconds at 200 deg. F. and the periphery of the hardened steel wheel polished with No. 00 emery paper, a wear impression of about 4 mm. in length is given when the sample is 70–30 brass, the load 20 lb., and the time of running 15 minutes. The reduction in wear produced by additions made to such an oil is revealed by a reduction in the length of the wear impression; the lower this length the less the wear. The figures given in the following tables are the lengths of the wear impressions in millimetres given by the various oils. For the purpose of facilitating consideration of the results obtained, the added substances have been arranged in groups of similar chemical constitution.

A complete consideration of all the results is impossible in a general discussion of this nature, but a brief analysis of some of the more interesting results may throw some light on the part that the chemical and physical properties of added substances may play in lubrication and wear problems. The results given are purely experimental, and restricted to one pair of metals, and they are therefore put forward only as a contribution to the study of one limited aspect of wear and lubrication problems. Whilst this is so, experimental work covering a wider field indicates that when using other metals, oils, and conditions

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* I.C.I. Metals, Ltd., Birmingham.

† Jl. Inst. Metals, 1936, vol. 58, No. 1.

TABLE I. LENGTH OF WEAR IMPRESSIONS GIVEN BY VARIOUS OILS

The figures in the tables give the length of wear in millimetres.

Mineral oil, Redwood viscosity at 200 deg. F. = 40 sec.
Wheel of hardened steel with smooth No. 00 emery finish.
Sample, " 70–30 " brass; speed, 500 r.p.m; time, 15 minutes; load, 20 lb.

Added substance	Percentage of added substance								
	100	10	5	2	1	0·5	0·1	0·01	0·005
Rape oil	0·97	1·10	1·34		4·24				
Lard oil	1·01	2·80	3·45		4·20				
Castor oil, 0·90 per cent free acid	0·84				4·04				
Castor oil, 0·14 per cent free acid	0·89								
Ricinoleic acid	0·85		1·07		3·0				
Oleic acid	1·10		1·41		3·2		4·35		
Stearic acid					1·30	2·35	3·76		
Butyric acid							1 00	4·00	
Acetic acid							1·00	3·50	
Formic acid, 90 per cent							0·90	0·90	0·95
Copper oleate					4·65				
Zinc oleate					4·60				
Aluminium oleate			1·39	2·91	4·30				
Lead oleate					4·50				
Tin oleate				1·68	2·87	3·10			
Chromium oleate			1·23	2·68	4·33				
Copper stearate					4·45				
Zinc stearate					4·26				
Aluminium stearate				2·31	2·64	3·01			
Lead stearate					4·51				
Tin stearate			1·40	1·65	2·74				
Chromium stearate			1·52	2·16	3·00				
Ammonium formate								1·29	

of test, the results obtained have much in common with those now put forward for discussion.

The main interest of these experiments centres around the chemical and physical characteristics of those substances which have a marked effect on wear reduction, even when added in small quantities of 1 per cent or less. Many of these substances are acidic in character, or give rise to acid decomposition products under the temperature and pressure conditions present at the seat of friction. In many cases such decomposition is facilitated by the addition of small quantities of water. The

TABLE 1, concluded

Added substance	Percentage of added substance						
	100	10	5	1	0·5	0·1	0·01
Chloroform	1·30			1·00	2·32	3·82	
Carbon tetrachloride	1·45			1·00	2·51	3·90	
Tetrachloroethane	1·20			1·06	2·35	4·00	
Hexachloroethane				1·00	1·54	3·76	
Monofluorotrichloroethane				1·18			
Trifluorotrichloroethane				1·50			
Trichloroethylene	2·20	1·00	1·05	1·40	3·40	4·22	
„ +H_2O					1·50		
Tribromoethylene				1·20			
Cereclor No. 1	1·10	0·95	1·00	2·90			
„ +H_2O				0·92	1·73	4·21	
Anglamol A	1·10	1·00	1·05	4·00			
p-Dichlorobenzene				4·30			
„ +H_2O				1·00			
Hexachlorobenzene			1·50	3·55			
Dichloroethylether	1·30	1·00	1·10	3·80			
„ +H_2O				1·00	1·13	1·74	3·69
Chloral				0·98		1·51	4·00
„ +H_2O						1·10	3·00
Trichloroacetic acid						1·00	4·00
Trichlorophenol		1·00	1·15	2·90			
„ +H_2O				0·96	1·94	3·72	

+H_2O indicates that water was added in quantity equivalent to that of the added substance.

most active acids are those of a volatile nature or readily decomposable, and therefore capable of setting up vapour or gas pressure between the rubbing surfaces. Acids of low molecular weight are generally more active than those of high molecular weight. Stable fixed acids such as perchloric and phosphoric acids do not behave as wear-reducing additions. Sulphuric acid is anomalous in so far as it reduces wear when added in concentrated form, but not when diluted, which might be explained by the fact that reaction between strong sulphuric acid and the metals may take place with formation of sulphurous acid, which is an active wear-reducing addition. Pyrogallic acid is just as active when made alkaline as when added in the acid condition. Such considerations lead to the view that other properties besides mere acidity must be looked for to explain the activity of these acidic additions.

The reducing nature of a number of the more active substances calls

TABLE 1, continued

Added substance	Percentage of added substance							
	100	10	5	2	1	0·5	0·1	0·01
Methyl alcohol	2·18			2·60	4·10			
Ethyl alcohol	1·56		1·60		3·16			
Ethyl ether			4·15					
,, +1 per cent H_2O			1·45		3·42			
Formaldehyde (40 per cent)	3·40				1·00		1·12	4·15
Acetaldehyde					1·45	3·32		
Ethyl formate	1·83	1·08	1·12		1·05	1·00	0·96	4·20
,, $+H_2O$								3·85
Ethyl acetate	2·10	1·15	3·05					
,, $+H_2O$			1·21		2·75	2·86	3·45	4·16
Butyl acetate	2·21	2·17	3·23					
Amyl acetate	2·03	2·97	3·54					
Acetone		3·95			3·65		4·01	
Petroleum ether		4·45	4·20			3·71	3·85	4·01
Formamide					0·95	0·96	3·88	
Benzene		4·10	4·09		3·80		4·00	
Phenol			1·42		3·00			
,, $+H_2O$					0·95	1·00	2·97	4·07
Resorcinol							3·32	
Pyrogallol							1·10	4·00
,, $+H_2O$							0·95	3·25
Picric acid					1·10	1·00	3·05	
,, $+H_2O$						1·05	3·64	
Trinitrotoluene					1·95	3·55		
,, $+H_2O$					3·97			
Benzoic acid					1·55		3·85	

attention to the part that such characteristics might play in wear reduction. The very serious and accelerated wear that can result from active oxidation is shown by the results given by hydrogen peroxide additions, and it might be argued therefore that substances having reducing properties would act as protective agents against oxidation, and so lead to a reduction in wear. The behaviour of ammonia as an active addition agent for the reduction of wear is of interest in this connexion, since ammonia protects steel from oxidation, even in the presence of strong oxidizing agents, and it reacts with copper to form cuprous salts, powerful reducing agents. In addition ammonia is very

TABLE 1, continued

Added substance	Percentage of added substance						
	1	0·5	0·1	0·05	0·01	0·005	0·001
Hydrofluoric acid (40 per cent)			1·14	1·38	1·87	1·90	
Hydrochloric acid (35 per cent)					1·00	3·50	
,, ,, dry gas					1·05	0·95	
Perchloric acid (60 per cent) .	3·39	3·72	3·62		3·90		
Hydrobromic acid (46 per cent) .	1·44	3·92	4·30				
Hydriodic acid (65 per cent) .	1·20	3·22	3·92				
Chlorine		1·02			1·00	1·05	4·04
Bromine			1·13	4·05			
Iodine	1·41	1·36	2·95	3·90			
Aluminium chloride + H_2O .			3·90				
Zinc chloride + H_2O . . .	3·89		4·00				
Stannous chloride + H_2O . .			1·10		4·16		
Arsenious chloride . . .					1·13	3·53	
Arsenious iodide . . .	1·27	1·42	3·14				
Silver nitrate + H_2O . . .			1·30		3·60		
Sulphuric acid (conc.) . .					1·00	4·32	
,, ,, + H_2O . .					4·00		
Phosphoric acid . . .	4·05		4·00		4·00		
,, ,, + H_2O . .			4·05				
Selenious acid + H_2O . .	3·96		4·30				

volatile and may set up gas pressure at the seat of friction. The behaviour of pyrogallic acid would appear to be limited to its reducing properties.

The behaviour of the various gases is of special interest as tending to indicate that the presence of a small amount of practically insoluble gas in the oil has a marked effect on wear reduction. With a gas such as carbon dioxide it is difficult to conceive that its action is due to any other than a purely physical cause. The case of hydrogen sulphide is somewhat complicated by sulphide film formation, but the fact that 0·005 per cent of such a gas brings about wear reduction equal to that obtained by 100 times more sulphur or sulphur compound might reasonably be attributed to conditions associated with the presence of a gaseous constituent. That there is no chemical combination with the oil is shown by the fact that the gases are easily removed by blowing air through the oil, after which the oil gives normal wear impressions of about 4 mm.

TABLE 1, concluded

Added substance	Percentage of added substance							
	100	10	5	1	0·5	0·1	0·01	0·005
Carbon dioxide (gas)						0·03%	=1·18	
Ammonia (gas)						0·96	2·32	3·63
Ammonium carbonate						0·96	2·63	
Sulphur dioxide (gas)						1·00	1·20	2·80
Hydrogen sulphide (gas)								1·50
Ditto, after driving off H_2S								4·00
Sulphur			1·20	1·20	1·47	2·6	3·97	
Sulphur chloride				1·38	1·30	1·12	1·11	3·74
Sulphur iodide				1·67	1·53	1·28	4·00	
Floyd's base	0·85	3·12		3·52				
	Percentage addition							
	83	17	8·5	1·5	0·15			
Water	1·85	1·90	2·11	2·51	3·89			
Hydrogen peroxide (20 vols.)	8·00*	1·95	2·00	2·85	4·00			
(H_2O_2 per cent)	5	1	0·50	0·10	0·01)			

* 4 minutes test only.

Some evidence of the effect of films, both metallic and non-metallic, is provided by some of the substances used in the experiments. Wear reduction brought about by the addition of stannous chloride and arsenious chloride is of interest when compared with the negative effect given by aluminium and zinc chlorides. All these salts hydrolyse, giving free hydrochloric acid, but the results indicate that the effect of tin and arsenic chlorides is probably due to metallic film formation rather than to acidity produced by hydrolysis. On the other hand their action may be also associated with their reducing nature. The active effect of silver nitrate, which is neither decomposed by hydrolysis nor has reducing properties, goes to confirm the metallic film explanation of their action. The behaviour of sulphur and sulphur compounds is generally attributed to sulphide film formation, and although this may be so, the active effect of sulphur dioxide cannot be ignored, and it may be that this is formed in sufficient quantities to have some effect on wear reduction. The effect of chlorinated hydrocarbons has been attributed to chloride film formation, but it may be that their behaviour as wear inhibitors is also associated with the reducing properties of such films and the protection they therefore afford against oxidation of the metal surfaces.

That a neutral inert substance such as water has some effect on reducing wear can only be explained by its acting in a physical manner, probably that of exerting vapour pressure at the seat of friction. Some other inert substances such as alcohol behave in a similar manner. The vapour pressure exerted by such substances will depend not only on their volatility, but also on the solubility of their vapours in the oil, and it will be noted that the most active additions are those of low solubility in the oil; acetone, petroleum ether, and benzene, although volatile, having no appreciable effect on wear reduction. Most of the active substances are inert chemically in so far as any reaction between them and the oil is concerned, the effect they produce disappearing completely on their removal.

When reviewing these experimental results it is evident that no single characteristic of these many substances can be put forward as the cause of their behaviour. Some are undoubtedly decomposed under the temperature and pressure conditions existing at the seat of friction, and their activity is therefore probably due to their decomposition products rather than to the pure substances themselves, and in such cases both chemical and physical properties may play some part in the effect they produce. There is not only evidence that free oxygen can give rise to very serious accelerated wear, but also that the presence of reducing agents may inhibit wear. The behaviour of some of the substances may reasonably be attributed to purely physical characteristics such as vapour or gas pressure produced between the rubbing surfaces. Metallic or non-metallic films may lead to wear reduction. In some cases the desirable chemical and physical characteristics may be associated, as in formic acid, which is volatile, decomposable into gaseous constituents, of low solubility, and also is a reducing agent.

A point of interest is associated with decomposable substances such as chlorinated hydrocarbons, since, as might be anticipated, the greater the load the greater is the amount of decomposition product formed, as shown by the amounts of water-soluble chloride found in aqueous extracts before and after tests made under different loads.

Whilst the results discussed in their present form are purely experimental, they are not without some bearing on more general problems associated with lubrication and wear.

OILINESS IN RELATION TO VISCOSITY

By A. W. Burwell and J. A. Camelford*

It is well known that when the characteristics of thin films of lubricants are examined, phenomena present themselves which cannot be explained by purely hydrodynamic principles. This region of extremely thin film lubrication has frequently been referred to as that of "boundary lubrication". Hersey (1936) explains the term as referring to the mutual action of the two adsorbed layers when there is no free liquid intervening, or at least no liquid retaining its original bulk properties. The property of oiliness is most frequently associated with these phenomena, although there is no reason to assume that this is the only region in which oiliness effectively functions. The definition adopted by the Society of Automotive Engineers is as follows:—

> "Oiliness is a term signifying differences in friction greater than can be accounted for on the basis of viscosity when comparing different lubricants under identical test conditions."

It will be noted that phenomena of wear—abrasion or corrosion, temperature differences, and other modifying influences—are omitted from this concept of the fundamental property of oiliness.

The obvious means of studying the mechanism of thin film lubrication is by comparison of coefficients of friction, kinetic or static.

In the study of oiliness given in this paper, all determinations of coefficients of kinetic friction shown are obtained by the use of one instrument, which is a modification of the Moore-Carvin machine (Fig. 1). The machine consists essentially of a ½ h.p. motor with a hard carbon-steel journal keyed to the shaft. This steel journal forms the shaft of the test bearing; it is exactly 1·668 inches in diameter, and the surface is ground to a highly polished finish. Over this is fitted a steel housing with an inside diameter of 1·678 inches, which is supported by a lug and counterbalanced so that it floats freely on the shaft. The housing is hollow, so that it may be used as a constant temperature or heating bath. This housing supports the circular bearing halves and also provides a means for supplying oil to the system. The bearing halves are each of such a diameter, as to give a projected area of 0·503 sq. in., but a chamfer on the leading edge of each of the bearing halves reduces this area to 0·500 sq. in. They may be made of any bearing metal, but all results reported here are for bronze on steel. The bearing halves have their inner faces machined to the shaft diameter and

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* The Alox Corporation, Niagara Falls, N.Y.

are carefully run in to a smooth fit. The pressure applied to the bearing halves is altered gradually by means of weights attached to lever arms from zero to 5,000 lb. per sq. in. The bearing is lubricated by the oil under examination being fed by gravity, and free feed is maintained by means of a needle valve adjustment in the oil reservoir so that the bearing is flooded continuously. The temperature of the bearing, as read on the potentiometer, is not that obtaining at the actual bearing surfaces, but it does afford a reliable and accurate means of controlling and regulating this temperature. At the actual bearing surfaces the temperature very probably is much greater, but at present there is no means by which it may be readily ascertained (Bowden and Ridler 1936). During tests the motor speed is kept constant. Rigid control is obtained over every factor to be considered in coefficient of friction determinations—rubbing speed, bearing temperature, unit pressures, and lubricant feed. Bearing surfaces are as nearly perfect as accurate machine work with subsequent careful running-in can accomplish, and great care is exercised in keeping them in condition. Abrasion is never allowed to occur during the tests, as light fuses in the motor line blow under light overload such as would be obtained under conditions of high friction.

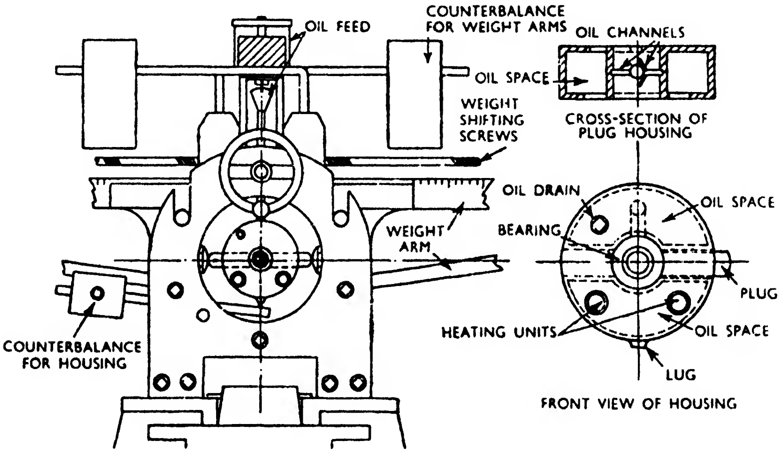

Fig. 1. Moore-Carvin Friction Machine

In plotting data observed when the bearing was kept at constant temperature with varying pressure, it was early seen that, in the range of pressures employed, the coefficient of friction–pressure curves were all similar in structure, although great differences in the numerical values of the coefficients of kinetic friction were observed when these curves were compared. Fig. 2 shows four typical curves of this nature. In each case there is : (*a*) a descending portion, where the coefficients of friction diminish with increase in pressure, followed by (*b*) an ascending portion where the coefficients of friction increase with increase in pressure. It may be accepted that the left-hand portions (*a*) of these curves are those of full fluid-film lubrication. The point at which (*a*) and (*b*) meet is obviously the critical point of minimum friction (Wilson and Barnard 1922). The right-hand portion (*b*) of these curves represents the region of thin film lubrication. In this portion :—

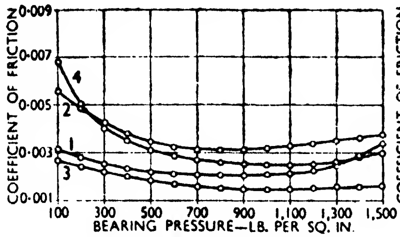

Fig. 2. Coefficient of Friction–Pressure Curves for the following Lubricants :—

Speed, 500 ft. per min.; constant temperature, 210 deg. F.

1. Mid-continent S.A.E.30.
2. Coastal S.A.E.30.
3. Pennsylvania S.A.E.30.
4. Mid-continent S.A.E.30.

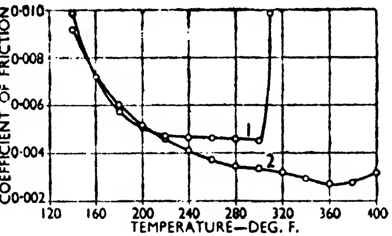

Fig. 3. Coefficient of Friction–Temperature Curves for Two Mid-Continent Oils

Constant pressure, 1,500 lb. per sq. in.

1. Mid-continent S.A.E.30.
2. Blended mid-continent S.A.E.30.

(1) The viscosity of the lubricant as a whole plays no part in the effective action of the lubricant. The metal surfaces of the bearing are separated efficiently by reason of the oiliness of the lubricant.

(2) The range of pressures throughout which the lubricant will act efficiently is not predictable beyond the limits of the observations.

(3) Increase in friction occurs, but no abrasion or metal-to-metal contact over the wide range of pressure employed.

Fig. 3 shows typical curves derived from data obtained in the examination of two samples of Mid-Continent oils, obtained in the open market. One is a straight petroleum oil, the other is known to be treated with an addition compound. With the former there is a viscosity-temperature effect in the region of full fluid-film lubrication between 140-200 deg. F. From this point to 300 deg. F., the system is operating in the region of thin-film lubrication, where the lubricant exhibits such oiliness as it contains, and, as demonstrated by the slightly diminishing values for the coefficient of friction, no metal-to-metal contact occurs. Between 300 deg. F. and the temperature of complete breakdown, the rapid rate of increase in the value of the coefficient of friction indicates metal-to-metal contact with frequent rupture and reformation of the lubricating film until the temperature becomes too high to permit this reformation to continue. Film rupture is then complete, and seizure of the bearing ensues. The curve for the compounded oil shows that between 280 and 310 deg. F., the lubricant acts in the region of thin films, where the oiliness of the oil is the most important factor in determining the path of the curve. Beyond 310 deg. F. up to 360 deg. F. the coefficient of friction again falls, which is rather remarkable when the operating conditions of the bearing are considered. The only rational explanation for this phase is that the thin film separating the metal faces under the given conditions must take the form of a plastic solid rather than of a viscous liquid and that upon further application of heat the melting point of this solid is reached and with further increase in temperature, viscosity effects upon this film are manifested by reduction of the value for the coefficient of friction. From 360 to 400 deg. F., the limit of the test, there is an increase in the value of the coefficient of friction, the rate of increase being progressively greater with rise in temperature.

The generalities to be deduced from these and other experiments are as follows:—

(1) Transition from full fluid film to adsorbed or oiliness film is normally gradual. Specific instances do occur where this broad conclusion does not hold true, for example, where a highly refined oil such as a white oil has been compounded with an additive agent to increase film strength or oiliness. Another notable instance is where Diesel fuel oils are "doped" to ensure adequate lubrication and protection of injection pumps.

(2) Adsorbed or oiliness films may be of any density or physical condition—solid, viscous, or extremely fluid.

(3) Whether or not a bearing is operating under conditions of full fluid-film lubrication the assumption that the viscosity of the lubricating film is identical with the pressure-viscosity of the lubricant as a whole cannot be justified (Everett 1937).

(4) A lubricant can function in three distinct regions without permitting complete breakdown:—

(*a*) Full fluid film region.

(*b*) Adsorbed film or oiliness region.

(*c*) A region in which the lubricating film is repeatedly ruptured and reformed, and metal-to-metal contact occurs without necessarily causing bearing seizure. In this region, however, abrasion occurs, resulting in eventual destruction of the bearing surfaces.

Selective Segregation of Oiliness Constituents by Finely Divided Metal Powders. In order to obtain a more complete picture of the effect of highly polar bodies on the action of lubricating oil and of the effect of finely divided metal on an oil treated with such bodies, a series of experiments were made, using an oil which, of itself, possesses no particular virtues as a lubricant. In the authors' experience, however, it is impossible to obtain a viscous petroleum fraction, no matter however drastically refined or treated, which is entirely devoid of oiliness. The oil used in these experiments was a technical white oil of 130 sec. Saybolt universal viscosity at 100 deg. F. and the tests were made on the Moore-Carvin apparatus. A quantity of the same oil was then treated with a compound available commercially for the purpose of enhancing oiliness, and its behaviour was observed under identical test conditions. Portions of 500 grammes of the compounded oil were treated with 50 grammes of a fine metallic powder. The metals used were lead, copper, iron-by-hydrogen, zinc, and copper and lead together. The treatment consisted in stirring the metal-oil mixture mechanically for one hour at 140 deg. F., and then filtering. A sample of the filtered oil was retained for chemical analysis; the remainder was submitted to friction tests as before.

TABLE 1. SAPONIFICATION VALUES OF THE EXTRACTS

Sample	Saponification value, mg. KOH per gramme
Technical white oil (A)	1·96
Technical white oil+compound (B) . . .	10·94
Sample B after treatment with Pb powder . .	0·56
,, ,, ,, Cu ,, . .	0·28
,, ,, ,, Fe ,, . .	1·68
,, ,, ,, Zn ,, . .	2·24
,, ,, ,, Cu and Pb powder .	0·73

The additive used was an ester of saturated fatty acids of high molecular weight, produced synthetically. Determination of saponification value (Table 1), therefore, gave a ready index of what had occurred chemically during treatment.

After the oils had been filtered, the residual metal powder was extracted with benzol and then with carbon tetrachloride. Saponification values of the extracts were obtained (Table 2).

TABLE 2. SAPONIFICATION VALUES OF THE EXTRACTS

Extract	Saponification value, mg. KOH per gramme
Extract from Pb powder	1·21
,, ,, Cu ,, ...	5·73
,, ,, Fe ,, ...	1·16
,, ,, Zn ,, ...	1·10
,, ,, Cu and Pb powder . . .	3·24

These analyses demonstrated that intimate contact between the lubricant and the metal particles, even in the absence of pressure or other extreme physical forces, leads to the segregation of some active constituents of the lubricant at the metal surfaces. This, of course, does not necessarily mean that the attraction of these polar bodies to the metal surface gives rise to adequate film formation and protection under the conditions which would be met in a bearing. But it does show that in a complex mixture such as any normal petroleum lubricating oil, some constituents of that mixture will be attracted to the metal under given conditions. It must be remembered that though petroleum lubricating oils consist principally of hydrocarbons, there are present in small quantities, other substances containing oxygen, sulphur, nitrogen, or other naturally occurring elements. These compounds, although they may be present only in minute quantities, influence the film-forming characteristics of the lubricants in which they occur.

Examination of the pressure–coefficient of friction curves (cf. Fig. 4) of oils which had been treated with metal powder showed that in every instance, in the higher range of pressures, the values for the coefficient of friction with the treated compounded oil were higher than those of the untreated compounded oil, and were lower than those of the original samples of white oil. Below 800 lb. per sq. in., in several cases (of which Fig. 4 is an example) the oils which had been treated with metal powders gave values higher than those of the white oil. These results are not attributable to any error or to an anomaly. It has already been pointed out that the white oil was not completely devoid of any oiliness characteristics, and it is conceivable that the lubricant originally contained some substances which were preferentially attracted to the metal surfaces upon exposure to the metal in powder form. The authors' laboratory has made numerous examinations of the chemical characteristics of many petroleum oils. In every case acid esters, alcohols, principally secondary and tertiary, alcohol-ketones, and ketones were present in the oil, and in some cases, free carboxylic acids in measurable amounts. In some

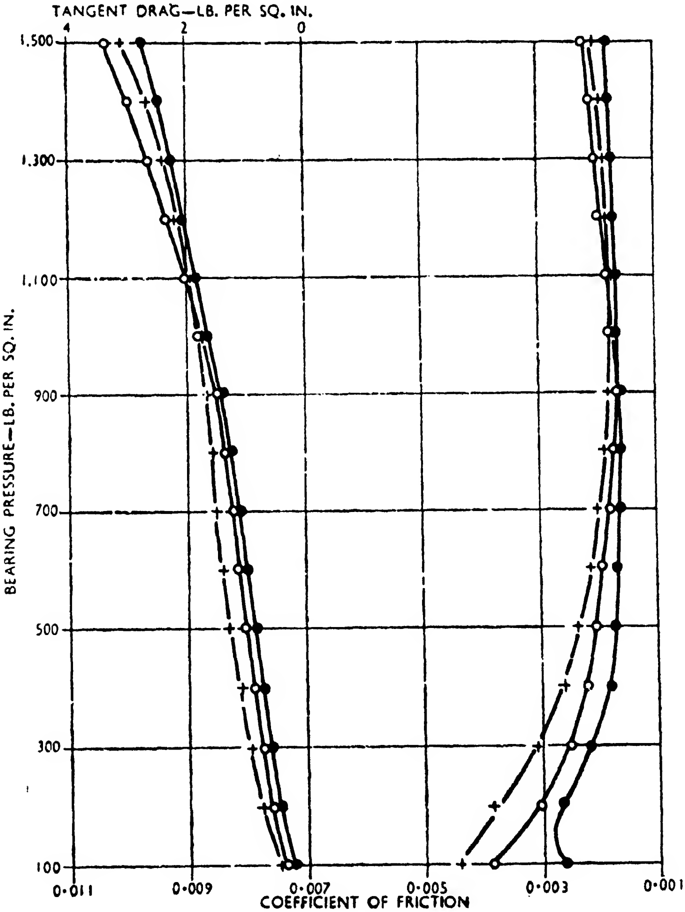

Fig. 4. Coefficient of Friction–Pressure Curves of Oils Treated with Metallic Powder

Temperature, 212 deg. F.; speed, 500 ft. per min.

○—○ Technical white oil.
●—● Technical white oil +0·5 per cent "oiliness" compound.
+—+ Technical white oil +0·5 per cent "oiliness" compound, then treated with 5 per cent copper powder and 5 per cent lead powder.

paraffin stocks which showed admirable lubricating ability these oxygen-containing compounds were present to the extent of from 10 to 15 per cent.

On working out the temperature–coefficient of friction characteristics of the oils, the original white oil showed an ordinary temperature-viscosity relationship until 170 deg. F. was attained (Fig. 5). Beyond this point, gradual increase in the value of the coefficient of friction points to early film rupture and bearing seizure at a temperature of only 250 deg. F. On treatment with the oiliness compound, the coefficient of friction fell steadily until a temperature of 250 deg. F. was attained; from this point to 400 deg. F. the curve is practically a straight line, parallel to the temperature axis. This portion of the curve is definitely in the region of thin-film or adsorbed-film lubrication, and no pressure-temperature-viscosity modifications of the lubricating film exert any influence upon the friction characteristics of the lubricant. Beyond 400 deg. F., the film begins to disrupt with accompanying increase in friction, and bearing seizure occurs at 440 deg. F. After treatment with various metallic powders, the coefficient of friction was higher in every case than before, and seizure of the bearing took place at temperatures varying between 370 and 410 deg. F.

These experiments would indicate that there occurs a selective segregation of polar bodies to metal surfaces which exerts a very pronounced influence upon the behaviour of lubricating films. This influence depends upon such factors as the material comprising the bearing surfaces, and the type and quantity of active substances present in the lubricant. The term "active" does not imply chemical reactivity, but the presence of polar groups which are attracted to the surfaces. Whether the segregation takes the form of a selective adsorption (A.S.M.E. Report 1919; Langmuir 1934), or an orientation of certain of the molecules of the lubricant attracted to the metal surfaces is not definitely decided (Trillat 1928; Clark, Lincoln, and Sterrett 1935).

The authors feel that more evidence has been adduced in favour of the latter hypothesis. In the course of examination of numerous substances proposed for the purpose of increasing the oiliness characteristics of lubricating oils, or of increasing their film strength, it has been found that the degree of efficiency was frequently in proportion to the polar strength of the molecule. Compounds with unattached or residual valencies invariably exhibit a strong attraction for the metal surfaces in a bearing. This property is by no means the sole criterion of the value of these additives. In the selection of compounds suitable for blending with petroleum oils many other factors have to be taken into account, for example, chemical stability exhibited both to the bearing metals and to the lubricant. In consideration of the latter the products of oxidation occurring in the petroleum oils under conditions of severe service cannot be ignored.

When considering the broad applications of oiliness in test or journal

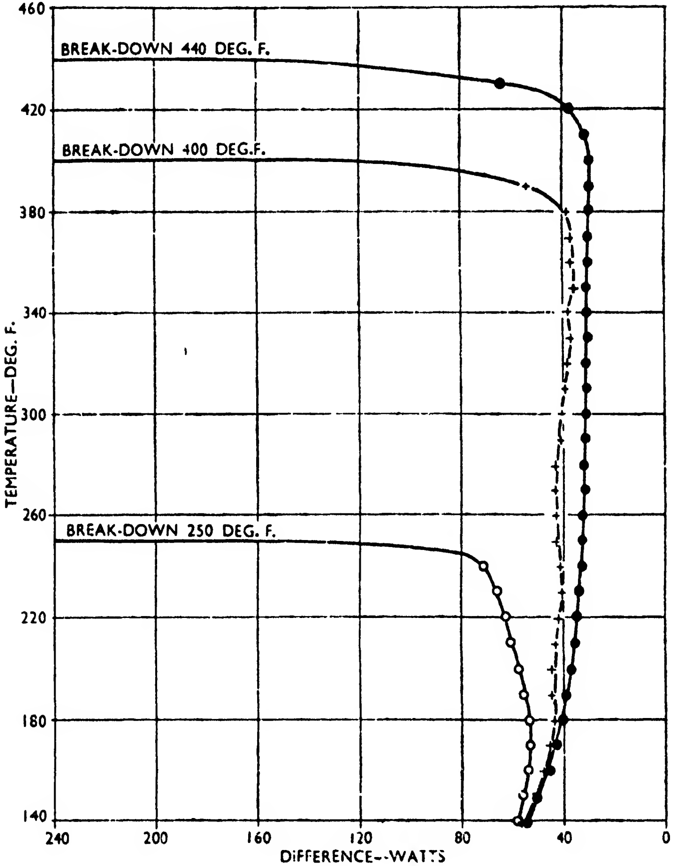

Fig. 5. Effect of Treating Compounded Oils with Various Metallic Powders

Constant load, 1,500 lb. per sq. in.; speed 500 ft. per min. The oils are the same as those listed under Fig. 4.

bearings the additional property of film strength is so closely associated with oiliness phenomena that the conclusion is inevitable that in certain phases of lubrication, film strength is synonymous with oiliness. For example, an asphaltic neutral oil is totally incapable of lubricating a test bearing operating at pressures above 1,000 lb. per sq. in. and temperatures exceeding 300 deg. F. If, however, a small percentage of some halogenated compound is added to this neutral oil, the bearing functions efficiently at pressures well exceeding the limit given above and at temperatures which induce thermal decomposition of the hydrocarbon medium. In this instance it is obvious that the value of the coefficient of kinetic friction for the neutral oil is infinite. With the oil that has been treated, the value of the coefficient of kinetic friction in all probability will not exceed 0·010. To conclude this argument summarily, every possible range of pressures, temperatures, and other controlling factors should be explored thoroughly before logical conclusions can be presented confidently.

The Effect of Segregation on Small Metal Capillaries. Further proof of the strong attraction of some constituents of lubricating oil towards metal surfaces is given by an experiment designed to determine the time required to clog small metal capillaries or the time required to attain a steady rate of flow. The work was suggested by experiments conducted by Wilson and Barnard (1922).

The apparatus used consists of a glass tube tightly packed with strands of copper wire of a uniform diameter of 0·0151 inch and uniform length. Oil was supplied at a constant head, and readings were taken (at 24-hour intervals) of the time required for 10 cu. cm. of oil to pass through the mass of wire. The customary corrections for the temperature-viscosity relationship were applied. As might be expected, rate of change of flow was rapid for the first five days, subsequently diminishing until at the expiration of about 15 days the rate of flow remained fairly uniform (Fig. 6). The oil used in this test was identical with that used in the previous tests. As the oil passed through between the columns of copper wire the polar constituents of the oil were attracted to the metal surface. This process continued until the metal surface exposed was covered with a continuous, uniform layer of polar bodies. Then the rate of deposition necessarily slowed down. Since the segregation could no longer occur directly upon the metal surface but had to build up as a secondary layer on the molecules previously deposited, the attraction due to the polar forces was to some extent inhibited. This process continued, but only until a sufficient thickness of these polar bodies had been deposited on and around the metal to insulate the metal completely from further polar attraction. It was then obvious that the rate of flow would continue at a uniform rate.

Segregation on metal surfaces occurs independently of conditions of high temperature or pressure, but while this attraction exists in materials showing oiliness it does not follow that all substances which exhibit this property of attaching themselves to metal surfaces will increase the oiliness of a lubricant.

Conclusions. To express the results of these experiments very briefly, it can be said that when a lubricant is at work in any system, regardless of any factors which determine its efficiency, the actual composition of the film is not identical with the composition of the oil mass. In other words, the characteristics of the lubricating film are determined by the relative attraction of the metals comprising the bearing for each component of the mixture comprising the lubricant. This is true whether extraneous material has been added to the lubricant or not.

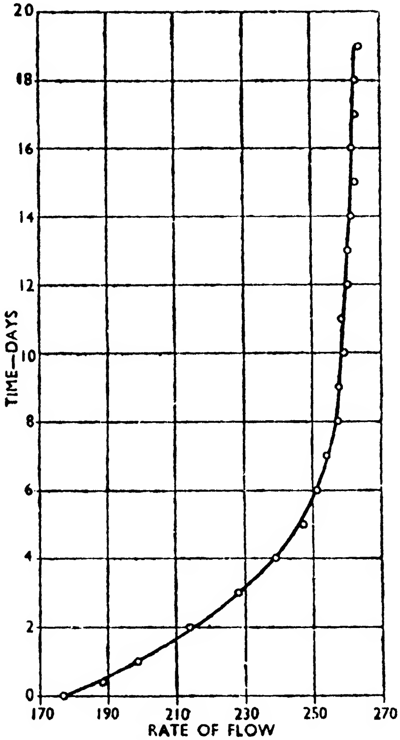

Fig. 6. Flow of an "Oily" Lubricant through Packed Copper Wire

Differences in coefficients of kinetic friction have been noted in many oils possessing the same general physical characteristics but of different origins. Similarly, variations in refining practice cause wide differences in the friction properties of lubricants which may have been derived from the same source. These differences are usually observed over the entire range of pressures and temperatures used. In some cases these variations in frictional qualities of films are of great importance, even in the region of full fluid film lubrication. Observations have been made in the authors' laboratories of several instances where high-viscosity lubricants, specifically airplane engine oils, produced such high coefficients of kinetic friction in the range of full fluid-film lubrication, that the internal friction of the films produced a gradual shearing-off on the bearing faces, resulting in bearing failure, and necessitating lengthy reconditioning of the test bearing. When tests were made, on these same oils, with identical bearing surfaces, but under greatly increased pressures and, particularly, temperatures, very low coefficients of friction were obtained. It may be argued that this represents the results predictable from consideration of the type of formula $f=\phi(ZN/P)$. When it is considered that many lubricants of comparable viscosity characteristics give very much lower coefficients of kinetic friction under identical conditions of test, it is clear that to define Z as the viscosity of the lubricant at the operating temperature (Fig. 3) is completely erroneous. Long ago when Hersey (1914) first suggested the dimensional notation leading to this equation the quantity Z was expressed as *the effective viscosity in the film*. This places an entirely different construction upon the equation, and is much more nearly in accord with experimental results obtained in the region of full fluid film lubrication.

The primary effectiveness of any lubricant lies in its ability, under conditions to which it is exposed in service, to maintain at all times complete separation of the metal surfaces between which it is interposed. But, in selecting lubricants for particular uses, many other considerations certainly have to be entered into. Unit pressures are high, temperatures exist which in some cases exceed the cracking temperature of the oil used, clearances in many cases are extremely small, and frequently bearings are designed which do not permit of lubrication by purely hydrodynamic means. With conditions like these becoming more prevalent, it is imperative that the study of thin-film lubrication be steadily pursued, that eventually a complete theoretical formulation may be accomplished which will continue on as firm a foundation as the classical hydrodynamic consideration of Beauchamp Tower (1883) and Osborne Reynolds (1886), Sommerfeld (1904).

The authors believe that valuable information, which could be translated into rational fundamental dimensional terms, would be gained from detailed experimental investigation of pressure-temperature-viscosity relations together with a study of the forces of attraction between bodies possessing known lubricating ability and the metals which commonly comprise bearing materials. Investigators are at present working upon each of the two subjects outlined, and though much of the projected research will be empirical, with a strictly utilitarian purpose of developing lubricants more suitable for the progressively more arduous demands made upon them by recent mechanical trends, this is not necessarily to be deplored. With the production of these lubricants will come a vast amount of service and physical data which will be of definite value in developing hypothetical considerations into soundly logical theory expressible in mathematical terms.

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THE USE OF THE FOUR-BALL EXTREME-PRESSURE LUBRICANT TESTING APPARATUS FOR ORDINARY LUBRICANTS*

By D. Clayton, B.Sc.†

Apparatus. In the four-ball apparatus (Boerlage 1933) supplied by the N.V. de Bataafsche Petroleum Maatschappij, a ½-inch diameter hard steel ball is rotated in the cavity formed by three other similar balls clamped in a cup containing the lubricant to be tested. Means are provided for recording the frictional torque on the balls in the cup.

Type of Test. Having applied the particular load, the short-circuit driving motor for the top ball is switched on, reaching its running speed of 1,500 r.p.m. very quickly. There is a small peak on the friction chart due to the initial movement of the cup and the plate on which it stands. The friction then remains constant until, if the load is high enough, seizure occurs with a rise of friction to a high value; normally there is recovery from the seizure, the friction falling to a steady value. The motor is switched off after one minute.

Impressions closely circular in shape are worn on the three clamped balls, and the diameters are measured. A ring is worn on the single ball. If seizure does not occur, these diameters are closely equal to those of the contact areas for the particular loads calculated by the Hertz formula. The mean diameters are plotted against the loads on logarithmic paper, and the Hertzian values plot as a straight line (cf. Fig. 2). When seizure occurs the impressions are larger, and the plotted diameters fall generally on a straight line above the Hertz line. If the load is made high enough the seizure ultimately becomes so severe that the balls weld together. The time to seizure decreases with increase of load. Measurements are also made of the friction at the various stages of the test, and the coefficients are deduced.

Particulars of Present Tests. The tests were generally made at loads increasing in steps of 10 kg., starting from 20 kg., up to loads at which the times to seizure became so short as to indicate a danger of welding seizure; normally, this severe seizure was avoided. Additional tests were then made in the neighbourhood of the lowest load at which seizure occurred to determine its value more definitely; this value is referred

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* Work carried out for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† Engineering Department, National Physical Laboratory.

to as the "breakdown load". There is normally some "scatter" of the plotted observations, as is to be expected because, when failure is imminent, the roughness of even bearing-ball surfaces must be large compared with the thickness of the oil film; the failure will thus depend on the particular configuration of the contact surfaces, which is certain to vary somewhat from test to test with different balls. The scatter appears to vary with different types of oils, and must be regarded as to some extent a characteristic of the oil. It is thought that the breakdown load, as shown by the mean line in the wear-load diagram, has been determined generally to an accuracy of ± 5 kg.

When the load (on the single ball) is 20 kg., the maximum pressure at the centre of each contact area, calculated according to the Hertz theory, is 27,600 kg. per sq. cm. (175 tons per sq. in.); when the load is 100 kg., the maximum pressure is 47,100 kg. per sq. cm. (300 tons per sq. in.). The mean pressure over the contact area is two-thirds of the maximum.

The balls used were all of one make. There have been three separate consignments, but no difference was found between them that was greater than the variation in the results with any one consignment. Most of the oils were tested with balls from more than one consignment, so that some averaging has occurred in any case.

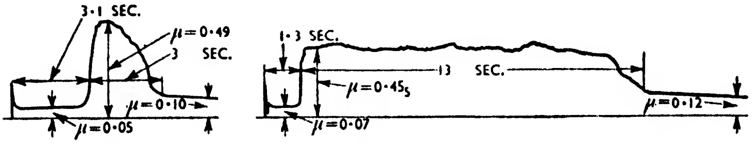
The balls were cleaned with soap and water until the water would spread freely on the surface, and then dried with clean linen. Actually, however, it was not possible to detect any difference from the results when the balls were cleaned with petrol, degreased with trichlorethylene, or even tested as supplied with a slight film of grease. Apparently, the relative motion at the contacts results in the very quick removal of any contaminant.

It was suggested at one stage that there might be some influence due to the variation of speed at the beginning of the tests when starting under load. Trials with castor oil showed, however, that there was no difference in the results when the load was imposed after starting; the practice of starting under load was, therefore, continued.

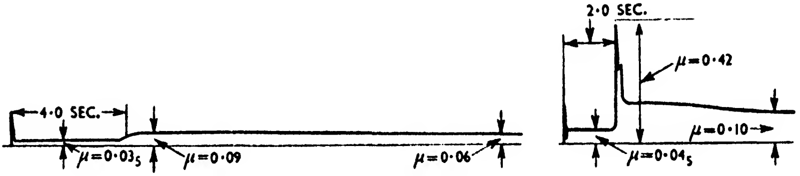
RESULTS

Because of the interesting differences found in the behaviour of lubricants in this apparatus, the tests have been extended to cover other substances besides the extreme-pressure lubricants for which the apparatus was designed. It has been inferred that no fluid pressure film is formed between the balls. This freedom from the effect of viscosity is regarded as an important advantage for research into the qualities of lubricants in general, and further justified the extension of the tests, for there is usually difficulty in ensuring separation of the "oiliness" and fluid film aspects of the behaviour of lubricants in testing machines. Generally, in any practical application, both aspects are material, either in combination or consecutively, but it is considered that progress towards understanding the action of different lubricants lies in the separate examination of the different qualities; then ultimately they can be chosen according to the combination of qualities required.

Gear Lubricants. The behaviour of various gear lubricants, including extreme-pressure lubricants, is described in a paper in



a Castor oil, 104 kg.   *b* Mineral oil A, 81 kg.



c Glycerin, 81 kg.   *d* Golden Syrup, 150 kg.

Fig. 1. Friction-Time Charts

Group III.* It may be mentioned here that the extreme-pressure lubricants had a relatively high breakdown load, but were also characterized by a very quick recovery from seizure, with corresponding low wear. Note may also be made of the superior behaviour of a mineral gear oil which had been refined only to a limited extent.

Castor Oil. A typical friction chart for "first-pressings" castor oil is shown at Fig. 1*a*. The times to seizure and wear results are plotted in Fig. 2. In the diagram (Fig. 2) giving the time to seizure, the arrows near the top indicate that in these tests there was no seizure within the 60 seconds of the test; where a number is added, this shows the number of seconds to seizure, the point falling above

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* See vol. 2, p. 34.

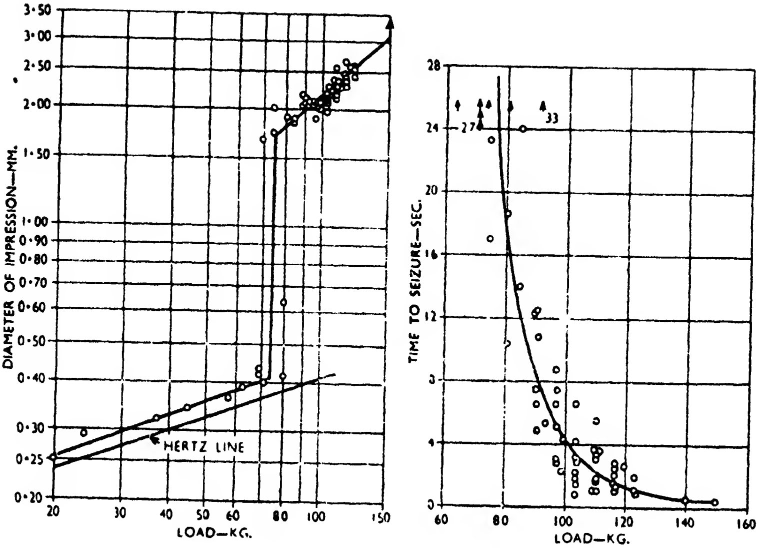

Fig. 2. Castor Oil

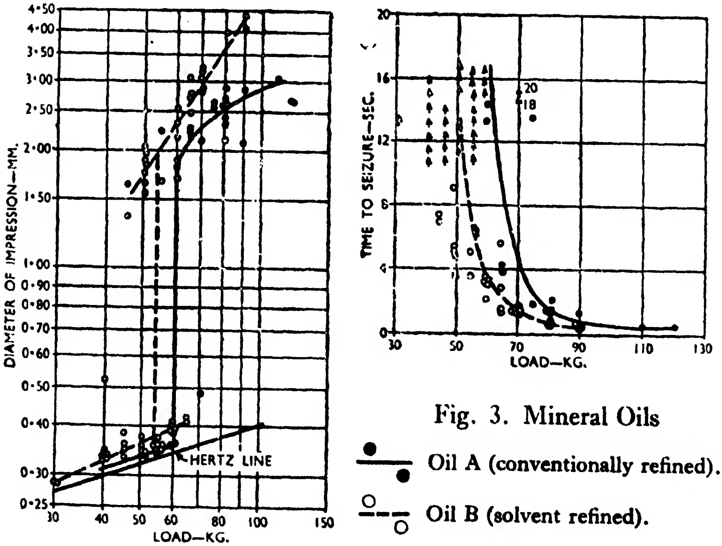

Fig. 3. Mineral Oils

● —— ● Oil A (conventionally refined).

○ - - ○ Oil B (solvent refined).

the limit of the diagram. A welding seizure occurred at 150 kg. The maximum coefficient of friction at seizure varied from 0·45 at low loads to nearly 0·6 at the highest loads.

Mineral Oils. The results for two mineral summer-grade motor car engine oils are given in Fig. 3. Oil A, a conventionally-refined oil, is no longer marketed, having been replaced by B, a solvent-treated oil. The breakdown load was definite with A, but was particularly indefinite for B. Many tests were made with this oil, in fact Fig. 3 shows the results of two separate series of tests; it was concluded that this variation was characteristic of the oil. The mean value of the breakdown load was 55 kg. as against 60 kg. for A; these values are below that for castor oil.

A typical friction-time chart is shown in Fig. 1*b*. The irregularities were greater at higher loads and rather less at lower loads. The marked difference from castor oil in time to recovery is apparent. The time for A ranged from 5 to 15 seconds, and for B from 15 seconds to beyond the full 60 seconds of the test, the long times being reached only at the highest loads. The wear with the latter oil was accordingly greater than with the former, and the curve steeper, and the wear for both oils was greater than for castor oil at corresponding loads. The maximum coefficient of friction at seizure for B varied from 0·35 at breakdown to 0·5 at the highest loads; oil A gave equal values at equal loads.

Oil B shows in the seizing-time–load curve a characteristic of practically all the ordinary mineral oils tested, namely, that a normal seizure is not obtained within the one-minute duration of the test if seizure does not occur within about 7 seconds, often 5 seconds, of the start. Only occasional tests are exceptions; oil A shows more exceptions than any of the other mineral oils. This is in marked contrast with the fatty and extreme-pressure oils, which will give a normal seizure peak after as long as 15 seconds or more.

Graphited Mineral Oil. The addition of 0·2 per cent of colloidal graphite to oil A reduced the breakdown load to 50 kg., and increased the wear with seizure slightly, corresponding with the increase of time to recovery to 15–30 seconds; the wear curve was intermediate between those of oils A and B.

Water in Mineral Oil. Tests of samples prepared for the journal bearing tests described in another paper,* showed that the addition of 0·02 and 0·05 per cent of water reduced the breakdown load of the dry oil from 50 to 40 kg. approximately. With 0·5 and 1·0 per cent water, however, the breakdown load became higher, at 55 kg., than that of the dry oil. It is possible that this curious result is associated with the visible emulsion which was present with these higher proportions of water.

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* Fogg and Jakeman. "The Influence of Water on the Lubricating Value of a Commercial Motor Oil," vol. 1, p. 465.

Fatty Acid Addition to "B.P." Paraffin. In view of the beneficial influence of small proportions of fatty acids on mineral oils in friction tests, it was thought that it would be interesting to see whether similar additions had an effect on the behaviour of a mineral oil in this apparatus. B.P. paraffin was chosen as the mineral oil. The effect of adding 2 per cent of oleic acid was only small, so a 10 per cent solution was also tested. The beneficial influence of the acid was just appreciable on the breakdown load (Fig. 4), and the wear with seizure was slightly less with the 10 per cent solution. There is thus rather a contrast with the marked influence of small proportions of fatty acid in static, and certain kinetic, friction tests, and it may be that the adsorbed layer of oleic acid molecules, which is thought to be obtained in the latter cases, cannot be maintained or re-formed quickly enough under the test conditions of this apparatus. The breakdown load for the plain B.P. paraffin was lower than that of any of the other mineral oils tested. The wear in the seizure region was closely the same as that of oil B.

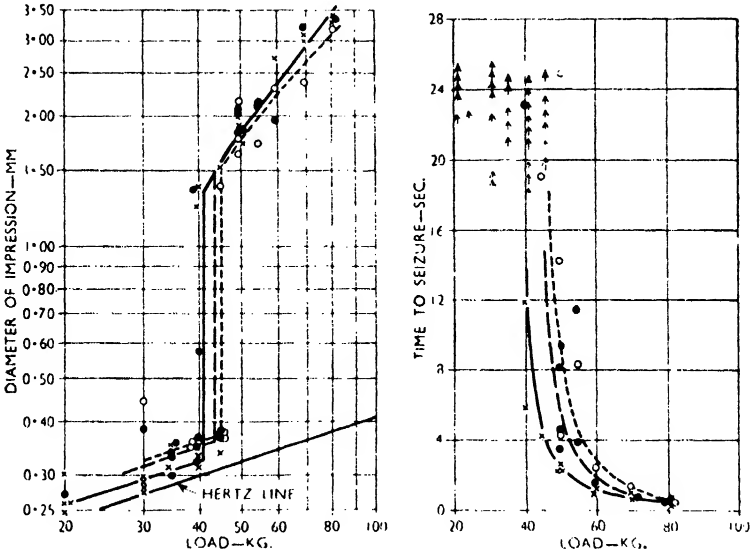

Fig. 4. B.P. Paraffin with Oleic Acid

× ———— Plain oil.

● — — ● 2 per cent oleic acid.

○ - - - - ○ 10 per cent oleic acid.

Compounding. The wear-load and seizing-time–load diagrams for an express engine oil, consisting of 25 per cent rape oil and 75 per cent mineral oil, and for the constituent rape and mineral oils, are combined in Fig. 5. The breakdown load for the mineral oil was higher than that of B.P. paraffin but lower than that of oil B (Fig. 3). The wear with seizure was just a little higher than with the latter oil. The breakdown load for the rape oil was 10 kg. less than that of castor oil, and a little higher than that of mineral oil A. The wear with seizure was higher at low loads but lower at high loads than with castor oil.

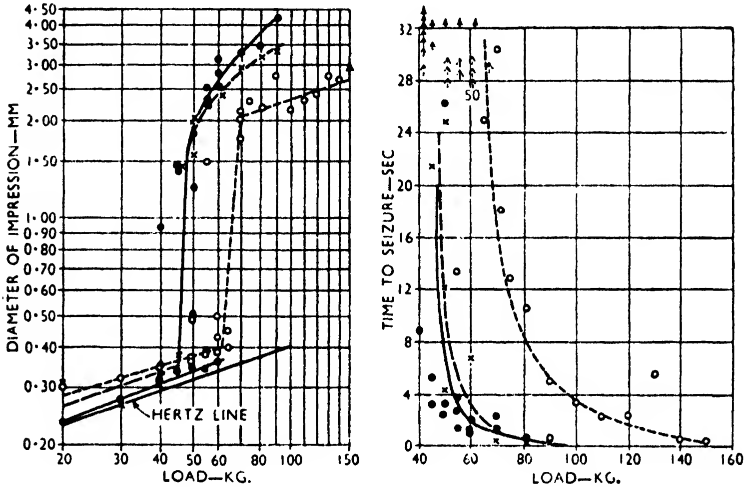

Fig. 5. Compounding

● ——— Plain mineral oil.

○ - - - - - Plain rape oil.

× — — — Express engine oil.

The effect on the mineral oil of adding 25 per cent of rape oil, in preparing the express engine oil, was not apparent as regards breakdown load, but the time to seizure at corresponding loads was rather greater, and the wear at higher loads was less. Related with this last feature were the lower coefficient of friction at seizure and the shorter time to recovery at the higher loads. The coefficient of friction at seizure for the plain mineral oil varied from 0·25 at the lowest seizure loads to 0·45 at the highest loads, and that for the compounded oil reached a maximum of only 0·40. The values for the plain rape oil were similar to those for the compounded oil at corresponding loads, but reached nearly 0·60 at the highest loads used; a welding seizure was obtained at 150 kg. The time to recovery from seizure for the plain mineral oil varied from 15 seconds at the lowest loads to over 60 seconds at the highest, as for mineral oil B (p. 83); for the compounded oil it never exceeded 35 seconds. With the plain rape oil the times were much shorter, ranging from 3 to 10 seconds, compared with 3 to 5 seconds for castor oil.

Glycerin and "Golden Syrup." With B.P. glycerin the type of seizure varied with the load. Between loads of 60 and 90 kg. a series of mild seizures was obtained, the friction rising only slightly to about 0·10; the friction usually decreased again to a slightly smaller value over a period of 10 seconds or so. A typical friction-time chart is shown in Fig. 1 at *c*. The wear was small and increased with increase of the load (Fig. 6, A). At about 90 kg. seizures of more normal type were obtained. In some cases the seizure peak was short, recovery occurring in about ½ second, and the maximum coefficient of friction was about 0·33; the wear was relatively small (Fig. 6, B). In the other cases the recovery was not so quick, the peak of the friction-time chart resembling that for castor oil (Fig. 1*a*), whereas previously the width of the peak was less than half of this; the maximum coefficient of friction was now about 0·45. The wear was accordingly higher, the impressions being twice as large as in the previous case (Fig. 6, C). At the highest loads no recovery occurred in the full minute of the

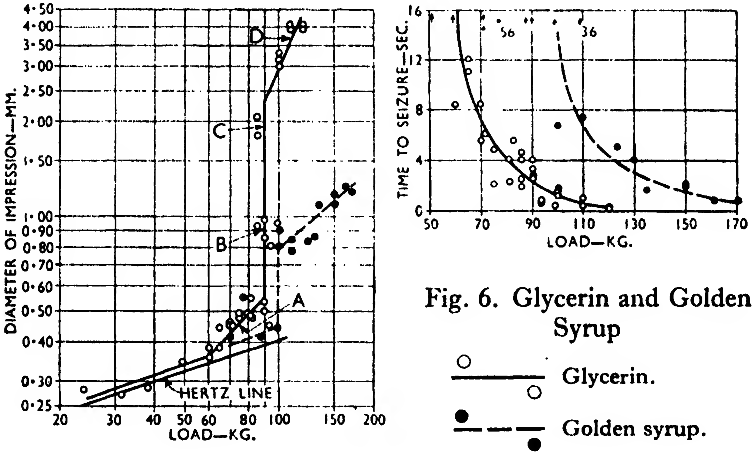

Fig. 6. Glycerin and Golden Syrup

○ —— ○ Glycerin.

● — — ● Golden syrup.

test run, the maximum coefficient of friction being high, from 0·60 to 0·70, and the friction-time chart being very wavy. The wear was correspondingly high (Fig. 6, D), making this portion of the mean curve very steep. The main breakdown occurred at about 90 kg., i.e. higher than with castor oil; the mild seizures, however, started at the lower load of 60 kg. and the seizing-time-load diagram lends support to the deduction that this was the true breakdown load.

"Golden Syrup" was chosen for test because of its very high viscosity. A high load could be taken without seizure, the breakdown load (Fig. 6) being about 100 kg., i.e. as high as those of some of the extreme-pressure lubricants and exceeding those of the mineral and fatty oils. The typical friction-time chart (Fig. 1*d*) shows the seizure peak to be different in character from those of the other substances tested. It consists of a sharp peak, the friction falling in about ¼ second to a moderate value, and then more slowly (over about 4 seconds) falling to what may be regarded as the true recovery value. Corresponding with the quick return of the friction to a lower value at seizure, the wear was small, being only slightly greater than the mean for the best extreme-pressure lubricant tested. The maximum coefficient of friction at seizure varied from 0·3 at the low loads to 0·45 at the highest.

Viscosity. The results generally do not show any direct relation to viscosity at ordinary temperatures. Moreover, the wear impressions when seizure does not occur are closely the same at the same load whatever the viscosity; even with syrup the values correspond with the Hertzian values. Tests were carried out with a viscous extreme-pressure lubricant at 70 kg. to see whether the marking of the balls was due to starting under load; it was found that the impressions without seizure were the same whether the machine was started under load or not, and even when the load was removed again after 5 seconds. It seems, therefore, that there is at any rate very little tendency towards the formation of a load-carrying fluid film between the balls.

No definite explanation of the mechanism of the recovery is offered. The fall of the friction to a low value points to the re-establishment of some kind of effective lubrication. The surfaces are very rough so that the formation of a complete fluid film would appear to be impossible, and it is difficult to see how the wedge-shape for a load-carrying film can be formed over the contact area. Very different temperature conditions must exist with different lubricants at the stage at which recovery occurs. It is probable that some kind of thermal equilibrium is reached, in conjunction with boundary and, possibly, partial fluid lubrication.

Coefficients of Friction. The values of the maximum coefficient of friction at seizure have already been mentioned. At corresponding loads the values vary greatly with the different kinds of lubricants, but taken in relation to the breakdown loads the general order of values is 0·35 to 0·55. The upper limit is uncertain, as the range of loads varies. The rate of increase of this coefficient with load is greatest with the mineral oils, as is also the rate of increase of wear. It is thought that there is possibly some significance in these values being of a similar order to the static and kinetic friction values for clean dry steel on steel. Tomlinson's (1929) static value is 0·39, and Hardy and Doubleday's (1922) 0·74; Beare and Bowden's (1935) kinetic value is 0·42.

The determination of the friction before and after recovery from seizure is not very precise, as there is some variance in the recording mechanism. It is thought that the relative average values are, however, in the right order, but the absolute values may not be quite correct. The coefficients before seizure were: mineral oils, 0·07–0·08; rape oil, 0·06; castor oil and golden syrup, 0·05; and glycerin, 0·04. This low value for glycerin is in accord with the results obtained by Merritt (1935) with a disk-type machine. After recovery the coefficients were: mineral oils, 0·10–0·12; castor oil and golden syrup, 0·10; rape oil, 0·09; and glycerin, 0·08–0·10.

Importance is attached to the coefficient of friction before or without seizure, as at any rate related to the kinetic boundary friction of the oils, i.e. an indication of quality, and also from the point of view of heat generation, which must play a large part in the ultimate breakdown. A further investigation will therefore be undertaken to determine more accurate values for these coefficients.

Although interpretation of the coefficient of friction results is difficult owing to the number of factors involved, it is noteworthy that there is apparently no variation with load of the coefficient of friction before seizure; also there is constancy of the value at any load during the time before seizure occurs, in spite of the heat generation, and after recovery. With those lubricants which give long seizures, the coefficient of friction is usually of the same order throughout, although the area of the impression must be gradually increasing.

The Significance of Certain Results. Little comment is required on the importance of the breakdown load. The qualities of the oils which result in quick recovery from seizure are desirable to reduce the amount of damage which would be done should failure occur in practice. They are probably also of importance for running-in. For, when high point contacts occur, with local cohesion, the removal of material is much more likely to remain local with an oil giving quick recovery. Similar behaviour may be expected with regard to particles trapped between the surfaces. It may be that the breakdown load itself also depends on this feature.

The slope of the seizing-time–load curve is of importance in that, within the range of loads which would lead to seizure, the oil which takes the longer time to seizure will allow a longer time for the contacting parts of the surface to change and so avoid a seizure. It will also confer a greater immunity from the effects of odd shock loads. The feature of fatty and extreme-pressure oils, mentioned above, that they give normal seizures after longer times than do mineral oils, makes the breakdown load effectively higher than is shown by the wear-load curves.

Conclusions. It appears that "oiliness" properties are being tested, as apparently there is no formation of a load-carrying fluid film. The order of increasing quality as regards breakdown load and wear with seizure is: mineral oils, fatty oils, and extreme-pressure lubricants. Doubtless related to the wear is the time to recovery from seizure, which is long with the mineral oils (say, 10 to over 60 seconds), intermediate with fatty oils (3 to 6 seconds), and short with extreme-pressure lubricants (1 to 2 seconds). The coefficient of friction generally rises at seizure to 0·35–0·55.

Glycerin has intermediate properties, whilst "Golden Syrup" behaves like an extreme-pressure lubricant. A mineral oil, only moderately refined and presumably containing a relatively high proportion of unsaturated compounds, behaved in some respects like a fatty oil. The addition of colloidal graphite or small quantities of water to a mineral oil has a slight deleterious effect, but with larger quantities of water the breakdown load increases again. Additions of oleic acid and rape oil to mineral oils have a small beneficial effect.

Acknowledgement is made of the efficient assistance of Messrs. M. J. Wilkie and S. A. Hunwicks in carrying out the tests.

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METALLIC SOAPS IN LUBRICANTS

By E. A. Evans and A. L. Kelman*

The prolongation of the life of lubricating oils and the diminution of deposits from such oils are problems which have claimed the attention of chemists and engineers, and still continue to do so. These problems can be divided into chemical and engineering phases. The chemist devotes his energies to the treatment of the product to render it less prone to decomposition, whilst the engineer endeavours to design an engine in which the lubricant is operating under favourable conditions.

The greatest efforts of the chemist have been in the field of refining. In this he has been handicapped in not knowing sufficient about the hydrocarbons with which he has to deal, and in not knowing the mechanism of their decomposition or chemical change. Working under these disadvantages he has been compelled to make various assumptions and adopt arbitrary tests. Is it surprising therefore that one school is arguing that the deposit on the piston head of an internal combustion engine is due to the residual coke left from the burning of the oil, whereas another school proclaims that the deposit is an oxidized product from the oil ? Considering the fact that the deposit contains a considerable quantity of combined oxygen it is difficult to appreciate the coke theory in its rawness. It may be that at the high temperature existing in the piston ring area accelerated oxidation of the oil occurs, followed by vaporization of the oil, leaving a residue of solid oxidation products. If this hypothesis were true, then the coke value as normally determined would not truly represent the conditions. The fact that the solid products in a used oil contain approximately the same amount of oxygen as the deposit on the piston head lends colour to the oxidation theory; in fact, it suggests a link between the two solids. It leaves one wondering whether the oxidation occurs in the crankcase, or within the vicinity of the rings, or indeed within the combustion space.

If one adheres to the oxidation theory one should look to developments in the art of refining to reduce the propensity for oxygen absorption, and the resultant reduction in solid formation. If this path is pursued the final issue is a colourless oil apparently very repellant to oxygen, but not serviceable as a lubricant in general. Hence comes the difficulty of deciding how far to carry this theory. There are too many exceptions to air-blowing tests giving criteria for practical running results to accept them unreservedly. Even combinations of

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* C. C. Wakefield and Company, Ltd.

air-blowing tests and coke tests fail to supply the missing factor. Much has been written about oxygen absorption, but too little about its correlation with engine results. It is little wonder that the chemist turns again to running an engine under controlled conditions to supply the requisite data.

That oxidation does play a prominent part in oil deterioration cannot be denied. The oil in the crankcase of an internal combustion engine has an average temperature of about 60 deg. C., though in some cases it rises to 150 deg. C., and in other cases never rises above 50 deg. C. Is this temperature sufficiently high to promote oxidation ? As the oil is atomized and exposed in thin films oxidation is induced, but surely the most likely place for real activity is on the cylinder walls, and in piston ring areas where the temperature is high. In these parts the catalytic effect of the ferrous metals is most likely to be active, yet there is an inclination to conduct oxidation tests at lower temperatures and almost a disinclination to conduct the oxidation test in the presence of iron.

In many industries inhibitors are used to reduce the velocity of reaction. It is not surprising therefore that the chemist should suggest inhibitors to give resistance to lubricating oils. Unfortunately, organic inhibitors have not so far proved to be of value in lubricants working at high temperatures, though it is possible that for temperatures up to 60 deg. C. they can be used. β-Naphthol, phenyl-α-naphthylamine, tertiary butyl cresol, guaiacol, amongst others have been suggested.

Compounds of metals were advocated fifty years ago by Wass (1887). He added ashes of lead or zinc or tin to a preparation comprising an animal, mineral or vegetable oil, to which was added a small quantity of asphaltum and glycerin. The metallic oxides were added to minimize any tendency to gum. The Villiers London Company protected (1922) the addition of metallic stearate to castor oil to render it less liable to carbonization. Evans (1926) showed that electrical insulating oils could be rendered more resistant to oxidation through the agency of some metallic soaps. Amongst others he instanced tin oleate and nickel stearate. Following upon the use of tetraethyl lead in petrol to prevent too rapid oxidation it is not surprising to find Callendar, King, and Mardles (1927) using tetraethyl lead, iron carbonyl, etc., to prevent the formation of organic peroxides of oil. Then Helmore (1932) combined the ideas of Evans and Callendar and used tin oleate in conjunction with tetraethyl lead. Metallic compounds act not only as inhibitors to oxidation, but also as running-in compounds when present in vegetable lubricating oil (King 1932). A further use of such compounds was shown to exist by the Standard Oil Development Company (1931) in depressing the pour point of hydrocarbon oils.

Turning from the patent literature to scientific journals, Hatta (1926) has described tests in which tetraethyl lead was added to turbine oils to reduce sludge formation. Yamada (1930) differentiates between sludge forming and anti-sludge forming metallic soaps in transformer oils.

When discussing inhibitors it is postulated that deterioration of the oil and the production of solid deposits are due principally to oxidation. If oxidation does occur in an internal combustion engine it is probable that it is not a perfectly straightforward affair. It may occur in the crankcase at temperatures up to 80 deg. C., and in exceptional cases up to 150 deg. C. It may occur in the piston ring area at 200–250 deg. C., or on the cylinder walls at even higher temperatures. The oxidation may take place on the surface and, or alternatively, in the interior of the oil. A whole molecule may be completely oxidized at one range of temperature and one set of conditions, or it may be successively changed. Finally, do the metal surfaces or the metallic compounds present play any part in the chemical change ? Simple though all these considerations may appear, they are far from being satisfied. It is little wonder therefore that the various tests which are available to the chemist are so incomplete and unconvincing when endeavouring to assess the resistance of an oil to deterioration. With all its difficulties, the engine test is still the final court of appeal.

The simple air-blowing test has many critics of its procedure, and many unconvinced supporters of its interpretation, yet the critics are not entirely opposed in principle to an oxidation test. Since the introduction of inhibitors the problem has intensified, because some of them accelerate the oxidation in the oxidation test and reduce the manifestation of chemical change in the engine. The Air Ministry test and oxygen absorption tests have produced the same anomaly. In these methods of test the air or oxygen is bubbled through the oil and the conditions are different from those in the engine, where large surfaces are exposed either through atomization or in thin films. The expedient of exposing an ever-changing thin film to air or oxygen in a rotating tube in an oven was tried, but the results were equally unproductive. In some cases the repeatability was poor, and in others the correlation with engine tests was bad.

Measurements of oxygen absorption can be exceedingly misleading unless the conditions under which the determination is made are carefully analysed. The use of a closed tube gives results which at first show a rapid absorption until a maximum is reached, followed by no further absorption, indicating complete saturation. If the tube is refilled with oxygen, absorption continues at approximately the same rate, and by further refillings of the tube the reaction proceeds until the oil solidifies. Needless to say the rate of oxygen absorption varies with different oils, and is modified by the addition of various compounds. Lead tetraethyl in sufficient amount will accelerate the reaction, which can be decelerated by a suitable amount of tin oleate.

Obviously the reaction between oil and oxygen in a glass container must be modified if better correlation is to be obtained. This disability is somewhat removed by heating the oil in the presence of metallic iron in a Petri dish. Whilst the increase of acidity and the increase of solid oxidation products are of practical importance, the viscosity of the oil after oxidation appears to give closer correlation with engine results. It is not suggested here that a new method of appraisal is to be found in viscosity measurements, but it cannot be ignored that viscosity modifications obtained by heating the oil in Petri dishes in the presence of iron do produce results which are in closer correlation with engine tests than do blowing and oxygen absorption methods, at least in the study of oils containing metallic soaps.

TABLE 1. INHIBITORY EFFECT OF CERTAIN ORGANO-METALLIC COMPOUNDS ON OXIDATION

Oil	Untreated	Tin oleate, 0·2 per cent	Tin oleate, 1 per cent	Lead tetraphenyl, 0·2 per cent	Tin tetraphenyl, 0·2 per cent	Lead tetraethyl, 0·05 per cent
	per cent	per cent	per cent	per cent	per cent	per cent
A	0·3	—	Nil	—	—	—
B	8·9	7·4	—	8·2	7·9	—
C	1·5	—	—	—	—	0·95

The air blowing test does reveal the inhibitory characters of certain organo-metallic compounds.

Tin oleate, lead tetraphenyl, tin tetraphenyl, and lead tetraethyl, when added to oils A, B, C, in the proportions shown in Table 1, reduce the amount of solid oxidation products, but it must not be assumed that when they are used in other oils and in other proportions a reduction will occur. The method finally adopted was to heat 6 grammes of oil in a Petri dish at 180 deg. C. For purposes of comparison in this paper the same oil was used.

In each case the addition of the metallic soap led (Fig. 1) to increased oxidation of the oil as shown by viscosity, insoluble matter, and acidity. The effect of tin and chromium, although they do in fact behave as accelerators, is small, however, compared with that of iron and copper. These curves show certain features. In each case, after the addition of a certain quantity of metallic soap, larger quantities of soap have little or no effect on the oxidation. With copper, and with iron in particular, the effective quantity of soap is greater than with chromium and tin. As the period of oxidation is prolonged the effective quantity of metallic soap becomes greater. If the soap had a direct oxidation effect on the oil, then further additions of soap would lead to a proportional increase in the oxidation of the oil. As it does not, then the effect of the soap must be limited by some other factor. The oxidation of the oil is primarily autocatalytic. It is therefore possible that the products of oxidation, which themselves give rise to the catalytic nature of the reaction between oil and oxygen, co-ordinate with the metallic soap. These new products, in the cases considered here, have a greater activity than the parent body, thus giving increased oxidation of the oil in the presence of the metallic soap. On the other hand, any excess of metallic soap which has not reacted with oil oxidation products has no effect on the oxidation of the oil.

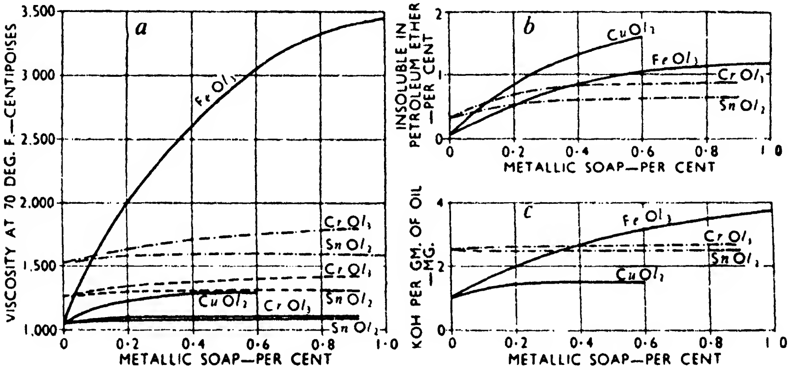

Fig. 1. The Effect of Metallic Soaps on the Oxidation of Oil at 180 deg. C.

——— 6 hours' oxidation.
- - - - - - 9 hours' oxidation.
-·-·-·- 12 hours' oxidation.

If there is a strong metallic activator present (e.g. iron oleate) the resultant increased oxidation will lead to the production of larger quantities of active oxidation products which will combine with the iron oleate, making larger quantities of metallic soap effective in the reaction. Similarly, under the influence of time, more oxidation products will be generated, and the reaction will proceed as before.

This working hypothesis serves to explain the results, but it may also be assumed that during the reaction period there is always present metal soap to which the oxidizable material diffuses, developing new chain centres. Finally, a layer of polymer is formed which serves as a growing obstacle to the formation of such chain centres. If, however, the polymer can be visualized as peeling off and exposing a fresh surface the reaction will carry on. The difference between the behaviour of the various metal soaps is possibly due to the formation of different types of unstable complexes between the catalyst and the polymerizing material. Much more work will be needed to establish whether a chain reaction is in progress. Whether the accelerator increases the chain length is by no means clear, but it does serve the purpose to believe that an inhibitor is a chain breaker. This dilemma may perhaps be avoided by assuming that atoms migrate from one portion of the chain to another. A change in orientation may have to be assumed.

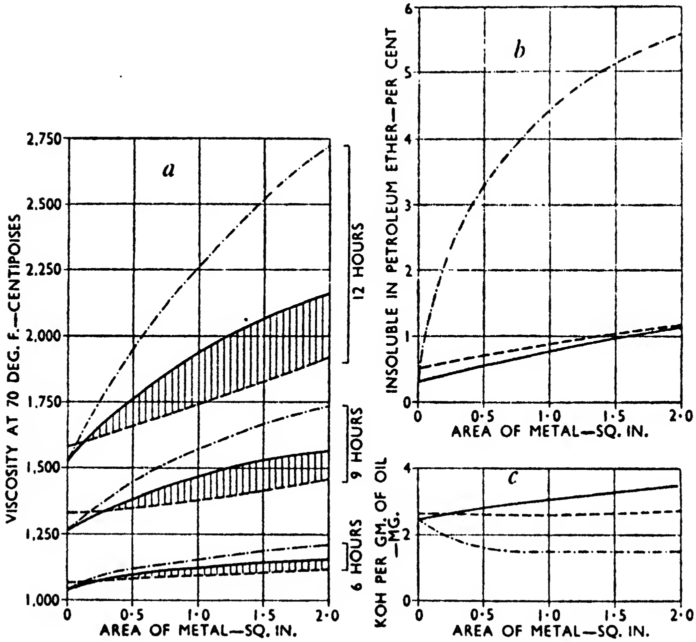

Fig. 2. The Effect of Metals on the Oxidation of Oil at 180 deg. C.

-·-·-·- Oil in the presence of copper.
——— Oil in the presence of iron.
------ Oil+0·25 per cent tin oleate in the presence of iron.

Sodium oleate behaves differently from the before-mentioned metallic soaps in that the viscosity increases more than proportionately to the quantity of soap present. The increase in viscosity may be attributed to the auto-oxidation of the oil, together with the structure of the soap gel.

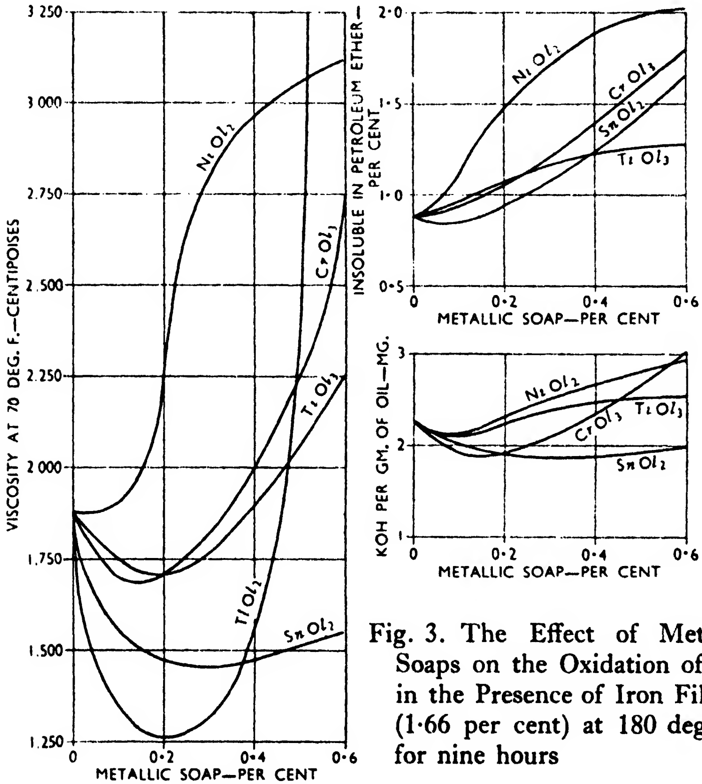

Fig. 3. The Effect of Metallic Soaps on the Oxidation of Oil in the Presence of Iron Filings (1·66 per cent) at 180 deg. C. for nine hours

In most cases lubricating oil operates in close contact with metal surfaces. As iron and copper predominate, nearly all lubricants are exposed to an active catalyst in addition to an oxidizing atmosphere. The problem is to produce an oil by refining which will resist atmospheric oxidation. That problem is being actively pursued, knowing full well that over-refining is detrimental. Experiments with metal soaps have revealed (Fig. 2) that some of them can inhibit the catalytic action of iron. Iron has been selected because it is used so extensively in engineering.

The effect of iron on the oxidation of the oil is progressive, but not proportional to the amount present. When tin oleate is added to the oil the effect of the iron is proportional to the area exposed. Part of the increase in oxidation due to the iron has been inhibited by the tin oleate. This is represented by the shaded portion in Fig. 2. Obviously, if there were total inhibition, the results would be represented by horizontal lines. The simple explanation would appear to be that products of auto-oxidation are formed, which attack the iron and produce FeX. If the FeX is the active catalyst, then the tin oleate links with it giving a less active catalyst or, alternatively, inhibits the formation of FeX. On the other hand, the free iron may induce oxidation, but it may be dimmed by adsorption of metallic soap at the metal interface. This graphic description should not prejudice any views which may be based on chain reactions.

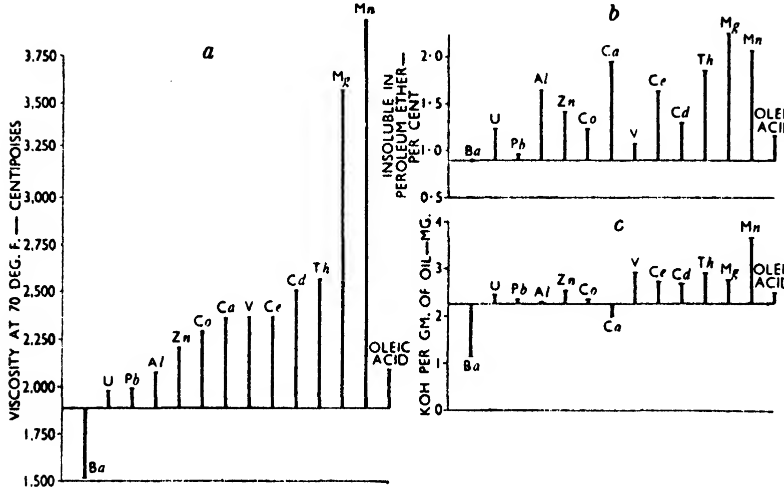

Fig. 4. The Effect of Metallic Soaps (0·2 per cent) on the Oxidation of Oil in the Presence of Iron Filings (1·66 per cent)

The horizontal line indicates the oxidation figures of the untreated oil. The experiments lasted nine hours at 180 deg. C.

Another aspect of the effect of metallic soaps in inhibiting the accelerating action of iron on the oxidation of the oil is obtained by oxidizing samples of oil containing different proportions of metallic soap in the presence of a fixed quantity of iron (Fig. 3). Thus, with tin oleate, the maximum inhibitory effect is observed when the oil contains 0·25 per cent of metallic soap. When the surface of the iron is decreased by using metal squares instead of filings, the optimum quantity of tin oleate is found to be independent of the area of metal. On the other hand, the inhibitory effect of the soap is magnified in the presence of increased quantities of iron. Chromium shows a similar type of inhibition, but less actively than tin oleate. Thallium oleate is very interesting, but it is very costly.

From the results it is clear that approximately 0·2 per cent of metallic soap produces the inhibitory action, consequently when a general survey of metallic soaps was being made 0·2 per cent of soap was used in the presence of iron filings.

Barium presents some attractive characteristics, but its use is debarred by other considerations. In general, the only inhibitory metallic oleates contain tin, chromium, nickel, thallium, or titanium. The inhibitory powers of the metallic soaps are not materially affected by a change of acid radical.

Laboratory tests without proper regard to engine performance would be of little value, therefore a series of engine tests was made under well-controlled standardized conditions. A single-cylinder petrol engine was run under load at a temperature at which the maximum amount of deposit could be produced in the combustion space. The oil which was used was the same as that used in the laboratory experiments. It is well known that it is virtually impossible to reproduce engine results on oils even though precautions are taken to standardize conditions; consequently frequent check tests were made using the untreated oil. The deposit produced by the untreated oil was given the arbitrary value of 100; all the following results were compared with this standard:—

Oil	Deposit
Untreated oil	100
Oil +0·1 per cent tin oleate	90
,, +0·1 ,, ,, ,, triphenyl-*iso*-butyl	85
,, +0·1 ,, ,, ,, naphthenate	90
,, +0·2 ,, ,, ,, oleate	92
,, +0·2 ,, ,, chromium oleate	78
,, +0·8 ,, ,, ,, ,,	72
,, + 0·025 per cent tin oleate 1·0 ,, ,, chromium oleate	91
,, + 0·1 ,, ,, tin oleate 0·4 ,, ,, chromium oleate	62
,, + 0·2 ,, ,, tin oleate 0·4 ,, ,, chromium oleate	81
,, + 0·1 ,, ,, tin oleate 0·8 ,, ,, chromium oleate	58
,, + 0·1 ,, ,, tetraethyl lead	81
,, + 0·1 ,, ,, tin oleate 0·1 ,, ,, tetraethyl lead	91
,, + 0·5 ,, ,, tin oleate 0·5 ,, ,, tetraethyl lead	60

The laboratory tests indicate that 0·25 per cent of tin oleate gives the greatest inhibitory effect, whilst the engine shows it to be 0·1 per cent.

For chromium oleate the respective results are 0·17 and 0·8 per cent. Most encouraging results were obtained in the engine when exploring mixtures of tin and chromium. Efforts to obtain similar criteria by laboratory methods were not so successful. From the engine results it is quite clear that the correct amount of tin oleate to use is 0·1 per cent and the amount of chromium oleate between 0·4 and 0·8 per cent. The correlation of laboratory results with engine tests does bring into bold relief the advantage of using iron in the oxidation test.

The authors wish to express their thanks to Messrs. C. C. Wakefield and Company, Ltd., for their permission to publish these results.

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THE STUDY OF WEAR AND LUBRICATION BY ELECTRON DIFFRACTION

By Professor G. I. Finch * and Dr. F. D. Zahoorbux *

Now that the modern tendency for high engine speeds and loading is giving rise to new problems of wear and lubrication, it is fortunate that developments in theoretical physics should lead to the discovery of a new tool peculiarly adapted to the direct study of surface phenomena. The confirmation by Davisson and Germer (1927) and by Thomson and Reid (1927) of de Broglie's prediction (1924) that moving particles should interact with matter as if they were guided by wave systems has given us in the phenomena of electron diffraction a new and powerful means for the study of surface structure.

Unlike X-rays, electrons, owing to their charge, are so easily deflected by the atoms in any aggregate of matter on which they impinge that their penetration is limited to the surface layers. Thus any information about the surface conditions obtained by the diffraction of electron waves is limited to a revealing of the structure to a depth of only a few atoms. In practice, a beam of electrons moving at a velocity of the order of half that of light is caused to glance off the surface under examination, and the directions of the diffracted rays leaving the surface are recorded photographically as a so-called electron diffraction pattern. Since these directions are determined by the manner in which the atoms are arranged in the surface layers, it is possible to derive the structure of the surface from the diffraction pattern. The pattern in its details usually reflects the degree of order of the atomic arrangement in the surface. For example, a smooth single crystal surface in which all the atoms are regularly arranged at the points of a space lattice gives rise to a pattern of spots and regularly arranged parallel pairs of black and white lines (Fig. 1) from which the atomic arrangement can be deduced. A rough single crystal surface, on the other hand, may yield only a regular pattern of spots, the sharpness of which will depend on the shape of the projections rising from the surface (Fig. 2). A polycrystalline surface in which the crystals are disposed at random will give rise to a pattern of rings, sharp (Fig. 3) or more or less diffuse (Fig. 4) according to the texture of the surface. The degree of disorder is at a maximum in a liquid surface, and the atomic size, i.e. the nearest distance of approach of the atoms to each other, is practically the only feature of regularity; in this case the

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* Imperial College of Science and Technology, London.

diffraction pattern consists of rather faint, very diffuse halos superimposed on a dense background of general scattering (Fig. 5).

Cutting or abrasive methods of smoothing a metal surface leave its structure crystalline. The surface of a honed cylinder sleeve of hardened steel, for example, gives a sharp ring pattern characteristic of the crystal structure of the metal (1935). A mild steel surface finished with fine emery paper gives a similar pattern characteristic of α-iron (Fig. 6) (1935). A similarly treated cast iron surface yields a pattern of rings or broad spots, according to the fineness of the abrasive; here, however, the pattern is not due to the metal but to minute crystals of graphite freed from the pores and spread out over the surface (1935). When the graphite crystals are disordered, as they will be if the abrasive is relatively coarse, the pattern consists of rings (Fig. 3); but when the abrasive is very fine the graphite crystals tend to arrange themselves with their slip planes parallel to the surface, and the pattern then consists of a row or rows of broad spots (Fig. 7).

The surface finish put on a metal by hammering, burnishing, and polishing is quite different in its structure from that afforded by abrasive methods (1937). In these proccsses very high surface temperatures are attained so that localized melting occurs; the surface is not abraded but smoothed to a state of high finish by a smearing over the surface of molten or viscous material formed at the "high spots" as a result of intense heating due to high localized stresses. This fluid material congeals in stress-free regions so rapidly that the atoms have no time to rearrange themselves into crystal units, but remain in disordered array just as they would be in an instantaneously frozen liquid. This non-crystalline amorphous layer, known as the Beilby layer (Beilby 1921), gives rise to the diffuse halo pattern characteristic of a liquid surface.

The Beilby layer is harder than the corresponding crystalline form of the material; its energy is also greater and it therefore represents an unstable condition. In some cases moderate heating of the Beilby layer on the surface to far below the normal melting point of the material suffices to bring about its transformation into the crystalline state, and in others recrystallization occurs spontaneously even at room temperature (1937). Such a transformation gives rise to surface cracks and unevennesses such as do not appear on a liquid or vitreous surface. The high energy of the Beilby layer also explains the phenomenon of the dissolution, even at room temperature, of crystalline metals in the polish layer on a metallic surface which in the crystalline state exhibits no such action (1934). Hence it is also reasonable to suppose that mutual alloying together of the two metal surfaces will

Fig. 1. Electron Diffraction Pattern from a Natural Face of a Diamond

Pattern characteristic of the smooth crystalline surface of a single crystal.

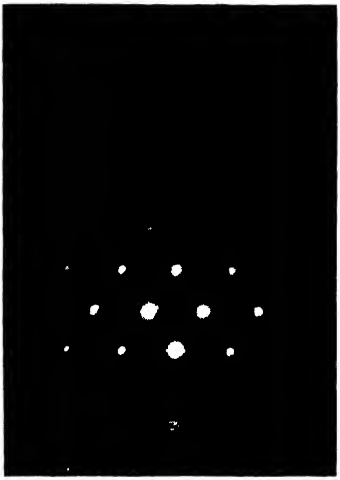

Fig. 2. Pattern Given by a Single Crystal from which Project many Minute Spikes

The spot arrangement is characteristic of zinc blende.

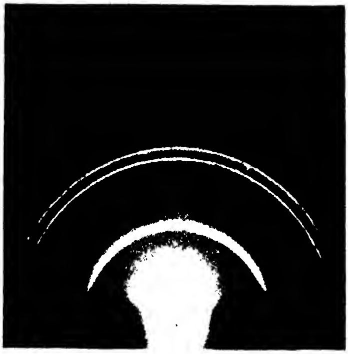

Fig. 3. Sharp-ringed Pattern Yielded by a Surface covered by many Chaotically Disposed, Very Thin, but Relatively Broad Crystalline Spikes

The ring diameters are characteristic of graphite.

Fig. 4. Non-sharp Rings Indicating Unusually Short Random Crystalline Projections

[I.Mech.E., 1937]

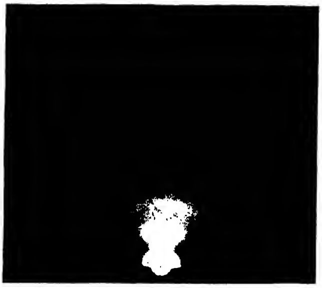

Fig. 5. Diffuse Halo Pattern, Typical of Liquid and Amorphous Surfaces

Fig. 6. Pattern Due to α-Iron

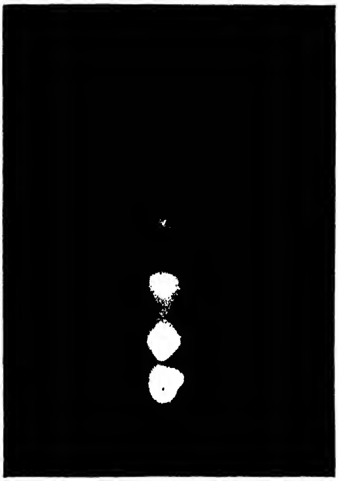

Fig. 7. Pattern Given by Graphite Crystals with their Slip Planes Parallel to an Electron Beam Grazing a Plane Surface

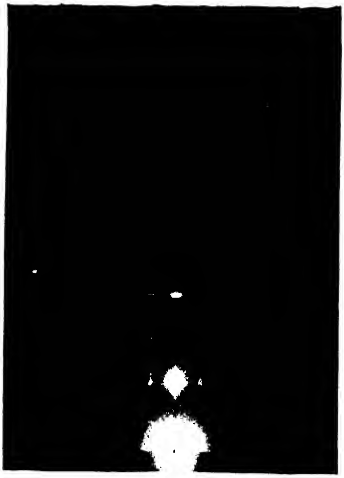

Fig. 8. Pattern of Parallel Broken "Layer Lines" Due to Long-chain Hydrocarbon Oil Molecules Standing Vertically on the Surface Grazed by the Electron Beam

The faint straight lines pointing in various directions are due to the quartz single crystal used as substrate.

occur more readily when they are in the amorphous condition than otherwise. This shows the importance of selecting for bearing surfaces metals which do not readily dissolve in each other, even though their mutual friction may give rise to Beilby layers. Often this can be ensured by selecting metal pairs of widely different melting points.

Electron diffraction shows the surface of new bearing surfaces to consist of a multitude of minute peaks of relatively soft crystalline material projecting above its mean level. Shaw (1934) has examined the contour of surfaces after various finishing processes. His observations are of a macroscopic, or at most microscopic, nature, and can take no account of the many submicroscopic crystalline projections which electron diffraction reveals as present on the visible rugosities of a freshly finished surface, but they clearly show the crude "ridge and valley" formation of even the finest turned finish as opposed to the apparently flat surface produced by honing, which actually consists of a profusion of minute "humps" and "pits". The electron beam grazing the surface penetrates only to a very small, wholly submicroscopic depth below the peaks of these large-scale projections, so that with its aid only the true bearing surface is examined, and the valleys or pits which add nothing to the loaded area make no contribution to the diffraction pattern.

Even under the light loads of normal running-in, the bearing stresses concentrated on the submicroscopic crystalline peaks at the tops of the large-scale projections result in intense local heating which causes flow over the top and down the sides of these projections, where the flowed material freezes, on escaping from stressed regions, into the non-crystalline amorphous condition. As running-in progresses, the depressions eventually become filled with amorphous material until, on completion of an ideal running-in process, the whole bearing surface is covered by a relatively thick, hard, amorphous layer of "flowed" metal (Finch, Quarrell, and Roebuck, 1935). Even more important than the gain in hardness is the advantage of the change in surface texture brought about by running-in, for the load is no longer localized at a few projecting spots but distributed over wide areas of contact, with a consequent reduction in the temperature of the loaded areas and the risk of alloying together of the bearing surfaces, i.e. "pick-up".

Owing to the manner in which oil molecules orientate themselves on a surface, the production of an amorphous layer on the bearing surfaces, with the corresponding increase in effective contact area, is essential for the maintenance of a stable oil film. The electron diffraction pattern of a long-chain hydrocarbon, deposited as a monomolecular layer over a really smooth flat substrate, consists of a series of "layer lines" all parallel to the shadow edge of the substrate (Fig. 8). This fact shows that the orientation of the hydrocarbon molecules is such that they stand up vertically on the surface. The separation of the "layer lines" corresponds to 2·54 Å., which is the distance between alternate carbon atoms in the aliphatic chain; this indicates further that the hydrocarbon molecules have a zig-zag configuration of carbon atoms. The number, length, and sharpness of the "layer lines" is a rough measure of the perfection of orientation. Since the paraffin hydrocarbons are non-polar, their ability to orientate in the above manner on smooth surfaces must be due to lateral adhesive forces distributed along the length of the chains, rather than to attractive forces between the ends of the molecules and the substrate.

A monomolecular film of a long-chain hydrocarbon on an optically smooth surface formed by careful polishing gives a sharp "layer line" pattern. A bimolecular film affords only a rather diffuse "layer line" pattern, and a trimolecular film does not, as a rule, give a pattern at all. This means that the orientation of the superficial layer of molecules, which, with a sufficiently long-chain compound, is the only one that contributes to the formation of the electron diffraction pattern, decreases as the film gets thicker. It is probable that as boundary lubrication conditions are approached these poorly orientated layers are readily swept away, whilst the well orientated layer in contact with the substrate resists rupture by virtue of the lateral attractive forces which bind the molecules together like the sticks in fascines. Metal-to-metal contact occurs only when the bearing load is so great as to overcome this force and drive the molecules apart, thereby rupturing the film. Therefore the more truly plane the surfaces, the greater will be the load which the boundary oil film will support without collapse.

Electron diffraction shows that the chemical nature of the substrate has no appreciable effect on the orientation of long-chain molecules. Highly polished metals, smooth crystal facets, and lens surfaces all give equally good monomolecular oil "layer line" patterns when used as substrates. The real smoothness of the surface, however, is very important in deciding the extent to which the direction of orientation with respect to a true plane is the same for all molecules. Thus the degree of perfection of orientation of the hydrocarbon molecules in a single layer on a copper surface depends greatly on the finishing process used in polishing the copper, although no difference in finish may be apparent to the eye. The more highly orientated the oil film, the greater is the load resistance of the lateral adhesive forces, and the less likely is the film to rupture. Anything tending to prevent the formation of a well orientated film, such as crystalline projections on the surface, must be avoided; hence the production of an amorphous Beilby layer is of paramount importance.

The effect of a substituent group at the end of the hydrocarbon chain is to diminish the angle at which the molecules are orientated with respect to the surface to an extent which depends mainly on the actual size of the group. A pure aliphatic alcohol exhibits this effect in the diminished sharpness and number of the "layer lines" visible in the diffraction pattern, whilst an aliphatic acid, such as palmitic, does not give a "layer line" pattern at all. Here the long chains do not stand erect on the surface but are tilted over to some definite angle in all possible azimuths, the degree of tilt away from the perpendicular increasing with the size of the substituent group. The tilt can be quantitatively explained by assuming that there is a very strong tendency for the aliphatic chains to set themselves at a characteristic distance apart as a result of the lateral adhesive forces already alluded to. If there is a large group at the end of each of the chains, these cannot at the same time come close enough together and still remain erect. They must therefore tilt over in order to satisfy the side spacings. This is true only of pure compounds; if mixed with a normal straight-chain hydrocarbon, the large groups at the ends of the substituted chains are separated from each other by the normal hydrocarbon chains, and the side spacings can thus be satisfied with all the chains erect as with the pure hydrocarbons, provided the concentration of the substituted compound is not too great. This has been confirmed by electron diffraction, which shows that mixtures of a long-chain hydrocarbon and palmitic acid are perfectly orientated in mixtures containing up to as much as 50 per cent of palmitic acid.

It has been pointed out above that the cast iron element of a bearing which has been well run-in becomes tenaciously coated with a thin orientated layer of graphite, the so-called "graphoid" layer. Electron diffraction shows that such a layer provides an admirable temporary substitute for an orientated oil film. Owing to its peculiar layer-lattice structure with a direction of singularly easy slip which in an orientated film is also the direction of shear for the bearing, graphite has lubricating properties of its own. Under conditions of adequate oil lubrication the graphoid layer will be completely covered with an orientated oil film which suffices to prevent well run-in surfaces from coming into contact either with each other or with the graphoid layer or layers with which they may be covered. Should the oil film become ruptured, the orientated graphoid layer is virtually as effective as the lubricant in permitting ready shearing between the two bearing surfaces. It is thus possible for a bearing with graphitic surfaces to run dry for considerable periods without seizure. When two freshly machined bearing surfaces are run-in together, large compressional stresses occur between "high spots" which engage momentarily. The presence in the lubricant of a little clean graphite of suitably small crystal size ensures the formation at the points of contact of an orientated graphite layer which facilitates the relative motion of the surfaces by a relatively small component of shear. The danger of "pick-up" occurring or the stripping of projections, with consequent damage by small particles of metal carried in the oil, instead of their being gradually "flowed" to form a smooth Beilby layer, is thereby materially decreased. In this way the production of a deep polish layer extending over the whole area of the bearing surface is greatly assisted. These conclusions are borne out by the work of Shaw (1934). Also, the authors have shown that if a thick layer of graphite deposited from colloidal solution is compressed normally without shear against another similar layer, then although both surfaces show originally a random arrangement of small crystals, after compression very strong orientation is seen. There seems to be a critical pressure which must be exceeded before the slip planes can be made to lie in the direction of shear. Under shear stress, of course, the graphite crystals orientate readily.

The presence of graphite in cast iron accounts for its self-lubricating properties, but in certain cases this graphite supply is inadequate and can only be replenished by the opening of fresh pores as the result of wear, unless an external source of graphite is furnished. Indeed, it has become usual to add small amounts of colloidal graphite to the lubricating oil for graphite-free metals as well as for cast iron. An orientated graphite layer is soon formed on a bearing surface run-in with such a graphitized oil. The authors have found that a bearing, neither of whose elements is cast iron, but which has been lubricated with graphitized oil, can run without damage or seizure in a dry condition for a much longer period than one previously lubricated only with plain ungraphitized oil. Since under normal running conditions the oil film at some point or other continually undergoes momentary rupture, the chief function of the graphite is to prevent "pick-up" during these brief periods of oil-film breakdown.

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MEASUREMENT OF BOUNDARY FRICTION IN AN OSCILLATING BEARING MACHINE*

By A. Fogg, M.Sc.†

Complete boundary conditions or perfect fluid film conditions probably do not occur to any great extent in practice, the friction of most lubricated surfaces apparently being a combination, in some proportion, of these two types of friction, at least during some part of the time of operation. In most mechanisms, viscosity of the lubricant appears to be one of the factors determining the frictional resistance. Boundary conditions in some form, however, occur at the instant of starting although the value of the coefficient of friction appears to depend to some extent on the time the mechanism has been at rest. With a journal bearing, the static boundary condition on starting may be replaced by fluid film conditions when continuous rotation ensues. The surfaces of an oscillating bearing are relatively at rest at the beginning of each stroke and in this position the static boundary condition may be reached. If, in the subsequent motion, operating conditions are such that the boundary layers remain undisturbed, the friction during the stroke will be kinetic boundary friction.

This paper gives the results of measurements of the friction of an oscillating bearing under a variety of operating conditions, and it is suggested that, under certain conditions, the machine appears to give a measurement of boundary friction. The apparatus employed was originally designed by the late Sir Thomas Stanton, but with modifications made during the course of this work.

Description of the Machine. The general layout of the machine is shown in Fig. 1. The journal J, 1·500 inches diameter, is of hardened cast steel formed in the middle of a shaft mounted in two ball bearings B_1 and B_2. The steel bush A is pressed into a steel holder H, which is loaded by means of weights and the lever L, so that the lower part of the bush presses on the journal. The bush and journal are completely submerged in oil contained in a bath O, fixed to the oscillating shaft. The journal is driven through a torque meter from a shaft S, mounted in roller bearings B_3 and B_4. A driving disk E is attached to the other end of the shaft S. A pin on the disk E is coupled to a pin on a revolving crank disk driven through gearing from a variable

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* Work carried out for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† Engineering Department, National Physical Laboratory.

speed electric motor. The throws of the two crankpins are dimensioned to give a total angle of oscillation of 30 deg. to the shaft S. The oil in the bath O is heated electrically and the temperature is measured by means of a thermocouple T. A camera box P is placed over the torque meter and carries a ground glass screen N on which a light trace from the torque meter may be observed. This screen is replaced by a photographic plate when taking records.

The torque meter (Fig. 2) consists of a torque bar F, one end of which is fixed at the bottom of a bored hole in the shaft S. The other end of the torque bar is fixed to the shaft carrying the test journal, and near this end, a diametral pin Q is attached. A mirror M_1 and an arm L are fixed to a bush, free to rotate on the pin Q. The tubular part G of the shaft S is free from the torque bar and carries at its end an arm K. A hardened steel ball on the end of the arm L makes contact with a hardened plane surface on the arm K. The two parts are kept in contact by means of a rubber band R. A twist of the torque bar causes the arm K to rotate the arm L about the pin Q and to tilt the mirror M_1 so that a ray of light impinging on the mirror is deflected

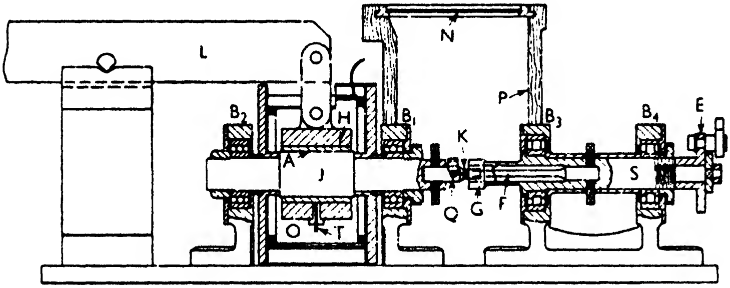

Fig. 1. Apparatus for the Measurement of Boundary Friction

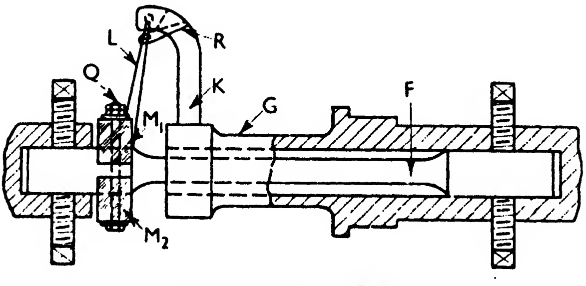

Fig. 2. Torque Meter

in a direction parallel to the axis of the machine. The oscillation of the shaft S causes the ray of light to be deflected in a direction perpendicular to the axis of the machine, so that the light traces on the screen N a closed figure on which distances in one direction are a measure of the angular position of the test journal and distances in a direction at right-angles to this are measurements of the twist of the torque bar, i.e. the frictional resistance of the bearing. A fixed mirror M_2 on the torque bar gives deflexions due to the angular motion of the shaft only and thus traces out a zero line of friction load on the screen. The machine is calibrated by applying known torques to the test journal shaft whilst it is moving very slowly through one cycle. The calibration chart thus obtained is placed over the friction diagrams and the friction forces are read off directly. The frictional resistance of the ball bearings B_1 and B_2 is included in the torque measured and it is assumed that this does not introduce an appreciable error. The bush is made of hardened steel to avoid distortion and wear.

To minimize the likelihood of a continuous fluid film being formed in the bearing, the bush has eight spiral grooves cut in the surface, four right-hand and four left-hand, the actual bearing surface thus consisting of a number of diamond-shaped facets.

Friction Measurements. Measurements of the friction during a cycle of oscillation have been made with a heavy mineral oil (viscosity at 25 deg. C.=620 centipoises) at loads of 100 to 500 lb. per sq. in., at frequencies of oscillation of 1 to 15 cycles per minute and at temperatures from air temperature up to 110 deg. C. The results are shown in Table 1.

(1) *Friction during a Cycle of Oscillation.* From the two sections of Table 1, it will be seen that the friction is constant throughout the cycle at a frequency of one cycle per minute at any load and temperature. At higher frequencies of oscillation, however, there is a fall in friction at the middle of the stroke through the temperature range 20–70 deg. C. This fall almost disappears at 90 deg. C. and completely disappears at 110 deg. C.

(2) *Effect of Frequency of Oscillation.* From 20–60 deg. C. the friction at the start of the stroke shows a small decrease as the frequency increases, the fall becoming less as the temperature rises. At 70 deg. C. it has become independent of the frequency and remains so at higher temperatures. The friction at the middle of the stroke falls considerably with increase of frequency up to a temperature of 70 deg. C. At 110 deg. C., however, it has become quite independent of frequency.

TABLE 1. COEFFICIENTS OF FRICTION

Temp., deg. C.	Load, lb. per sq. in.	At start of stroke, cycles per min.					At middle of stroke, cycles per min.				
		1	3	5	10	15	1	3	5	10	15
20	100	0·11$_5$	0·11$_5$	0·11	0·10$_5$	†	0·11	0·10	0·09	0·07	†
	200	0·12	0·12	0·11$_5$	0·11$_5$	†	0·11$_5$	0·11	0·10$_5$	0·09	†
	300	0·11$_5$	0·11$_5$	0·11$_5$	0·11	0·10$_5$	0·11$_5$	0·11	0·10$_5$	0·08$_5$	†
	400	0·11$_5$	0·11$_5$	0·11$_5$	0·11	0·10$_5$	0·11	0·10	0·10$_5$	0·09	0·07$_5$
	500	0·11$_5$	0·11$_5$	0·11	0·11	0·10$_5$	0·11	0·10$_5$	0·09	0·09	0·07
30	100	0·13	0·13	0·13	0·11$_5$	0·11	0·13	0·12$_5$	0·08	0·06	0·05
	200	0·12$_5$	0·11$_5$	0·10$_5$	0·10	0·09$_5$	0·11$_5$	0·11	0·08	0·06	0·05
	300	0·12	0·11$_5$	†	†	†	0·11$_5$	0·10	†	†	†
	400	0·12	0·12	0·11$_5$	0·11$_5$	0·11$_5$	0·11$_5$	0·10	0·09	0·07	0·05$_5$
	500	0·12	†	†	†	†	0·11$_5$	†	†	†	†
40	100	0·13	0·12$_5$	0·11$_5$	0·11$_5$	0·11$_5$	0·13	0·11$_5$	0·09$_5$	0·07	0·05$_5$
	200	0·13	0·13	0·13	0·12	0·12	0·13	0·11$_5$	0·10$_5$	0·09	0·07$_5$
	300	0·13	0·12$_5$	0·12$_5$	0·12	0·11$_5$	0·12	0·10$_5$	0·09$_5$	0·08	0·06$_5$
	400	0·13	†	†	†	†	0·12	†	†	†	†
	500	0·12$_5$	0·12$_5$	0·12$_5$	0·12	†	0·12	0·11$_5$	0·10$_5$	0·09	†
50	100	0·13$_5$	0·12$_5$	0·11$_5$	0·11	0·11	0·13$_5$	0·11	0·09	0·06$_5$	0·05
	200	0·13$_5$	0·13$_5$	0·13	0·12$_5$	0·12	0·13$_5$	0·12$_5$	0·12	0·10	0·08
	300	0·13$_5$	0·13	0·13	0·12$_5$	0·12	0·13$_5$	0·12$_5$	0·11	0·09	0·07$_5$
	400	0·13	0·13	0·13	0·12$_5$	†	0·13	0·12	0·10$_5$	0·08	†
	500	0·12$_5$	0·12$_5$	0·12	0·11$_5$	0·11$_5$	0·12$_5$	0·11$_5$	0·10$_5$	0·09	0·07$_5$
60	100	0·15	0·14$_5$	0·14	0·14	0·14	0·15	0·13$_5$	0·11	0·08	0·06
	200	0·14$_5$	0·14	0·14	0·13$_5$	†	0·14	0·12	0·10	0·07	†
	300	0·14	0·13$_5$	0·13	0·13	0·12$_5$	0·14	0·12$_5$	0·11	0·08$_5$	0·07
	400	0·13$_5$	0·13	0·13	†	†	0·13$_5$	0·11$_5$	0·10$_5$	†	†
	500	0·13$_5$	†	†	†	†	0·13$_5$	†	†	†	†
70	100	0·14$_5$	0·14	0·14	0·14	0·14	0·14$_5$	0·13	0·11	0·08$_5$	0·07
	200	0·14$_5$	0·14$_5$	0·14$_5$	0·14	†	0·14$_5$	0·13	0·11$_5$	0·09	†
	300	0·14	0·14	0·14	0·14	†	0·14	0·12$_5$	0·10$_5$	0·09	†
	400	0·13$_5$	0·13	0·13	0·13	†	0·13$_5$	0·12$_5$	0·12	0·10$_5$	†
	500	0·13$_5$	0·13	0·13	0·13	†	0·13$_5$	0·13	0·12	0·11	†
90	100	0·10$_5$	0·10$_5$	0·10$_5$	0·10$_5$	0·10$_5$	0·10	0·10$_5$	0·10$_5$	0·10$_5$	0·10
	200	0·10$_5$	0·11	0·11	0·11$_5$	0·11	0·10	0·11	0·11	0·11	0·10
	300	0·11	0·11$_5$	0·11$_5$	0·11$_5$	0·11$_5$	0·11	0·11$_5$	0·11$_5$	0·11$_5$	0·11
	500	0·12	0·12	0·12	0·12	0·12	0·12	0·12	0·12	0·11$_5$	0·11
110	100	0·10	0·10	0·10$_5$	0·10$_5$	0·10$_5$	0·09$_5$	0·10	0·10	0·10	0·10$_5$
	200	0·10	0·10$_5$	0·10$_5$	0·10$_5$	0·10$_5$	0·10	0·10	0·10$_5$	0·10$_5$	0·10$_5$
	300	0·11	0·11$_5$	0·11$_5$	0·11$_5$	0·12	0·11	0·11$_5$	0·11$_5$	0·12	0·12
	500	0·12	0·12	0·12	0·12	0·12	0·12	0·12	0·12	0·12	0·12

† Accurate measurements could not be made owing to excessive vibration.

(3) *Effect of Load.* The effect of variation in load is very small. The friction at the start of the stroke at temperatures up to 70 deg. C. is almost independent of load, whilst at 90 deg. C. and 110 deg. C. there is a small increase as the load increases. At the higher frequencies, the friction at the middle of the stroke shows a small increase with increase of load throughout the range of temperature.

(4) *Effect of Temperature.* At a frequency of one cycle per minute the friction both at the start and at the middle of the stroke shows an increase as the temperature rises up to about 60–70 deg. C., after which the friction falls. At the higher frequencies of oscillation the friction at the centre of the stroke shows a steady increase through the whole temperature range.

Suggested Explanation of Results. Considered from a molecular standpoint the bearing surfaces consist of a series of hills and valleys, and in an unlubricated state actual contact would only be made on the high spots. When a lubricant is introduced between the surfaces, a film of varying thickness will be formed, which under pressure and with no relative motion between the surfaces will be squeezed out until limiting boundary layers remain between the high spots. The parts of the surfaces between the high spots, however, will be separated by pools of lubricant, sufficiently deep to allow a middle region to exist which will be free from the influence of the enclosing surfaces, and in which a state of viscous flow may exist when motion takes place. On this assumption a reasonable explanation of most of the results can be offered.

When the surfaces are at rest the force required to start motion, and hence the static friction, will depend on the boundary layers on the high spots, since the pools of lubricant will offer no viscous resistance until motion actually takes place. It is suggested that when motion takes place and the surface speed reaches a certain value, the original static film is modified by the establishment of a thicker film (experiments have shown that, with continuous rotation, a complete fluid film may be formed in less than a revolution) and the higher the frequency of oscillation the thicker this film becomes. Consequently, the number of high spots, in effect, becomes less and the extent of the pools of "free" lubricant increases, resulting in a substitution of boundary friction by viscous, or fluid film friction, the net result being a reduction in total friction. The variation of friction during any particular cycle may be accounted for similarly, the lowest friction being recorded at the middle of the stroke where the surface speed is greatest. Below 3 cycles per minute, however, where there is no variation of friction during a cycle, it appears that the viscous resistance of the pools existing under static conditions is insufficient to affect the friction appreciably. Under these conditions the friction appears to be kinetic boundary friction (although the possibility should not be overlooked that the value of kinetic boundary friction may be higher than that of static boundary friction and the increase offset by a partial substitution of boundary friction by viscous friction). Above 3 cycles per minute the variation in friction during the stroke becomes appreciable but the departure from the complete boundary condition appears to be gradual.

The small decrease in the friction at the beginning of the stroke with increase in frequency of oscillation may be due to insufficient time being allowed on reversal of motion for the thicker film built up during motion to be squeezed out, and the original static boundary layers to be established.

The very small variation of coefficient of friction with load is evidence of predominant boundary conditions, since Amonton's Law, that coefficient of friction is independent of load, has been found by Hardy to hold rigidly under limiting boundary conditions. At the higher frequencies of oscillation, the small increase in the friction at the middle of the stroke with increasing load may be attributed to a reduction in the thickness of the layer of lubricant, resulting, as before, in a change, over parts of the surfaces, from fluid film to boundary conditions.

A similar effect will occur at the higher frequencies when the temperature is increased, as the accompanying reduction in viscosity of the lubricant will oppose the formation of a thicker film and give an increase in the friction.

The friction of mixtures of "B.P." paraffin and palmitic acid has also been measured at a frequency of 1 cycle per minute. The addition of 2 per cent palmitic acid caused a reduction in the coefficient of friction from 0·18 to 0·12 at 25 deg. C. and from 0·21 to 0·16 at 110 deg. C. A mixture containing 5 per cent palmitic acid gave similar results, whilst with a 1 per cent mixture, the friction was intermediate in value. The optimum lubricating mixture had thus been reached with the addition of 2 per cent palmitic acid.

Conclusion. At the lowest frequency of oscillation any effect due to viscosity appears to be absent, whilst chemical composition of the lubricant has a considerable effect on the friction. Below 3 cycles per minute it is thought that the bearing is operating under complete boundary conditions throughout the cycle and the machine thus provides a means of measuring boundary friction.

VISCOMETRY AS APPLIED TO PETROLEUM PRODUCTS IN THE UNITED STATES

By J. C. Geniesse*

The Saybolt viscometer is the most commonly used commercial apparatus for measuring the viscosity of petroleum products and lubricants. It was introduced about 50 years ago, almost simultaneously with the Redwood and Engler viscometers. All three were developed in response to a need for comparing the viscosities of mineral oils with those of vegetable and animal origin. Their application was rapidly broadened to uses for which they were not intended. For heavy oils, they proved to be quite satisfactory; however, for light oils, they were not only found to be inaccurate, but also gave results which were not significant.

In the United States the need for more accurate viscosity measurement became urgent about five years ago when the combined effect of unusually cold winters and smaller engine clearances made it necessary to use lighter engine oils to assure satisfactory starting. Since starting difficulties arose principally during cold weather, this indicated the desirability of determining the viscosity of oil at low temperatures. Committee D2, of the American Society for Testing Materials (A.S.T.M.) was requested to develop a method for that purpose. In view of the known difficulties of actually measuring the viscosity at the lower temperatures it was decided to determine it more accurately at higher temperatures, and then extrapolate by means of a viscosity-temperature chart to lower temperatures. It was believed that kinematic methods using capillary tubes would probably supply a solution to the problem; therefore, they were investigated, and finally adopted.

Theoretical Considerations. Almost 100 years ago Poiseuille (1846) started his extensive investigation of viscosity, ultimately leading to the following formula for long tubes :—

$$Q=\pi P d^4 t/128\mu l \quad . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

where Q is the discharge in time t, in cubic centimetres; P the difference in pressure between the two ends of the tube; l the length, and d the diameter of the tube, in centimetres; and μ the absolute viscosity, in poises.

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* Research Chemist, The Atlantic Refining Company, Philadelphia.

For shorter tubes the equation for kinematic viscosity finally developed (Herschel 1917) was:—

$$\frac{\mu}{\rho}=\frac{\pi g d^4 t}{128Q(l+\lambda)}\left(h-\frac{mv^2}{g}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where ρ is the density of liquid, in grammes per cubic centimetre; g the acceleration due to gravity $=981$ cm. per sec. per sec.; λ the equivalent length addition due to end effects; h the head of liquid, in centimetres; m the kinetic energy correction coefficient; and v the mean velocity of liquid through tube, in centimetres per second.

If the volume of discharge is kept constant, equation (2) for a given apparatus may be simplified to the following:—

$$\mu/\rho=Ct-B/t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In order to reduce errors due to the factors λ and m it is necessary to use a tube with a high ratio of l/d and a reasonably low efflux rate. To assure accurate viscosity measurement it is then only necessary to measure the efflux volume and time accurately. Inspection and study of the methods meeting the above requirements led to publication for information in 1936, and adoption as tentative standard D445–37T in 1937, of the suspended level and the modified Ostwald methods for measuring kinematic viscosity.

Suspended Level Method. The suspended level apparatus as developed by Ubbelohde (1936, 1937) and as modified by Ogden FitzSimons (1935) are illustrated in Fig. 1 (from A.S.T.M. Tentative Methods, 1937). On the left is shown the Ubbelohde instrument, and on the right two FitzSimons modifications. The latter are more compact, and therefore easier to install in a vapour bath. The instrument on the extreme right has two sizes of capillary tube, thus increasing the total range of a given viscometer. In the centre, alternative detailed cross-sections are given for the shoulder at the lower end of the capillary tube. With the spherical shoulder the surface tension at the bottom of the capillary balances that at the top, thus making it possible to test liquids with different surface tensions without corrections. With the square shoulder the surface tension pull at the bottom only is eliminated, making it necessary to correct for this factor if liquids with differing surface tensions are tested. In operation, bulb A is filled with liquid through tube 1. Tube 3 is closed and suction applied to tube 2 until the necessary quantity of liquid is drawn into bulb C. When tubes 2 and 3 are opened to the atmosphere the oil in bulb 5 immediately drains into bulb A, producing a suspended liquid level at the lower end of the capillary tube. This suspended level increases the accuracy since it assures a more constant head of liquid. The observer then determines the time interval required for the upper meniscus to drop from the etched line above bulb B to that below. Any capillary diameter may be used provided the efflux time is greater than 80 seconds. This limit was established to reduce the errors due to kinetic energy losses. In the calibration of the instruments, B in equation (3) is assumed for all, except very small, capillary tubes. If capillaries below 0·75 mm. are used the constant must be determined by experiment.

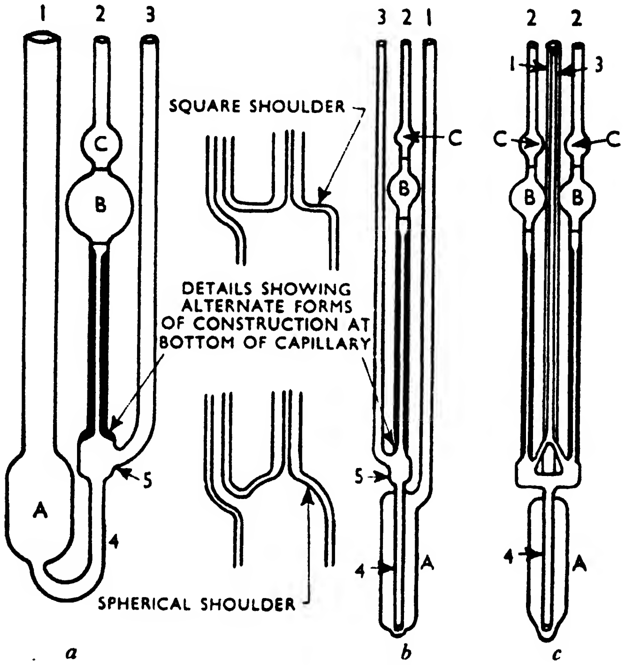

Fig. 1. Suspended-Level Viscometers

a Ubbelohde viscometer with one capillary tube.
b FitzSimons's modification with one capillary tube.
c FitzSimons's modification with two capillary tubes.

Modified Ostwald Method. This apparatus, shown in Fig. 2 (from A.S.T.M. Tentative Methods, 1937), was developed by Prof. Fenske (1935, 1936) at Pennsylvania State College. Bulb C is placed in line with bulbs A and B in order to reduce errors due to misalignment. Bulb A was added to reduce variations in volume of liquid introduced into the viscometer. This precaution, in conjunction with the use of a large diameter for bulb C, practically eliminates all errors that may arise from variations in liquid head. The dimensions of the instrument and the minimum outflow time are selected so as to make the kinetic energy effect insignificant. The constant B in equation (3) is assumed to be zero. In operation, the instrument is charged by inverting and drawing liquid by suction into tube 2 until the liquid level reaches the etched mark at the end of the working capillary. The viscometer is then adjusted in the bath. After the desired temperature is obtained the liquid is drawn back up into bulbs A and B, and the observer then determines the time interval required for the meniscus to drop from the etched line above B to that below.

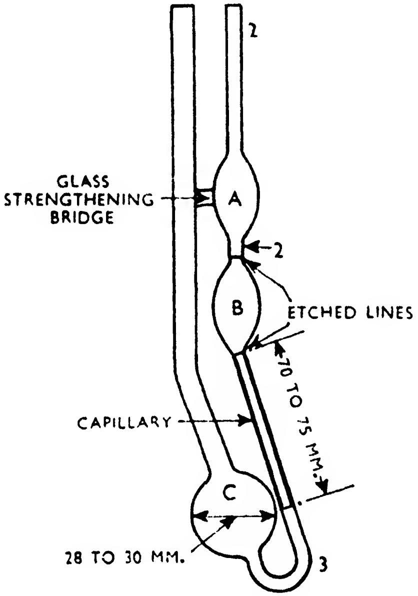

Fig. 2. Modified Ostwald Viscometer

Calibration and Accuracy of Kinematic Viscometers. In calibrating the viscometers a radical departure has been made from the practice now possible with the Saybolt instrument. In the Saybolt instrument a master tube serves as the standard. Secondary standards are calibrated by comparing outflow times for given reference oils. Commercial instruments are then calibrated in a similar manner, using the secondary standard tubes. This method is open to the objection that

TABLE 1. CONVERSION OF KINEMATIC TO SAYBOLT UNIVERSAL VISCOSITY

Kinematic viscosity at 100 deg. F., centistokes	Saybolt universal viscosity at 100 deg. F., sec.	Kinematic viscosity at 100 deg. F., centistokes	Saybolt universal viscosity at 100 deg. F., sec.	Kinematic viscosity at 100 deg. F., centistokes	Saybolt universal viscosity at 100 deg. F., sec.	Kinematic viscosity at 100 deg. F., centistokes	Saybolt universal viscosity at 100 deg. F., sec.
2	32·6	14	73·4	29	136·5	44	203·8
2·5	34·4	15	77·2	30	140·9	45	208·4
3	36·0	16	81·1	31	145·3	46	213·0
3·5	37·6	17	85·1	32	149·7	47	217·6
4	39·1	18	89·2	33	154·2	48	222·2
4·5	40·7	19	93·3	34	158·7	49	226·8
5	42·3	20	97·5	35	163·2	50	231·4
6	45·5	21	101·7	36	167·7	55	254·4
7	48·7	22	106·0	37	172·2	60	277·4
8	52·0	23	110·3	38	176·7	65	300·4
9	55·4	24	114·6	39	181·2	70	323·4
10	58·8	25	118·9	40	185·7		
11	62·3	26	123·3	41	190·2		
12	65·9	27	127·7	42	194·7	Above 70 centistokes: Saybolt sec.=centistokes ×4·620	
13	69·6	28	132·1	43	199·2		

loss of or damage to the master tube makes it necessary to set up a new standard which in all probability will not be exactly the same as the old one. Another difficulty arises from the fact that the kinetic energy correction is large in the Saybolt instrument, thus making it necessary to calibrate at several efflux times in order to obtain a high degree of reproducibility.

The kinematic viscometers are calibrated, using water as the ultimate reference liquid. The viscosity of water is assumed to be 1·007 centistokes at 68 deg. F. For this purpose, water is run through a fine capillary to obtain the constant for the instrument, then a more viscous oil is tested. This oil in turn is run into the next larger size capillary to determine the constant of this viscometer. In this way all the viscometers may be calibrated, using water as the ultimate standard, thus making it possible for any one familiar with ordinary viscometer technique to calibrate his own instruments. For the convenience of those who do not care to go through this tedious operation, certified oils have been prepared and are being furnished by the American Petroleum Institute and the National Bureau of Standards for calibration purposes.

Under "Theoretical Considerations" it was pointed out that, for great accuracy, it was necessary to measure time accurately. It is also equally important to control the temperature within close limits. Limiting the time of outflow to values not less than 80 seconds makes possible the accurate measurement of time. New thermometers calibrated to ±0·02 deg. F. reduce errors due to faulty temperature control to a minimum. Assuming the viscosity of water to be correct, indications are that the new kinematic methods should be accurate to within ±0·2 per cent.

Conversions. In view of the fact that industry will continue to specify and think in terms of Saybolt seconds for some time, it was necessary to obtain an official method for converting centistokes to Saybolt seconds. Accordingly, samples of oil were tested (Headington and Geniesse 1937) in the new kinematic equipment, and in Saybolt instruments with very small corrections. Data were obtained at 70, 100, 130, 210, and 300 deg. F. The data at 100 deg. F. permitted the calculation of the figures shown in Table 1.

The data at the other temperatures enabled the calculation of the conversion for temperatures other than 100 deg. F. The formula for calculating the Saybolt equivalent at any temperature from the corresponding value at 100 deg. F., given in Table 1, is as follows:—

$$\text{Saybolt seconds at } t \text{ deg. F.} = 1 + (t - 100)0{\cdot}000064 \times \text{Saybolt seconds at 100 deg. F.}$$

Thus, 10 centistokes at 210 deg. F. are equivalent to 58·8 × 1·0070 or 59·2 Saybolt sec. at 210 deg. F.

The fact that the volume of oil is measured after cooling below the temperature of test in the Saybolt instrument is the principal factor responsible for the variation of the conversion with temperature. The above was adopted as "Tentative Method for Conversion of Kinematic Viscosity to Saybolt Universal Viscosity", A.S.T.M. Designation: D446–37T.

Viscosity Temperature Chart. In 1932 the A.S.T.M. adopted method D341–32T for estimating viscosity by means of the viscosity temperature chart. At that time many charts were examined, resulting in the choice of one using as its basis the empirical equation proposed earlier by N. MacCoull of the Texas Company. The viscosity lines were spaced (Geniesse and Delbridge 1932) in accordance with the expression log log $(cs+0{\cdot}8)$ and the temperature linings proportional to the logarithm of the absolute temperature. Data plotted on that chart showed slight curvature with the concave side up at low viscosities. Revision was undertaken and new charts adopted as "Tentative Standard Viscosity-Temperature Charts for Liquid Petroleum Products", A.S.T.M. Designation: D341–37T. The new charts, which are obtainable from the American Society for Testing Materials, are :—

(A) Saybolt, full size.
Viscosity, 33 to 100,000,000 seconds.
Temperature, −30 to +450 deg. F.
Sizes, 16 × 20½ inches.

(B) Saybolt, abridged size.
Viscosity, 33 to 100,000 seconds.
Temperature, −10 to 300 deg. F.
Size, 8½ × 11 inches.

(C) High kinematic viscosity range.
Viscosity, 2 to 20,000,000 centistokes.
Temperature, −30 to 450 deg. F.
Size, 16 × 20½ inches.

(D) Low kinematic viscosity range.
Viscosity, 0·4 to 100 centistokes.
Temperature, −30 to 450 deg. F.
Size, 20 × 20½ inches.

In charts A, B, and C the constant in the kinematic viscosity scale is 0·6 instead of 0·8 as previously used. Charts A and B have Saybolt lines only which slope sufficiently to take into account the effect of temperature on the Saybolt viscometer. Charts C and D have only kinematic lines so spaced that the constant in C is 0·6 and in D it varies from 0·6 at the top to 0·75 at the bottom. Both charts are on the same scale so that one may be superimposed on the other over the overlapping range in order to make one chart covering the range 0·4 to 20,000,000 centistokes.

Standardization of Saybolt Scale. The Saybolt as well as the Engler and Redwood instruments depend upon master tubes for calibration, a very unsatisfactory situation since, as previously stated, injury to or loss of the tube makes it impossible to duplicate the viscosity scale accurately. Furthermore, the fact that all of these instruments are used under conditions of turbulent flow makes accurate calibration difficult. A percentage correction obtained at a given efflux time does not necessarily hold throughout the useful range of the instrument. The first difficulty may be eliminated by defining the Saybolt second scale in terms of centistokes. In view of the fact that the centistoke scale can be reproduced without the aid of a reference instrument this would give a reproducible Saybolt second scale. For extreme accuracy it would still be necessary to calibrate the Saybolt instrument at several times of outflow. To eliminate this tedious operation it would be necessary to discard the Saybolt instrument and to substitute kinematic equipment. The greater accuracy and reproducibility of the kinematic equipment makes it a decidedly more desirable instrument. The use of such equipment also opens the way for the more general use of the centistoke scale. The centistoke scale would not only provide a universal scale, simplifying international correspondence and specification writing, but would also give viscosity units which are more nearly in line with measurements of friction in bearings, pipe lines, and tubes. This scale would be far more significant to the average individual than the commercial viscosity scales now widely used.

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GRAPHICAL SOLUTIONS OF VISCOSITY PROBLEMS

By J. Groff*

Viscosity is not constant. It decreases as the temperature rises and vice versa. It is, therefore, necessary to state at what temperature the viscosity must be considered between certain limits compatible with the necessities of the lubrication problem concerned. On the other hand, viscosity is not expressed in the same manner in the various countries of the world. In England, it is given in seconds Redwood, in America as seconds Saybolt, and in France and Germany as degrees Engler. In France use is also made of the fluidity Barbey. The temperature-viscosity relations are sometimes based on a viscosity index such as that of Dean and Davis, whilst motor engine oils, gear oils, and back axle oils are classed according to certain viscosity ranges due to the Society of Automotive Engineers (S.A.E.).

Owing to these various methods of stating viscosity, there is often much confusion. As will be seen below the graphical method of the author's company, which has been in practical use for more than five years, admits of representing clearly everything connected with viscosity as a function of temperature, and of solving with a precision of the order of ±2 per cent, the problems which occur most frequently. A brief outline of some types of these problems is as follows:—

(1) To express the viscosity in whatever units at whatever temperature (between 0 deg. F. and 500 deg. F.).

(2) To determine at what temperature two oils have the same viscosity and which of them is the more viscous above or below this temperature (so-called temperature of equiviscosity).

(3) To find the Dean and Davis viscosity index of an oil as a function of any two viscosities of the oil.

(4) To determine the proportions of blends of two or more oils, to produce given viscosity characteristics and vice versa.

(5) To determine which oil or oils of given viscosities will conform to one of several given specifications, and to deduce therefrom concordance or discordance of two or more specifications.

(6) As a particular application of the preceding problem, to be able to find the S.A.E. number of an automobile oil.

(7) To determine the Ubbelohde pole heights and index *m*, knowing two viscosities or two relations between them.

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* Chief of the Technical Service, Texaco, Paris.

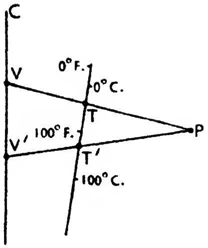


Fig. 2

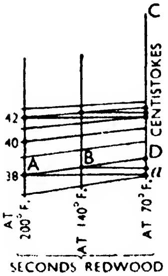


Fig. 3

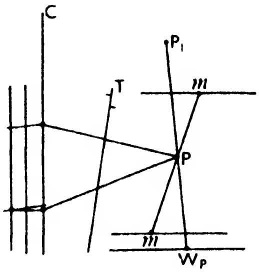
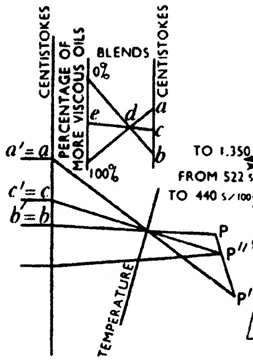


Fig. 4



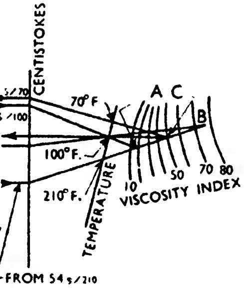


Fig. 5

Fig. 6

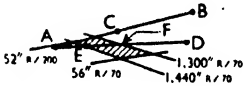


Fig. 7

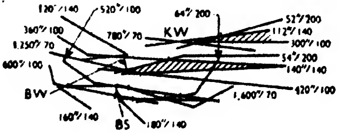


Fig. 8

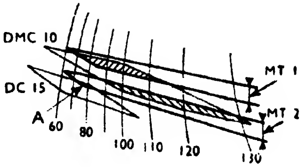


Fig. 9

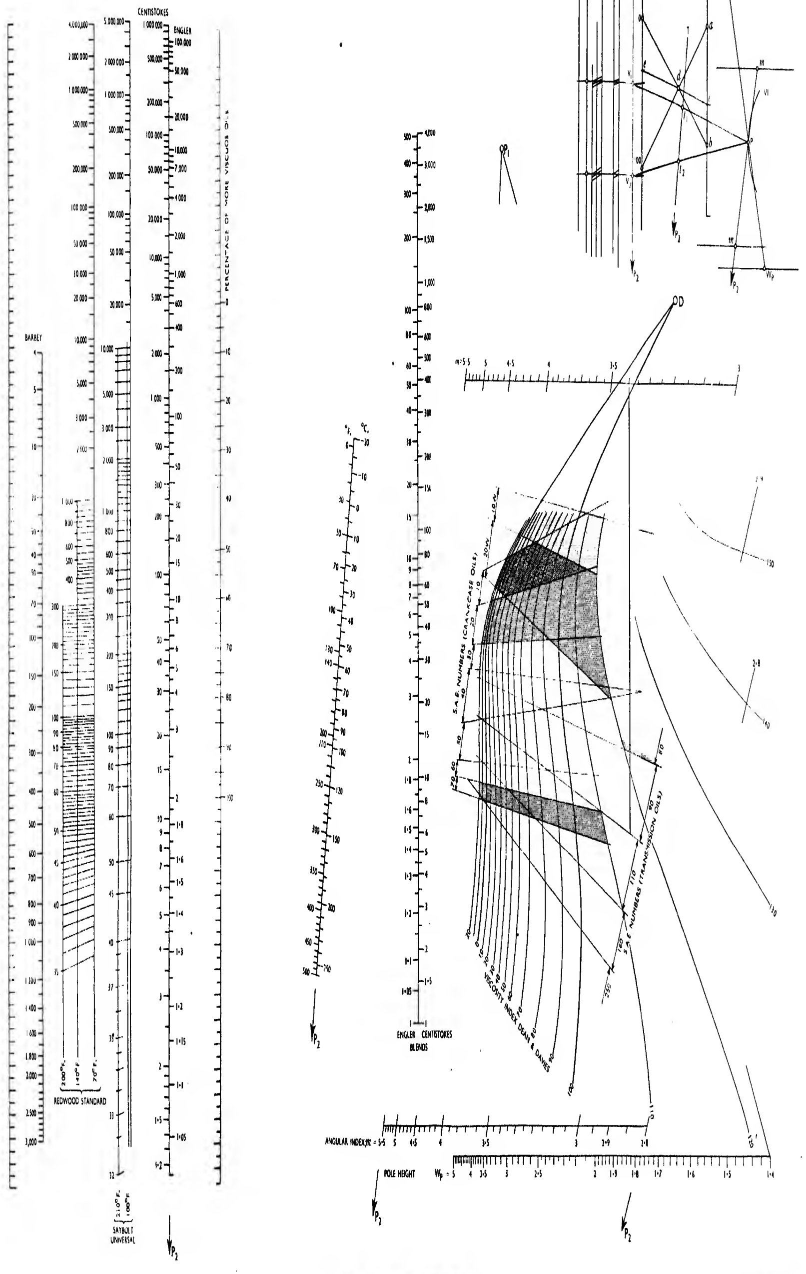


Fig. 1. Viscosity Conversion Chart

(Copyright by the Texas Company, New York.)

[*To face p.* 318]

The Chart. The chart comprises essentially two scales (Fig. 1). One scale C, graduated in centistokes, and the other a temperature scale T, graduated in degrees F. and degrees C. These graduations are determined by means of the equation of Walther. This equation is a linear function of two logarithmic functions of the kinematic viscosity in centistokes and the absolute temperature. For further details reference should be made to the proceedings of the Second World Petroleum Congress, Paris, 1937.

The viscosity V in centistokes at the temperature T is shown on the chart by the straight line VT in Fig. 2. For the same oil all the straight lines such as VT will meet at a given point P, and this point P on the chart will represent the particular oil in question, and can be called the point of origin of the oil. If it is desired to know the viscosity of this oil at a particular temperature T′, a straight line PT′ is drawn, which cuts the scale C at the point marked V′, which gives the required viscosity.

To convert from viscosities in centistokes to Redwood viscosities,* for example, all that is required is to place alongside the scale C, the seconds Redwood scale as shown. Since, however, correspondence of these viscosities is not rigorously independent of the temperature, it is necessary to show this scale of seconds Redwood as in Fig. 3. Thus, 38 sec. at A corresponds to *a* centistokes at 200 deg. F., at B to *b* centistokes at 140 deg. F., and at D to *d* centistokes at 70 deg. F. The slope of the lines, such as AD, diminishes as viscosity increases. At intermediate temperatures it is easy to interpolate the required centistokes.

The same procedure is applied in order to obtain the Saybolt viscosities at 100 deg. F. and 210 deg. F. A chart in common use in the United States, for example†, would show the seconds Saybolt scales placed alongside the centistokes scale. In France the Engler scale is used directly joined to the centistokes scale. Charts such as Fig. 1 have been prepared, where Saybolt centistokes and Redwood centistokes are the reference scales in place of the Engler centistoke reference scale.

The Redwood and Saybolt scales and the Barbey fluidity scale are placed on the left. The viscosities are converted in the same manner, i.e. by horizontal lines which pass through the point of the given centistokes scale. For greater facility the viscosity scales are enclosed by two vertical graduated lines, which assist in ensuring that the lines are traced rigorously horizontal without having recourse to drawing board equipment.

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* Jl. Inst. Petroleum Technologists, 1936, vol. 22, p. 21.

† Tentative Standards, Am. Soc. Testing Materials, Committee D-2, D 341-32T, 1936.

On the right of the chart curves of equal Dean and Davis viscosity index are placed from −20 to 150. As a variation of ±10 per cent is always possible in the viscosity index, sight interpolation is sufficient. The Dean and Davis index of a given oil is shown simply by the position of the point of origin describing the oil on the family of curves.

Fig. 4 reproduces the method of determination of the Ubbelohde index *m* and the pole height W_P*. In order to determine *m*, a line is drawn through the point P of the given oil to cut the horizontal scale *m* at an equal value of *m* above and below, and this figure is the value required. In order to determine the pole height, P is joined to the fixed point P_1, and the line is produced through the scale W_P; the point where the line cuts W_P is the required pole height.

Mixtures of Oils. Between the temperature scale and the centistokes scale (Fig. 1) there is a scale graduated in percentages. This scale shows the percentage of the more viscous oil in a binary mixture. Fig. 1, top right-hand side, shows how to obtain the viscosities of a mixture of two oils. Let *a* be the viscosity of the more viscous oil at a given temperature and *b* the viscosity of the less viscous oil at the same temperature. Join *a* to the point representing 100 per cent and *b* to the point of zero per cent, and let *d* be the intersection of these two lines. A mixture of *e* per cent of the more viscous oil with (100–*e*) per cent of the more fluid oil, has the viscosity *c* at the given temperature, obtained by drawing a line *ed* and producing it to meet the viscosity scale. In making this construction at various temperatures, all the various possible mixes of two given oils shown on the chart by the points P and P′ are represented by the straight line PP′. Similarly, it may be deduced that all the possible blends of three oils are represented by points situated inside a triangle which has for its apexes the points of origin of the three oils, P, P′, and P″. The following examples are given to show the utility of the graphical method of solution of viscosity problems:—

Question 1. Given an oil of viscosity 522 sec. Saybolt at 100 deg. F. and 54 sec. Saybolt at 210 deg. F., determine:—

(1) Its viscosity Saybolt at 70 deg. F. and 130 deg. F.
(2) Its viscosity Redwood at 70 deg. F., 140 deg. F., 200 deg. F., and 212 deg. F.
(3) Its viscosity Engler at 20 deg. C., 35 deg. C., and 50 deg. C.
(4) Its Dean and Davis viscosity index.
(5) At what temperature has this oil a viscosity of 1,000 sec. Saybolt and 1,000 sec. Redwood ?

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* Ubbelohde, "Zur Viscosimetrie", 1936 (Leipzig).

Answer. The point of origin of this oil is determined by the intersection of the two straight lines V–100 deg. F. and V′–210 deg. F., V being the number of centistokes corresponding to 522 sec. Saybolt at 100 deg. F. and V′ being the centistokes corresponding to 54 sec. Saybolt at 210 deg. F. This point of origin P being determined, all that is necessary is to join the points of the scale at the various temperatures stated, and to read off the Engler scale, centistokes, or degrees at the given temperatures. By drawing horizontal lines from these centistokes the viscosities in the other units are obtained:—

(1) 1,800 sec. Saybolt at 70 deg. F.; 210 sec. Saybolt at 130 deg. F.

(2) 1,550 sec. Redwood at 70 deg. F.; 138 sec. Redwood at 140 deg. F.; 52 sec. Redwood at 200 deg. F.; and 48 sec. Redwood at 212 deg. F.

(3) 58 deg. Engler at 20 deg. C.; 18·2 deg. Engler at 35 deg. C.; and 7·4 deg. Engler at 50 deg. C.

(4) Viscosity index=16.

(5) 1,000 sec. Saybolt corresponds to a certain number of centistokes, and the corresponding point, joined to the point P of the oil, gives a line passing through the temperature scale at the temperature required, namely, 29 deg. C. or 84 deg. F. Starting with 1,000 sec. Redwood, a temperature of 27 deg. C. (81 deg. F.) is obtained.

Question 2. Suppose it is required to raise the viscosity index of the oil mentioned in question 1 from 16 to 50, by blending it with another oil, whose viscosity index is 75, and which possesses the same viscosity at 210 deg. F., determine (*a*) the proportions of the blends of the two oils required and (*b*) the Saybolt viscosity of the blend at 70 deg. F. and 100 deg. F.

Answer. Fig. 6 shows the tracing obtained from the given data. The point of origin of the oil is determined as in the preceding example. A straight line A–210 deg. F. is produced until it cuts the curve of the viscosity index 75. This gives a point B, which represents the oil to be mixed with oil A in order to obtain the oil C of viscosity index 50, and the same viscosity as A and B at 210 deg. F. In order to obtain the proportions of A and B it is necessary to consider viscosity at a temperature as far as conveniently possible from that of the temperature of equal viscosity, which in this case is 210 deg. F. If the viscosity is taken at 70 deg. F., the answer is that 34 per cent of oil B of viscosity index 75 and 66 per cent of oil A of viscosity index 16, are necessary to produce the required oil. The required viscosity of oil C at 70 deg. F. is 1,350 sec. Saybolt, and at 100 deg. F. 440 sec. Saybolt.

Question 3. If an engine builder specifies oil of a viscosity of 1,300–1,440 sec. Redwood at 70 deg. F. and 52–56 sec. Redwood at 200 deg. F., it is required to know whether one of the three oils A, B, or C of the previous example satisfies the specification.

Answer. This question can be solved graphically either directly on the chart or tracing paper can be applied to the chart, and the lines drawn representing the given specification. The straight lines 1,300 sec. and 1,440 sec. at 70 deg. F., and those of 52 and 56 sec. at 200 deg. F. define a quadrilateral. Oils which satisfy the imposed conditions must be represented by points of origin within this quadrilateral. If a tracing is made of the points A, B, and C of the preceding example, all that it is necessary to do is to superimpose the tracing for the specification on that for the three oils, or read directly on the chart, whichever is preferred.

Fig. 7 shows the results obtained, and it is noted that none of the oils A, B, or C satisfies the specification. In order to obtain an oil to satisfy the specification it will be necessary to mix A with an oil D, whose viscosity at 210 deg. F. is greater than that of A. A straight line AD will then cut the quadrilateral, and the limiting admissible proportion of the satisfactory blends of A and D are given by the points E and F, where AD cuts the sides of the quadrilateral, and the required blends to produce these oils may be calculated as in question 2. As an example, Fig. 8 reproduces graphically a well-known specification BW, KW, and BS for Diesel engine lubricating oil, the specifications of the engine builders being given in Table 1, together with the corresponding specifications deduced from the chart.

The minimum and maximum viscosity indexes permitted by these specifications are 82–101 for BS, 59–118 for BW, and 95 minimum for KW.

Question 4. What are the oils of S.A.E. viscosity range which can satisfy the specifications mentioned in question 3 ?

Answer. The straight lines giving the limits of the different zones of S.A.E. viscosities can be traced on the chart and it will be seen in which S.A.E. zone the areas for the above specifications fall. Most oils which satisfy the BS specification are S.A.E. 30 oils of the upper viscosity limits, and a few S.A.E. 40 oils. Almost all oils satisfying the BW specification are S.A.E. 30, there being a few S.A.E. 20 oils with viscosities at the upper part of this range which satisfy the specification if their viscosity indexes are between 76 and 97. For specification KW only S.A.E. 20 or S.A.E. 20W oils would be satisfactory.

TABLE 1

Engine builder's specification		Specification deduced from chart, sec. Redwood
Temp. of test, deg. F.	Sec. Redwood	
Specification BS (*hot climates*):—		
70	1,600 maximum	1,600 maximum
100	520–600	520 minimum
140	160–180	180 maximum
200	64 minimum	64 minimum
Specification BW (*temperate climates*):—		
70	1,250 maximum	1,250 maximum
100	260–420	260 minimum
140	120–140	140 maximum
200	54	54 minimum
Specification KW (*cold climates*):—		
70	780 maximum	—
100	300 maximum	300 maximum
140	112 minimum	112 minimum
200	52 minimum	52 minimum

Question 5. Oils are required to meet both the users' and engine builders' specifications at one and the same time.

Answer. Fig. 9 shows M.T. 1 and M.T. 2 specifications of the French Navy and DMC. 10 and DC. 15 specifications of an engine builder. It will be seen that only the most fluid oils of the DMC. 10 range can satisfy M.T. 1 (the shaded part of Fig. 9), and only the most viscous oils of DMC. 10 can satisfy specification M.T. 2. Oils which satisfy DC. 15 will only very exceptionally be oils which will satisfy the M.T. 2 specification, since the only part overlapping of D.C. 15 with M.T. 2 is the small area shown at A. The viscosity characteristics of oils common to both specifications can be readily found by the methods given above.

Question 6. Given a "200 pale oil" of Redwood viscosity 73 sec. at 140 deg. F. and a bright stock of 3,200 sec. Saybolt at 100 deg. F. and 160 sec. at 210 deg. F., is it possible to mix these to provide a series satisfying the S.A.E. classification? Find the percentages of the blends, the approximate viscosity indexes, and the Engler viscosities at 50 deg. C. and 100 deg. C.

Answer. The points of origin A and B are readily obtained and the line AB is found to cross the S.A.E. zones from 10 to 70. Taking the middle of the S.A.E. ranges, and applying the mixture construction as above for 50 deg. C. or 140 deg. F. as examples, the results obtained are shown in Table 2.

TABLE 2

S.A.E. No.	Percentage composition		Viscosities, Engler		Index, Dean and Davis
	A	B	at 50 deg. C.	at 100 deg. C.	
10	100	0	3·4	1·42	25 to 30
20	82	18	5·1	1·57	55
30	66·5	33·5	7·5	1·78	68
40	51·5	48·5	10·6	2·04	72
50	35	65	16·8	2·53	74
60	17·5	82·5	25·2	3·23	76
70	8·5	91·5	32·5	3·8	78

Such results would be a ready guide to the laboratory and would avoid a number of trial and error experiments. From the chart also the Ubbelohde indexes *m* and the pole heights W_P may be readily found from the points of origin of any given oils.

As regards the S.A.E. 40 and S.A.E. 50 viscosities, the chart (Fig. 1) and Table 2 do not take into account the modification by which the Society of Automotive Engineers has carried the limit of Saybolt viscosity at 210 deg. F. from 75 seconds to 80 seconds, which is the limit separating the S.A.E. 40 and S.A.E. 50 viscosities.

PRACTICAL RUNNING TESTS WITH GEAR OILS

By Reg. Rat. Dr.-Ing. A. Hagemann *

The researches discussed in this paper constitute part of an investigation on the driving mechanisms of automobiles and concern the relations between the properties of the oils and the construction of the driving mechanism of power-driven vehicles. The object was to select oils which cause the least loss of power and minimize the stress of the metallic parts.

Oils are often tested in special testing machines, but the results obtained lack correlation with practice. This use of the testing machine depends on the assumption, in itself correct, that the lubricating value is the chief property for the evaluation of gear oils. The physical conditions—high pressure, small bearing surface, and so on—are such that fully fluid friction does not occur, so there is no clear relation of the lubricating effect to the viscosity of the oil. In fact, the use of gear oils lies rather in the region of boundary friction, a region in which "oiliness" plays an important if not decisive part. Oiliness still lacks a satisfactory physical definition, as well as an accurate method of test, especially in relation to conditions of pressure and temperature. In comparison with the lubricating value, the other properties of the oil are far less important, being related to matters of practical significance such as behaviour at low temperatures, corrosion, and so on. In this category falls the frothing of the oil, a property which is often observed but has apparently not been completely elucidated.

For these reasons, practical running tests were carried out on drives of types in current use. Road tests were neglected because they are not reproducible, while the results of individual tests are not comparable. The test bed, however, was designed so that the conditions approximated as closely as possible to those of the road. The investigations were carried out on a Krupp lorry with rear drive, as well as on several direct and oblique-toothed drives made by the Zahnradfabrik Friedrichshafen.

The test bed for the worm drive of the lorry, with the straight-toothed group drive, is shown diagrammatically in Fig. 1. An alternating-current motor, running at 725 r.p.m., and taking a maximum of 44 kW. at 147A. drives the group drive through V-belts (step-up ratio, 1/3·5). The group drive ran in bottom gear (gear ratio, 1/2·22), so that the worm shaft ran at 1,140 r.p.m., equivalent to a speed of 40 kilometres per hour. The four road wheels were replaced by dynamometers, by means of which the

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* Gruppenleiter im Heereswaffenamt, Berlin.

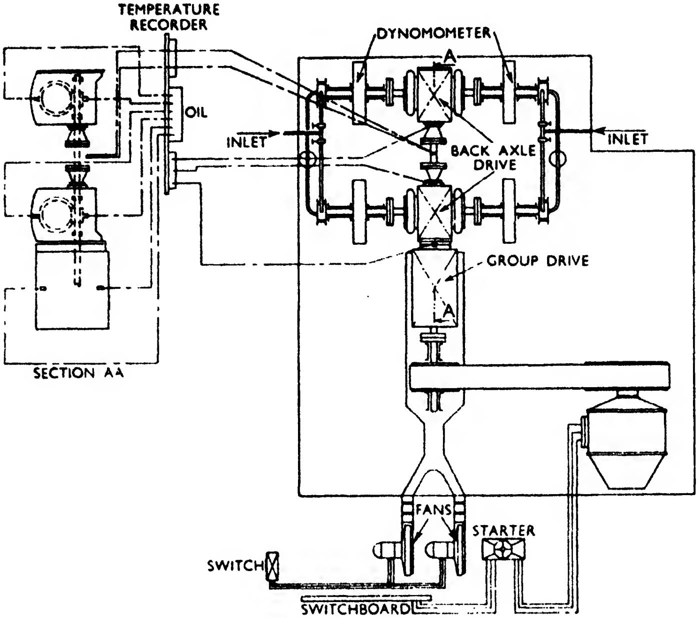

FIG. 1. Test Bed Arrangement with Worm Drive

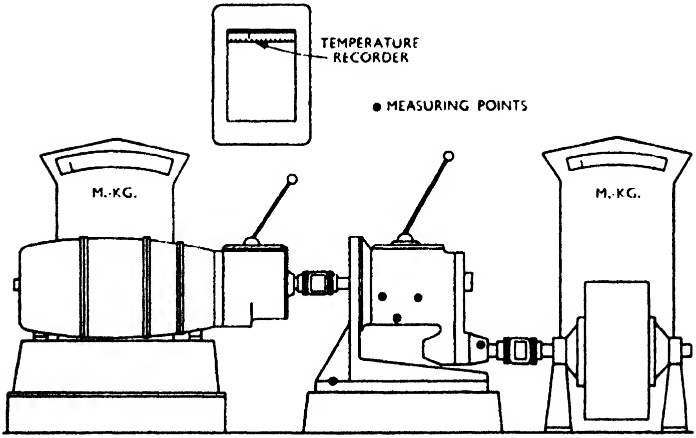

FIG. 2. Test Bed used for Testing Oils

worm drive could be given a continuous load of 100 per cent. The movement of air at a speed of 40 kilometres per hour was simulated by the action of two fans. Thermocouples were mounted 2 mm. above the worm, just below the wormwheel.

Another test bed (Fig. 2) was used to test oils in different drives made by the Zahnradfabrik Friedrichshafen. For one set of tests an oblique-toothed "G 25 Aphon" drive was used. The speed on the driving side was 3,000 r.p.m., against 1,790 r.p.m. on the driven side, the brake load being $N_B=29\cdot7–30\cdot3$ h.p. In a second set of tests another oblique-toothed drive of different construction was used with the second speed engaged. Here the driving speed was 2,920 r.p.m., the driven speed 753 r.p.m., and the brake load $N_B=29\cdot7–30\cdot3$ h.p.

The first experiments with the test bed shown in Fig. 1 were designed to disclose the best conditions for the series tests with oils of very different character. With 100 per cent sustained load, i.e. with a load on the Zahnradfabrik group drive of 60 h.p., the load on each axle drive being 30 h.p., there was an early occurrence of pitting owing to excessive stress, followed by greater wear and premature destruction of the back axle drive. Therefore the load was reduced to 75 per cent in subsequent tests. Special attention was paid to the sealing of all the drives. By using sealing rings made from "Buna N" synthetic rubber, perfect sealing was obtained even with oils of low viscosity (1·6 deg. Engler at 100 deg. C.) and at temperatures up to 150 deg. C. The Buna material remains perfectly fit to use whereas leather used under the same conditions is completely worn out (Fig. 3).

Fig. 3. Wear of Sealing Rings

a Leather ring after running for 390 hours, comprising 291 hours at 5,200 r.p.m., and 99 hours at 950 r.p.m.

b Ring of synthetic material after running for 820 hours, comprising 640 hours at 5,200 r.p.m., and 180 hours at 950 r.p.m.

It was found that the oil level had a considerable influence on heating due to churning, even under no load. The explanation is probably that owing to the pumping action of the gear teeth or the wormwheels when excess oil is present, an oil pressure is set up which produces a considerable braking effect so that, besides the rise in temperature, there is loss of power.

Tests were also carried out with various oils. The test with each oil lasted 24 hours, with changes every 4 hours from no load to three-quarters load. To avoid erroneous results, the oils were given a further test in a new series of experiments. The characteristics of the oils are given in Table 1, while the temperatures attained in the tests are summarized in Table 2. The ash contents of the oils after a 24-hour test are given in Table 3. The evaluation of the oils, derived from Tables 2 and 3, is set out in Table 4. The results of similar tests of oils 1, 3, 4, 5, 8, and 10, under various loads, are reproduced in Table 5.

As a result of the findings, tests were carried out with selected oils. A close-cut distillate of paraffin-base oil, and a close-cut naphthene-base

TABLE 1. CHARACTERISTICS OF THE LUBRICANTS TESTED

Oil	Nature	Viscosity, sec. Engler		Viscosity index	Resin content	Asphalt content	Ash, mg. in 100 grammes oil
		at 20 deg. C.	at 100 deg. C.				
1	Pure mineral oil prepared by the "Selective" process, containing bright stock—known winter motor oil	31·2	1·75	92	2·9	0·1	3·5
2	Pure mineral oil, prepared by the sulphuric acid process, not containing bright stock	50·0	1·97	91	3·3	0·02	2·2
3	Synthetic oil, made by the $AlCl_3$ process, from pure hydrocarbons; contains bright stock	55·8	2·02	90	3·0	0·02	0·8
4	Light extreme-pressure lubricant, consisting of a paraffin-base oil compounded with sulphurized fatty oils	82·9	2·51	104	9·4	0·3	7·1
5	Light extreme-pressure lubricant, consisting of a naphthene-base oil compounded with sulphurized fatty oils	167·0	2·78	79	8·8	0·1	8·5
6	Mineral oil, containing bright stock and organic phosphorus compounds	277·0	2·91	54	7·7	0·2	130·0
7	As No. 6	355·0	3·83	87	16·1	0·2	30·0
8	Mineral oil containing bright stock compounded with fatty oils	397·0	4·60	94	8·3	0·04	9·0
9	Mineral oil containing bright stock, fatty oils, and small amounts of a soap	448·0	5·24	100	8·0	0·02	3·2
10	As No. 9	537·0	5·28	94	8·5	0·1	1·2

TABLE 2. TEMPERATURES IN DEG. C. AT 75 PER CENT LOAD (AVERAGE VALUES)

The first ten tests were made with worm No. 5 and the second ten with worm No. 6.
A. No-load temperature less room temperature.
B. Temperature rise under load.

Oil	Group drive		Back-axle drive 1		Back-axle drive 2		Room temperature
	A	B	A	B	A	B	
1	41	12	20	41	13	48	23
2	42	20	20	49	14·5	55·5	21
3	43	16	20	42	17·5	45·5	20
4	52	16	25·5	43·5	23·5	43·5	19
5	50	17	21	52	20	52	24
6	49	15·5	23	46·5	23	47·5	23
7	48	13·5	23	42·5	19	47·5	23
8	55	10	27	41	24·5	47·5	23
9	59	20	30	56	29	53	23
10	58	10·5	28	41	26·5	42·5	22
1	32	23	13	65	7	70	25
2	39	23	19	68	12	75	22
3	40	20·5	19	65	12·5	74	23
4	45	19	22	41	13	51	25
5	60	35	39	59	24	78	25
6	50	16	23	51	15	58	26
7	54	18	25	63	16	79	25
8	59	16	31	45	23	52	25
9	57	12	29	55	21	65	25
10	59	12	34	50	26	60	24

TABLE 3. ABRASION BASED ON THE ASH CONTENT OF THE USED OIL, LESS THE ASH CONTENT OF THE UNUSED OIL

The figures show the amount of ash, in milligrammes, in 1 gramme of oil.

Oil	Group drive		Back-axle drive 1		Back-axle drive 2	
	Worm 5	Worm 6	Worm 5	Worm 6	Worm 5	Worm 6
1	3·1	3·0	18·0	168	8·5	129·5
2	0·1	1·6	282·0	168·5	48·0	118
3	3·2	0	12·5	124·5	8·5	79·5
4	3·3	3·0	14·0	150·0	5·0	267
5	13·0	6·0	58·0	250·0	24·5	215
6	12·0	11·0	50·0	78·0	34·0	75
7	26·5	21	600·0	2,000	550·0	2,200
8	0·2	4(5)	0·2	50(24)	0	105(75)
9	4·4	3	26·0	13·5	12·0	31·5
10	0	2·0	4·0	6·0	1·2	7·0

TABLE 4. EVALUATION OF OILS, BASED ON THE DATA OF TABLES 2 AND 3

Oil 1	Owing to the low viscosity, the temperatures are low with no load. Under load, there are signs of lack of oiliness and abrasion begins.
Oil 2	The abrasion increases owing to lack of oiliness. The question of how far the viscosity index or the composition of oils 1 and 2 is concerned with the difference in abrasion found with the two oils is under investigation.
Oil 3	The oiliness again is insufficient and abrasion is again evident.
Oil 4	Abrasion appears as a result of corrosion.
Oil 5	Comparatively high abrasion as a result of lack of oiliness. The temperature rise on changing from no load to load is comparatively high. The extent to which corrosion favours abrasion is not clear.
Oil 6	The abrasion is of the same order as with oils 1 and 3. The added organic phosphorus compound has but slight influence.
Oil 7	Very high abrasion. Owing to deficient oiliness, corrosion appears to have considerable influence on the abrasion.
Oil 8	The high viscosity implies high temperatures under no load, but the abrasion lies within reasonable limits.
Oil 9	Very low abrasion because the oiliness is satisfactory, but the temperatures developed are still too high.
Oil 10	Also very low abrasion, no corrosion, and good oiliness despite the high temperatures.

TABLE 5. OBSERVATIONS OF TEMPERATURES AND ABRASION

Oil	Steady temperatures		Temperature difference	Amount of ash, in mg., in 100 gm. oil, after deduction of the ash content of the unused oil
	No load	Under load		
1	36	91	55	14·9
3	37	91	54	11·8
4	39	94	55	32·3*
5	46	100	54	9·4
8	52	98	46	9·3
10	52	99	47	28·0*

* These results do not compare well with those given in Table 3, and the tests are to be repeated.

oil, with a viscosity (Table 6) of about 2 deg. Engler at 100 deg. C., were chosen as base oils. The third oil was a synthetic product of similar viscosity. These oils were compounded with various fatty oils and tested. The results are given in Tables 7 and 8.

The abrasion was judged from the ash content of the oils. Researches extending over many years have shown that the estimation of the ash content of the oil enables a much closer evaluation of the abrasion than is possible with direct measurement of the running surfaces. The resin content of the unused oil was determined by the method of Suida and

TABLE 6. CHARACTERISTICS OF SELECTED OILS

Oil	Nature	Viscosity at		Vis-cosity index	Resin con-tent	Amount of ash, in mg., in 100 gm. oil	Specific gravity
		100 deg. C.	20 deg. C.				
	Paraffin-base oil—						
21	without addition .	2·03	47·5	103	3·9	0	0·895
22	with fatty oil A .	1·94	45·2	95	5·7	0	—
23	with fatty oil B .	1·98	44·8	97	5·2	0	—
24	with lead soap .	1·98	47·0	96	3·5	75·0	—
	Synthetic oil, containing bright stock prepared as oil 3—						
31	without addition .	2·10	73·0	83	2·6	0	0·858
32	with fatty oil A .	2·15	70·0	93	8·5	0	—
33	with fatty oil B .	2·12	67·0	89	7·5	0	—
	Naphthene-base oil(Edeleanu process)—						
41	without addition .	1·93	74·0	51	0·6	1·0	0·902
42	with fatty oil A .	1·92	69·0	62	3·0	1·0	—
43	with fatty oil B .	1·88	67·0	52	2·8	1·0	—

Pöll. The resin content of the used oil was determined, but is not quoted; it slowly increases during use.

In considering the results it must be noted that the temperatures, particularly under load, and the abrasion with different worms vary greatly under the same conditions of test. The explanation is to be found in the suitability and play of the bearing surfaces and in the assembly of the worm gear. The differences found are of an order which is greater than the differences which occur with one worm and very different oils.

The results show that the size of the molecule is of great importance, at any rate for pure hydrocarbons, as is clearly shown by the experiments with oils 1, 2, and 3. Further, the viscosity-temperature curve, expressed as the viscosity index, plays an important part, as is shown by the tests with oils 21, 31, and 41.

Abrasion is reduced by the addition of fatty oils to the paraffin base and synthetic oils, whereas the oiliness of the naphthene-base oil is too low and leads to destruction of the gear despite the addition of fatty oils. As the results obtained with the naphthene-base oil can only be explained in this way, further tests are being carried out to confirm them.

From the content of resin it is concluded that the refining of the oil requires great care. Resin is in no way troublesome, but, on the contrary, useful, as it assists lubrication. The supposition that resin plays a part in lubrication is now being examined.

TABLE 7. TEMPERATURES (DEG. C.) UNDER 73·5 PER CENT AVERAGE LOAD WITH WORM 6

A. No-load temperature less room temperature. B. Temperature rise under load.

Tests	Oil	Group drive		Back-axle drive 1		Back-axle drive 2		Room temperature
		A	B	A	B	A	B	
II, III	21	38 (36)	12 (18)	28 (16)	31 (38)	10 (7)	49 (49)	27 (24)
IV, V	22	33 (34)	19 (14)	13 (25)	41 (32)	3 (3)	53 (63)	27 (30)
VIII	23	36	13	27	44	6	62	28
VI	24	41	15	30	38	8	66	27
XIII	31	30	13	26	43	7	71	30
VII	32	40	14	30	39	6	61	28
IX	33	37	15	26	45	5	63	29
XIV, XX	41	36 (26)	17 (17)	31 (24)	47 (65)	8 (1)	80 (104)	30 (35)
XVI	42	34	18	27	55	6	77	31
XV	43	30	14	29	61	8	80	31
I	2	38	24	17	70	13	73	23
X, XVIII, XXI	4	41 (38) (32)	19 (13) (17)	18 (25) (20)	45 (39) (56)	11 (7) (3)	54 (57) (101)	23 (32) (34)
XI	9	55	9	37	40	21	65	29
XII, XIX, XXII	10	53 (45) (44)	14 (14) (17)	26 (27) (26)	63 (50) (78)	17 (7) (8)	59 (61) (96)	25 (33) (34)
XVII	5	42	13	29	47	14	64	29

TABLE 8. ABRASION, BASED ON THE DIFFERENCE OF THE ASH CONTENTS OF USED AND UNUSED OIL

(Amount of ash, in milligrammes, in 100 gm. oil).

Test	Oil	Group drive	Back-axle drive	
			I	II
II, III	21	6 (0)	83 (65)	164 (25)
IV, V	22	5 (0)	50 (38)	85 (215)
VIII	23	0	36	170
VI	24	−27	−20	+24
XIII	31	3	40	96
VII	32	6	23	64
IX	33	0	24	41
XIV, XX	41	4 (34)	85 (205)	135 (164)
XVI	42	6	150	75
XV	43	14	210	82
I	2	4	974	756
X, XVIII, XXI	4	8 (8) (32·8)	275 (148) (243·6)	239 (185) (185)
XI	9	—	—	—
XII, XIX, XXII	10	1 (4) (25)	77 (43) (232)	122 (24) (63)
XVII	5	12	144	195

The wormwheels were rendered useless by pitting after tests covering periods corresponding to 30,000–40,000 kilometres actual running. It is not clear how the metal particles are detached from the running surface. Further, it is not known whether the occurrence of pitting depends on the metal of the worm alone or on the lubricant used. Explanation of these points is urgently required.

Present tendencies favour extreme-pressure lubricants, which act by producing superficial corrosion. Though this superficial corrosion hinders the steady wear of the rubbing surface—the corrosion products are almost infusible—the danger in using such lubricants is that at the oil temperatures due to friction (over 100 deg. C. under load) the corrosion is accelerated to such an extent that the life of the parts is considerably reduced. For this reason, the aim should be to use oil, which is, first, of low viscosity, in order that the temperature remains low under no load and that the lubricating value of the oil, which depends on temperature, is not impaired, and, second, which contains oily substances that do not cause corrosion, but form a film which bears the load. It will be all the easier to find such oils if the maker takes the properties of the lubricant into consideration in the design of his machines.

THE INFLUENCE OF CATALYSTS ON THE OXIDATION OF OILS

By T. K. Hanson, B.A., and Professor A. C. Egerton, F.R.S.*

The gumming of lubricating oils is one of the factors which limit the period between overhauls of aero-engines, while sludging gives rise to blocking of scraper ring holes and other troubles. The gumming and sludging characteristics of oils have to be tested, yet in spite of various methods devised to gauge such characteristics, behaviour in an engine may often not coincide with the results of such tests. The Air Ministry oxidation test is intended to provide a measure of the oxidizability of oils and does so with considerable success, for an oil which passes the specification usually gives satisfaction; but oils may fail in the oxidation test and yet behave well in an engine, while satisfactory oils may fail under severe conditions in an engine.

When an oil is submitted to oxidation, nothing appreciable happens for some time and then oxidation rapidly sets in. It is the same with oxidation of gaseous hydrocarbons: there is first an induction period when the gaseous mixture is preparing to react and then reaction suddenly sets in, perhaps with sufficient rapidity to cause ignition of the mixture. During the induction period small quantities of substances are formed which, by giving rise to branched reaction chains, according to the Semenoff theory of the induction period, reach a climax exponentially but may take considerable time to reach this state of rapid reaction. This is also probably what happens in the oxidation of a lubricating oil, the most easily oxidizable constituents forming the nuclei which increase exponentially. However, it is more complicated than this, for the oxidation is an auto-oxidation, the products of reaction influencing the primary reaction. The effect of inhibitors such as β-naphthylamine and of promoters such as powdered iron (or oxide of iron) is in accord with such hypotheses as to the nature of the oxidation.

As a lubricating oil is a complicated mixture of hydrocarbons one of which influences the oxidation of the other, its oxidation would be expected to be very easily influenced by small changes in composition or conditions and no doubt this is so. But the facts that measurements of the length of the induction period and the rate of oxidation are generally fairly repeatable in oxidations carried out on the same

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* Department of Chemical Technology, Imperial College of Science and Technology, London.

oil under specified conditions, show that the oxidation is more regular than might be expected. In an engine, however, behaviour is not so consistent as in a slow oxidation test; an oil which behaves well in one engine may not do so in another engine. This might be expected if conditions are such that the oil is brought to a state when the induction period is about to terminate: it would behave satisfactorily if still within the region of very slow oxidation, but would rapidly gum if the oxidation reaction sets in. Experience shows that when gumming starts, it rapidly increases; but this is no doubt due not only to the causes indicated, but partly to the gumming setting up ring sticking which permits the flame to have more effect on the oil round the rings. In fact, the influence of the combustion of the fuel on the rate of gumming has often been noted; thus if knocking occurs gumming sets in more rapidly.

This being so, and as considerable quantities of nitrogen oxides were found to be formed in the cylinder of an internal combustion engine (Egerton, Smith and Ubbelohde 1935), particularly under conditions of knock, it was of interest to find to what extent the presence of oxides of nitrogen influences the oxidation and therefore the gumming of the oil. In view of the catalytic influence of small quantities of nitric oxide on the gums formed from butadiene and other compounds in coal gas (Hollings 1936) and of the influence of nitrogen peroxide on the oxidation of hydrocarbon vapours (Bibb and Lucas 1929, Kane and Townend 1937), a considerable effect of oxides of nitrogen might be expected. This has been found to be true and the paper describes certain preliminary experiments which have been made and are being extended. The results have a bearing not only on the composition of the lubricating oil itself, and on the means of testing its behaviour, but also on the characteristics of the fuel and of the engine, for in the combustion of fuels different amounts of oxides of nitrogen may be formed; the question of the effect of nitrogen oxides on corrosion wear is also involved.

Further experiments are being made on the amount of nitrogen oxides formed at different stages of the engine cycle by the methods previously described and it has been found that in the compression-ignition engine the amounts of both nitric oxide and nitrogen peroxide rise to a maximum of about 6×10^{-4} mol fraction and then diminish, but that in the petrol engine the nitric oxide increases right to the end of the stroke; the results of these experiments on the amounts of nitrogen oxides formed are being published elsewhere. The quantities of nitrogen oxides obtained increase very considerably under conditions of knock, and they are also influenced by the lubricating oil, the cylinder jacket temperature, etc.

Effect of Nitrogen Oxides on the Oxidation of Lubricating Oils. Preliminary experiments on the effect of oxides of nitrogen on the formation of gum by the oxidation of oils have been made in a modified form of the Air Ministry oxidation test (D.T.D. 109). 20 cu.cm. of oil held at 218 deg. C. in a naphthalene bath were blown with air in a glass test tube (6 inches × $\frac{7}{8}$ inch) at the rate of 15 litres per hour, and nitrogen peroxide was added by various methods to the air stream. The concentration of nitrogen peroxide obtained was about two parts in 10,000 parts of air, and was slightly less than the maximum concentration of nitrogen peroxide found in the cylinder gases. A concentration of one part of nitric oxide in 1,000 parts of air was used. At this concentration the rate of reaction $2NO + O_2 \rightarrow 2NO_2$ is slow, and the experiments indicate that the NO must first react to give NO_2 before it can affect the rate of absorption of oxygen by the oil. Sometimes with mineral oils nitric oxide had an accelerating effect, in other cases no effect was observed, e.g. with the oil containing much castor oil. Some results are shown in Table 1 which gives the "oxidation number," i.e. the viscosity after blowing divided by the viscosity before blowing. The time of oxidation was three hours and the viscosities were measured at 100 deg. F.

The nitrogen oxides act catalytically, for although they are absorbed by the oils, their effect on the absorption of oxygen is out of all proportion to their concentration. The presence of nitrogen peroxide tends to remove the induction period, reaction starting much sooner than in its absence.

Other simple inorganic gases, such as chlorine or hydrochloric acid, which might be present in traces in the combustion chamber of an engine, do not similarly affect the rate of oxidation of the oils. But it is probable that the nitrogen oxides in the cylinder gases may affect in a secondary manner the behaviour of sulphur dioxide and this point is being investigated.

The effect of some inhibitors on the catalysis of oxidation by the nitrogen peroxide has also been investigated. Inhibitors can act either (*a*) by stopping or reducing the catalytic action of nitrogen peroxide, or (*b*) by absorbing the nitrogen peroxide and so forming a compound which has no catalytic effect. The effect of some such substances is shown in Table 2.

These results indicate that nitrogen peroxide has considerable influence on the initial stage of the oxidation. Some tests were made in a standard D.T.D. 109 apparatus at 220 deg. C. to investigate the effect of nitrogen peroxide (mol fraction 2×10^{-4}) on the later stages of the oxidation of mineral oil. The results are shown in Fig. 1. The air containing the nitrogen peroxide causes the oil to deteriorate

TABLE 1. OXIDATION NUMBERS OF VARIOUS OILS

Oil	Treatment	Oxidation number (measured at 100 deg. F.)
Paraffin-base mineral oil	Air only	1·52
„	Air + NO	1·68
„	Air + NO_2	1·77
Oil containing large proportions of castor oil	Air only	2·83
„	Air + NO	2·83
„	Air + NO_2	20·4
„	Air + 2 per cent NO_2 (½ hour)	solid at at 100 deg. F.
„	Air + HCl	2·83
„	Air + Cl_2	2·84
Oil E	Air only	1·30
„	Air + NO	1·29
„	Air + NO_2	1·55
Oil A	Air only	1·26
„	Air + NO	1·40
„	Air + NO_2	1·50
Oil B	Air only	1·31
„	Air + NO_2	1·44
Oil C (compounded)	Air only	1·26
„	Air + NO_2	1·47
Oil I (compounded)	Air only	1·05
„	Air + NO_2	1·28
Oil I (mineral)	Air only	1·00
„	Air + NO_2	1·03
Oil G (mineral)	Air only	1·08
„	Air + NO_2	1·33
Oil G (compounded)	Air only	1·01
„	Air + NO_2	1·11

TABLE 2. EFFECT OF INHIBITORS ON OXIDATION BY NITROGEN PEROXIDE

Oil	Treatment	Oxidation number
Oil containing large proportions of castor oil	Air + NO_2	20·4
Ditto + diphenylurea	Air + NO_2	3·20
Ditto + Fe_2O_3	Air + NO_2	2·50

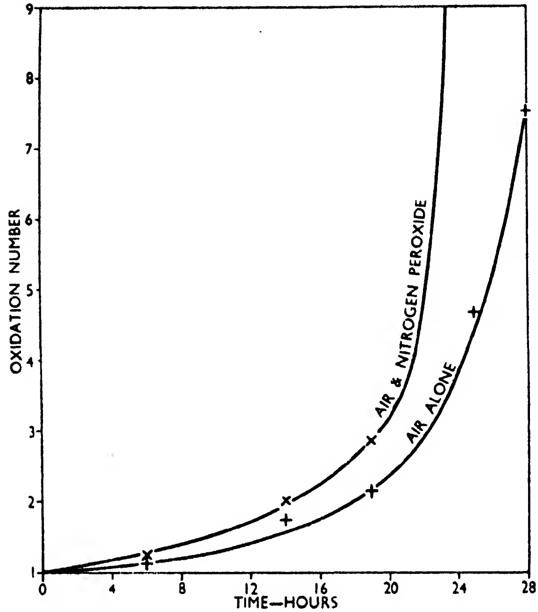

Fig. 1. The Effect of Nitrogen Peroxide on the Later Stages of the Oxidation of Mineral Oil

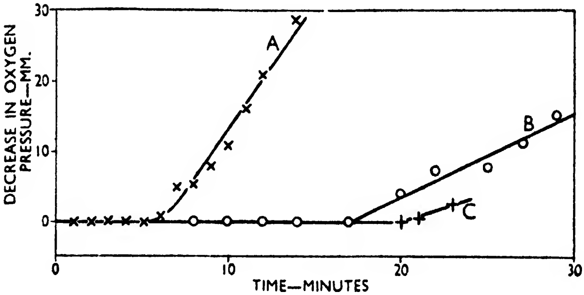

Fig. 2. The Effect of Nitrogen Peroxide on the Rate of Absorption of Oxygen by Castor Oil at 218 deg. C.

and the effect is very marked after 20 hours in comparison with the oil blown with air. Thus nitrogen peroxide not only shortens the induction period but also increases the rate of oxidation.

In order further to determine the influence of nitrogen peroxide on the oxidation of oils, the rate of absorption of oxygen by vegetable oils has been investigated. Fig. 2 shows the effect of nitrogen peroxide (curve A) on the rate of absorption of oxygen by castor oil. The shortening of the induction period is quite clear and the slope of the curve shows an increase in the rate of oxidation, as compared with the oil in contact with oxygen alone (curve B). The effect of a small quantity of inhibitor (0·1 per cent diphenylurea) in delaying the oxidation is shown by curve C.

Both in the blowing tests and in the oxygen absorption experiments, when nitrogen peroxide is present a thick gum always separates at the surface of the liquid. The oil darkens much more rapidly in presence of nitrogen peroxide even when the viscosity is very little changed. The separation of gum on the glass tubes in which an oxidation test is carried out is a disadvantage because such substances are detrimental but may not be included in the effect on the viscosity of oil. Different results would be expected in metal tubes, particularly iron tubes, as iron oxide is an inhibitor and iron itself may accelerate the reactions. In an engine, the surface of oil exposed to oxidation is mainly an oil-to-gas surface and the time of contact is short; these points are being further investigated. It is noteworthy in connexion with the absorption of nitrogen peroxide by the oil that the carbon deposits in an engine show the presence of nitrogen.

These experiments prove that the rate of absorption of oxygen by oils at 220 deg. C. is greatly influenced by the presence of nitrogen peroxide in the concentrations found in the engine cylinder, and as different oils behave differently in this respect, this fact must be taken into account when ascertaining the suitability of an oil. Since the effect is to shorten the induction period, the presence of nitrogen peroxide may help to explain the rapid deterioration of the oil when once gumming starts. It also explains the increased tendency to gum when knocking occurs, and indicates why engine tests on oils give very different results under different conditions and why the results do not always accord with the oxidation characteristics of the oil obtained in tests where no account is taken of the catalytic effect of nitrogen oxides.

Further experiments are in hand to determine the effect of nitrogen oxides on the oils in an engine cylinder; and to study the relation of the nitrogen oxides to cylinder wear, the oxidation reactions which are catalysed by the nitrogen peroxide, and the effect of substances which tend either to remove the nitrogen peroxide or inhibit its action.

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TESTING OF BALL BEARING GREASES

By B. Kjerrman, Dr.-Ing.*

Ball bearing greases are of three principal kinds:—

(*a*) Lime-soap grease.
(*b*) Alkali grease.
(*c*) Lime-alkali grease.

Lime-soap grease is the cheapest and most generally used; its use, however, is limited to temperatures below 45 deg. C. It is therefore not wise to use this grease where there are doubts about the maximum temperature which might be attained, particularly where expensive bearings or machines are concerned. The danger of hot running is much greater when using lime-soap grease than when either of the other two kinds is used. When melted, lime grease decomposes into its constituents. Even at temperatures much below the melting point the grease tends to disintegrate and this is progressive as the temperature rises, because the water, essential to retain the grease characteristic of the lubricant, escapes by evaporation. In lime greases the soap is thought to form cells, like those in beeswax, in which the oil is trapped. If the water disappears from the walls of these cells, they break down, letting the oil out, and the remainder of the cells become brittle. When this takes place in a ball bearing, the separated oil escapes from the working surfaces, leaving behind a more or less hard soap which possesses no lubricating value but rather the reverse. In consequence, the temperature of the bearing will rise rapidly due to the lapping effect of the soap which gradually carbonizes, causing complete collapse of the bearing.

The chief advantage of lime-soap grease is its low cohesion. This is why the bearing tends to run at a lower temperature, as the work to separate the grease itself, when attached to different surfaces which continually have to make and break contact, is smaller the lower the cohesion. Owing to the rather small adhesion, however, lime greases are fated to lubricate insufficiently, the grease being easily extruded from the lubricated surfaces.

Alkali grease can, compared with lime-soap grease, be used within a wider range of temperatures. The ranges are also larger or smaller between much wider limits of the extreme temperatures than for lime soap greases. This depends upon the fact that the compositions of alkali greases vary considerably, some being purely sodium soaps, others containing more or less potassium, of which the greater the

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* Aktiebolaget Svenska Kullagerfabriken (SKF), Sweden.

content the weaker the grease. If an alkali grease melts, it usually recovers after cooling.

The main advantage of alkali grease is the wide range of temperatures in which it may be used, but its better adhesion is also essential as well as its property of recovering after cooling. If a bearing is running hot because the bearing and housing have been filled up with too much grease, it may liberate itself if alkali grease is used. Due to running hot, part of the grease will leave the housing, thereby reducing the quantity available to the bearing. This causes the temperature to fall, and the grease left will be restored. The danger of running hot because of too much grease is greater the higher the cohesion. In general this property is higher for alkali grease than for lime-soap grease. The adhesion of alkali grease is in general high, which is all to the good provided the grease is not too abundant.

Lime-alkali grease is a mixture of the two kinds of greases mentioned above. It might be possible to pick out the best properties of both kinds and make them useful in this mixed grease. The ideal lime-alkali grease should have a low cohesion, because of the lime soap, and a high adhesion, because of the alkali soap. The drop point will be higher than for lime-soap greases, thus allowing a wider range of temperatures in which it may be used.

The use of lime-alkali grease is, however, comparatively rare. Practical experience is therefore not yet sufficient to allow of proper appreciation.

The chief laboratory tests differ according to whether a lime-soap or an alkali grease is to be examined. Most of the tests are, however, common to both. These are: the determination of the starting drop point and the drop point, the content of ash and chlorine, the acidity and alkalinity and, most important of all, the behaviour of the grease in bearings. Lime-soap grease is sometimes examined for its water content, the alkali grease more rarely and only when its appearance makes a test necessary.

Lime-soap greases may be tested for consistency by a penetration test. A loaded and guided cylindrical steel rod is allowed to sink in the grease which is held at different temperatures. The rapidity and depth of penetration is measured. The tests may be performed on the same grease also after it has been worked or kneaded in a special machine for different periods. The American Society for Testing Materials specifies the use of a different apparatus. Although this test can be applied to sodium-soap greases, warning is required against the practice of judging the results on the basis of lime-soap grease estimations. In the author's opinion the best alkali grease is, when delivered, sufficiently stiff to give a good seal. Working, however, makes it very weak, which is good when carried out in the bearing itself, giving good lubrication. It has, however, happened that this extremely good grease has been rejected by purchasers because its weakness has been determined by the penetration test in a worked condition.

There are many other determinations, which are neither necessary nor useful. Some of these may be mentioned:—

> For lime-soap grease the contents of lime, fat, fatty acids, free fatty acids, glycerides, and combined and free lime are determined.
>
> For alkali grease the corresponding determinations are made, as well as that of the total content of potassium and sodium, etc.

In determining the moisture absorption of ball bearing greases, it is presumed that the absorption shall be as low as possible. This may be true for some greases and for some purposes. It is, however, not to be accepted generally. A good alkali grease may absorb a very large quantity of water in a moist atmosphere and quickly lose it when transferred to a dry atmosphere. It may be very useful that the grease can absorb moisture, forming against rust a protective emulsion, which certainly cannot be regarded as a drawback. A lime-soap grease that absorbs water in a moist atmosphere does not lose its water content nearly so rapidly as an alkali grease after having been transferred to a dry atmosphere. As the lime-soap grease does not form a protective emulsion against rust, the water content, if too high, may be dangerous. One therefore must protest against this moisture absorption test, if a good grease generally is required to absorb no water, or very little.

It is very useful to examine the general appearance of the grease, which should be quite uniform and free from harder and weaker parts, i.e. the consistency must be the same throughout. To obtain this consistency the greases are worked in special machines after having solidified. It is, however, still more important that the grease shall be free from foreign particles such as iron chips, sand, or dust. To ascertain this, the grease is dissolved in a solvent, filtered, and the foreign particles on the filter given as a percentage of the total grease in the dry state. The precipitate is ignited in order to find, by difference, the content of organic matter, which certainly is not so dangerous as the inorganic matter. Next the residue is examined and may be analysed for iron, silica, etc. Because of the ignition, the iron is oxidized to iron oxide, so that the precipitate must be examined before ignition, using a magnet or a microscope, if iron chips are present. The foreign matter, especially the steel chips, is derived partly from the oils through rusty barrels or containers, and partly from the apparatus used for stirring the grease during manufacture. These foreign substances, especially steel chips and silica, are extremely dangerous for ball and roller bearings, causing dents in the track, which later on become flakings. For greases for ball and roller bearings the content of foreign matter is certainly one of the most important estimations.

Among the most important determinations the running test or the behaviour of the grease in the bearing takes first place. It is, however, also very important to determine the starting drop point and the drop point. In some greases these two points are very near to each other. Then it is not so necessary to determine the starting drop point, but, when the points are further apart, the first point is the more important and must be determined. It is also very important to determine the chlorine content. The author's experience has shown that the chlorine content must be kept very low, if the grease is not to be a source of danger because of rust formation. This point is often neglected, because even if a grease contains a dangerous amount of chlorine it may nevertheless be useful for many purposes. If, however, the grease has to work under arduous conditions, and especially if the bearing is subjected to large variations in temperature, grease containing much chlorine will rapidly corrode steel. The author therefore maintains the chlorine content at or below 0·001 per cent, calculated on the grease itself. It is generally recognized that the most important test is the investigation of the behaviour of the grease in bearings. The test is, however, not standardized. The author's solution of the problem is described below.

The machine used for testing the life of bearings has a shaft with two test bearings, supported in two plummer blocks and in two overhanging, auxiliary loading bearings on the ends. The bottoms of the plummer blocks contain electrical heating units, which heat the housings to the desired temperature. In this way the grease can be tested at temperatures higher than those obtainable through the working of the bearing itself. To test a grease the bearing is filled, but only the lower part of the housing; no seals are used, thus giving the grease free exit if it melts or if it has such a large adhesion that it may creep out. With this arrangement it is possible to test not only the lubricating quality of the grease but also its sealing properties. If the grease has the expected properties it must remain in the housing just as it was placed before the start of the test and form a free path for the moving parts. If it begins to rotate and slip out of the bearing, the adhesion is too high. These requirements are also related to certain temperatures which the grease may reach.

A general examination cannot be described, as the test has to be directed in accordance with requirements. In general, however, a long run is required if the capacity for lubrication is tested. Using a bearing on a 60 mm. shaft 53 million revolutions are run, which corresponds to the number of revolutions of a railway bearing having run for 93,400 miles. After this time the lubrication must still be good and the bearing not worn, i.e. the rollers must not have lost their shape, and there must be no pathways marked on the rings.

The above-mentioned tests do not include any examination of the special compounds (graphite, zinc oxide, etc.) mixed in the grease. If such additions are made they must be controlled by obvious methods. It must be pointed out, however, that, as far as it has been possible to find, no advantages are given by special additions. On the contrary it is always dangerous to add solid compounds to a grease, as they may cause dents and therefore scaling in the bearings. It has not been found possible to use at high temperatures greases containing graphite. Further, the presence of zinc oxide in the grease has not given any better resistance against corrosion.

To summarize, for all kinds of ball bearing greases it is necessary to know the starting drop point and the drop point, the ash content, and the chlorine content, calculated on the grease itself, also the amount and kind of foreign particles. Running tests should be made under conditions corresponding as closely as possible to the working conditions intended for the grease. For lime-soap greases a penetration test may be used. The alkali greases may be examined for their power to form an emulsion which protects against rust.

THE PROBLEM OF "OILINESS"

By S. Kyropoulos, Ph.D.*

The confusing situation of the "oiliness" problem ten years ago and the consideration by various authors of "oiliness" effects entering into friction experiments, where hydrodynamic lubrication should exclusively prevail, led to the hypothesis of viscosity drop by molecular flow orientation (Kyropoulos 1928, 1929). This hypothesis was confirmed later by viscosity measurements by means of a machine designed by Prandtl (Kyropoulos 1932), with which such measurements can be made under bearing operating conditions. Measurements on various motor oils (SAE 30–60) under varying shearing stresses demonstrated that the optical anisotropy created by flow is accompanied by a decrease in viscosity of 5–17 per cent. This effect, small though it is, compared with the effect of temperature on viscosity, is real, as a computation of heat transfer in the apparatus also shows and certainly accounts for discrepancies between measurement and theory. The same measurements showed also that hydrodynamic lubrication obtains at least down to film thicknesses of about 2×10^{-4} cm., while the thickness of a monomolecular layer of oil will be about of the order of 10^{-6} cm.

These measurements were not supposed to penetrate to the region of boundary lubrication (Hardy 1928), but were intended rather to extend and supplement our knowledge of hydrodynamic lubrication. The hypothesis of the viscosity effect of flow orientation was the result of considerations appertaining to molecular physics, and of the often unsatisfactory introduction of physico-chemical hypotheses *ad hoc* for the explanation of lubrication phenomena and an attempt, *first*, to push the hydrodynamic theory to its limits.

The great difficulties which hinder studies in the region where the application of the hydrodynamic theory becomes questionable are illustrated best by a review of the literature and the criticism which all work on boundary lubrication arouses sooner or later, criticism which goes sometimes even so far as to deny all practical importance to boundary lubrication or "oiliness" problems. There is a natural reason for this. The problem is by nature a physical one and as soon as molecular dimensions are entered, most of our methods for testing properties of films, which are, ultimately, "mechanical" properties, tend to fail, or their results tend to be influenced by irregularities of the surfaces and by impurities which are negligible in the measurement of

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* California Institute of Technology, Pasadena, California.

bulk properties or in the continuous state on which the methods were originally based. Moreover, in many experiments made on thin film lubrication with commercial oils under conditions simulating practical bearing operation, we are faced with physico-chemical complications which tend to modify the materials handled. To avoid all these difficulties engineers had recourse to machines supposed to test "film strength" directly by increasing the load on a model bearing operated with different oils until *seizure* occurred. The information obtained with such machines is obviously incomplete because the results obtained with various types of machines on the same series of oils do not always agree closely, which fact, among others, may be due to the rather extreme nature of any seizure test.

From the point of view of method, this failure of the seizure test means that the question it was supposed to answer was not put correctly. The mistake is twofold. On the one hand it consists of the violation of the old scientific rule, never to use an effect, the mechanism of which is unknown (seizure), as a start for an investigation with the object of obtaining general and fundamental information. On the other hand, it consists in adhering—consciously or unconsciously—to a notion ("oiliness") which, from its very introduction, had to be considered as an exclusion principle which *ex*cludes all hydrodynamic properties and *in*cludes all unknown, and could not be treated like a well-defined physico-chemical property of matter.

Thus, any attempt at a general approach to the "oiliness" problem should start with the lubrication problem proper, *both* on mechanical *and* physico-chemical lines and the limitations of its treatment as a problem of the mechanics of the continuous state of liquids. To recognize these limitations we must avail ourselves of our present knowledge of the liquid state. Our conception of this state is entirely different from what it was a few years ago. The mechanics of liquids are adequate for gaining a deeper insight into lubrication problems so long as the liquids may be regarded as being in a state of molecular disorder. This treatment is, however, at most descriptive when the liquid presents a "structure" or a state of order. Flow orientation is an example to this point. While internal friction is a statistical phenomenon in a state of disorder, phenomenologically satisfactorily described by the coefficient of viscosity, and adequately accounting for the phenomena of hydrodynamic lubrication, there is no close analogy to this in the cases of extreme order as presented by the problems of boundary lubrication, in which we are dealing with forces much more analogous to the lattice forces in crystals.

A comparison of the coefficients of viscous and static friction demonstrates this. The former is a constant of the liquid, depending closely on temperature, because with increasing temperature the distances between the molecules increase and the molecular forces operating between the molecules decrease with high powers of the distance. Moreover, in the liquid state the internal mobility of each molecule increases considerably with temperature. Both effects strongly favour molecular disorder. In the case of extreme order—the adsorbed state of the boundary layer (static friction)—the increase of the distance between the molecules with rising temperature is limited by the thermal expansion of the adsorbing surfaces. For the same reason the internal mobility of the molecules is limited. There is no translatory movement of individual molecules or groups of molecules as in the liquid state, and there will be at most the possibility of an interchange of anchoring places on the surface between the adsorbed molecules, similar to the interchange of places in the lattices of mixed crystals. The orientated state on a surface is, in general, a state of minimum potential energy for each temperature and there are fewer possibilities of similar energies than in the liquid state. In contradistinction to the comparative freedom of the molecules or bundles of molecules in the liquid state, their firm attachment to a solid surface has the result that this surface influences the coefficient of static friction. As will be seen later, to understand this influence does not necessitate any special assumptions as to the range of action of the surfaces. The determining factor of this influence is the atomic spacing of the surface. As a second factor which might influence the running properties of the surface material we have to consider the ease of slip of this material and how this slip is influenced by the lubricant. Such an influence is well known in the process of cold-drawing and the use of graphite as a lubricant is an extreme case of imparting special running properties to a bearing by the use of a surface material which slips extremely easily.

Summarizing: while fluid lubrication still lends itself to the conventional mechanical and phenomenological treatment resulting in an "understanding" of the phenomena sufficient for technical applications, this treatment becomes insufficient when the effect of flow orientation is considered. The awkward term "oiliness" is apt to mislead engineers into the belief that it denotes something which must ultimately be a tangible physical property. As a matter of fact it is only an exclusion principle, comprising all the lubrication phenomena which do not enter into the hydrodynamic theory and are not secondary mechanical effects. The origin of the "oiliness" phenomena is the discontinuous atomic and molecular structure of matter. This fact and their occurrence in dimensions of molecular order, where states of order and boundary phenomena prevail over states of disorder and bulk phenomena, call for a molecular-physical treatment of all these problems along with the conventional phenomenology. "Oiliness" phenomena are due to molecular interactions which constitute an intermediate type between classical "mechanical" and chemical interaction. They occur between molecular aggregates of an intermediate type between the liquid and the crystalline state and must be treated fundamentally as problems of molecular structure, molecular forces, and crystallography.

For the discussion of the fundamentals reference is made to earlier papers and the literature quoted there (Kyropoulos 1936). It will be useful, however, to illustrate the previous general section by a few examples.

Flow Orientation. In order to understand the viscosity drop owing to flow orientation we consider the liquid state. The more anisotropic in shape molecules are (e.g. normal paraffin chains), the more are favoured, in the liquid, states of order constituted of bundles of molecules in an arrangement of minimum potential energy. The cohesion between neighbouring molecules is proportional to their "area of contact", i.e. their length. A picture very close to reality may be this: from the central part of such a bundle, where the molecules may or may not be arranged at the exact distances of the crystalline state, this molecular distance becomes gradually larger until it becomes so large owing to the oscillatory motion of the molecules that molecules occasionally leave and again stick to the "complex". Thus while preserving the *general* order of the crystal, the bundle is distinguished from the crystal by its indefinite lattice spacing. There is evidence that similar bundles or "groups" of molecules also persist in the molten state of certain metals (e.g. bismuth) (Goetz 1934). Now, under all circumstances, the "heads" of the molecules which form a bundle are much farther apart from their neighbours than the carbon atoms of an individual chain, the distances of which, moreover, remain nearly constant. Thus, the residual fields of force of the heads—which are already larger for another reason—are still larger because they are less saturated by neighbours than the residual fields perpendicular to the chain. An oil composed of such oblong molecules and their bundles is easily orientated by flow because it is, under these circumstances, in a state of minimum potential energy. For, in the flow-orientated state, many more "chain-atoms" will be exposed to mutual interaction with their *smaller* molecular forces than "head-atoms", as compared with the state of random orientation. It can readily be seen from this consideration of the molecular forces in conjunction with molecular structure, shape, and distance, how identical atoms—carbon atoms—may bring about different frictional effects.

The Adsorbed State. These simple considerations can be immediately applied to the adsorbed state. While it was long doubted that the normal paraffins orientate themselves on adsorbing surfaces, it was shown later that well-defined orientation with the chain axes perpendicular to the surface follows from Hardy's friction experiments, as should also be expected on account of their shape (Kyropoulos 1933). Still later this was also demonstrated by means of electron diffraction by Trillat and Motz (1935). Taking mobility as a criterion, the adsorbed state is a solid state, a solid fixation of the bundles of molecules of the liquid on a surface. However, a consideration of the molecular forces involved immediately shows that, supposing a single crystal to be the surface, the distances between the molecules will be *definite* and *fixed* in comparison with the liquid state and determined both by their mutual fields of force and the force distribution in the surface. There is no reason why the adsorbed, orientated "primary" film, should have the spacing of its matter in the crystalline state. As a matter of fact, Hardy and Nottage (1928) showed that a long-chain compound crystallized between two metal blocks forms a primary film on each metal surface and crystallizes between the films in a form which is certainly not a continuation of the primary film. The same was shown by Trillat (1926) by means of X-ray patterns. This means that the primary film does *not* inoculate the molten mass of the long-chain compound above it and has, consequently, a different molecular spacing. The experiment of Hardy quoted above shows, moreover, that this spacing must be different on different metals because its mechanical properties are different. This result is very important, because it shows conclusively that the adsorbed state is not a fixation of the *crystalline* state of the matter on the adsorbing surface, which means that it is *not* the lattice forces of the long-chain compound (in its crystalline state) which determine the lateral packing of the molecules, but, ultimately, the lattice forces of the adsorbing surface. It is known that the mechanical properties of the materials are functions of these forces and of the atomic distances in the lattice and it is known how they vary. Consequently, there is no reason why the adsorbed molecules should have the same density of packing on any surface, as is, in general, implicitly assumed. Applying the same considerations as in the case of flow orientation, it can be seen why *identical* "free" ends of molecules must have varying residual fields of force. This means a coefficient of static friction, varying with the surface through the variation of molecular spacing or *one* of the factors constituting the running properties of the surfaces. Thus, correlating molecular-physical experience with engineering experience, there is no need for the explanation of contradictory assumptions like action at a distance through the adsorbed molecules, which action, as is known (e.g. from spectroscopic data), does not go beyond at most three atoms in the chain.

The decrease of the frictional coefficient with increasing chain length is, in general, correlated with the mechanical properties of the molecules, elasticity, etc. From the present point of view this increase in "elasticity" with increasing chain length should be interconnected with the increase of internal molecular mobility. The carbon atoms of the zig-zag chain molecules are free to rotate around the carbon bonds under preservation of the valency angle. The longer the chain, the greater is the number of such "joints", and with it the number of positions of similar potential energy between neighbour molecules which should result in a greater mobility of the "free" ends. It is this greater freedom of lateral movement which, generally, makes the difference in behaviour of the adsorbed molecules as compared with their natural lattices.

The same general considerations that apply to the non-polar paraffin molecules, apply also to polar molecules, like fatty acids, etc. These contain a polar head (carboxyl group, etc.) the size of which is also a factor determining the spacing of the adsorption lattice. In mixtures of such polar compounds the question may arise whether one of the components will be preferentially adsorbed to constitute the boundary layer. Any preferential adsorption will *inter alia* depend on the closest approach to the surface possible. The question can be decided experimentally by means of a simple apparatus which also permits the measurement of the static coefficient of friction free from stray effects (Kyropoulos and Shobert 1936, 1937). The case of mixtures is the most important in practice and justifies a brief discussion.

One type of mixture is mineral oil with a polar addition which forms the boundary layer, a behaviour characteristic of polar additions. Nevertheless, the presence of a polar group in a molecule is *insufficient* to bring about this effect. There is still another condition to be fulfilled, namely, that the polar group should be "accessible," that it can approach the surface sufficiently. There is a simple criterion for this accessibility which can always be measured and sometimes predicted from molecular structure. The criterion is whether the particular polar molecules form complexes by *dipole* association in the liquid state or not.

The second and most common type of mixture comprises the commercial lubricating oils. The components are not so different in this case with regard to their molecular forces that there should be *one* composition of the adsorption layer of outstandingly low potential energy, as in the case of polar additions. Such a boundary layer, though fairly definitely orientated, should be expected to vary continually slightly in composition. It is known that the *longest* molecules will be preferentially adsorbed, that they should impart to the mixture the best frictional characteristics, and that their concentration should be *as high as possible*. But our ordinary methods fail as criteria of molecular *length*. The only methods which provide measurements of molecular anisotropy *and* concentration of the longest molecules are measurements of the Kerr constant and of flow-orientation. Of these methods, the mechanical one (Kyropoulos 1932) is the only one directly applicable to coloured oils. The lubrication engineer accustomed to the hydrodynamic treatment of his problems may see with satisfaction a hydrodynamic method contribute thus to the solution of the "oiliness" problem in the most frequent practical applications and, incidentally, notice that most representations of the matter in literature refer to mixtures with polar additions which are, for the time being, rather the exceptions.

This general survey may show that the methodically sound way to approach the technical problem of "oiliness" from the point of view of friction, and put the correct questions and obtain unambiguous answers, is to study the physico-chemical properties of the oils concerned, and the molecular forces involved, on a broader basis and to apply the available knowledge from these fields of research in constructing working hypotheses and planning experiments.

"Oiliness" research, as it had developed from engineering work during the last fifteen years, showed sometimes a tendency to develop a physical chemistry of its own, disregarding well-established experience and theories of physical chemistry. The attempt to base "oiliness" research on the fundamental phenomena and to connect pure science with engineering practice may involve some sacrifice of originality. However, without restricting intuition it promotes clarity and has demonstrated time and again its methodical soundness in other fields of engineering research, such as mechanical and electrical engineering, metallurgy, and electrochemistry.

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THE OXIDATION BEHAVIOUR OF INTERNAL COMBUSTION ENGINE LUBRICANTS

By E. W. J. Mardles, D.Sc., and J. E. Ramsbottom, D.Sc., Ph.D.*

Lubricating oils in the internal combustion engine are subjected over a wide range of temperature to oxidizing conditions by aeration. Empirical oxidation tests, devised for specification purposes, give reliable results, provided the oils under examination belong to the same class. Under some conditions of test it is possible to obtain oxidation results which do not appear to agree with the engine behaviour (1930)†; this has given rise to the view that there is only one satisfactory test for an oil and that is to try it in the engine for which it is intended. Engine tests are expensive and for many reasons laboratory oxidation tests are welcome if they can indicate the possible engine behaviour of an oil and discriminate between oils of good and bad quality. Although the rise in viscosity, the development of acidity and sludge, etc., attend to a greater or less extent the oxidation of all oils, each oil has its own oxidation behaviour, the oxygen consumption being the same or similar. The oxidation behaviour of the hydrocarbons of high molecular weight, found in commercial lubricating oils, is indeed complex and no single laboratory oxidation test appears to be adequate in deciding entirely the merits or demerits of any particular oil.

Apparatus and Methods. The apparatus and methods employed were as follows: In method (*a*) the oil was incubated in open dishes and was examined from day to day for acidity, sludge formation, etc. In method (*b*) a stream of air or oxygen was bubbled through a column of the heated oil in a tube. In method (*c*) a quantity of oil was heated isothermally on the walls of a rotating bulb and the oxidation changes were noted by chemical analysis and manometry, whilst in method (*d*) the vapour or spray of the oil was passed with air through a heated tube and the products examined.

The work has been divided into three sections, namely, oxidations (1) below 125 deg. C., (2) between 125 deg. and 225 deg. C., and (3) above 225 deg. C.

I. OXIDATIONS BELOW 125 DEG. C.

These tests were carried out mostly in open dishes over periods up to 500 or 1,000 hours. Some results obtained with oils aged in dishes

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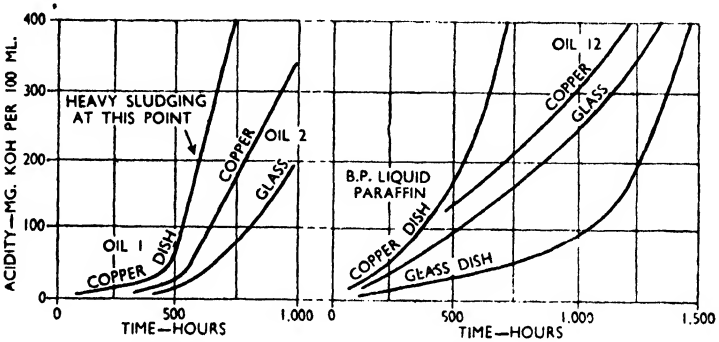
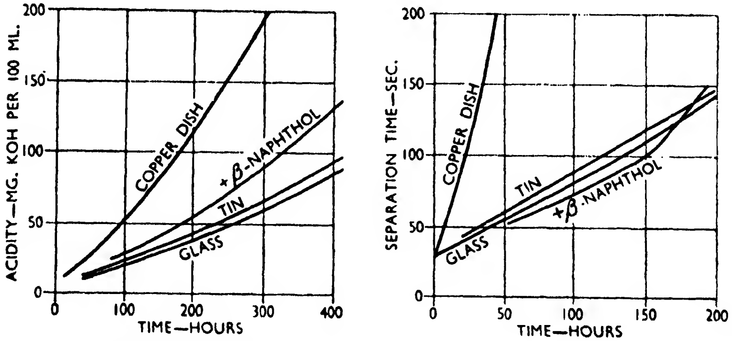
* Royal Aircraft Establishment, Farnborough, Hants.

† Lederer and Zublin, *National Petroleum News*, 1930, vol. 22, No. 33, p. 63.

are shown in Fig. 1. Oil No. 1* was an unstable light mineral oil which developed free sulphuric acid on ageing; heavy black sludging occurred at 85 deg. after about 500 hours. With oil No. 2, a better acid-refined Russian light oil, no sludging occurred up to 750 hours, although there was a considerable increase in acidity. It will be seen (Fig. 1*b*) that B.P. liquid paraffin is sensitive to copper, but oil No. 12, which is a mineral oil blended with 5 per cent vegetable oil, is less so.



a Acidity and sludge at 85 deg. C. *b* Acidity at 100 deg. C.



c Acidity at 100 deg. C. *d* Emulsivity-separation time, at 100 deg. C.

Fig. 1. The Oxidation of Oils in Air in Open Dishes

Tests *c* and *d* were carried out with oil No. 1 mixed with one-third volume of Pennsylvanian oil

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* See the list of oils in the Appendix, p. 366.

The results of incubation of various oils at 100 deg. C. in open glass dishes is given in Table 1.

TABLE 1. RESULTS OF INCUBATION OF OILS IN OPEN DISHES AT 100 DEG C.

Oil	Acid values, milligrammes of KOH per 100 millilitres				Remarks
	7 days	14 days	21 days	28 days	
No. 4	45	70	105	—	Sludge visible after 14 days
5	55	105	140	—	Heavy deposit after 14 days
8	30	55	70	93	Slight sludge after 26 days
9	55	85	125	—	Sludge visible after 21 days
11	30	50	55	120	No sludge

Oxidation changes by the bubbling method, Air Ministry Specification, as would be expected, are more rapid than in the dish; the following results (Table 2) illustrate this difference.

TABLE 2. OXIDATION CHANGES AT 100 DEG. C.: DISH AND BUBBLING METHOD (STANDARD) COMPARED

	Viscosity changes (percentage increase) after 100 hours	
	Glass	Copper
Oil, No. 3, in dishes . . .	2·4	2·6
,, ,, bubbling method .	7·5	11·7
Oil No. 14, in dishes . . .	12·3	25·7
,, ,, bubbling method .	15·7	38·7

The Absorption of Oxygen by Oils. The measurement of the quantity of oxygen taken from the air by a lubricant in a closed bulb can be made by manometry or by an analysis giving the oxygen depletion.

Results obtained by the closed-bulb method are given below:—

As regards the action of surfaces on the rate of oxygen absorption, tin dust exerted an inhibitory action; gold had little or no effect; whilst iron and aluminium increased the rate of oxidation. The

Table 3. Absorption of Oxygen from the Air by Different Oils at 125 deg. C.

Oil	Oxygen consumed, milligrammes per gramme of oil			
	1 day	2 days	4 days	5 days
Proprietary oil, No. 3	0·9	1·5	3·2	—
" 109," No. 10	0·8	1·3	2·6	—
Venezuelan, No. 14	1·5	—	5·1	6·7
Pennsylvanian, No. 15	0·8	1·1	—	4·3
Russian oil, No. 16	0·7	1·0	—	2·0
Castor oil, B.P., No. 17	14	—	—	—

accelerating action of some metal surfaces (5 sq. cm. per gramme of oil) at 125 deg. is shown in Table 4.

Table 4. Effect of Metal Surfaces on Oxygen Absorption by Oils at 125 deg. C.

Oil and conditions	Oxygen consumed, milligrammes per gramme of oil					
	Glass	Iron	Copper	Nickel	Aluminium	Lead
Oil No. 3, 2 days at 125 deg. C.	1·7	11·5	9·8	3	3·5	8·6
,, 16, ,, ,,	1·0	6·5	5·6	—	—	—
,, 15, ,, ,,	1·1	10·0	—	—	—	—
,, 10, 3 ,, ,,	2·0	12·0	6·4	3	7·6	—

II. Oxidations in the Temperature Range 125–225 deg. C.

The rate of oxidation of an oil increases enormously over this range. Some idea of the temperature coefficient of oxidation can be gained by the fact that whilst at 110 deg. the Russian oil No. 16 took about 300 hours to consume 1 milligramme of oxygen for each gramme of oil, at 125 deg. it took 48 hours, and at 200 deg. just 10 minutes.

Temperature Coefficient of Oxidation. The data obtained by means of the closed-bulb method for oxygen absorption of several oils are given in Table 5.

TABLE 5. TIME, IN MINUTES, TAKEN BY 1 GRAMME OF LUBRICATING OIL AT DIFFERENT TEMPERATURES TO REACT WITH 5 MILLIGRAMMES OF OXYGEN FROM THE AIR

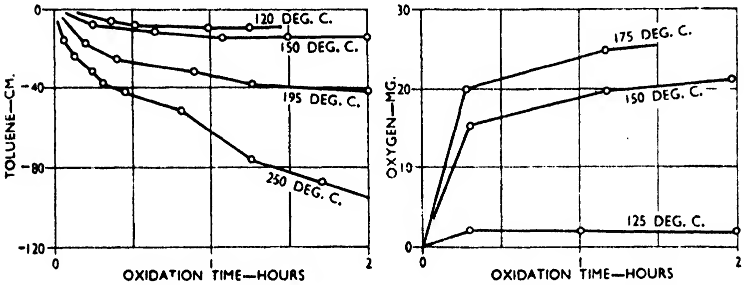
Oil	Temperature, deg. C.							
	110	125	135	150	200	250	275	300
No. 3 .	—	8,600	1,600	—	—	—	—	—
No. 10 .	28,000	10,000	2,000	125	—	—	2·5	—
,, 11 .	—	—	—	175	48	5	0·5	—
,, 12 .	—	12,000	2,200	170	46	17	4	0·4
,, 14 .	24,000	5,500	900	95	25	9	1	—
,, 15 .	48,000	12,000	2,300	180	55	23	5	0·7
,, 16 .	85,000	15,000	2,500	180	50	21	—	—
,, 17 .	—	480	45	20	5	2	1	0·5

The temperature coefficient of the oxidation rate thus differs considerably from oil to oil, whilst above 250 deg. there is a critical temperature above which an oil is liable to break down rapidly under oxidizing conditions.

Oxygen Absorption and Chemical Changes in Oils. Some idea of the influence of temperature and oxygen absorption on sludging, etc., is given by the behaviour of oil No. 3; when heated in the closed bulb for eight days at 180 deg. the amount of sludge formed was about 2 per cent of the oil, whilst at 250 deg. with two hours' heating, the amount was 5·5 per cent and at 325 deg. with ten minutes' heating, 4·2 per cent. The bulb method was used to determine the time taken for an oil to become oxidized to a solid. Oil No. 11 at 150 deg. required over eight weeks; at 200 deg., three days, at 250 deg., four hours; and at 300 deg., 30 minutes. No well-defined correlation was found between oxygen absorption and the degree of sludging, rise in viscosity, acid formation, etc.; all that can be said is that those oils which consume oxygen readily tend to become acid and form sludge quickly, but two different kinds of oils may absorb the same amount of oxygen and yet show decidedly different oxidation behaviour; any test, therefore, based on oxygen absorption needs supplementing by sludge tests, etc.

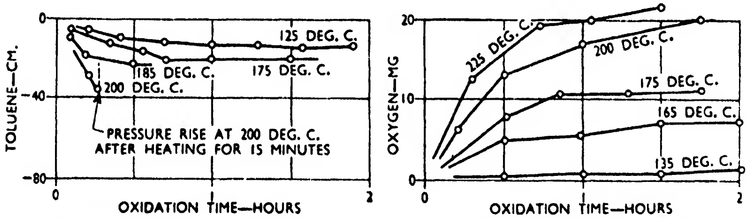
Manometry. A simple means of following oxygen absorption continuously is to measure the decrease in pressure in the closed bulb. Some results are shown in Fig. 2, together with results obtained by chemical analysis of the oxygen absorbed from the air. At higher temperatures the absorption is offset by pressure due to the oxidation products.

Sludging, Viscosity Rise, and Other Changes not Proportional to the Amount of Oxygen Absorbed. Sludging is more noticeable with some oils than others. Oil No. 4, for example, began to form black deposits at 100 deg. after ageing a fortnight, whilst on the other hand oil No. 15 even at high temperatures was exceptionally resistant to sludging, although it showed the usual increase in viscosity and acidity rise after oxidation. The amounts of sludge, acidity, and deposits are not usually proportional to the time of heating. There is often a sudden development of these changes during heating and so it is not possible to compare quantitatively the degree of deterioration from one set of points on the curve relating time and oxidation changes.



a Pressure decreases in closed bulb.

b Oxygen used by 1 gramme of oil.



c Pressure decreases in closed bulb.

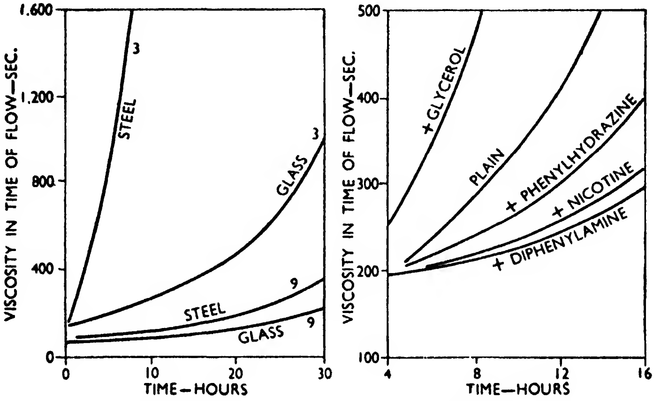
d Oxygen used by 1 gramme of oil.

Fig. 2. The Oxidation of Oils at Different Temperatures

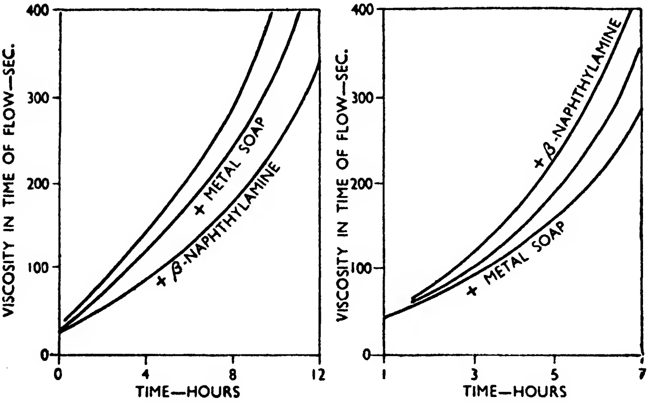
a and *b* castor oil. *c* and *d* asphaltic base oil No. 14.

Sludging and Temperature. Usually at 200 deg. the mineral oils required at least 24 hours before they accumulated 1 per cent of sludge. At 225 deg. sludging becomes more rapid and 1 per cent accumulates within a few hours and at 250 deg. in a few minutes.

Oxidation of Oils at 200 *deg.* Lubricating oils at 200 deg. C. kept under oxidizing conditions deteriorate within a few days to a thick black plastic material. The rate of viscosity rise at 200 deg. for some oils blown under standard conditions is shown in Fig. 3.



a 200 deg. C *b* 200 deg. C.



c 220 deg. C. *d* 250 deg. C.

Fig. 3. Rate of Viscosity Rise under Oxidizing Conditions

The numbers refer to the oils listed in the Appendix. Graphs *c* and *a* refer solely to oil No. 10.

Table 6 reproduces some viscosity results (in time of flow in seconds at 20 deg. C.) from an oil tested in accordance with the Air Ministry Specification D.T.D. 109, and blown at different temperatures by the standard method, in the presence of case-hardened steel.

Table 6. Viscosity Rise after Air-Blowing at Different Temperatures

Original time of flow, 43 seconds.

Time of blowing, hours	Temperature, deg. C.			
	160	200	220	250
4	—	—	121	140
6	—	91	225	330
7	—	—	—	620
10	—	123	389	>1,000
15	—	205	>1.000	—
25	—	550	—	—
30	55	>1,000	—	—
300	287	—	—	—
350	450	—	—	—

Table 7 gives results by the dish method for viscosity rise (time of flow in seconds) and development of acidity (milligrammes of KOH per 100 millilitres) with oil No. 11.

Table 7. Changes during Incubation in Dishes

(*a*) Viscosity rise. (*b*) Acidity in milligrammes KOH per 100 millilitres.

Time, days	Temperature, deg. C.					
	100		150		200	
	(*a*) sec.	(*b*) mg.	(*a*) sec.	(*b*) mg.	(*a*) sec.	(*b*) mg.
0	55		—		—	
1	—		—		72	
4	—		—		91	
6	—		—		108	
8	—	15	—	25	>500	90
20	58	48	—	112	—	
40	—		73		—	
100	87		—		—	

When the temperature of oxidation is raised from 200 to 225 deg. C. the oxidation rate is approximately doubled and the appraisement of oils can be made in a few hours. Some results obtained by the bulb method at 225 deg. are given in Table 8. The viscosity ratio is the final viscosity at 35 deg. divided by the initial viscosity.

TABLE 8. RESULTS OF OXIDATIONS AT 225 DEG. C. FOR SIX HOURS

Oil	Viscosity ratio	Sludge, milligrammes per gramme	Deposits on walls, milligrammes per gramme
No. 3	1·6	4	14
4	High >10	23	55
5	1·9	10	23
6	2·4	12	15
7	High >10	22	56
9	2·4	8	10
11	2·2	2	12
12	1·7	2	10
14	High >10	24	52

Results by the Bulb Method, Bubbling, and Dish Methods Compared. Differences observed in the appraisement of oils by the bubbling, bulb, and incubation methods respectively, especially when inhibitors were present, may be due to the difference in the ratio of surface to liquid. It was found, however, that there were similarities in appraising oils by the three methods, taking into account sludge formation as well as viscosity rise. Thus with a selection of oils taken at random the appraisement according to merit by the bulb, dish, and bubbling methods was as shown in Table 9.

TABLE 9. APPRAISEMENT OF OILS

(*a*) Appraisement of lubricants according to resistance to viscosity rise in order of merit.

Order	Bubbling with oxygen at 225 deg. C. after 4 hours		Bulb method, 6 hours at 225 deg. C.		Dish method, 3 days at 200 deg. C.	
	Oil	Viscosity ratio	Oil	Viscosity ratio	Oil	Viscosity ratio
First .	No. 8	1·5	No. 3	1·6	No. 3	1·3
Second .	„ 3	1·6	„ 12*a*	1·8	„ 4	1·35
Third .	„ 12*a*	1·65	„ 8	1·85	„ 8	1·7
Fourth .	„ 4	2·4	„ 5	1·9	„ 5	>5
Fifth .	„ 5	2·5	„ 4	2·0	—	—

(*b*) Appraisement of lubricants according to resistance to sludging, in order of merit.

Order	Bubbling method, 3½ hours at 225 deg. C.		Bulb method, 6 hours at 225 deg. C.		Dish method, 3 days at 200 deg. C.	
	Oil	Sludge, milli-grammes per gramme	Oil	Sludge and deposits, milli-grammes per gramme	Oil	Sludge, milli-grammes per gramme
First .	No. 12*a*	11	No. 12*a*	11	No. 8	50
Second .	„ 8	33	„ 8	14	„ 3	75
Third .	„ 3	53	„ 3	25	„ 4	235
Fourth .	„ 5	125	„ 5	33	„ 5	270
Fifth .	„ 4	157	„ 4	78	—	—

The chief difficulty in making comparisons between the methods is in the choice of the time of oxidation; the oxidation changes are not proportional to the time of heating and the rate and reactions of sludging or viscosity rise are not necessarily the same in both cases. This is shown by some curves reproduced in Fig. 4.

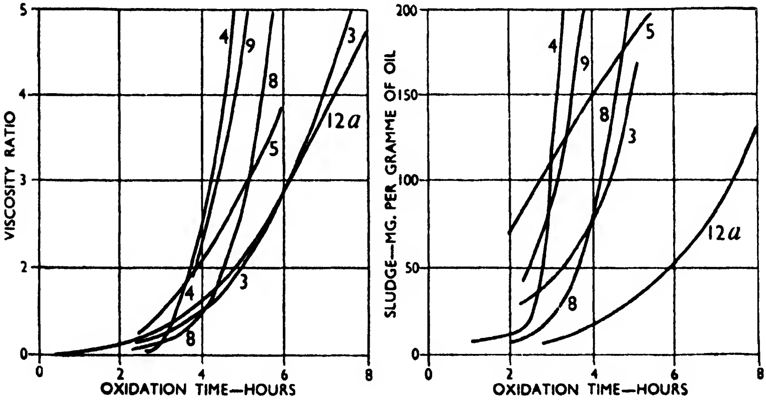


Fig. 4. Oxidation of Six Lubricating Oils with Oxygen under a Pressure of 40 lb. per sq. in. at 225 deg. C.

III. Oxidation of Lubricating Oils above 225 deg. C.

Oxidation of lubricating oils above 225 deg. C. occurs very rapidly and deep-seated molecular changes occur; water, carbon dioxide, formaldehyde, etc., are formed and the oxidation proceeds with chemiluminescence. Sludging is an important feature of high-temperature oxidations; this is shown by Table 10.

TABLE 10. SLUDGING OF OIL NO. 12 AT 250 DEG. C. BY BULB METHOD

Time of oxidation, hours	1	2	3	4	5	6
Sludge, milligrammes per gramme	40	80	140	170	180	190
Deposits on walls, milligrammes per gramme	22	84	125	150	160	160
Total	62	164	265	320	340	350

Blowing at High Temperatures. It is interesting to compare the behaviour of a collection of oils when blown at 250 deg. with regard to their flow properties at high temperatures when highly oxidized. Blended oils compare favourably with mineral oils in that they tend to remain liquid for a longer period.

Formation of Sludge at Temperatures above 250 *deg. C.* A "pot" method was devised for measuring sludge formation at high temperatures. Some results obtained by the method are given in Table 11.

TABLE 11. SLUDGE AND DEPOSIT FORMED IN 30 MINUTES FROM OIL NO. 3 AT DIFFERENT TEMPERATURES

Temperature, deg. C.	Sludge, milligrammes per gramme	Deposits, milligrammes per gramme
275	27	100
300	77	150
325	172	210
350	203	220

The amount of sludge formed at 300 deg. was roughly proportional to the time. Distinct differences in the amount and character of the sludge were observable with different kinds of oils. Some examples are given in Table 12.

TABLE 12. SLUDGING WITH DIFFERENT OILS AT 300 DEG. C.

Oil	Sludge, milligrammes per gramme		
	After 15 min.	After 30 min.	After 45 min.
No. 3	40	77	115
,, 11	45	75	140
,, 12	45	69	99
,, 16	57	85	175
,, 15	28	53	62
,, 14	—	98	160

The oils at temperatures above 250 deg. C. were sprayed or trickled through a glass tube, and the oxygen absorption from the air was determined. It was found that a mineral oil during its rapid passage down the tube showed but little change until a certain critical temperature was reached. At this point the oil began to discolour and give off a fog, consisting of water, aldehydes, etc., whilst deposits formed on the walls of the tube; the oxygen was also rapidly absorbed from the air. This breakdown temperature differed considerably amongst the various oils and substances tried. Thus, B.P. paraffin

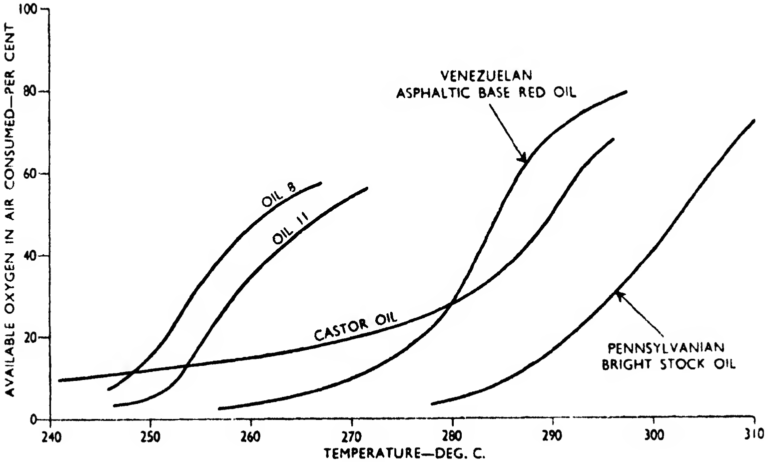


Fig. 5. Oxygen Consumed at Various Temperatures by Different Oils

oxidized rapidly at about 260 deg., oil No. 11 at 250 deg., dekalin at 280 deg. The results for oxygen absorptions at different temperatures with different oils are shown in Fig. 5. Self-ignition usually occurred when vigorous oxidation set in, about 50 deg. C. above the breakdown temperature.

APPENDIX

List of Oils Used in the Experimental Work

No. 1. Light mineral oil, liquid SO_2 extracted, sp. gr. at 20 deg., 0·877.
2. Light Russian oil (water white), sp. gr. at 20 deg., 0·889.
3. Proprietary oil.
4. Ditto, as received from National Physical Laboratory. Seizing temperature 225 deg. C. in prepared bush.
5. Ditto, as received from National Physical Laboratory. Seizing temperature 281 deg. C. at 50 hours.
6. Ditto, as received from National Physical Laboratory. Seizing temperature 175 deg. C. after 56 hours.
7. Ditto, as received from National Physical Laboratory. Seizing temperature 110–107 deg. C. for 50 hours.
8. Ditto (medium), as received from National Physical Laboratory. Seizing temperature 255 deg. C. in prepared bush.
9. Ditto, as received from National Physical Laboratory. Seizing temperature 222 deg. C. in prepared bush.
10. R.A.F. oil, summer grade.
11. Proprietary oil, thick grade.
12. Proprietary oil, thick grade.
12A. Proprietary oil, ordinary grade.
13. B.P. liquid paraffin.
14. Venezuelan asphaltic base distillate, sp. gr. at 60 deg. F., 0·938.
15. Pennsylvanian cylinder stock, sp. gr. at 60 deg. F., 0·892.
16. No. 1 Russian distillate, sp. gr. at 60 deg. F., 0·909.
17. B.P. castor oil.
18. Paraffin base oil (light), sp. gr. at 60 deg. F., 0·862.
19. Proprietary oil.
20. R.A.F. oil, sample 1.
21. R.A.F. oil, sample 2.
22–26. Proprietary oils.

OXIDATION TESTS FOR MOTOR OILS

By B. H. Moerbeek *

The motive for this paper lies in the fact that at the World Petroleum Congress in Paris in June 1937 the number of oxidation tests, which is already considerable, again had several added to it. These tests have much in common with certain patent medicines for rheumatism; the fact that their name is legion is enough to inform even the uninitiated that there is a great need for a remedy, and at the same time shows that the real thing has not yet been found.

Unfortunately, it is beginning to look as if the same undesirable state of affairs as prevails for transformer and turbine oils will soon arise with respect to motor lubricants. The need for an oxidation test for the former was very marked, and it was for these oils that the first tests were planned. They have now been consolidated to a certain extent, that is to say, in many countries a certain method has come to be recognized as official. Most countries use their own method, so that there are more or less official English, German, Dutch, Italian, Swedish, Swiss, etc., tests for transformer oils, besides the sometimes very good methods that have no official standing.

These different methods often test the oil for contradictory requirements, so that it happens that an oil that gives very satisfactory service is accepted in one country, but is rejected in another on the strength of the test acknowledged there. It may also be stated in passing that there are, moreover, some tests whose technical utilizability is greatly reduced by their poor reproducibility. It is to be hoped that this undesirable state of affairs may be avoided in the case of other lubricants, but it looks very much as if this is no longer possible.

In 1924 the American Society for Testing Materials (A.S.T.M.) took up the Sligh oxidation method; in 1935 it was given in the edition of the A.S.T.M. methods, but only as a "proposed method". This means that it does not even have the standing of a "tentative" test but is published for information only. The opinion generally held in technical circles in America is that the test is valueless; in a recent article (1934) this is proved on the strength of figures collected. Accordingly, the A.S.T.M. no longer give the method; it was omitted from the 1936 issue. But from a recent abstract (1937) it appears that this test, with a few slight alterations, is accepted in Sweden as the official standard.

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* Amsterdam Laboratory, N. V. De Bataafsche Petroleum Maatschappij, Royal Dutch Shell Group.

A similar thing has occurred with the British Air Ministry test. The value of the test for judging motor lubricants is doubtful. Thus, the very limited value of this test, even for examining aviation oils, for which it is really intended, is frankly acknowledged by Pye: "a check on the chemical stability of an oil, both against oxidation at 200 deg. C. and decomposition at the higher temperature is aimed at, and to some extent achieved. Undoubtedly there are examples of oils which would fail to meet the specification and yet might stand up excellently during prolonged tests in service. . . . Other points of weakness in the oxidation test are: that it depends upon decomposition in the presence of air, whereas in an engine the oil is subjected to its most severe temperature conditions round the piston rings, in an atmosphere deficient in oxygen; that it makes no allowance for the effect of the metal surfaces upon the oil, and that it endeavours to bring about in a short time by more severe temperature conditions, effects which in practice would normally take very much longer to produce."

Yet, in spite of all this, a more or less official body like the Testing Bureau of the Netherlands Royal Air Force have adopted this test, after making such inevitable alterations as replacing the determination of the Ramsbottom coke number by the Conradson carbon residue (van Gorkum 1932). But this, at least, was for aviation oils. The Netherlands Government Mines went a step further, however, and made this test, which is already severer than actual practice, even harder for engine oils, etc., for which it was never intended, by prescribing 5×6 hours' aeration instead of 2×6 hours' aeration. It is therefore sufficient if an official body somewhere publishes a test, to get it accepted somewhere else without a murmur even if the original body realizes the shortcomings of the test in question.

All this is scarcely to the advantage of the consumer, for the supplier now has to provide him with an oil that in the first place meets the test; whether it is also suited to the purpose it is wanted for is a secondary consideration. It is certainly not an imaginary danger that, by complying with exaggerated oxidation tests, the actual lubricating and protective capacity of the oil may be endangered. Usually this point must be specially allowed for, a fact which is by no means always taken into account by suppliers.

A recent paper by the U.S. Bureau of Standards (1937) aptly illustrates how difficult it is to find an oxidation test really correlating with actual conditions. Unlike the usual terminology of an inventor of a new oxidation test, the opinions of the Bureau are expressed very cautiously (as is only to be expected). Usually an inventor of a test carries out experiments with one engine and, on the strength of the fact that a certain combination of a few conditions of an oxidation test correlate with these experiments, draws the conclusion that *the* oxidation test for actual practice has been discovered. The Bureau of Standards, however, state emphatically that their conclusions as regards correlation only apply to the engine in question, though they, moreover, (unlike most investigators) had tested such a large number (22) of representative oils that they could justifiably have spoken of correlation. "It appears, therefore, that heating of oils in the laboratory with an exposed surface but without aeration for a period of 24 hours at a temperature of 175 deg. C. will produce changes which take place in a Pratt and Whitney 'Hornet' aero-engine."

Generally, in oxidation tests, ample use is made of air, indeed even of oxygen, and also of fairly high temperatures. In the publication just referred to, after long experimentation the conclusion is drawn that this is a mistake when applied to the above-mentioned air-cooled aero-engine: "Methods involving aeration of aviation oils are much too severe for oil stability tests and do not show correlation with engine data." The reason for this is the generally acknowledged fact that the oil, at the moment when and in the places where it is heated, does not come into contact with air but with combustion gas, which contains very little oxygen. It moreover appears from the experiments of the Bureau of Standards that from oxidation tests almost any order of lubricating oils used can be found by choosing the proper test conditions ! Surely this should compel one to tread warily when laying down conditions that are intended to show correlation with actual service conditions.

Most publications on this subject submitted at the World Congress show few signs of this caution, and, to begin with, come to entirely different conclusions among themselves. There is only one point on which they all agree, and that is, that *previously* published methods show *no* correlation with *actual* service conditions.

On the strength of engine tests with one particular Diesel engine, Noack asserts that the Indiana oxidation test shows *no* correlation with practice (i.e. with *that* Diesel engine) and concludes that "The ageing conditions (of the hitherto customary ageing methods) are too strict as compared with those conditions prevailing in practical use". He suggests a test with one hour's aeration at 250 deg. C.

On the other hand, Maillard, on the strength of engine tests on one Diesel engine, concludes that the temperature in oxidation tests must be very high, and arrives at 8 hours' aeration at 300 deg. C., easily holding the record for exaggerated conditions of oxidation tests. This is probably due to the fact that he mistakes incomplete combustion products of the fuel for products formed from the lubricating oils, consequently judging the lubricating oil used for his engine test far too unfavourably, which makes his laboratory test much too severe.

In a recent publication Baader (1937) warns investigators against applying such high temperatures. By working up the temperature, not only is oxidation accelerated, but its nature is completely altered.

It is therefore not to be wondered at that the large oil companies have little or no confidence in oxidation tests, but when selecting base materials and refining methods for lubricating oils, are rather led by extremely expensive and lengthy practical tests in actual Diesel, petrol, and aero-engines. They are not satisfied with just one type of engine, but feel obliged to include all the important makes of engines in their experiments. Experience has shown that certain constructions are accompanied by certain difficulties, which have usually no connexion with good stability in oxidation tests. For instance, certain American Diesel engines on the market very soon show ring-sticking. The better the oxidation stability of the oil used, the worse the ring-sticking. In this case it is only possible to lubricate with an oil of medium oxidation stability, which is even given a slightly lower stability by deliberately adding a material to it that prevents ring-sticking. The results obtained with this oil in these engines are excellent, however. Investigators have not yet succeeded in imitating this phenomenon in the usual small trial engines, nor in finding a laboratory test for the purpose. The only way of testing an oil in this case is to apply long-time tests with the actual engine.

It therefore seems impossible that an oxidation test will ever be devised that will show reasonable correlation with all classes of engines, and it appeares quite unlikely that a test will be found upon which a reasonable percentage of a whole class, say, of aero-engines, or Diesel engines, can be tried. The conclusion that must therefore at present be drawn is the same as that expressed (1937) by W. S. James, President of the Studebaker Company (as such to be regarded as a representative of oil consumers): "In conclusion, it would seem that this paper has dealt with everything but the subject mentioned in the title. It is the opinion of the author that the best way to select the motor oil from the standpoint of the consumer is to consult with representatives of the oil and car manufacturers and follow their advice. This may seem to be a strange conclusion to present to a society for testing materials, but the author has been forced to it by the complexity of the problem and the obvious necessity of specialized knowledge of the oils from which a selection is to be made, the particular characteristics of the cars considered and the driving habits of the particular customer. It is to be hoped that increased knowledge of oils and of car characteristics as well as a freer and more complete interchange of this information between the two industries will soon make it really possible to tell the consumer how to select his motor oil".

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LABORATORY TEST FOR THE STUDY OF THE AGEING OF LUBRICANTS

By MM. R. Moutte, G. Dixmier, and P. Lion*

The part played by the lubricant in the engine is to enable rubbing members to move without excessive friction. In internal combustion engines this aim is attained either by a very thin layer of lubricant between the metal surfaces, thus realizing lubrication by a molecular film, or by a relatively thick liquid layer which realizes lubrication by a liquid film or fluid friction. In the second case, only the hydrodynamic properties of the fluid, and especially the viscosity, are utilized. The exact parts played by these two methods of lubrication in an engine can be distinguished only with difficulty. The properties required in the lubricant to ensure lubrication by a molecular film include a series of physical and chemical properties which are usually grouped under the word "oiliness," which expresses the intrinsic properties of the oil as well as those of the system constituted by the contact surfaces and the lubricant. Oily lubrication is difficult to define and, moreover, is practically uncontrollable so that the motor manufacturer and user try to ensure that a fairly thick film is present wherever possible, so as to obtain perfect lubrication. The continuity and upkeep of this film depend on the ease with which the liquid circulates, and therefore on the absence of any deposit which can separate from the lubricant and obstruct the piping. Engine troubles are caused largely because the oils fail to arrive at the parts requiring lubrication. Lubricants undergo oxidation and polymerization in the engine, resulting in the formation of insoluble deposits which settle in the lubrication system at points where circulation is slowed down by bends and so on, and, by hindering the passage of the oil, lead to engine trouble.

It is this problem of the physico-chemical ageing of oil that is considered here with the object of foreseeing, before carrying out a running test, how the lubricant will behave with regard to deposits. The authors believe that the real test of a lubricant is one carried out on the engine which it should lubricate, but engine tests are long and costly and should be as few as possible, while it is desirable to avoid the damage to the engine that might be caused by a test of an unknown lubricant. The tests described in this paper, and the conclusions arrived at, apply to the case of a pure mineral oil.

In the engine the oil undergoes two different kinds of change. In

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* Under the direction of M. Dumanois, Inspecteur Générale de l'Enseignement et des Recherches Aéronautiques, Ministère de l'Air, France.

the explosion or combustion chamber, in the cylinder bore, and on the piston, oxidation at a fairly high temperature, evaporation, and even partial combustion, take place, while in the sump and piping oxidation at low temperature (100 deg. C. and below), and polymerization occur. What is the part played by each of these changes ? When it is desired to study the oxidation or polymerization products of mineral oil, these products are separated from the oil by extraction with petroleum ether or standard petrol. The deposits thus obtained are separated according to their solubility or insolubility in chloroform. Tests carried out on used oils from aero-engines and motor car engines show that when oil has been in use for even a few hours (say, 10), there is a considerable formation of products which are insoluble in chloroform. Further, no matter how long the running has continued, the matter insoluble in chloroform is always predominant, representing on an average 80 per cent by weight of the matter insoluble in petroleum ether. The oxygen content of the chloroform-soluble material is always less than 20 per cent by weight and over 20 per cent by weight in the chloroform-insoluble material. By "oxygen" is understood the difference between the total content of carbon and hydrogen and 100. From 10–20 per cent of metallic oxide is also present, though its origin is not clear.

In 1927 the authors developed a method for studying the formation of oil deposits by heating samples in an oven in an oscillating trough. This apparatus was standardized in 1932 (Fig. 1). The temperature employed did not exceed 140 deg. C., and a strip of pure copper was used as a catalyst to accelerate the production of deposits. None of the tests made at low temperature (below 140 deg. C.) on mineral engine oils gave products which are insoluble in chloroform except for a few rare cases of particularly unstable oils, no matter what method of ageing, time of heating, or catalyst was employed. The products of ageing contained roughly 14 per cent of oxygen. It was found that artificial ageing in the oven at 100 deg. C. without a catalyst resulted in the same increase in viscosity as that obtained by use in the engine for the same time. Further, the amount of deposit thus obtained was always higher, but remained of the same order as the amount of matter insoluble in petroleum ether, but soluble in chloroform, which is produced in the engine during the same period of time. This implies that artificial ageing below 140 deg. C. only gives part of the ageing which the oil really undergoes in the engine. This conclusion is confirmed by tests by M. Maillard (1937)*.

It thus appears that the production of deposits insoluble in chloro-

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* Maillard, A., Englander, and Martin. Second World Petroleum Congress, Paris, 1937, June, section 2, paper 183, "Sur l'oxydation des huiles minérales à température élevée".

form requires the use of higher temperatures. The authors have worked at temperatures between 200 and 400 deg. C. because, at higher temperatures, some cracking takes place which seems to have little influence, according to observations made on an engine. For these tests an oscillating oven was used to ensure that the oil, which was sealed in a 13 cu. cm. tube, was properly mixed (Fig. 2). The quantity of oxygen available varied from 0 to 2 per cent by weight of the oil under test, while heating was carried out at from 200 to 417 deg. C. for from 2 to 48 hours.

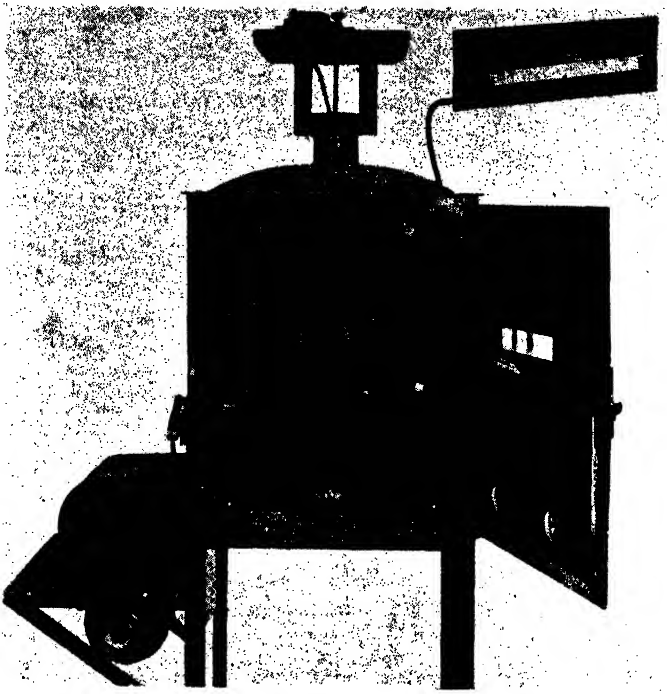

Fig. 1. Ageing Oven with Oscillating Troughs

Between 200 and 340 deg. C. no products were formed which were insoluble in petroleum ether or chloroform, though at about 350 deg. C. a small amount of matter was obtained which was insoluble in chloroform. Heating had to be extended to 400 deg. C. to obtain products which were insoluble in petroleum ether and partly soluble and partly insoluble in chloroform, the last-named insoluble portion being very small in amount compared with the soluble portion. In the experiments carried out at 400 deg. C. a difference of 3 or 4 deg. C. has a definite effect on the quantity of decomposition products formed. Below 300 deg. C. the aged oil increases in viscosity, whereas above that temperature, the commencement of cracking seems to make up for the increase in viscosity.

Variations of the quantity of air present appeared to have no noticeable effect, but the replacement of air by pure oxygen or nitrogen had an important influence on the formation of ageing products, as the amount of matter insoluble in petroleum ether is doubled (from 0·13 to 0·28 per cent) when air is replaced by oxygen in tests lasting two hours at 400 deg. C. Further, replacement of air by nitrogen together with preliminary de-aeration of the oil reduced the content of matter insoluble in chloroform from the amount of 0·5 per cent (formed on heating for 16 hours at 410 deg. C.) to traces.

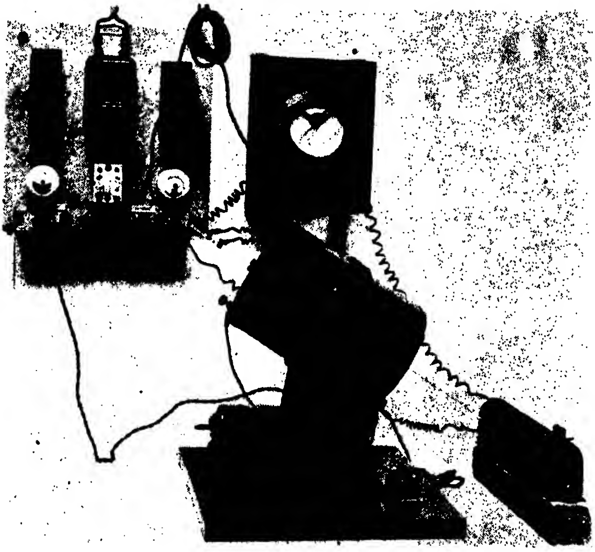

Fig. 2. Oscillating Ageing Oven

Copper acts clearly as a catalyst at 100 to 140 deg. C., but acts very weakly at 300 to 400 deg. C., as was found by Maillard (1937).

The duration of heating influences the formation of decomposition products, as the amount produced increases with the length of heating. In general, it was found that all the matter obtained contained very little oxygen, a maximum of 5 per cent in the chloroform-soluble products, which thus contain more oxygen the lower the temperature at which they have been formed. The chloroform-insoluble products, which could only be obtained above 400 deg. C., were very poor in oxygen (1 per cent) and very rich in carbon. Here, again, it must be admitted that the quantities and qualities of the products of ageing were not comparable with those found in oils which had been used in an engine.

It was, however, realized that the oil, because of its circulation in the engine, is affected alternately by the two changes which were mentioned above, i.e. successive changes at low temperature and at high temperature. For this reason, a new series of ageing tests at high temperature was planned, utilizing oils which had first been aged at low temperature by the first method described in this paper. Oil was thus heated in the oscillating furnace only after it had been aged at 140 deg. C. for 120 hours in the oven with oscillating troughs.

These tests have not yet given results equivalent to those obtained in the engine, but they show a distinct tendency to approximate to them. Oils which have been aged at 140 deg. C. are distinctly sensitized for the production of decomposition products when they are heated in a confined atmosphere (in the sealed tubes mentioned above) at 392 deg C. for 24 hours. Under these conditions certain oils yielded 4·6 per cent of matter insoluble in petroleum ether, comprising 3·2 per cent soluble in chloroform and 1·4 per cent insoluble in chloroform. The insoluble matter contained roughly 11 per cent of oxygen whereas the matter soluble in chloroform only contained 5–6 per cent of oxygen.

Thus, by ageing at 392 deg. C. the authors have obtained products which are insoluble in chloroform and contain more oxygen than the products soluble in chloroform which were obtained at the same time. The content of oxygen is not yet that found in products due to ageing in an engine, but the method seems to promise still closer results. The fact that the chloroform-soluble products formed at low temperature can be transformed into insoluble products, whereas it was very difficult to obtain those products directly, and the fact that the chloroform-insoluble products formed directly by heating at a high temperature were always very poor in oxygen, whereas those obtained by the high-temperature conversion of soluble products formed at low temperature were always richer in oxygen, both suggest that the products obtained in the engine are formed by a similar process, i.e. by oxidation for a fairly long time at low temperature giving products fairly rich in oxygen (less than 20 per cent) and soluble in chloroform, followed by very rapid heating at high temperature (350 to 400 deg. C.), thus transforming the chloroform-soluble into chloroform-insoluble products rich in oxygen (over 20 per cent). The remarkable agreement between the proportions of the chloroform-soluble and chloroform-insoluble products obtained in an engine appears to confirm the authors' theory of the chemical ageing of mineral oil in an engine.

Conclusions. As a result of the tests described above, the authors conclude that no method of artificial ageing is yet available which gives products identical with those obtained in an engine. However, despite this, the method of artificial ageing described makes it possible to distinguish oils which have a strong tendency to form deposits. Even though artificial ageing is imperfect, it provides a necessary protection against oils of poor quality. It may be hoped that in the future, a more complete ageing will make it possible to differentiate between oils which show little or no difference when aged artificially by the present methods.

To obtain a laboratory process of ageing which gives results similar to use in the engine, two methods can be suggested :—

(1) Heating the oil to 300–400 deg. C. after it has been sensitized by heating at a low temperature.

(2) Alternate heating at low temperature (100–140 deg. C.) and at high temperature (300–400 deg. C.), the heating at low temperature being more prolonged.

The problem is rather to simulate the ageing process than to accelerate it, for the average length of service of a molecule of oil in an engine in service is roughly 10 hours. Further, the fact that engines put into service very soon contain all the ageing products shows that the time factor relates to the quantity and not to the quality of the products. Various factors which play a part in the engine will have to be neglected in seeking a method of ageing and it will have to be sufficient to utilize the oxygen content, the temperature, and catalysts in various combinations. Other factors, particularly the action of the fuel, will have to be neglected in an approximate synthesis of the ageing process.

A criterion on which is based the suggestion that a method of ageing is available which produces changes in the oil similar to those in the engine, should not depend simply upon the separation of products which are soluble or insoluble in chloroform. That separation is insufficient and should be completed, either by chemical identification or by analyses of the ageing products obtained in the two cases.

FRICTION AND WEAR COMPARISONS FOR LUBRICATED SURFACES

By G. L. Neely*

In an investigation of addition agents for lubricating oils, one of the most striking observations was that the great majority of oil-soluble compounds will reduce the coefficient of friction for some operating condition. It is therefore a simple matter to find an addition agent which merely improves "oiliness". The effectiveness of such compounds on the wear-reducing value of the lubricants may often be considerably different in degree, and sometimes in direction, from the effect on friction. It is the purpose of this paper to present some friction and wear data obtained concurrently in a special machine, with the hope that these experiments may in some measure contribute to a better understanding of boundary lubrication.

The present day interest in the subject of wear relates primarily to the cylinder walls and piston rings of internal combustion engines and their lubrication. During the last year there have been many reports and papers presented pertaining to engine wear and the influence of the type of lubricant used. Most of the lubricants used in the engine wear tests reported were selected from friction or film strength tests, but not from wear tests. A review of the literature shows a paucity of data illustrating fundamental relationships between friction, wear, and operating variables, especially for lubricated surfaces. This paper does not pertain to engine wear or other practical applications directly, but presents some data on friction-reducing value as obtained in a special laboratory machine designed solely for experimental purposes.

The Testing Machine. The machine used is called the kinetic oiliness testing machine, "Model B."† It is similar to model A, which was described in 1932.‡ The model B machine is shown in Figs. 1 and 3. The frictional elements are shown separately in Fig. 2 and consist of two opposing surfaces, one in the form of a track having two flat-topped, concentric rails and the other consisting of three small buttons with flanged rims forming single flat-topped rails. This machine was designed to minimize the effects of viscosity in order to investigate lubricants under conditions where oiliness effects predominate. Another feature of the machine is that the surfaces are

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* Research Engineer, Standard Oil Company of California.

† U.S. Patent No. 2,020,565.

‡ Proc. Am. Petroleum Inst., 1932. Mid-Year Meeting.

automatically maintained at a fairly uniform degree of smoothness by the lapping action produced by the combined rotating and sliding motion of the buttons.

The buttons 1 (Fig. 3) are located by and loaded through the spindle shafts 2, secured in the spindle head 3. The pivotal member of each spindle shaft is a steel ball 4, which rests at the centre of the buttons 1 in the hardened adaptor cup 5. The buttons are driven at spindle speed by means of universal connectors 6. The spindles and spindle head are driven by gearing 7; and spindle speeds up to 3,000 r.p.m. and spindle head speeds up to 200 r.p.m. are obtained through the change gearing 8. The track 9 rests in a circular track-holder 10, which is supported at the centre by means of the torsion wire 11, secured by nut 12. Frictional torque on the track caused by the rotating and sliding buttons is indicated by the scale 13 on the track holder 10. Variations in frictional torque which are caused by slight irregularities in the smoothness of the rubbing surfaces are effectively damped out by means of the dashpot 14 attached to the track holder 10 through the flexible connexion 15. The desired degree of loading is obtained through the weights 16 acting downward on the buttons through the spindle head assembly. Upon dismantlement, the dashpot 14 is disconnected, the nut 12 is removed, releasing the track holder 10; and the arm 17, which supports the electric heater 18, is lowered and

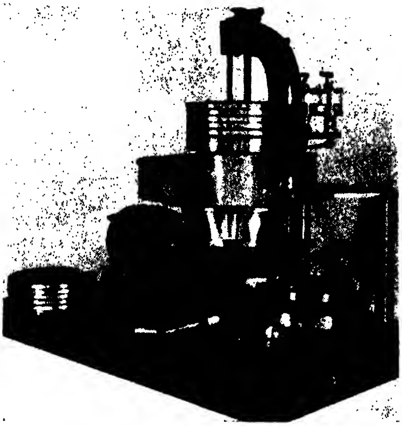

Fig. 1. Kinetic Oiliness Testing Machine (Partially Dismantled), Model B

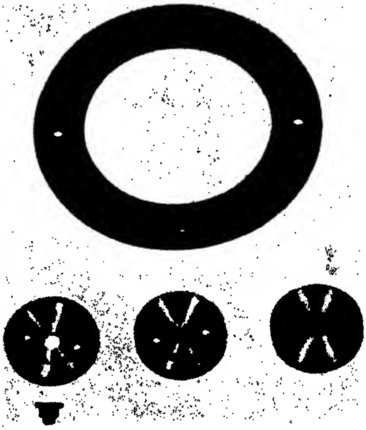

Fig. 2. Frictional Surfaces used in the Kinetic Oiliness Testing Machine

swung to one side as shown in Fig. 1. Test oil is supplied to the track through tubing 19. The test elements of the machine are enclosed by means of the cylindrical glass 20 in order that tests may be made in various atmospheres.

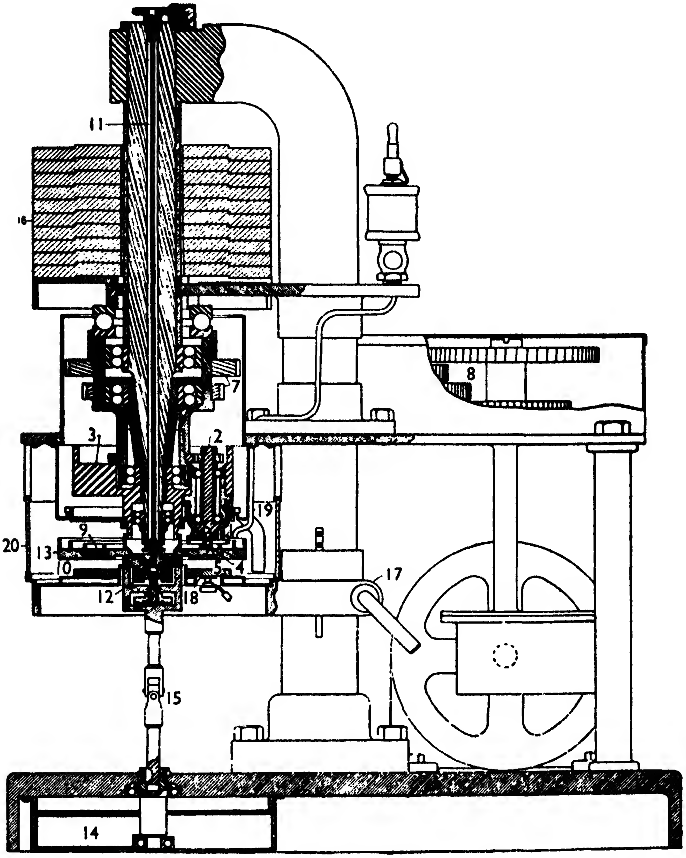

Fig. 3. Cross-Section of Kinetic Oiliness Testing Machine, Model B

Test Procedure. The procedure is very simple: the machine is assembled and operated with the test lubricant for a given set of conditions of load, speed, and duration. Measurements of friction are obtained during each test. While testing, a small amount of make-up lubricant is fed from the sight-feed oiler. Friction and wear values are obtained at various loads and speeds, using an oil temperature of

175 deg. F. The maximum dead load used was 41 lb., which corresponds to a unit loading of approximately 750 lb. per sq. in. The metal combination used was a ground, hardened steel track and "S.A.E. No. 64" phosphor-bronze buttons. The steel track showed no measurable wear in these tests, whereas the wear or amount of metal removed from the buttons was of sufficient magnitude to be easily measured and was found to be proportional to the operating time.

Effect of Load. Typical test data are plotted in Fig. 4, in which variations in the frictional resistance and wear with load are shown for two lubricants. It will be noted that both the frictional resistance and the wear vary directly with load. The fact that friction varies directly with load, or in other words, that the frictional coefficient is not affected by load (at least at moderate loads), is of interest since it complies with Amonton's law, which states that the coefficient of friction is not influenced by the intensity of loading.

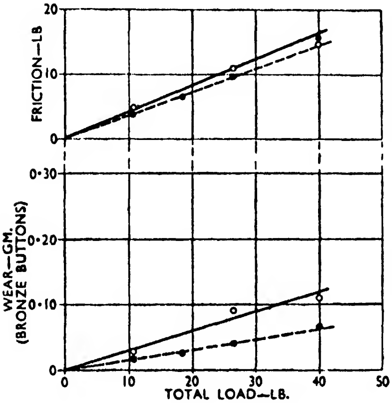

Fig. 4. Variations in Frictional Resistance and Wear with Load

Speed, 3·6 ft. per min.; surfaces, bronze on steel; temperature, 175 deg. F.; duration of test, 8 hours.

○ Lubricant No. 1
● „ No. 2

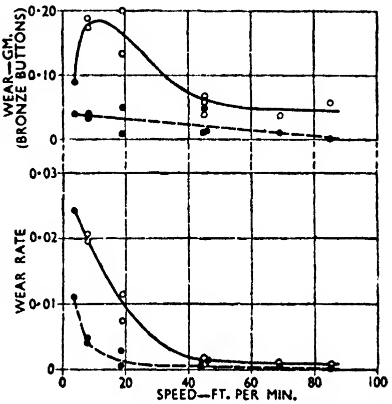

Fig. 5. The Effect of Rubbing Speed on Wear

Surfaces, bronze on steel; load, 26·7 lb. (500 lb. per sq. in.); temperature, 175 deg. F.; duration of test, 8 hours.

○ Lubricant No. 1.
● „ No. 2.

Effect of Speed. Fig. 5 shows the effect of rubbing speed on wear. The speed used was the resultant of the rotating and sliding speed of the buttons. The curve for total wear is of special interest since it shows that with lubricant 1 there is one particular speed at which the greatest amount of wear takes place in a given period of operating time. On the other hand, with lubricant 2, it would seem that the maximum amount of wear occurred near the minimum speed shown. It is also shown that the rate of wear, which equals the total wear divided by the rubbing speed (distance), decreases as the speed is increased. Thus the point of maximum wear per unit of time may occur where the speed equals a certain value, whereas the amount of metal removed per unit of distance travelled is greatest at the lowest speed. This is obviously of interest with respect to reciprocating mechanisms.

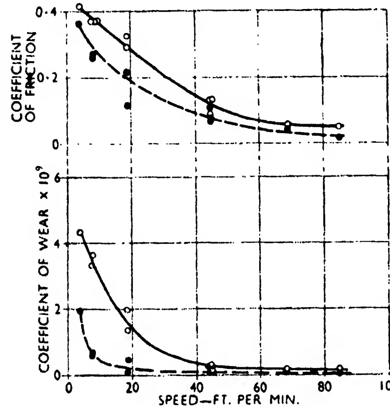

Fig. 6. The Effect of Speed on the Coefficients of Friction and of Wear

Surfaces, bronze on steel; unit load, 500 lb. per sq. in.; temperature, 175 deg. F.; duration of test, 8 hours.

○ Lubricant No. 1.
● „ No. 2.

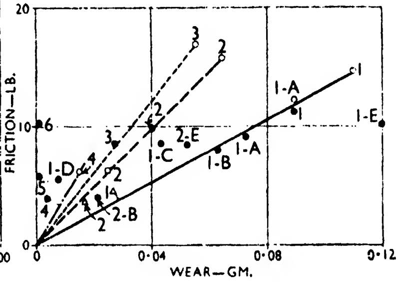

Fig. 7. The Effect of Addition Agents on Friction and Wear

Surfaces, bronze on steel; speed, 3·6 ft. per min.; temperature, 175 deg. F.

A numeral identifies the lubricant, a letter identifies the compound.

Unit loads per sq. in.:—

△ 200.	● 500.
□ 350.	○ 750.

Fig. 6 shows the effect of speed on the coefficient of friction and on a wear factor which involves the total wear divided by the load and the total resultant distance travelled (by the buttons). This factor is described as the "coefficient of wear" and appears to be fundamental for the data obtained thus far on this machine for the steel-on-bronze combination used. As may be noted, the factor is of extremely small magnitude. Dividing wear by load produces a coefficient similar to the coefficient of friction. Wear, however, takes place continuously and is a function of time. Therefore, in order to allow a cross comparison of wear values obtained at different loads and in tests of different duration, the "coefficient of wear" must also include a time factor which has been included as distance for convenience.

Inasmuch as considerable time is required to conduct each wear test, the investigation of the relation of wear to load has not yet been completed for the entire speed range of the machine.

Effect of Addition Agents. The effects of small quantities of various addition agents on friction and wear are shown in Fig. 7. The effect of adding compounds A and B was the same as reducing the load, as is shown by the fact that the points for the oils containing these compounds, 1A and 1B, fall on the line joining the points obtained at various loads with lubricant 1. Thus the friction-reducing value indicated that these two compounds possessed wear-reducing value. However, compound D had entirely different characteristics in that the wear with lubricant 1D was reduced to a far greater degree than was the friction. Compound C had an intermediate effect. On the other hand, the addition of compound E to lubricants 1 and 2, while slightly reducing the friction, greatly increased the wear. This result is shown by comparing points 1E with 1, and 2E with 2. Lubricants 5 and 6 are of interest in that they show it is possible to obtain extremely small amounts of wear with lubricants possessing relatively poor friction-reducing properties. These data illustrate the futility of attempting to predict wear-reducing value from friction tests.

Fig. 8 presents the same data shown in Fig. 7, but the scheme of plotting coefficient of friction versus the author's "coefficient of wear" is utilized. Perfect correlation between these two coefficients was not obtained, though the points for the same lubricants at different degrees of loading are fairly well grouped.

Discussion. While this paper discusses relationships between friction-reducing value and wear-reducing value, the term "oiliness" is employed in some cases with reference to friction-reducing value. There is considerable difference in opinion about the definition of "oiliness". Broadly, the author's use of this term refers to friction-reducing value under test conditions such that the frictional coefficient is independent of the intensity of loading. Under such test conditions it is assumed that fluid-film effects are substantially eliminated and that the occurrence of uniform and continuous wear under such conditions provides evidence that metallic contact, rather than fluid-film conditions, is in control.

"Oiliness," as described by the above definition, is markedly influenced by operating conditions, including the combination of metals used, temperature, and rubbing speed. In regard to speed effects, for example, early data showed that the properties of lubricants affecting "oiliness" at extremely low rubbing speeds differed from those that had effect in the high-speed ranges. Therefore, since "oiliness" effects depend upon several operating factors, it was found convenient for test purposes to consider the zone of "oiliness" operation as that where the frictional coefficient is independent of the intensity of loading.

While the data presented relate to the kinetic oiliness testing machine, it is believed that in all bearing operations in which there is solid metal contact, the frictional coefficient is independent of the intensity of loading for those portions of the bearing surfaces which constitute actual areas of contact, and that for these areas, the general findings obtained with the kinetic oiliness testing machine apply.

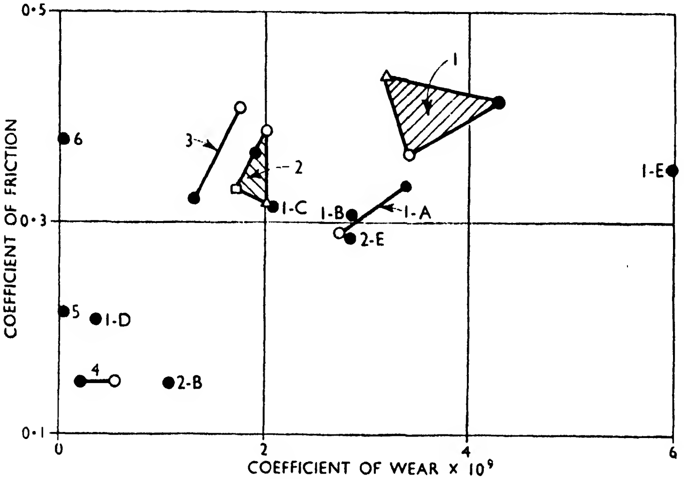

Fig. 8. The Effect of Addition Agents on Friction and Wear

The conditions are the same as those of Fig. 7.

One of the important observations in these wear tests was the effect of the surface condition of the hardened steel track. When the surface had a smoothly ground finish, data of the type shown on the charts were obtained, whereas with a mirror-finished track surface, the friction was somewhat reduced, though wear was reduced to a much greater degree. Furthermore, with a mirror-surfaced track, the influence of addition compounds was much less pronounced. This effect of surface smoothness for the steel on bronze combination applies only to the steel surface, as the bronze buttons attained a mirror-like smoothness in a very short operating time and wore at a uniform rate for a given set of operating conditions, whereas the steel track did not wear perceptibly even over long periods of time.

The author's previous paper related principally to "oiliness" effects on cast iron surfaces. It is significant that with this metal combination the wear obtained with lubricated surfaces was too small to be measured, even for extended periods of operating time. The present kinetic oiliness testing machines do not provide high enough load ranges to make them suitable for wear testing for the cast iron metal combination, and a third design is now being arranged where loads of much greater magnitude can be employed.

Insufficient data concerning wear have been obtained to evaluate adequately the effects of such factors as temperature and viscosity. The time required for each test precludes the possibility of rapid progress in this field of lubrication study. It is therefore hoped that this paper will stimulate research work relating to wear testing in order that the combined efforts of a large number of investigators may serve to accelerate the growth of fundamental knowledge on this subject.

In conclusion, it is believed that the data presented definitely show that friction and wear are not correlated and that a high degree of "oiliness" does not necessarily result in low wear.

SWEDISH PRACTICE IN THE STANDARDIZATION OF LUBRICATING OILS

By Professor Evert Norlin*

With the wider application of physics and chemistry to the manufacture of lubricating oils, the older system of buying lubricants according to their trade marks is being gradually superseded by methods of selection based on quality tests. In this connexion the buyer either wants to select the grade of oil best suited to his purpose and available at a price below a definite limit, or he desires to purchase the cheapest oil which will give full satisfaction for the purpose in question. In the former case a point calculation is made in the specification. It is also possible to use a sliding price scale, in which case the price is put into relation to the point sum. In the latter case limiting conditions are set up with which the oil must comply, and the cheapest oil is then chosen.

Both these methods have their advantages and their drawbacks. In the author's opinion, the method using fixed limit values is more advantageous than the method based on point calculation, for the reason that the latter method can result in giving too much weight to some comparatively insignificant properties of the oil, so that it may lead to oils which are not very suitable for practical use. In drawing up a specification, a compromise always must be made between the various requirements and properties. For instance, from the point of view of starting the engine, a winter oil must be made thinner than is advantageous for the lubrication of the hot engine. As a specification with fixed limit values for the requisite properties of the oil leads to a grade of oil which is satisfactory in all important respects, provided that the requirements are properly adjusted, this form of specification is used in Sweden as well as in most other countries, although the point calculation method is also employed here and there. Moreover, different classes of quality may be included in the specification, so that machines running at lower speed or subjected to moderate loads are lubricated with oils of lower quality, while a class of high-quality oils is provided for machinery operating at very high speeds or under heavy loads.

In working out oil specifications, it happens that their scope varies considerably. Sometimes they contain a large number of regulations relating to almost all properties which can be controlled by laboratory tests. From various points of view, however, it is better to confine the

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* Government Testing Institute of Sweden, Stockholm.

number of stipulations to the most important properties of the oil. In that case, the control tests will be cheaper and may be carried out more quickly. The specifications must be practical, and they should be adapted to the needs of engineering and to the possibilities of the oil refineries. One should seek to obtain, for each particular purpose, a fully satisfactory grade of oil ensuring a sufficient margin of safety, but it is not necessary to use expensive grades of unnecessary quality.

In this connexion the question often arises whether the suitability of an oil grade for practical needs can be ascertained by laboratory tests, as it is often thought that these properties can only be estimated by practical experience. In the author's opinion, the importance of laboratory tests is underestimated, while the value of practical experience is largely overestimated. Comparative tests of lubricating oils made in the laboratory as well as in practical operation, show that it is very difficult to obtain from practice reliable results which may be utilized for the purpose of comparison. The conditions of practical use are so complex that it is difficult to obtain figures for those factors or materials which are to be tested, and measurements often tend to be expensive and uncertain. A large portion of so-called " practical experience " is based on faith and imagination, not on knowledge and well-grounded information. In order to be conclusive, practical experience must be based either on controlled special tests or on comprehensive numerical data subjected to competent statistical treatment.

There are many grades of oil on the market which bear different names, and lubricating oils with fairly divergent properties are sometimes recommended for one and the same purpose. For most uses a comparatively small number of standard oil grades may be used, and the standardization of lubricating oils for some of the principal uses is necessary and useful. In carrying out this standardization, the oils should be classified in a rational manner. Among the various properties, the viscosity may be regarded as the most suitable basis for classification, and from an international standpoint it is advantageous to use the absolute kinematic viscosity at a definite temperature, e.g. 20 or 50 deg. C. This system is employed in Sweden for the Standards for Motor Lubricating Oils issued by the Swedish Association of Engineers and Architects. The oils are denoted by a letter and a figure, namely, the letter M for motor oils and the letter V for rear-axle and gearbox oils, while the figure indicates the kinematic viscosity of the oil in centistokes at 50 deg. C. Lubricating oils for motors are divided into the following classes: M 20, M 40, M 65, M 90, M 115, and M 150. The tolerance allowed in the specification is ±5 centistokes for oils having a viscosity from 20 to 65, and ±10 centistokes for oils with a viscosity from 90 to 150. Oils for gearboxes are classified according to viscosity in two groups, V 200 and V 350, and the tolerance for kinematic viscosity is ±20 centistokes for the group V 200 and ±30 centistokes for the group V 350. The viscosities given in the symbols for motor lubricating oils are selected so as to correspond to the viscosity of most lubricating oils for motor cars existing on the Swedish market. Where two different classes of oils must be given in the specification, this can be indicated in the symbols, e.g. by using a small letter for the lower grade and a capital letter for the higher grade.

Specifications of more or less official character using this method of notation for lubricating oils other than oils for motor cars, are not yet adopted in Sweden, but, following a proposal put forth by the author, this method of notation has been adopted by a large Swedish industrial concern for all mineral oils of some importance. The author intends, moreover, to utilize the above-described system of notation for lubricating oils in further standardization work.

Those grades of lubricating oils and greases which may require standardization, are as follows:—

(1) Dark machine oils (railway carriage oils and black oils).
(2) Spindle oils.
(3) Light machine oils: (*a*) thin; (*b*) thick.
(4) Cylinder oils: (*a*) for saturated steam; (*b*) for superheated steam.
(5) Lubricating oils for internal combustion engines: (*a*) carburetter engines; (*b*) ball-ignition engines; (*c*) Diesel engines.
(6) Compressor oils.
(7) Refrigerating machine oils.
(8) Steam turbine oils.
(9) Gearbox oils: (*a*) enclosed gears; (*b*) open gears.
(10) Transformer oils.
(11) Fatty oils and solid grease (vegetable and animal).
(12) Compounded oils (marine oils and other grades).
(13) Emulsifying oils (oils for drills and milling cutters).
(14) Solid lubricants: (*a*) solid grease and ball bearing grease; (*b*) rolling mill grease and briquettes; (*c*) crude vaseline.
(15) Anti-corrosive oils and greases.

It may be noted that the same oils may be used in several groups mentioned above.

In Sweden the standardization of lubricating oils for motor cars (group 5) has made the greatest progress, and most extensive experience is available in this respect. Therefore, these oils will be taken as an example and dealt with at some length in what follows. The conclusions drawn from the considerations below, may also be applied to other lubricating oils and greases.

The "Standards for Lubricating Oils to be used for Internal Combustion Engines, Gearboxes, and Rear Axles" were published by the Swedish Association of Engineers and Architects in 1935. The technical stipulations laid down in these standards, which have acquired fairly wide popularity in Sweden, are summarized in Table 1.

TABLE 1. SPECIFICATIONS FOR LUBRICATING OILS

	Lubricating oils for internal combustion engines						Lubricating oils for gearboxes and rear axles	
	M 20	M 40	M 65	M 90	M115	M150	V 200	V 350
Specific gravity at +20 deg. C., maximum	0·925	0·925	0·930	0·930	0·930	0·935	0·940	0·945
Viscosity at +50 deg. C., centistokes . .	20	40	65	90	115	150	200	350
Tolerance . . .	±5	±5	±5	±10	±10	±10	±20	±30
Viscosity quota +50/+100 deg. C., maximum	4·5	5·5	6·5	7·2	7·7	8·2	—	—
Flash point (Pensky-Martens), minimum, deg. C.	170	170	180	190	200	200	180	180
Pour point. The oil must flow out of a test tube 20 millimetres in diameter at the temperature, in deg. C.	−20	−15	−10	−10	−0	−0	−0	+5
Coking residue according to Conradson, maximum percentage	0·3	0·3	0·4	0·8	1·0	1·2	—	—
Asphalt insoluble in standard petrol, maximum percentage	—	—	—	—	—	—	0·2	0·2
Oxidation test: Sligh number, maximum	20	20	20	15	15	10	—	—

As regards laboratory control of conformity to requirements given in the specification and the practical value of the results, the author's experience is as follows.

The *Specific Gravity* is particularly easy to determine and may be used with advantage to ascertain whether the quality of the various portions of the same delivery of oil or of the amounts of the same oil delivered at different times is uniform. The viscosity being constant, a grade having a lower specific gravity will give a more advantageous viscosity-temperature curve than a grade of higher specific gravity, and it is, therefore, better in this respect.

The *Viscosity*, or rather the *viscosity curve*, is the most important property of lubricating oils. The influence of the temperature on the viscosity has hitherto been expressed in the specification by the viscosity quota at 50/100 deg. C. This method is unsatisfactory as the viscosity quota is not constant. As is shown in Table 1, it varies considerably with the viscosity, so that in the revised standards it is intended to replace the viscosity quota either by the viscosity index or by the polar height. The former index will probably be used, since the percentage figure in the viscosity index is more instructive for the practical man than the corresponding figure for the polar height. The use of the viscosity index is rendered possible by the computation of this index from the kinematic viscosity at 50 and 100 deg. C., which was made by Mr. Einar Molin, Director of the Laboratory of the Swedish State Railways, Stockholm.* Experience from various sources shows convincingly that where well-refined oils are concerned, those oils which have a high viscosity index give greater resistance to wear in the process of lubrication than oils with a low viscosity index. In the revised specification, a high viscosity index will, among other things, be prescribed for high-grade oils.

The *Flash Point* is in itself of comparatively little practical value and it must be regarded as a very crude criterion for the tendency to evaporation of the oil. This obsolete testing method ought to be eliminated from specifications for lubricating oils. With motor and cylinder oils, it is the direct evaporation of the oil under conditions prevailing in internal combustion engines and cylinders of steam engines, which offers real interest. It appears that the evaporation test devised by Dr. Noach of Berlin gives much better results. A weighed quantity of oil is heated to 250 deg. C. for one hour in a special apparatus, while a current of air is drawn by suction over the oil surface. In this test evaporation figures varying from 10 to 20 per cent have been obtained with motor lubricating oils. This testing method will be examined and, if it proves suitable, adopted for the revised specification.

The behaviour of oil in cold weather (*pour point or congealing point*) is of primary importance for the starting of motors in winter and is, therefore, of special interest for Swedish conditions, particularly as regards Central and North Sweden. Up to now the simple but

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* For a preliminary report, see Teknisk Tidskrift Kemi, 1936, p. 81.

comparatively primitive pour point test in test tubes has generally been used. Vogel's viscometer for low temperatures has also been tried, but so far it is not certain that the pour point test will be replaced by the determination of viscosity at low temperatures, since the former method gives results which are quite satisfactory for practical purposes, and the determination of viscosity at two or three temperatures between −10 and −30 deg. C. is a rather lengthy and expensive procedure.

Coking Residue According to Conradson. It is well known that the practical value of this test is rather questionable. Some experts believe this test to be more or less valueless, whereas others, e.g. the Deutsche Versuchsanstalt für Luftfahrt, have found the coke numbers run in parallel with the amounts of coke formed in the engine. As a result of comparative laboratory tests and examination of engines, the author concludes that the limiting values for the Conradson test should not be regarded too rigorously, though it appears to be certain that oils with a high coke number (2 to 3 per cent) tend to form obnoxious deposits in the engine under unfavourable conditions. It is possible that the permissible values for the coke number will be somewhat increased in the revised specification.

Oxidation Test—Sligh Number. There is no doubt that oils having a high Sligh number are more liable to oxidation by hot air in the crankshaft casing, so that soluble pitch-like substances of slightly acid character are produced, together with asphalt-like components—asphaltenes—which are insoluble and are subsequently deposited in the shape of sludge. The technique of determination of the Sligh number has been improved by the Government Testing Institute of Sweden; * the results obtained are accurate and in good mutual agreement. Refining tests have shown that the Sligh number diminishes when the refining process is carried on. Engine tests show that, for a given mileage, oils with low Sligh numbers contain smaller amounts of oxidation products than oils with higher Sligh numbers. It is to be noted that the Sligh number of motor oils of the same origin belonging to the same oil series, but differing in viscosity, diminishes with increasing viscosity, whereas, on the other hand, the coking residue increases. In the revision of the specification it is desirable and possible to prescribe lower Sligh numbers for higher grades of oil.

The question may now be raised whether the properties included in the specification are sufficient or whether some essential property of the

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* Communications, No. 75, 1937.

oil has been neglected. One might perhaps think of the adhesiveness of the oil to metal surfaces, or, in more general terms, its "oiliness," a concept which is being studied at the present time. It appears, however, that the specific pressures occurring in engine bearings and cylinders are comparatively so low that the oiliness of commercial lubricating oils for internal combustion engines is amply sufficient and does not need to be checked. Amongst machines recently devised for the determination of oiliness, the Timkens and Wells machines deserve special mention, but it is not yet known whether they will be suitable for routine tests.

In Sweden a special committee set up by the Swedish Association of Engineers and Architects, is dealing with the revision of standards for lubricating oils for internal combustion engines. Another committee appointed by the Swedish Standardization Commission is working on the standardization of testing methods for lubricating oils and on Swedish oil nomenclature. Each of these committees will draw up more or less official Swedish conditions of delivery for some important grades of lubricating oils and grease. Such conditions of delivery are used by various State institutions and large private companies, so that experience is available in respect of standardization. In order to be able to arrive at a rational system of standardization of conditions of delivery, it will first be necessary to standardize the testing methods.

Finally, it is to be observed that the conditions of delivery for lubricating oils should not be considered as fixed for a long time to come. On the contrary, they must be revised from time to time and should be adapted to technical development taking into account both the requirements of practical use in machinery, etc., and the possibilities of oil refineries. In this connexion it is also to be noted that the designers of machinery, etc., should not set up too severe requirements for lubricating oils. The lubricating devices should be designed in such a way that the oil is not subjected to excessively heavy mechanical stresses or to the action of the air at too high temperatures.

STATIC FRICTION AND BOUNDARY LUBRICATION

By M. E. Nottage, B.Sc.*

Friction between smooth polished surfaces is caused by the unbalanced forces of molecular attraction of the surfaces acting across the interface; the function of a lubricant is to neutralize these unbalanced forces. When the surfaces are separated by a layer of lubricant which is thick enough to place the plane of slip beyond the influence of the enclosing solids, the frictional resistance is due solely to the internal friction or viscosity of the lubricant and the lubrication is said to be of the "complete" or "fluid" type. But when the lubricating film is so thin that every part of it lies within, and has its properties modified by, the strong attraction forces of the solid surfaces, the frictional resistance at the surface of slip is due mainly to these attraction forces and the lubrication is said to be of the "incomplete" or "boundary" type.

Boundary lubrication gradually merges into fluid lubrication as the lubricating film increases in thickness. The work of the late Sir William Hardy and his colleagues (1922 a, b; 1925) has shown that it is possible to distinguish two regions in boundary lubrication:—

(*a*) A region within which the lubricating film is wholly under the influence of the attraction forces of the solid surfaces—it is probable that this region extends only to a molecular thickness beyond the solid surface.

(*b*) A transitional region lying between that described above and that in which the lubrication is complete, a region in which the characteristic properties of the limiting film are gradually lost.

In a series of papers published recently, King (1933) has produced experimental results which suggest that boundary conditions in journal-bearing lubrication are not entirely hypothetical. He suggests that under conditions which promote oxidation in the oil, orientated layers of oxidized molecules are built up on the bearing surfaces, the plane of slip lying between them, and, as a result of this, seizing, which occurs when the bearing surfaces come into contact through failure of the oil film, is postponed until a higher temperature is reached.

It is evident that from a practical point of view the most important property of a lubricant under boundary conditions is its ability to resist film destruction. This property is a function of the chemical composition of the lubricant, and experiments carried out by Hardy and his

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* Research Worker, Building Research Station, Garston, Watford.

colleagues (Hardy and Nottage 1929; Nottage 1934) have opened up a very interesting field for investigation. The experimental method was simple. Clean steel surfaces (consisting of a plane plate and a spherical slider) were lubricated with the oil to be tested, and the force needed just to set the slider in motion was measured when the whole system (bearing surfaces and oil) was heated to various temperatures between 18 deg. C. and 106 deg. C. and in the two atmospheres, air and nitrogen. Hence the friction considered was the static or starting friction and it was measured on the limiting adsorbed layer.

Commercial lubricating oils consist of complex mixtures of chemical compounds whose lubricating properties vary within wide limits. To investigate the effect of variation in the relative proportions of good and bad lubricants in a mixture, preliminary experiments were made with synthetic mixtures of two substances which were each of known lubricating properties and were incapable of reacting with one another chemically. Examples of such mixtures are caprylic acid and undecane, lactic acid and water. The results showed that when the proportion of the more polar or surface-active substance is greater than about 0·7 per cent, it dominates the adsorption layer and the coefficient of friction of the mixture approximates to that of this constituent. This well-known phenomenon is expressed by the statement that the more polar or surface-active substance is selectively adsorbed by the solid surface. For this reason the composition of the adsorption layer may be, and generally is, very different from that of the mixture in bulk. The fact is of great importance when commercial lubricants become subjected to boundary conditions. It means that if an oil contains even a low percentage of a surface-active constituent its coefficient of friction will be lower, and its resistance to film destruction higher, than if it contained no such constituent.

The statement that the coefficient of friction on the limiting absorbed layer has no temperature coefficient is true for a pure substance both above and below its melting point; it ceases to be true when the chemical composition of the adsorbed layer changes. Such a change is produced when the relative proportions of the various constituents composing the adsorption layer depend on the temperature; also when these constituents undergo polymerization or oxidation.

The investigation of the effect of temperature on the composition of the adsorption layer was first carried out with simple synthetic mixtures. As a base for these mixtures the medicinal paraffin of the British Pharmacopœia was used. This oil, henceforward referred to as oil B.P., is the residue left after removal of the surface-active constituents from a mineral oil and, although undoubtedly it is a mixture of many constituents, it behaves as a single chemical substance so far as its lubricating properties are concerned and gives a coefficient of friction of 0·228 which is constant between 18 deg. C. and 106 deg. C. and is independent of the atmosphere (air or nitrogen) in which the system is heated. Varying small amounts of (*a*) non-polar and (*b*) polar constituents were added to this oil and the results are shown in the friction-temperature curves (Figs. 1 and 2).

The addition of a non-polar constituent such as a wax or a paraffin hydrocarbon (Fig. 1) produces no effect until its melting point is approached, when the lubricating value of the mixture changes quickly to some value intermediate between that of the oil B.P., and that of the wax or paraffin. It is possible that, below the melting point, the degree of dispersion of the wax or paraffin in the oil B.P. is so low that the molecules possess very little free surface energy and are incapable of being drawn into the adsorption layer; above the melting point the dispersion increases and with it the free surface energy. On cooling the system to the original temperature the reverse effect is produced.

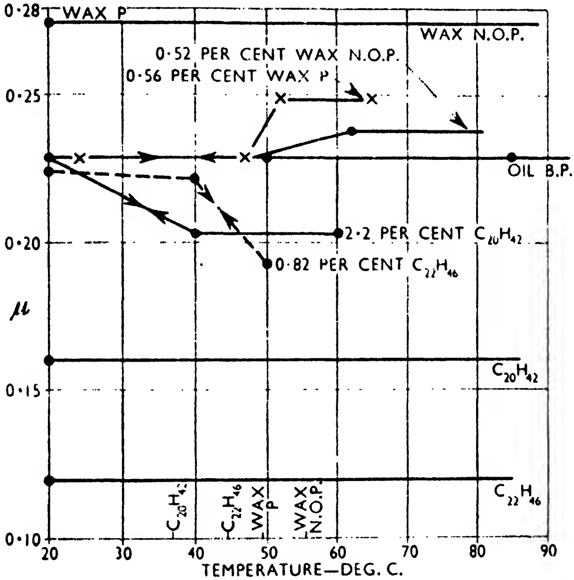

Fig. 1. Friction-Temperature Curves Obtained with Oil Containing Non-polar Constituents

When a polar constituent such as palmitic acid is added to oil B.P. (Fig. 2), the coefficient of friction decreases at first rapidly and then more slowly with rise of temperature until a value is reached which

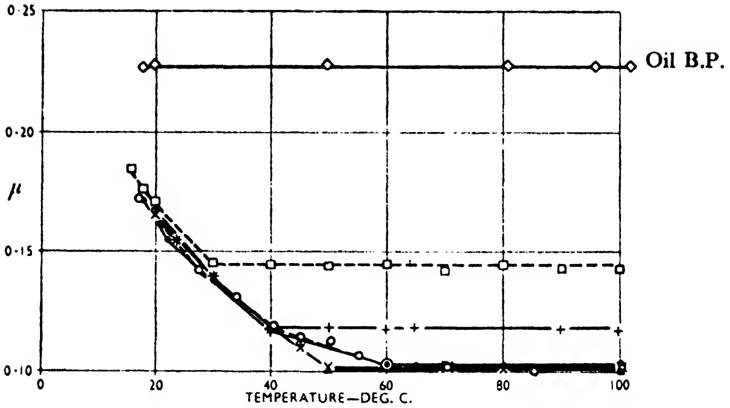

Fig. 2. Friction-Temperature Curves Obtained with Oil Containing Polar Constituents

Percentage of Palmitic Acid :—

—— × ——	6·08	—— ● ——	0·97
—— △ ——	2·41	—— + ——	0·33
- - - ⊙ - - -	1·50	- - - □ - - -	0·014

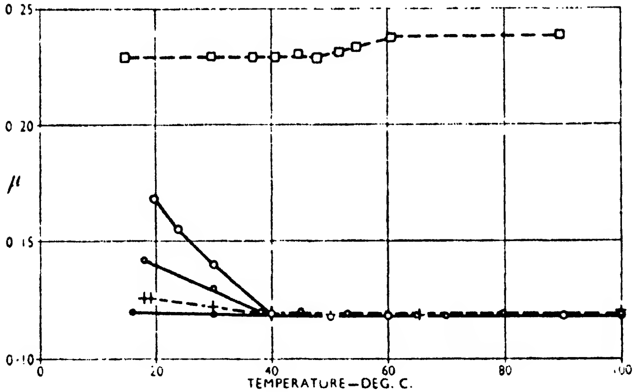

Fig. 3. Effect of Adding Wax (melting point 54–57 deg. C.) to a Mixture Containing Polar Constituents

Percentage composition :—

	B.P. oil	Palmitic Acid	Wax
- - - - □ - - - -	99·48	0	0·52
——— ○ ———	99·67	0·33	0
——— o ———	99·09	0·34	0·57
- - - - + - - - -	96·97	0·34	2·69
——— ● ———	90·06	0·34	9·60

remains steady up to 106 deg. C. The curves for all mixtures containing more than 0·97 per cent of the acid are practically coincident, i.e. the maximum saturation value of palmitic acid in the adsorption layer is reached when the mixture contains this percentage of acid. These mixtures differ from those containing non-polar constituents in that no reversion to the original lubricating value occurs when the system is cooled down to the original temperature.

The addition of wax to the mixture of oil B.P. and palmitic acid is instructive (Fig. 3). Although the coefficient of friction of the wax ($\mu=0{\cdot}275$) is higher than that of either of the other constituents (oil B.P. $\mu=0{\cdot}228$; palmitic acid $\mu=0$) its effect on the mixture is to produce a considerable increase in the lubricating value at temperatures

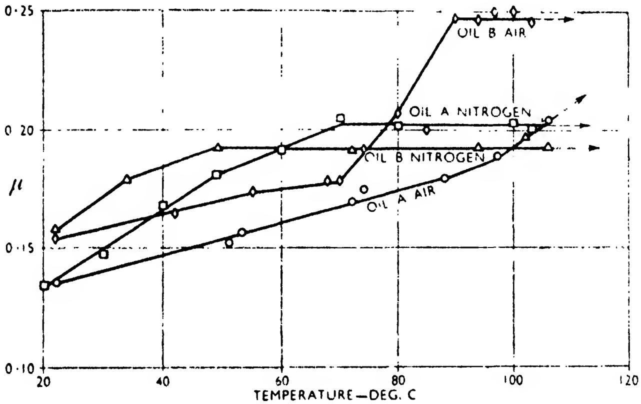

Fig. 4. Friction-Temperature Curves of Two Mineral Oils

below 40 deg. C., i.e. it eliminates the variation of friction with temperature. It is possible that the wax increases the degree of dispersion of the acid in oil B.P. and so enables the palmitic acid to gain full access to the adsorption layer at a lower temperature. This experiment serves to illustrate the significant fact that it is not possible to predict the lubricating properties of a mixture from a knowledge of the coefficients of friction of its constituents.

The failure in practice of many commercial oils is due to the formation of gummy substances on the bearing surfaces when the latter become heated. Ordinary physical and chemical tests made on the oil in bulk may fail to reveal this tendency to gum when the oil forms a thin film on the solid surface. The case may be quoted here of two different mineral oils of approximately the same viscosity and density which were found to behave very differently under test. Oil A under a pressure of 2,500 lb. per sq. in. gave a minimum coefficient of friction of 0·001 at 150 deg. C. and a seizing temperature of 224 deg. C., whilst oil B gave a minimum coefficient of friction of 0·0018 at 80 deg. C. and a seizing temperature of 94·5 deg. C. (Fig. 4). Since the curves obtained by heating the two oils in nitrogen are very similar, whilst that obtained with oil B by heating in air shows a rapid increase in friction above 70 deg. C. which does not occur with oil A, it is evident that oil B contains some constituent which forms a gum when a thin film of the oil is heated in an oxidizing atmosphere. The effect of the addition to oil B of the highly polar compound palmitic acid is shown in Fig. 5. When present in sufficient quantity, the acid completely prevents the formation of gum on the bearing surfaces. It is probable that the acid

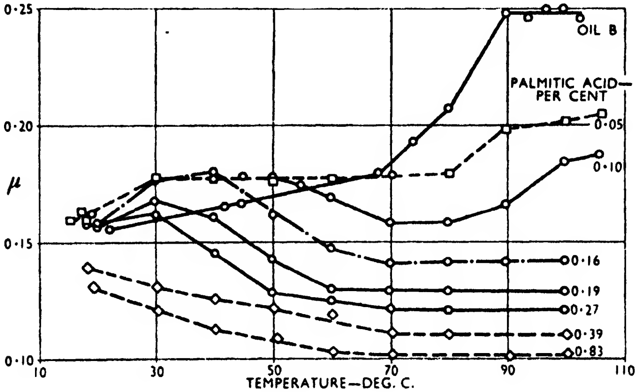

Fig. 5. Effect of Added Palmitic Acid on the Friction-Temperature Curve of a Mineral Oil

Readings in air.

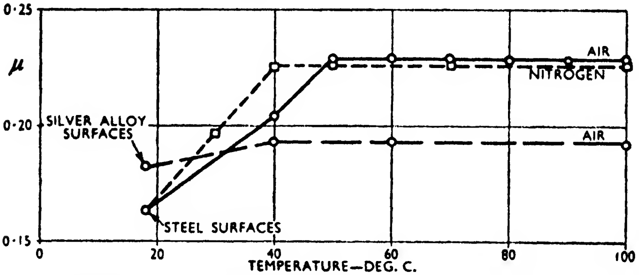

Fig. 6. Effect of the Nature of the Bearing Surface on the Constitution of the Adsorption Layer

is adsorbed from the mixture in preference to the gum-forming constituent. This effect of resistance to film destruction produced by the addition of a fatty acid to a mineral oil forms the basis of the "germ process" for the improvement of mineral oils, due to Messrs. Southcombe and Wells.

, Since boundary friction is due to the cohesive forces of attraction of the bearing surfaces transmitted through the adsorption layer formed by the lubricant, its magnitude is governed by two factors, namely, (*a*) the constitution of the adsorption layer, and (*b*) the intensity of the attraction field.

The experiments described above were directed towards the investigation of the first of these factors, the second being kept constant by the use of steel surfaces throughout. To elucidate the effect due to a variation in the intensity of the field, the friction-temperature curves were compared when bearing surfaces of steel and of silver-alloy were lubricated with the same oils. The results are shown in Fig. 6.

The friction-temperature curve for a pure substance is linear and parallel to the temperature axis and hence a change from one bearing surface to another merely shifts the curve parallel to itself. This is not always true, however, when the lubricant consists of a complex mixture such as a mineral oil. It is evident that the nature of the bearing surface may influence not only the constitution of the adsorption layer but also the changes which occur in it when the system is heated.

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AN EXPERIMENTAL STUDY OF LUBRICATION UNDER CONDITIONS OF EXTREME PRESSURE

By James E. Southcombe, M.Sc., Justin H. Wells, B.Sc.,
A.I.Mech.E., and John H. Waters*

When high-speed gearing, running under loads and speeds which give rise to very intense pressures, is lubricated with pure mineral oils a certain amount of scuffing of the teeth occurs. It is found, however, that when the oil is compounded with an organic derivative containing sulphur or one of the halogens, great improvement is observed in the lubrication of such gears, there being diminished "scuffing" or seizure. The extreme-pressure lubricants prepared in this way have been described as possessing "high film strength", and the following paper presents the results of some lubrication experiments which were made under conditions of extreme pressure with a view to throwing light on the property of film strength.

G. A. Tomlinson (1927) investigated the effect produced by rubbing steel balls against a plane surface with very small displacements and observed the formation of debris consisting of oxide of iron. Applying the Hertz equations, he calculated the mean intensity of pressure for various loads on balls of different diameters. When a curved surface is pressed on to a plane surface without translation the deformation is elastic and no permanent mark visible to the microscope is left after the application of pressures up to the elastic limit of the material, but Tomlinson found that very slight motion of translation produces a permanent mark at relatively low loads and this "scarring" is influenced by the presence of a contaminating film on the solids. When a polished steel ball was rubbed to and fro on a polished steel plate smeared with castor oil, no oxidation debris was formed when the intensity of pressure was 30,000 lb. per sq. in.; but 60,000 lb. per sq. in. caused the surface to show wear. Tomlinson calculated the intensity of pressure for spheres pressed on to a plane and determined the coefficient of friction at various intensities for glass on glass and steel on steel. Referring to a fused glass sphere on a plane surface of the same material he says, "The first point of interest is that there is a critical intensity of about 65,000 lb. per sq. in. at which the film appears to suffer rupture. Below this stress nearly all the observed values of the coefficient fall between limits 0·09 and 0·11, the mean being closely 0·10. With higher stresses a clear majority of the points

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* Germ Lubricants, Ltd.

lie well above 0·11 and above the critical stress the values obtained are far more erratic. The inference is that the rupture of the film is in irregular patches the form and size of which vary widely in different experiments. In the case of hard steel on steel the results with lower stresses than the critical value are more erratic and it is thought that occasionally a certain degree of solid cohesion occurs with stresses well below 65,000 lb. per sq. in. Some experiments may be mentioned in which a smear of castor oil was used and the rupture of the film was judged by a microscopic examination of the surface afterwards; it was found that in certain cases with a stress as low as 30,000 lb. per sq. in. there had occurred some definite solid cohesion."

The data published on the friction of lubricated surfaces in relation to small contact areas at high pressure, appear to be scanty, but it has been stated that the area of a scar produced by the traction of steel balls against a plane surface is of the order of the calculated contact area formed by elastic deformation. It appeared desirable, therefore, to obtain further experimental evidence on this point.

For the preliminary experiments the investigation was confined to a study of the effect of rotating a cylindrical surface against a stationary plane by means of the Timken wear and lubricant tester. In this machine a polished steel rectangular block $\frac{3}{4}$ inch long by $\frac{1}{2}$ inch wide, carried on a system of levers, is pressed against a rotating disk of polished steel 2 inches in diameter, giving a line of contact $\frac{1}{2}$ inch in width; means are provided for varying the load and speed, and the area of the wear scar is measured with a microscope.*

When a cylindrical roller is pressed against a plane surface, values for the elastic contact area and the maximum intensity of pressure can be obtained on applying the formula of Timoshenko (1934), namely, $b=2{\cdot}15\sqrt{PD/E}$, the maximum intensity being $0{\cdot}59\sqrt{PE/D}$, where b denotes the breadth of rectangular contact surface, P the pressure per inch length of roller, D the diameter of the roller in inches, and E the modulus of elasticity (40×10^6 for Timken steel). The "average" intensity along the line of contact is given by P/A, where A is the calculated contact area (elastic). It should be noted that the wear area on the Timken block is not rectangular but wedge-shaped. The area has been calculated from the mean width and as this is not always strictly accurate the measurements of area can only be considered as close approximations. Fig. 1 shows the relationship of beam load to scar area, maximum intensity, and average intensity. Applying these formulæ to a Timken cup pressed against a plane test block of 60 deg. Rockwell hardness, the contact area (elastic), the values for

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* For comparison, the wear of the block only was taken.

the average intensity of pressure, and the maximum intensity for any applied load can be found. For a beam load of 10 lb. (115 lb. dead load) the calculated elastic area of contact is 0·0036 sq. in. and the average intensity of pressure is 32,000 lb. per sq. in., the maximum intensity at the centre of contact being 41,000 lb. per sq. in. This load was applied without motion or translation and no visible scar appeared. The surfaces were then flooded with castor oil and the roller slowly rotated through the distances given in Table 1, the scar which then formed being measured.

TABLE 1. RESULTS WITH 10 LB. BEAM LOAD (115 LB. DEAD LOAD) AT 100 DEG. F.

Distance moved, inches	Wear scar area		Average intensity, lb. per sq. in.	
	Castor oil, sq. in.	Mineral oil A, sq. in.	Castor oil	Mineral oil A
1 . .	Blurred	0·006	—	19,300
6* . .	0·0048	0·008	24,170	14,400
12 . .	0·0074	0·0116	15,500 (approx.)	10,000 (approx.)
30 . .	0·0076	—		
60 . .	—	0·0112		
120 . .	0·0072	—		

* See Figs. 13 and 14.

It is clear that a small traction of only 6 inches produces a wear scar somewhat greater than the area of elastic deformation, probably owing to lack of perfect rigidity in the machine. When sufficient wear occurs to reduce the intensity to, say, 16,000 lb. per sq. in., the castor oil appears to prevent further wear. It should be noted that in this experiment the calculated intensity of the applied pressure (32,000 lb. per sq. in.) is above the limiting stress found by Tomlinson for castor oil and the results seem to comfirm his statement that the film can rupture at this stress. The rate of wear scar formation was then measured on the Timken block with different lubricants at 400 ft. per min. with 215 lb. dead load. The calculated elastic maximum intensity was 56,000 lb. per sq. in. and the area 0·0049 sq. in.=44,000 lb. per sq. in. average intensity. Fig. 2 shows that with a wide range of gear oils there is almost instantaneous wear and at the end of a couple of minutes the block has worn to a mean contact area of 0·00175 sq. in. for several oils. After this area is produced the wear may become negligible. This includes oils containing so-called extreme-pressure addition agents and apparently if the extreme-pressure agent is very active the wear scar is greater before equilibrium is established.

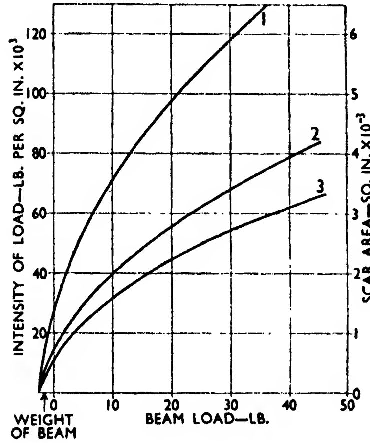

Fig. 1. Relationship of Beam Load to Scar Area

1 Calculated scar area (elastic).
2 Maximum intensity (elastic).
3 Average intensity (load/area).

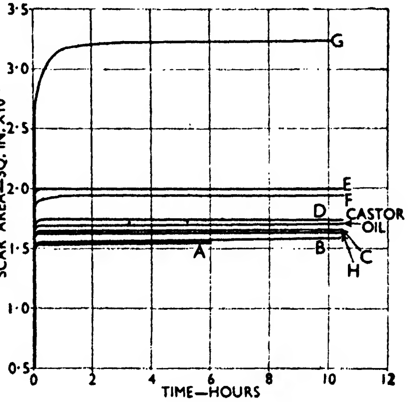

Fig. 2. Rate of Wear

Temperature of oil, 100 deg. F.; speed, 800 r.p.m.; load on beam, 20 lb. (215 lb. dead load).

For key to reference letters, see Appendix I, p. 410.

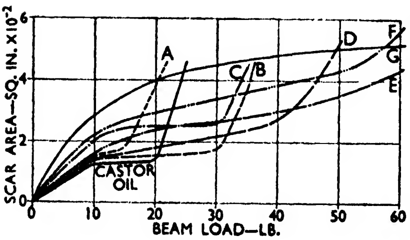

Fig. 3. Effect of Load on Wear after One Hour's Running

Temperature of oil, 100 deg. F.; speed, 800 r.p.m.

For key to reference letters, see Appendix I, p. 410.

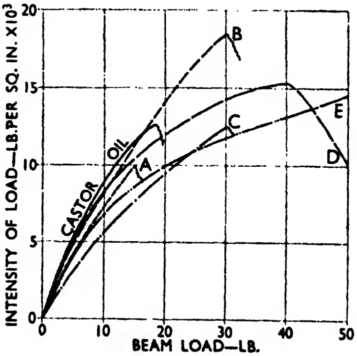

Fig. 4. Relationship between Beam Load and Load Intensity after One Hour's Running

For key to reference letters, see Appendix I, p. 410.

Interest attaches to the curve for oil A (Fig. 2). As this is a pure mineral cylinder oil, which normally will not carry without seizure a 20 lb. Timken beam load applied for 10 minutes, it was run at a beam load of 10 lb. for 2 hours; the scar area was 0·016 sq. in., and the intensity 7,250 lb. per sq. in. The beam load was now increased to 20 lb., but the scar area did not increase (intensity 13,500 lb. per sq. in.) after a further run of one hour.

Figs. 3 and 4 show the effect of load on the scar from one hour's run with various lubricants. Inspection of these graphs shows that at 400 ft. per min. four oils of different constitution lie closely together for all beam loads up to 10 lb., giving a mean value for the contact area at this load of 0·014 sq. in. and a mean intensity of 8,300 lb. per sq. in. With increments in load the wear remains fairly constant until a point is reached where any further addition to the load results in seizure; the point where the curve rises steeply is described as the maximum permissible load (Table 2).

TABLE 2. RELATIVE INTENSITIES AT MAXIMUM PERMISSIBLE LOAD

Oils*	Maximum permissible dead load, lb.	Scar area, sq. in.	Intensity, lb. per sq. in.
A	165	0·018	9,170
Castor	195	0·016	12,200
B	315	0·02	16,250
C	315	0·025	12,600
G	No seizure at 615	0·052	12,000
A (sulphurized)	415	0·05	8,300
F	465	0·046	10,500

* For specifications of the oils, see Appendix I, p. 410.

It is noteworthy that in no case can the average intensity exceed about 17,000 lb. per sq. in. without seizure. As soon as the loading approaches this average intensity, either seizure occurs or relief is found by wear. If the oil does not contain an extreme-pressure agent, seizure will occur at a point well below 17,000 lb. per sq. in. average intensity. An oil which carries heavier dead loads does so simply by allowing wear to take place smoothly. The property of certain lubricants which has been called "high film strength" does not appear to exist, and the authors conclude that an extreme-pressure lubricant must be defined as one which permits the metal to wear away smoothly when the intensity approaches a certain limiting value. This conclusion applies to a plane surface at rest which is rubbed along a narrow line. Considered geometrically, a pair of gear teeth may be represented as two cylinders, in contact and rotating with some slip, the percentage of slip to rolling motion varying with the design. It is realized that this form of relative motion involves considerations which complicate the treatment of the simple case of a rotating cylinder on a plane surface, but in the present paper only the latter case is considered.

Composition of Detritus. In the experiments carried out by the authors under conditions of extreme pressure using the Timken machine, the Faville Levally machine, and a machine recently designed by one of the authors, there is generally found floating in the oil a minute amount of wear detritus in the form of very fine powder. This has been collected and examined chemically. When lubricating with pure mineral oils this detritus consists of oxide of iron and steel dust. Tomlinson also noted the formation of oxide of iron resulting from the traction of steel balls on slip gauges lubricated with castor oil and with vaseline.

The wear detritus produced when steel surfaces are lubricated with an extreme-pressure oil blended with organic sulphur compounds always contains appreciable amounts of ferrous sulphide, and oils containing chlorinated hydrocarbons or organic chloro-compounds leave ferrous chloride in the detritus. Bowden and Ridler (1936) have shown that when two metals are rubbed together at speeds varying from 1 metre to 10 metres per minute very high temperatures are developed locally by friction of the "high spots", and Blok (1937) confirms this and observes local temperatures as high as 1,000 deg. C. on highly stressed gear teeth. At such temperatures desulphurization or dehalogenation of the organic molecules will occur, the sulphur or halogen liberated instantly combining with the metal to form a "flux" of ferrous salts which prevent seizure, as Professor Earle Buckingham (Hersey 1936) has suggested.

TABLE 3. EFFECT OF SPEED ON SEIZING LOAD

Speed, ft. per min.	Breakdown loading (seizure) on beam load, lb.		
	Castor Oil	Oil C	Oil D
100	50+	50+	50+
250	40	40	50
400	20	35	50
550	10	15	30

Apparently the breakdown load on the Timken machine depends somewhat on the rate of decomposition of the organic chloro-compound. Lincoln, Byrkit, and Steiner (1936) have shown, for example, that the very stable compounds in which chlorine is fixed in a benzene ring, give lower Timken values than do those less stable compounds containing the halogen atom attached to a side chain.

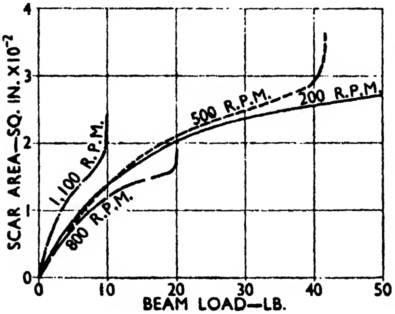

Fig. 5. Influence of Speed on Wear after Ten Minutes' Running

Castor oil; temperature, 100 deg. F.

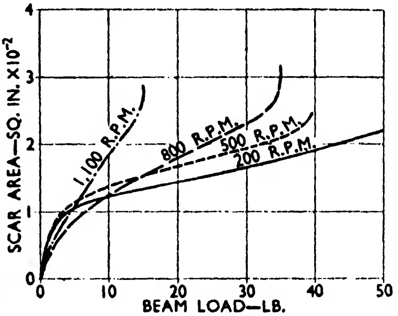

Fig. 6. Influence of Speed on Wear after Ten Minutes' Running

Oil C; temperature, 100 deg. F.

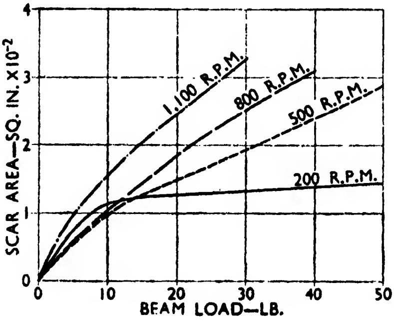

Fig. 7. Influence of Speed on Wear after Ten Minutes' Running

Oil D; temperature, 100 deg. F.

Influence of Speed on Wear of the Timken Block. Figs. 5, 6, and 7 show the influence of speed on wear of the Timken block at various loads for castor oil and for two proprietary extreme-pressure oils. Up to 10 lb. beam load (115 lb. dead load) the area of the wear scar is fairly similar for all three oils at speeds up to 800 r.p.m. (400 ft. per min.) the intensity falling to between 9,000 and 11,900 lb. per sq. in. At higher loading there is a spread of the curves with increasing speed which is particularly noticeable with oil D. The seizing load becomes lower as the speed increases, as is shown in Table 3 (p. 405).

Influence of Oil Bath Temperatures. Figs. 8 and 9 show that within the limits of 70–200 deg. F. the temperature of the oil bath has but little influence on the wear or seizing load. Above a certain limit the viscosity of the oil does not appear to have much influence.

Coefficients of Friction. A few years ago one of the authors (Wells 1929) designed an apparatus for measuring the kinetic coefficient of friction in the "boundary" or semi-fluid state, and a modification of the original machine is shown in Fig. 10. In this device two circular disks immersed in an oil bath are pressed together under a known load, the bottom member being rotated by a motor while the frictional torque on the top surface is measured. The area of contact of the top surface can be varied from 0·1 sq. in. down to point contact of three balls, giving an extremely wide variation in intensity of load. The test pieces and oil bath can be heated electrically to 250 deg. C.

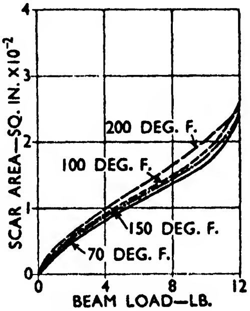

Fig. 8. Effect of Load on Wear at Various Temperatures, after Ten Minutes' Running

Oil A; speed, 800 r.p.m.

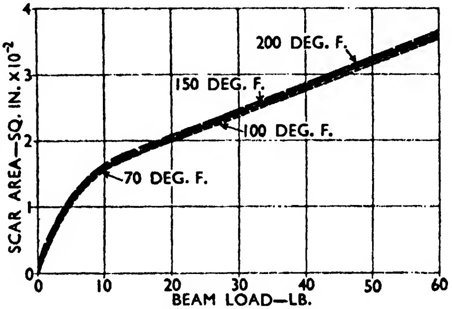

Fig. 9. Effect of Load on Wear at Various Temperatures, after Ten Minutes' Running

Oil D; speed, 800 r.p.m.

———— 70 deg. F.
— — — — 100 „
—— · —— 150 deg. F.
—— —— 200 „

Figs. 11 and 12 show the effect of temperature on the coefficient of friction for a typical mineral gear oil and the same oil containing extreme-pressure addition agents under two widely differing intensities of load, i.e. 130 lb. per sq. in. and approximately 80,000 lb. per sq. in. At low load-intensity there is a sharp rise in the friction at about 180 deg. C. for the mineral oil and the addition of a chloro-compound has no marked influence on the temperature–coefficient of friction diagram until about 180 deg. C. is reached.

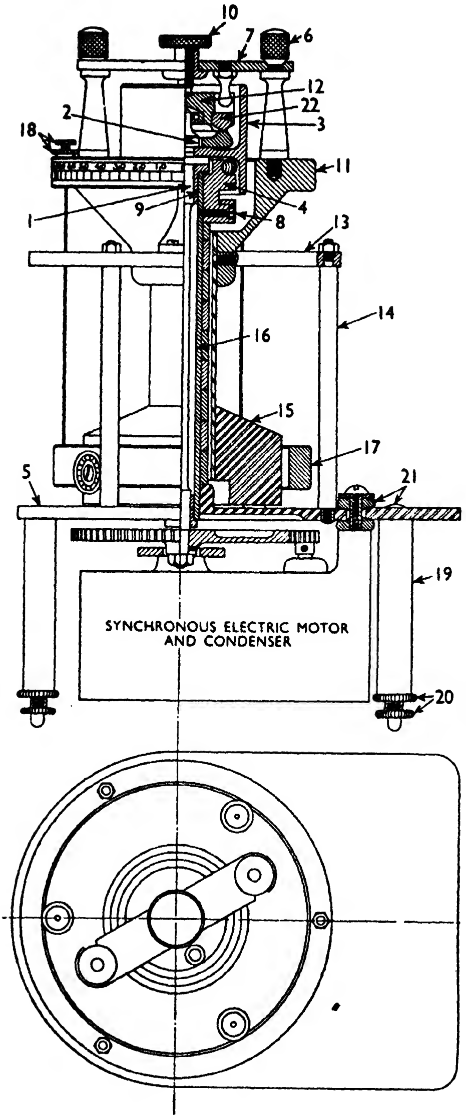

Fig. 10. "Germ" Oiliness Comparator

1 Vertical driving spindle.
2 Centring peg for bottom test piece.
3 Oil bath container.
4 Electric heater element.
5 Baseplate.
6 Clamping nuts for cross piece.
7 Cross piece.
8 Fixing screw for electric heater element.
9 Brass bush.
10 Lifting screw.
11 Brass head with scale.
12 Holder for top test piece.
13 Brass ring holding pointers.
14 Mild steel legs for brass ring.
15 Suspended weight.
16 Mild steel centre tube.
17 Aluminium torsion ring.
18 Adjusting screws for suspension threads.
19 Brass legs supporting base.
20 Levelling screws.
21 Rubber insertion washers.
22 Top test piece.

Heating the mineral oil with 1 per cent of sulphur gives a product showing much higher friction than the original oil; the extreme-pressure agent does not improve the efficiency or oiliness of the lubricant at low load. The characteristic curve for an oil containing a trace of fatty acid is included for comparison. At very high intensity of load the coefficient of friction (Fig. 12) is much higher for a mineral oil and for an oil containing sulphur, but the presence of a chloro-compound brings about a fall in the coefficient at about 150 deg. C., at which temperature the oil loses chlorine, hydrochloric acid being evolved.

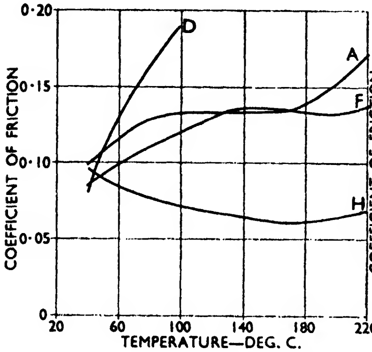

Fig. 11. Effect of Temperature on Coefficient of Friction: Low Loading

Intensity of load, 130 lb. per sq. in.

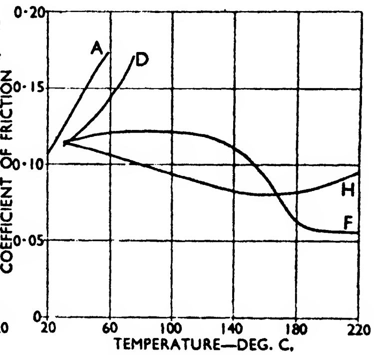

Fig. 12. Effect of Temperature on Coefficient of Friction: Intense Loading

Intensity of load, about 80,000 lb. per sq. in.

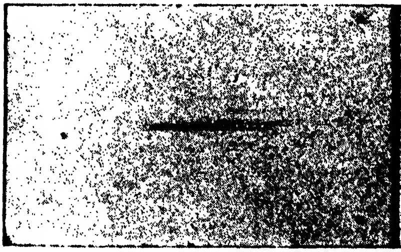

Fig. 13. Wear Scar with Castor Oil

Movement, 6 inches.

Fig. 14. Wear Scar with Oil A

Movement, 6 inches.

At the end of an experiment, a wear spot appears on the balls. In a typical experiment, using mineral oil, this area of wear was 0·00011 sq. in., giving for the three balls an average intensity of 18,000 lb. per sq. in.

The addition of a chloro-compound increases this area to 0·0002 sq. in., giving an intensity of only about 10,000 lb. per sq. in., thus confirming the experiments described above with the Timken apparatus.

All the extreme-pressure addition agents studied by the authors are characterized by considerable chemical instability at high temperatures, and it would appear that this is essential to their operation.

The authors wish to thank the directors of Messrs. Germ Lubricants, Ltd., in whose laboratory these experiments were carried out, for permission to publish these results.

APPENDIX I

SPECIFICATION OF THE OILS TESTED

Oil	Composition	Viscosity, sec. Redwood at 140 deg. F.
A. . . .	Mineral oil	950
B. . . .	Sulphurized mineral oil compounded with glycerides	970
C. . . .	,,	740
D. . . .	,,	920
E. . . .	,,	510
F. . . .	Mineral oil plus 5 per cent chlorinated paraffin wax	950
G. . . .	Mineral oil containing sulphur and lead soap	890
H. . . .	Mineral oil containing free fatty acid	950

APPENDIX II

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PHYSICAL PROPERTIES OF LUBRICANTS

By Yoshio Suge *

With the object of ascertaining the physical meaning of "oiliness", the following physical properties of lubricants have been investigated.

VISCOSITY AT HIGH PRESSURES AND TEMPERATURES

The viscosities of thirty well-known brands of oils, including lubricating oils of mineral, vegetable, and animal origins and also fuel oils, were measured up to a pressure of 2,000 kg. per sq. cm. at various temperatures between 0 and 150 deg. C. The falling-sphere method was adopted, the true viscosity η_0 being given by the formula $\eta_0=\eta(1-x)^{\frac{2\cdot 1}{1-x^2}}$, where η is the apparent viscosity given by Stokes's formula, and x the ratio of the diameter of the sphere to that of the tube. This formula is valid for values of x up to $\frac{1}{3}$. The time of fall was observed with the aid of a high-frequency circuit, based upon the principle that the fall of the metallic sphere caused a change of capacity. The apparatus is illustrated in Figs. 1 and 2, and the circuit is reproduced in Fig. 3. Some of the results are shown in Fig. 4, in which the ordinate η_p/η_0 represents the ratio of the viscosity coefficient at a pressure p to that at normal atmospheric pressure. In Fig. 4, the curves are numbered according to the numbers of the samples.

An empirical formula $\eta_p=\eta e^{\alpha p}$ was obtained which applies at any temperature and in which η is the coefficient of viscosity at atmospheric pressure, η_p that at a pressure p, and α a constant depending upon the nature of the oil so far as the temperature is kept constant. This constant α, the pressure coefficient, varies with the temperature as expressed by $\alpha=\alpha_0 e^{\beta\left(\frac{1}{T}-\frac{1}{T_0}\right)}$, in which α_0 is the pressure coefficient at an arbitrary standard temperature T_0 abs. (18 deg. C., that is 291 deg. abs., was chosen for all oils examined), and T abs. is any other temperature. The values of β are smaller for animal and vegetable oils than for mineral oils. At high temperatures, as is seen from the formula, α tends to a finite value. Although the values of α_0 and β are different for different oils, it is found that the ultimate values of α for vegetable, animal, and mineral oils are almost the same. This implies that the viscosity of all oils at high temperatures is similarly affected by the pressure, as is shown by the formula $\eta_p=\eta e^{\alpha p}$ where α is constant.

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* Kinoshita Laboratory, Institute of Physical and Chemical Research, Tokyo.

The relation between the temperature and the viscosity is expressed by the formula $\eta=\eta_0 e^{b\left(\frac{1}{T-\theta}-\frac{1}{T_0-\theta}\right)}$ where b is a constant, η_0 the coefficient of viscosity at T_0 deg. abs. and at atmospheric pressure. The temperature θ deg. abs. was found to be very nearly the same for all oils, and therefore, for all practical purposes, it may be considered as a constant by putting $\theta=200$ deg. abs.

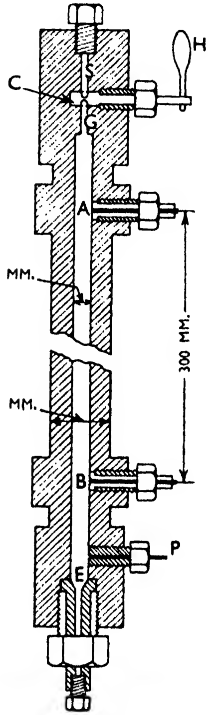

Fig. 1. Falling-Sphere Apparatus

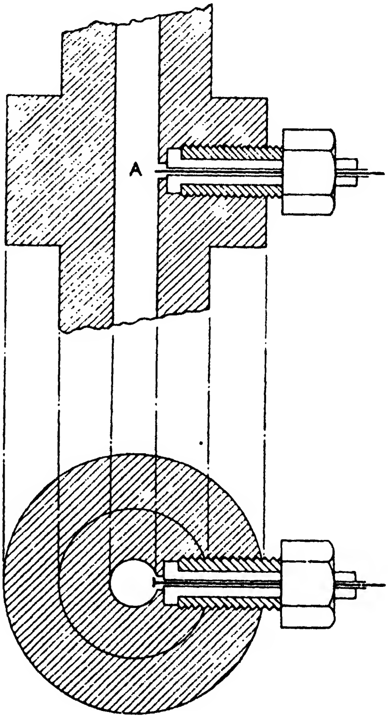

Fig. 2. Details of the Micro-condenser

Since the viscosity at high temperatures is only slightly affected by both temperature and pressure, its values at temperatures higher than 150 deg. C. can be extrapolated from the above formulæ with fairly good accuracy.

From the above three formulæ, the viscosity of any oil at any temperature and at any pressure can be calculated, provided that the values of the constants α_0, β, b, and η_0 for the oil are known.

Solidification under High Pressure. Some motor oils showed a tendency to solidify under a pressure of about 1,000 kg. per sq. cm. They showed increasing values of viscosity with time, although the

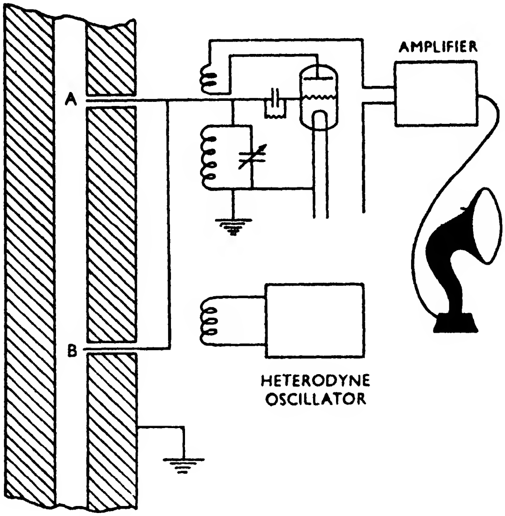

Fig. 3. High-Frequency Circuit

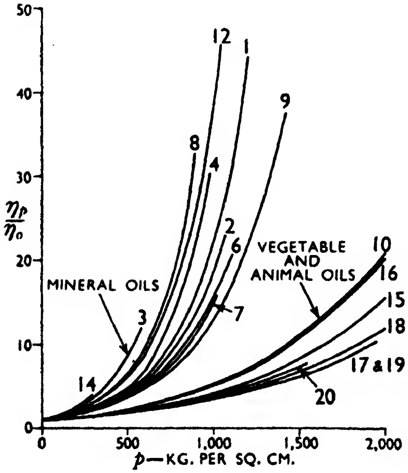

Fig. 4. Coefficient of Viscosity at High Pressure

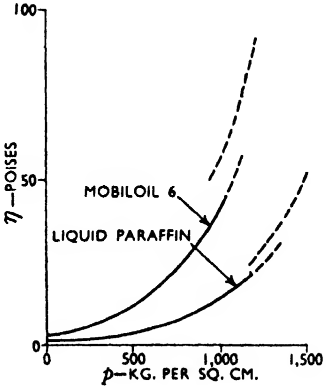

Fig. 5. Solidification under High Pressure

pressure applied was kept constant. With oil No. 6 and liquid paraffin this ambiguous state appeared at 1,100 kg. per sq. cm. and 1,300 kg. per sq. cm., although the absolute values of viscosity were not so great. This cannot be interpreted as complete solidification. Partial solidification was expected, and this was tested at much lower pressure with liquid paraffin in which solid paraffin was dissolved. At lower temperatures, this phenomenon was much more noteworthy. Fig. 5 shows the results obtained at 18 deg. C.

Viscosity of Compounded Oil. Straight mineral oil was compounded with different percentages of rape oil. The change in viscosity of compounded oil of various compositions was tested under various temperatures and pressures. A new empirical formula has been developed: $\eta=(\eta_A-\eta_B)e^{-\gamma x}+\eta_B$ where η_A and η_B are the viscosities of the constituents A and B respectively, γ is a constant, and x the ratio of the amounts of B and A. This formula was tested under as high a pressure as 1,000 kg. per sq. cm., and it was found that γ was almost constant throughout the whole pressure range.

COMPRESSIBILITY

The compressibility of oils was determined by measuring the supersonic wave velocity. A beam of monochromatic light, falling normally to the direction of propagation of the supersonic waves in the oil, is diffracted (Debye-Shearer and Lucus-Biquards's method). From the distance between the lines of any order and that of zero order, the supersonic velocity is determined, whereby the compressibility of the oil can be calculated. The frequency of the supersonic oscillator was 4,940 kilocycles per sec. In a viscous medium, the supersonic velocity becomes greater because of the viscosity. This increase in velocity, however, is less than 0·0005 in a medium with a viscosity of 10 poises, and is, therefore, negligible with ordinary oils. On the other hand,

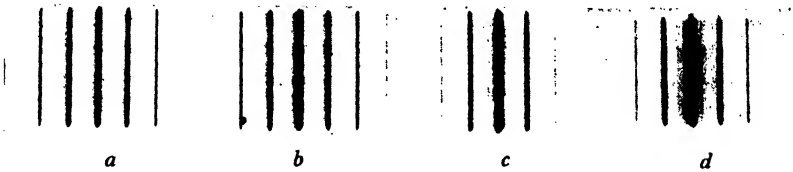

Fig. 6. Diffraction Patterns of Various Oils

a Ice machine oil. *b* Transformer oil. *c* Camelia oil. *d* Soya bean oil.

the damping is very great and the vibrations become hardly detectable at a small distance from the oscillator. Fig. 6 shows typical examples of the patterns. In general, the compressibility of vegetable and animal oils was smaller than that of mineral oil by about 10 per cent.

It is known that, at constant temperature, the product of the surface tension and the coefficient of compressibility of a liquid is almost constant. This observation was tested and the results are given in Table 1. The compressibility of an oil can thus also be estimated approximately by measuring its surface tension.

TABLE 1. SURFACE TENSION AND COMPRESSIBILITY

	T Surface tension	β Compressibility, $\times 10^{-6}$	$T \times \beta$ $\times 10^{-6}$
Rape oil	33·6	60·9	2,050
Olive oil	32·9	59·8	1,970
Transformer oil	31·4	65·0	2,040
Liquid paraffin	31·5	64·4	2,030
Ice machine oil	30·7	67·4	2,070
Motor oil No. 16	32·0	64·0	2,050
Cottonseed oil	33·5	60·9	2,040
Cod liver oil	33·2	60·9	2,020

THERMAL CONDUCTIVITY

The thermal conductivity of oils was determined by the parallel plates method. Some of the results are given in Table 2. Amongst vegetable oils, drying oils had the greatest value, semi-drying oils a medium value, and non-drying oils the least value, although the differences were not very great. The conductivity of mineral oils and commercial motor oils is at least 15 per cent less than that of the

TABLE 2. THERMAL CONDUCTIVITY (C.G.S. UNITS)

Linseed oil	0·00042
Rape oil	0·00041
Castor oil	0·00043
Olive oil	0·00040
Motor oil No. 1	0·00035
,, No. 3	0·00030
,, No. 6	0·00034
,, No. 13	0·00033
Liquid paraffin	0·00029
Transformer oil	0·00027

vegetable oils tested. The additive law holds good for the compounded oils, as was expected.

SURFACE STRUCTURE

The surface structure of oils upon metal surfaces was investigated by the electron diffraction method. A small piece of metal, cleaned by washing with benzene and ether, was smeared with a small quantity of the oil under investigation and subjected to the friction test machine. The oil film was rubbed vigorously for 10 hours under a pressure of 5 kg. per sq. cm., the coefficient of friction being recorded at intervals until the wear of the metal seemed to begin. The metal piece was then removed from the machine and its surface condition was studied by means of an electron camera. The materials tested included vegetable oil, commercial heavy motor oil, and grease spread over the surface of copper, steel, and Babbitt metal. The oil film in each case may have been of the order of 10^{-6} cm. thick. Grease alone exhibited a spotted parallel line pattern, showing a strong orientation, while the rest gave diffuse ring patterns. Although the friction coefficient in every case reached twice its initial value, there was no difference between the diffraction patterns taken from the used and the unused oil.

THE ADSORPTION OF OILS IN RELATION TO LUBRICATION

By Professor J. J. Trillat *

The interfacial tension between two liquids in contact can be measured by pulling away a platinum ring, of known diameter, placed exactly at the interface between the two liquids (Lecomte du Noüy, 1924). The ring is pulled by means of a torsion thread and once the apparatus has been calibrated by weighting the ring, the interfacial tension can be measured directly. Certain precautions are required when using this method, but it is one of the best available and it enables the important problem of interfacial adsorption as a function of time to be investigated.

The method was used to determine the oiliness of an oil and its adsorption by metallic surfaces. Oiliness is characterized by the greater or lesser aptitude of the lubricant to form a strong adherent film on metallic surfaces (Woog 1926). If the molecules can be adsorbed by the metal and orientate themselves on it, they form surface layers which can be revealed by X-rays (Motz 1935; Trillat 1927, 1928, 1929, 1933, 1935; Murison 1934). It is thus believed that when oil is placed on a metal surface, certain of its molecules—particularly those which possess a permanent electrical moment—are adsorbed on the surface and form that boundary film which is the seat of "oiliness". Oiliness also depends on the nature of the molecules adsorbed. Thus, vegetable oils are much more oily than mineral oils, whereas a mineral oil consisting chiefly of straight saturated hydrocarbons has little or no oiliness. The spread of oil on metal also depends on adsorption; an inactive oil (saturated hydrocarbons) will spread on the metal and can be easily forced away by pressure, whereas an oily, active oil will spread with difficulty but will form a highly resistant film. This procedure will be reversed on a layer of water.

The study of the interfacial tension of oil and water gives useful information for the control and acceptance of lubricants, but it is the study of the interfacial tension of oil and water as a function of time which is of most interest to the physicist, for it is related to the speed of diffusion of the active molecules and their adsorption at the interface.

The interfacial tension of any mineral oil in contact with an aqueous solution changes in time, the change depending on the hydrogen-ion concentration of the aqueous solution (Weiss and Vellinger 1929).

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* Director of the Institute of Physics, Faculty of Science, University of Besançon.

The author, with Leprince-Ringuet (1933, 1934), has studied the changes in the interfacial tension between pure water and various oils, in relation to impurities added to the oils and to changes induced in the oils by heat, ultra-violet rays, and use in an engine. When paraffin oil (saturated hydrocarbons) is placed in contact with very clean water and the interfacial tension is measured as described, the value obtained, about 45 dynes per cm., varies but little with the time of contact (Fig. 1, curve 1), but if increasing amounts of oleic acid (in concentrations varying from 10^{-2} to 10^{-5}) are dissolved in the oil, there is an initial fall in the tension which is the more marked the greater the content of oleic acid. Further, all the curves for interfacial tension in relation to time show a notable curvature at the beginning.

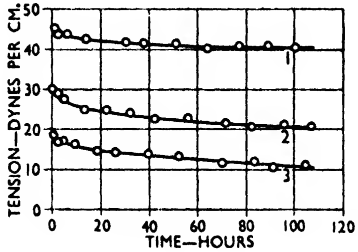

Fig. 1. Interfacial Tension between Paraffin Oil and Water as a Function of Time

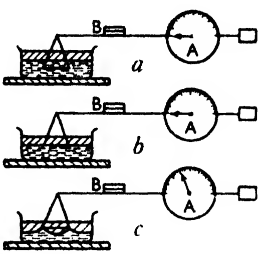

Fig. 2. Modified Platinum Ring Method of Determining Interfacial Tension

Systematic study of this phenomenon showed that molecules dissolved in the oil and possessing a permanent electrical moment, are adsorbed selectively on the surface of the water, on which they become orientated, being fixed by their active group. The initial curvature in representative curves represents the time required for the formation and development of these adsorbed layers (Trillat 1933).

The nature of the molecule and the value of its electrical moment play an important part, as can be shown by dissolving either cyclohexanol (with a permanent electrical moment) or cyclohexane (with no electrical moment) in paraffin oil: in the first case the interfacial tension falls, but in the second case nothing happens. This result agrees with those obtained by X-ray examination of the orientation of molecules in contact with water (Trillat and Nowakowski 1931). The method is very sensitive as it enables the detection of 1/100,000 part of active impurity dissolved in paraffin oil.

These results can be explained as follows: If a small amount of foreign matter is dissolved in a pure liquid, the interface between the liquid and another medium (air, solid support, or another liquid) will usually be the seat of a change in the concentration of the body dissolved. With substances which raise the surface tension of the solvent, the surface layer is less rich in solute than the main bulk of the liquid. The contrary takes place with substances which lower the surface tension. When the solution is sufficiently dilute, the adsorption can be expressed by Gibbs' equation: $U = -c/RT \times d\gamma/dc$, where c is the concentration in the bulk of liquid, U the excess of substance in grammes per cubic centimetre in the surface layer, γ the surface tension, T the absolute temperature, and R the gas constant. As U and $d\gamma/dc$ are of opposite sign, if the surface tension varies inversely as the concentration, the surface layer will contain more solute than the mass of the liquid (positive adsorption). If a very small quantity of fatty acid (e.g. 1 drop oleic acid in 500 cu. cm. of oil) is added to a pure paraffin oil consisting of saturated hydrocarbons, and the oil thus activated is poured on a clean surface of pure water, the molecules of fatty acid, on diffusing through the surface layer, will be fixed there by their active COOH group. As the fatty acid, like any other active molecule, reduces the interfacial tension between oil and water, the oil-water tension should decrease as a function of the time of contact. This decrease will be greater the higher the concentration of the active material in the oil. The curves (Fig. 1) show this clearly; their initial curvature corresponds to the time required for the complete formation of the adsorbed layer.

As the platinum ring method presents certain disadvantages, it has been modified as follows (Trillat and Vaille 1936): The clean ring is plunged in water and 15 cu. cm. of oil is poured gently on the water (Fig. 2*a*). The ring is then brought to the interface by lowering the receptacle containing the liquids. If the adjustment is correct, the beam of the balance should be horizontal (Fig. 2*b*). A constant vertical force is applied to the ring by a steel thread. This force (24 dynes in the experiments) should be less than the force required to rupture the water film on the oil, so that the interface is only deformed (Fig. 2*c*). As interfacial adsorption takes place, the interfacial tension decreases, and the ring rises gradually. This movement is observed at regular intervals by means of the sight B, and the readings are plotted, curves being obtained which characterize a given oil for a given temperature. The method enables the deformations of the interfacial film to be followed very accurately. If the interfacial tension is sufficiently high, the ring will tear away without external help, and at that moment the value of the interfacial tension is obtained by applying Gibbs's equation. The apparatus, adapted to give automatic registration, is now used by the French Air Ministry (Fig. 3).

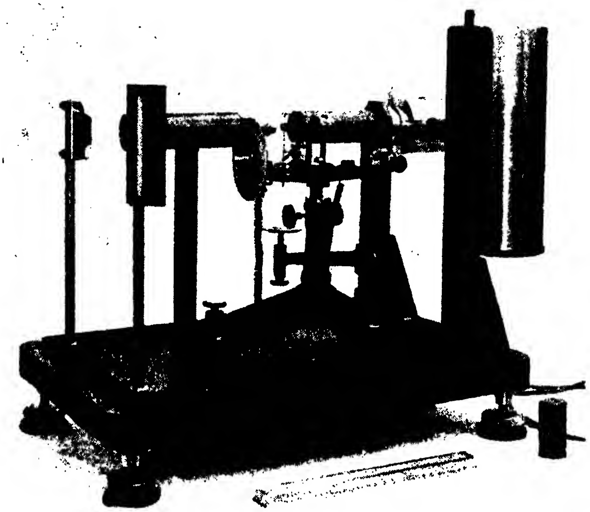

Fig. 3. Recording Interfacial Tensiometer

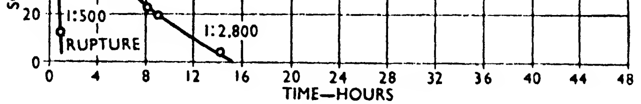

Fig. 4. Effect of the Strength of Paraffin Oil–Oleic Acid Mixtures on Interfacial Deformation

Paraffin Oil–Oleic Acid Mixture. Oleic acid was added to pure paraffin oil at the rate of 1 part of the former to 500, 2,800, 8,360, 16,800, and 100,000 parts respectively of the latter. Curves were plotted (Fig. 4) for a temperature of 18 deg. C., the experiment lasting

48 hours. When pure inactivated oil is examined, the deformation of the surface is insignificant, but when the oil contains oleic acid the change is much more rapid. The curves (Fig. 4) thus show the stretching of the interfacial film, which is related to the interfacial adsorption of the molecules and their orientation. The curves, however, do not give the absolute value of the interfacial tension, which is measured only at the end of the experiment, when the ring tears away. With oil containing 1/500 of oleic acid, the ring tears away after one hour; the tension was then 24 dynes. With oil containing 1/2,800 of oleic acid the tear occurs after 14 hours. The method is much more sensitive than any purely chemical method as the curve for oil of a concentration of 10^{-5} is quite distinct from that for pure oil.

The molecules of oleic acid, having a permanent electrical moment, are thus selectively adsorbed on the surface of the water. Adsorption is more rapid at the beginning of contact, as is shown by the curves, and as the interface becomes saturated with oleic acid, the number of molecules retained per unit of time decreases and the curves tend to become flat. This occurs after a period depending on the concentration: contact for 100 hours is required for the flat portion of the curve to be reached with an oil of 1/16,800 strength. At that moment the absolute value of the interfacial tension also remains constant.

Adsorption in Contact with Metal. An oil in contact with a metal surface gives up certain of its molecules, chiefly those which have a permanent electrical moment. The metal thus becomes covered with an extremely thin, adherent film. It is obviously useful to be able to study quantitatively a phenomenon so important for lubrication, to measure the thickness of the boundary film or "épilamen" (Woog), and compare the oiliness of different oils. The interfacial tension between oil and metal cannot be measured, but the difficulty can be overcome by calculating the strength of adhesion W_{MH} between metal and oil. Applying Dupré's formula it can be shown that $W_{MH}=\gamma_H(1+\cos\theta)$, where H is the surface tension of the oil, and θ the angle of contact of oil and metal. An oil is thus more adherent the greater H and the smaller θ. The method gives a general indication of the adherence but reveals nothing about the molecular mechanism of the phenomenon, the selective adsorption of certain molecules on the metal, or the thickness of the adsorbed layer (boundary film), so important in lubrication.

The author therefore employed the method solely to measure the activity of the oil. Paraffin oil activated by oleic acid is placed in contact with a clean metal surface (calibrated steel balls). If the oil loses some of its molecules of oleic acid to the metal, there will be a decrease in the concentration of oleic acid in the oil, and a new curve should be obtained when that oil is placed on double-distilled water. Experiment shows that this does occur (Fig. 7) and by referring to the curves of Fig. 4, the new concentration can be determined by extrapolation. It is now possible to calculate the number of molecules adsorbed and the thickness of the boundary layer retained on each ball. Further, the formation of the layer can be studied by varying the conditions of contact. In short, a new method is available to study what happens when oil comes into contact with a metal surface. The measurements carried out in contact with water only serve as intermediate steps and need only be relative. Examples of the adsorption of a mixture of paraffin oil and oleic acid will now be considered.

I. Study of Various Cases of Adsorption

Filtration of an Activated Oil. On filtering 17 cu. cm. of a mixture of paraffin oil and oleic acid (1/2,800), much of the oleic acid was retained by the filter paper (Fig. 5). Comparison with the curves of Fig. 4 showed that the concentration had fallen from 1/2,800 to roughly 1/30,000. The initial 17 cu. cm. contained $(17\times 6{\cdot}06\times 10^{23})/2{,}800\times 282=13{\cdot}6\times 10^{18}$ molecules (molecular weight of the fatty acid=282). As 2 cu. cm. was retained by the filter paper, therefore the remaining 15 cu. cm. (concentration 1/30,000) contained

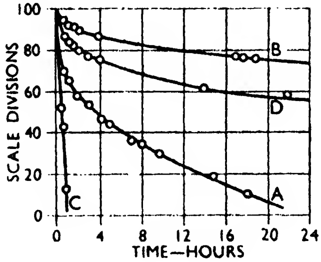

Fig. 5. Effect of Filtration on an Activated Oil

A Before filtration.
B After filtration.

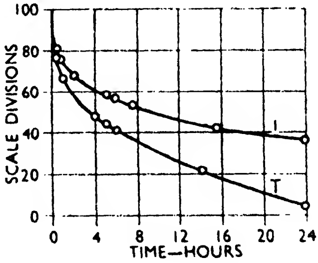

Fig. 6. Effect of Wick Filtration on an Activated Oil

T Before filtration.
I After filtration.

$$(15\times 6{\cdot}06\times 10^{23})/(30{,}000\times 282)=1{\cdot}1\times 10^{18}$$

molecules of acid. The difference represents the number of molecules adsorbed, i.e. of the order of 12×10^{18}, corresponding to 5·7 milligrammes of oleic acid adsorbed. Simple filtration thus brings about a profound change in the molecular constitution of an oil, removing the greater part of its active constituents, which are the most important as regards oiliness.

Passage of Activated Oil through a Cotton Wick. The activated oil was passed by capillarity through a cotton wick 24 cm. long and 2 mm. in diameter. Filtration is very slow: it requires some 25 days to collect 20 cu. cm. of oil. It is known that oil treated in this way is considerably changed (Woog 1926) as is shown by the two curves (Fig. 6) before and after filtration. On account of this result the author thinks that wick lubrication should be rejected where oily lubrication is required and the length of the wick is considerable (central oiling with wicks). Similar experiments carried out with glass wool as adsorbent have given the same general results.

Passage of Activated Oil over Metal Balls. This case is particularly interesting as it is directly related to oily lubrication, while, if calibrated metal balls are used, it is possible to calculate the total surface and from this the thickness of the layer of acid adsorbed. A similar procedure was utilized by Lecomte du Noüy (1924) to determine the adsorption of serum. Clean, calibrated steel balls (3·16 mm. in diameter) were placed in a clean tube of Pyrex glass, 18 mm. in diameter, closed below by a wide-meshed nickel grid. The height taken up by 3,821 balls in the tube was 29·5 cm. 25 cu. cm. of oil activated with oleic acid (concentration 1/1,500) was poured into the tube and allowed to drain through completely, 18 cu. cm. of oil being collected and studied by means of the torsion balance (Fig. 7). The oil thus obtained contained roughly 1/4,000 of oleic acid. Using the same method of calculation as before, the number of molecules of acid retained by the balls was 5×10^{18}, equivalent to 0·0023 gramme. Let s be the section of the oleic acid molecule, then the total surface of the balls being 1,200 sq. cm., $s \times 5 \times 10^{18} = 1{,}200$ sq. cm., and $s = 2{\cdot}4 \times 10^{-18}$ sq. cm. This value is much too low, since other methods show that the molecular section of the acid is 30×10^{-16} sq. cm. The adsorption layer must thus be multimolecular. Therefore the number of layers is $30/2{\cdot}4 = 12$, and as the length of the molecule of oleic acid is 23 Å, the thickness of the adsorbed layer is roughly 28 $\mu\mu$. It is assumed that the active molecules in the 7 cu. cm. of oil retained by capillarity between the balls are entirely adsorbed on the metal. Account should also be taken of the number of molecules adsorbed on the glass walls, though the surface of the glass is small in relation to that of the balls (170 sq. cm. as against 1,200 sq. cm.). Nevertheless the figure obtained agrees closely with results obtained by other methods. It is known that the active ends of the molecules (COOH groups) are adsorbed on the metal, forming a monomolecular layer and in doing so they become orientated perpendicularly to the metal. The following molecules are still attracted through the first layer and form a second orientated layer and so on. But as the field of attractive force due to the metal weakens rapidly, the orientation will become poorer and poorer and, at a distance equal to a few molecular layers, the molecules will be arranged at random. At this point begins the boundary layer, the seat of oiliness. The total thickness of this layer varies according to the activity of the molecules and the attractive force of the metal, and this gives a good explanation of the varying degrees of oiliness observed with different combinations of metals and oils. The author (1929, 1933, 1935) has shown that for saturated fatty acids, stratification occurs in some tens of molecular thicknesses and this is confirmed by electron diffraction.

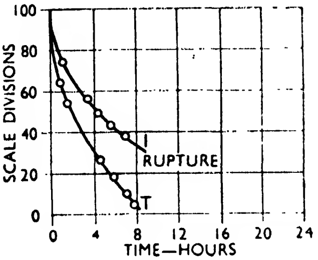

Fig. 7. Effect of Adsorption on Metal Balls

T Before adsorption.
I After adsorption.

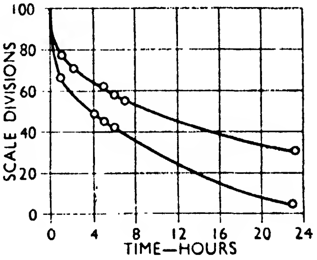

Fig. 8. Variation of the Composition of an Activated Oil with Time

Tests with lead balls gave similar results, while the determinations of the thickness of the adsorption layer were of the same order as those for steel balls, thus providing a check on the method.

Variation in Composition of an Activated Oil in Time. When an oil activated with oleic acid is placed in contact with water immediately after preparation, the curve obtained is similar to those of Fig. 4. But if the oil is allowed to stand long enough (50 days and more) in the dark, in a flask with a ground-glass stopper, a different curve, indicating a diminution in the oleic acid content is obtained, probably owing to the adsorption of acid on the glass (Fig. 8). It is therefore necessary when measuring the adsorption to work out a calibration curve for a control oil to show the true concentration.

II. Applications to Commercial Oils

The attempt has been made to apply these results to a commercial lubricating oil. If 15 cu. cm. of fresh oil is poured on double-distilled water and the movements of the pointer are recorded with the time, curve T of Fig. 9 is obtained. This curve falls rapidly with time as does the curve for paraffin oil rich in oleic acid, though the molecules which cause the decrease in interfacial tension are not molecules of oleic acid but a complex mixture of active molecules. By comparing curve T with the curves obtained with the paraffin oil–oleic acid mixture, the activity of the commercial oil can be expressed in terms of oleic acid. Thus curve T corresponds closely to the curve obtained with paraffin oil containing 1/3,000 of oleic acid. As commercial oils having the same viscosities can be clearly differentiated one from another, the curve provides a new means of control as it is directly related to the molecular structure of the lubricant. Further, the shape of the oil-water curve changes according to the age of the oil, its storage at different temperatures, any modification it has undergone, and the degree to which it has been refined (Trillat and Leprince-Ringuet 1933, 1934). Two examples follow:—

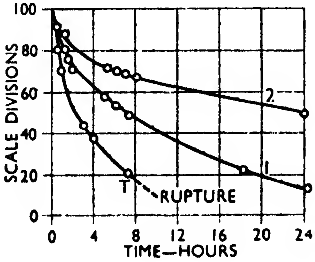

Fig. 9. Effect of Filtration on a Commercial Oil

T Pure oil.
1 Oil filtered once.
2 Oil filtered twice.

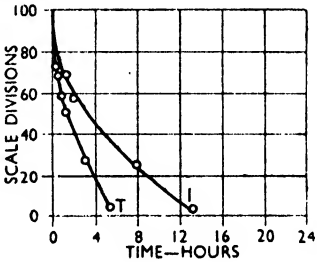

Fig. 10. Effect of Passage over Metal Balls

T Fresh oil on water.

(1) *Filtration through Filter Paper.* Fig. 9 gives the curves obtained with 15 cu. cm. of a commercial oil which has passed through filter paper (curve 1); the same oil, filtered through a second filter paper,

gives curve 2, whereas the untouched oil gives curve T. There is considerable adsorption after the first filtration, whereas the second filtration leads to the retention of most of the remaining active molecules. This result counter-indicates the use of filtration for the cleaning and recuperation of used oils.

(2) *Passage over Metal Balls.* Using the same method a commercial oil gave results (Fig. 10) similar to those obtained with activated paraffin oil. There was considerable adsorption by the metal, and oil that had passed over metal was found to be different from fresh oil. No chemical method will reveal this important change, which enables oils of different degrees of use, refinement, etc., to be compared.

Conclusions. The experiments described show the occurrence of selective adsorption, by metals and other bodies, of certain molecules normally present in, or added to, lubricants. They throw light on the mechanism of adsorption and make it possible to measure the approximate thickness of the boundary film and, in certain cases, to bring about the selection of given molecules or to measure the surface of adsorbents. The method provides a new and very delicate method for defining the quality of an oil as regards its activity or oiliness.

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COMPARISON OF THE BEHAVIOUR OF VARIOUS EXTREME-PRESSURE LUBRICANTS IN DIFFERENT TESTING MACHINES.

By Dr. J. L. van der Minne*

The problem of extreme-pressure lubrication resulted from the advent of the hypoid final drive in automobiles (Wolf 1936; Griswold 1936, 1937†), and this problem has since grown owing to the ever-increasing specific tooth-pressure in automobile gearboxes. Not only the specific tooth-pressures, but also the relative sliding speeds between two meshing teeth, especially in hypoid drives (Wolf 1936), are high in modern automobile transmissions. This caused the teeth to score and even to weld locally together, as long as ordinary lubricants were used. It became necessary to resort to very special lubricants of a kind already used for lubricating cutting tools. Admixture of sulphur or chlorine compounds with mineral oil was known to ensure good lubrication even under conditions of extreme pressure, such as exist at the cutting edge of many tools.

Not only does the high specific pressure in the small surface of contact lead to scoring, but also very high temperatures occur (Bowden 1936) on the sliding surfaces owing to the prevailing high pressure and velocity, and the two factors, high pressure and high temperature together bring about scoring or welding. The term "extreme-pressure lubricant" is therefore not quite correct; it should be " extreme-pressure and temperature lubricant".

It may be assumed that the sulphur and chlorine compounds referred to react with the metal surfaces where they come into contact at high temperature and that these reaction products prevent closer contact between the metal parts. It should be borne in mind that high pressures can never entirely expel lubricants from the area of contact; for one thing there may be pores filled with lubricant in even the smoothest surface. It follows from this that metal is used up by this chemical reaction and great wear may result. Experience has, however, shown that wear occurring under these conditions of chemical lubrication, need not necessarily be abnormally high and that contrary to general opinion, even low coefficients of friction may prevail.

The mere fact that certain substances impede scoring does not in

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* Amsterdam Laboratory, N. V. de Bataafsche Petroleum Maatschappij.

† See also the paper by H. Blok, vol. 2, p. 14.

itself fit them to rank as extreme-pressure lubricants. The following factors have to be taken into account:—

(1) Wear: the degree of wear (quantity of metal worn off) and its nature, i.e. whether the surface is smooth or grooved or whether pitting*, discoloration, etc., occur.
(2) The coefficient of friction.
(3) Exhaustion of the extreme-pressure lubricant: owing to chemical reaction, extreme-pressure lubricants deteriorate with varying rapidity in use.
(4) Corrosive nature: at high temperature a certain capacity for reaction is considered essential, but at low and moderate temperatures it is not desirable.
(5) Foaming: should not be excessive.
(6) Fluidity: should be adequate at low temperature.

Obviously, experiments for rating extreme-pressure lubricants for hypoid gears could be carried out on hypoid gears themselves, but there are other uses for extreme-pressure lubricants and more can be foreseen for various kinds of machinery. Further scientific treatment, in which the practical problem is dissected into several elementary problems which are then studied individually, is to be preferred. Many investigators have adopted the latter procedure, building apparatus in which certain rotating parts were variously loaded. Examples of these are Almen's apparatus, the four-ball tester designed by Boerlage, the Timken tester, the Floyd tester, and the machine of the Society of Automotive Engineers (S.A.E.). A brief description of these machines follows.

Four-Ball Tester (Boerlage 1933; Van Dijck and Blok 1937). Three ½-inch steel balls (Fig. 1) are clamped tightly together so that they cannot move. On top of these a fourth ball revolves in the oil under test under a pressure which can be varied. The test is usually carried out at 1,500 r.p.m. The coefficient of friction during the test and the wear of the balls can be measured. The initial pressure per square centimetre is very great, but as the balls wear this pressure gradually diminishes.

Timken Tester (1934 *Petroleum Times*). A steel block (Fig. 2) is pressed against a revolving ring for a given period under a certain pressure. The coefficient of friction and the wear of the block are

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* "Pitting" is often regarded as caused by fatigue. It appears, however, that the nature of the lubricant affects pitting, and this makes this explanation doubtful.

measured. The lubricating oil may be fed in between block and ring at any desired temperature. The test is made under different pressures and the pressure for imminent scoring (which has been called the "O.K. value") is recorded. The standard test starts at 100 deg. F. and lasts 10 minutes at 800 r.p.m. The initial specific pressure is high and diminishes gradually as wear develops during the test.

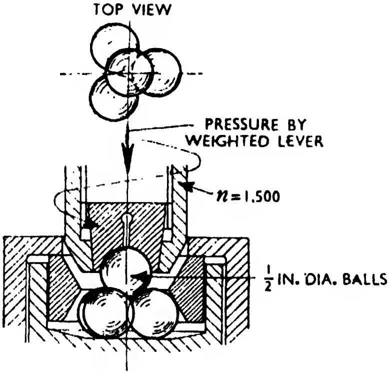

Fig. 1. Four-Ball Tester

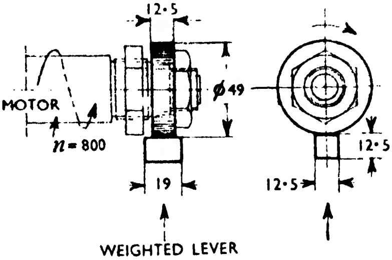

Fig. 2. Timken Tester

Dimensions in millimetres.

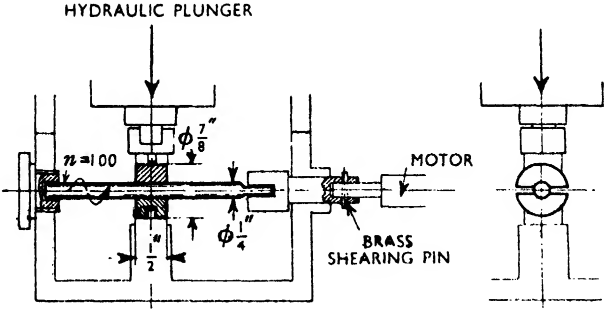

Fig. 3. Floyd Tester

Floyd Tester. A steel shaft (Fig. 3) revolves in two half-bushes of steel. The top bush is loaded hydraulically. Normally, the test is carried out at room temperature and the load on the bush is increased every 10 seconds by 25 lb. up to 200 lb., after which the machine has to run under this pressure for 5 minutes. The test is then repeated at 200 deg. F. This time the load is increased in the same way to 325 lb., under which load the machine has again to run for 5 minutes. No other measurements are made. The criterion is that a brass shearing pin that transmits the motive power shall not break. With this machine, the specific bearing load remains the same all through the test, namely, 1,757 kg. per sq. cm. for the applied load of 325 lb. For those oils that withstood the test, the load was further increased until the pin finally broke.

S.A.E. Machine (1936 *S.A.E. Journal*). Two ring-shaped test pieces (Fig. 4), 1·91 inches in diameter, are used, having cylindrical exterior surfaces ½ inch wide, and conical interior surfaces. These test rings are similar to regular bearing cups except that the cylindrical surfaces are ground to a special finish. The two test rings are fitted to two revolving shafts, one above the other, so that the faces of the rings roll over each other as the shafts revolve, but not at the same speed. The lower shaft is geared to the upper drive shaft by a set of changeable gears which give speed ratios of 3·4/1, 10·4/1, 14·6/1, and 20·7/1. The sample of lubricant is placed in an oil cup into which the surface of the lower test ring dips as it revolves. The oil is carried between the two rings by the revolving motion of the shafts. The upper ring is completely outside the oil cup and is lubricated only by the oil which is carried up to it by the lower test ring. The load on the bearing surfaces is increased gradually from a very light contact to the point at which the oil film fails and the bearing surfaces score. The load at failure is read from a spring scale fastened between the oil cup trunnion lever and the loading mechanism screw. If the bearing surfaces do not score before the highest attainable load is reached (6,000 lb. on the bearing cups=600 lb. on the scale), the machine is

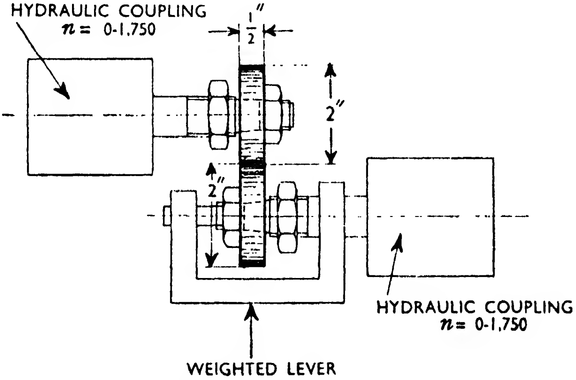

Fig. 4. S.A.E. Tester

stopped and the load carrying ability of the lubricant is expressed as >600 lb. The specific pressure is independent of wear phenomena. A definite method of testing with this machine has not yet been generally agreed upon; the author therefore records a specification issued by the General Motors Corporation for testing hypoid gear lubricants: main shaft speed, 750 r.p.m.; rubbing ratio, 14·6/1; rate of loading, 83·5 lb. per sec. (8·35 lb. on the scale). An oil, to pass the specified test, will not fail below a loading of 600 lb.

The principal details of these four testers are given in Table 1, together with the specific pressures and the sliding velocities; details for a hypoid gear are also recorded. The differences between these values clearly show the difference in scope of the four instruments.

The lubricants used are listed in Table 2. Of these samples No. 7 contains no "dope" and the mineral oil used in making up Nos. 2, 3, 4, 5, and 6 is the same as No. 8. The quantity of dope added varies for the different lubricants from 2 to 25 per cent.

Timken Tester. The results from the Timken tester will be discussed first, because this instrument aims at a single criterion, that is to say, the "O.K. value," according to which the lubricants have been arranged in a descending order of merit (from 1 to 8). This value is determined by the load (amounting to one-tenth of that to which the block is subjected) at which scoring is imminent; wear has then already developed. The width of the scar is recorded in millimetres. These particulars have been collected in Table 2, which also gives the coefficient of friction at the end of the test and the load in kg. per sq. cm., also at the end of the test. It will be seen that the rating of the lubricants, according to this final load, is in fair agreement with the "O.K." rating.

Four-Ball Tester. Fig. 5 shows the relation between load and scar diameter on the balls when the test is carried out as specified. The numbers correspond to the various lubricants. The sign ↑ signifies that this was the last point at which the balls had not yet welded, for if the test is continued at higher pressure, the four balls weld together into a single unit. The bottom line (contact surface according to Hertz) represents the size of the contact surfaces if the balls were to touch without wearing (calculated according to Hertz). A smaller scar size than this cannot be obtained. It will be noted that the best lubricant, No. 1, and the worst, No. 8, occupy the same extreme positions according to this standard too. For the other lubricants this tester does not give ratings which are in agreement with those of the Timken tester. Different criteria for interpreting the results of the

TABLE 1. DETAILS OF TESTING MACHINES

Machines	Moving parts and material		Kind of movement	Shifting of contact surface†		Sliding velocity, metres per min.	Duration of test, minutes	Specific pressure on contact surface, kg. per sq. cm.	Temperature of the oil at start	Particulars
Four-ball tester	Ball on three balls	"SKF" balls, chromium steel	Sliding	Yes	No	35	1	Initial, about 56,000 (load 600 kg.). Final, 1,000–10,000	Room temp.	Balls wear. Wear and coefficient of friction measured
Timken tester	Block on steel ring	Block and cup, nickel-molybdenum steel, S.A.E. 4620, carburized and hardened 65 deg. Rockwell, C.	Sliding	Yes	No	122	10	Initial, about 6,000 (load 110 lb.). Final, 500–1,200	100 deg. F.	Block wears, producing either smooth surface or scoring. Highest pressure at which no scoring occurs is recorded
Floyd tester	Split bushes around revolving shaft	Bushings: S.A.E. 1112 (0·12 per cent carbon). Shaft: carbon steel (0·9–0·98 per cent carbon)	Sliding	Yes	No	2·1	5	1,760 (load 325 lb.).	Room temp. 200 deg. F.	Brass shearing pin remains intact or breaks:— (*a*) owing to high coefficient of friction; (*b*) owing to scoring
S.A.E. tester	Two steel rollers, one on top of the other	See Timken tester	Sliding and rolling	Yes	Yes	109*	—	About 20,000	Room temp.	For a given rate of loading the load at which scoring occurs is noted
Hypoid gear (Wolf 1937)	—	Hardened alloy steel	Sliding and rolling	Yes	Yes	Up to 300–600	—	Up to about 25,000	—	Though these high values prevail only during a small part of the lifetime of the gears, obviously under these conditions substantial wear or scoring cannot be tolerated

† The contact surface of a contacting part may either be stationary with respect to that part, or may shift on it.

* Used according to General Motors Corporation test conditions: 750 r.p.m.; rubbing ratio, 14·6/1; rate of loading, 83·5 lb. per sec.

four-ball tester might be applied; for instance, the load at which welding occurs (↑) or where the lines begin to diverge from the Hertz line, or where the indications of the dynamometer clearly point to a change in friction during the test; yet all these criteria still fail to lead to agreement with the Timken tester data and their results do not agree mutually. The author therefore merely mentions the details of the wear and gives in Table 2 the coefficient of friction and the specific pressure towards the end of the test.

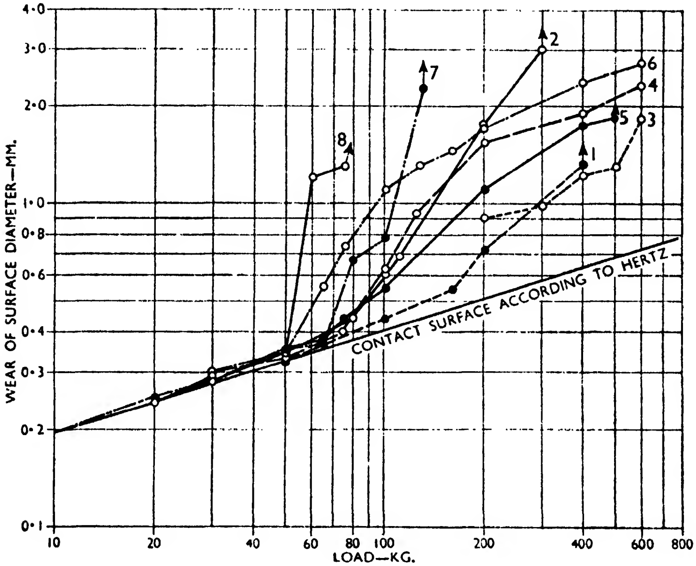

Fig. 5. Relation between Load and Wear on the Four-Ball Tester
Oil 8 seizes at 200 kg.; the arrow at 75 kg. is misplaced.

Floyd Tester. This tester is far less exacting (cf. Table 1), and it will be seen that the first four extreme-pressure lubricants pass the test right up to the highest pressure. Again the ordinary mineral oil (No. 8) produces by far the worst result, but there is one noteworthy fact: product No. 5 (containing active sulphur and lead soaps), being at present the only type of lubricant generally considered to be suitable for hypoid gears, is in this test inferior to fatty oil (No. 7). The fatty oil withstands a pressure of 525 lb. and lubricant No. 5 only 325 lb.

S.A.E. Tester. This is an apparatus at the end of the scale of requirements, making very rigorous demands upon the material. Table 2 shows that, practically speaking, only lubricants No. 1 and

TABLE 2. RESULTS FROM

Lubricants	Four-ball tester				
	Load at which seizure occurs, kg.	Load, kg.	Wear surface diameter, mm.	Coefficient of friction	Final specific pressure, kg. per sq. cm.
1 Special lubricant containing phosphorus but no mineral oil	500	400	1·53	0·075	11,500
2 Lubricant containing sulphur	400	300	3·00	0·161	1,750
3 Lubricant containing copper	>600	600	1·87	0·042	9,000
4 Lubricant containing sulphochloride	>600	600	1·96	0·056	8,000
5 Lubricant containing sulphur (in corrosive form) and lead soap	600	500	1·84	0·095	7,500
6 Lubricant containing suspension "dope" (the mixture contains organic compounds of little chemical activity, in insoluble form)	600	500	1·80	0·075	8,000
7 Fatty oil (olive oil)	150	125	2·45	—	1,100
8 Mineral oil (bright stock); viscosity, 480 centistokes at 50 deg. C.	200	150	3·45	0·093	650

* Used according to General Motors Corporation test conditions.
† Mineral oil viscosity, 75 centistokes at 50 deg. C.

No. 5 meet the General Motors Corporation specifications and that the other products lag far behind.

The remarkable difference in the ratings obtained on the various

Different Testing Machines

Timken tester				Floyd tester		S.A.E. tester
"O.K. value"	Coefficient of friction	Scar width, mm.	Final specific pressure, kg. per sq. cm.	Load at which pin breaks, kg.	Temp. of oil cup at the beginning of the test, deg. F.	Load at failure, kg., weight on lever*
>110	0·061	2·99	1,200	>600	200 deg. F.	>600
80	0·11	2·4	1,200	550	,,	120–140
62	0·101	2·21	1,000	>600	,,	100–128
43	0·120	2·04	760	>600	,,	64–70
42	0·036	3·20	380	325	,,	600
18	0·119	0·98	700	225	,,	60–52
12	0·12	0·65	760	525	,,	40–45‡
12	0·12	1·0	490	200	room temp.	57–60†

‡ Castor oil gave 39 lb.; rape oil 37 lb.

testers suggests that there are two ways in which non-fluid lubrication may be effected:—

(*a*) A chemical attack upon the surface at the points where, owing to high pressure, high sliding velocity, and great generation of heat, the temperature has risen to a high figure. As the result of the increase in temperature, the chemical reaction is appreciably accelerated. The formation of reaction products prevents welding together of moving parts. The coefficient of friction in this case may be very low.*

(*b*) An adsorption of, preferably, polar compounds at the surface, as a result of which the coefficient of friction remains low and there is less generation of heat ("oiliness"). Generally, the phenomena peculiar to oiliness (which may be studied with the aid of the apparatus designed by Herschel, Deeley, or Boerlage and Blok's four-ball top† only occur at much lower pressures and velocities than those with which this paper is concerned.

Some lubricants prove to be more effective in the sense of (*a*), namely, the typical extreme-pressure lubricants which contain chemically active products (among which are lubricants Nos. 1, 2, 4, and 5); others, like lubricant No. 7, are active rather in the sense of (*b*). An occasional lubricant, notably No. 1, may display both effects, producing good results on all the machines.

A typical extreme-pressure lubricant like No. 2 when tested in an oiliness-testing apparatus (like the four-ball top) does not show a lower coefficient of friction than a mineral oil like No. 8. A fatty oil like No. 7, however, when tested in the four-ball top does show at between 50 and 150 deg. C. a lower coefficient of friction than No. 8 oil.

On some testing machines it is necessary for the products to have the effect described under (*a*); in ascending order these machines are: the Timken tester, the S.A.E. machine, and the four-ball tester at the highest loads. In this order the temperature at the contact surface is higher (Van Dijck and Blok 1937). In the Floyd tester and in the four-ball tester at low loads, on the other hand, it is the oiliness (*b*) that comes more to the fore. On the former machine, olive oil (No. 7) shows up well, whereas a typical extreme-pressure lubricant does not. This may be explained by the fact that this machine operates at lower speed and under less pressure. On the other hand, olive oil produces very bad results on the other three testers, which run at great speed and under great pressure.

As already stated, oiliness lubrication (*b*) would seem to prevail under conditions of low speed and pressure. Oiliness, however, seems

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* See in Table 2 the low coefficient of friction found with high pressures on the four-ball tester and Timken tester.

† The difference between the "four-ball top" and the "four-ball tester" is that the balls used for the former are smaller and the pressures and especially the speeds on it are far lower.

not to improve lubrication at high speed and pressure. To explain the latter fact the following reasons are suggested:—

(1) Owing to the great pressure and speed and also to the resulting high temperature, an adsorption layer cannot be formed. In the region of low speeds and pressures the effect of oiliness often diminishes at greater speeds, independently of hydrodynamic effects.

(2) Even were this adsorption layer to be formed and the coefficient of friction to be diminished, the generation of heat would nevertheless be too great (and the resulting temperature too high), so that seizure would occur just the same.

(3) The adsorption layer might act as an inhibitor to the setting up of a chemical reaction by shielding the metal surface.

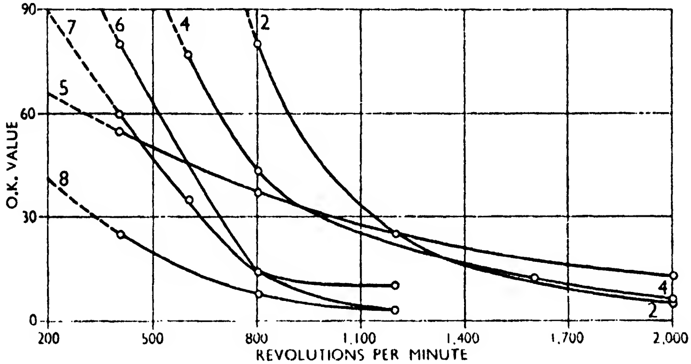

Fig. 6. Relation between Speed of Rotation and "O.K. Value" of the Timken Tester

Although, in certain circumstances (e.g. at no excessive pressure), oiliness may be an advantage to extreme-pressure lubrication, the adsorption layer may be capable of preventing or retarding action (*a*) owing to reasons (2) and (3); in that case oiliness in the extreme-pressure lubricant would be an undesirable quality.

A few points follow to substantiate the foregoing:—

(1) The heat generated increases with the speed of rotation. Fig. 6 shows the Timken "O.K. value" as a function of the speed of rotation for most of the lubricants (the normal speed for the Timken tester is 800 r.p.m.). The typical extreme-pressure lubricant No. 5 used at present in hypoid gears is superior at the higher speeds, but not at

lower speeds; at lower speeds olive oil (No. 7) gives a better result. To the right of Fig. 6 the lines provide an indication of the typical extreme-pressure lubricating action as required on the S.A.E. tester and hypoid gear (*a*), while to the left they are an indication rather of the oiliness (*b*) of the lubricants.

Unfortunately, the range of velocity and pressure on the Timken tester is limited. It was therefore impossible to extend the examination of products 2, 4, and 5 as far as was wished. Since it is not known what further course the curves follow at lower velocities, they are left out of consideration. Further investigation of this relation between velocity and "O.K. value" will undoubtedly yield interesting results.

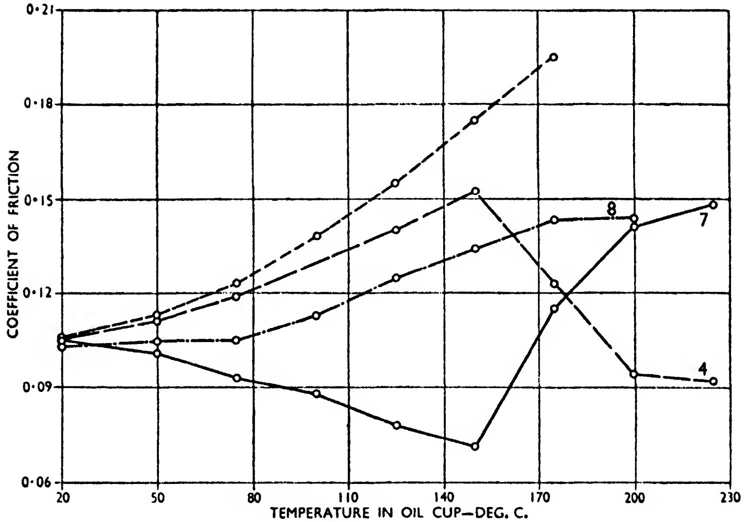

Fig. 7. Relation between Coefficient of Friction and Temperature on the Four-Ball Top

The unnumbered curve (– – –) relates to a mineral oil of 75 centistokes viscosity at 50 deg. C.

(2) The elementary influence of temperature is confirmed when this temperature is artificially raised. Fig. 7 illustrates how fatty oil No. 7 shows an increase in coefficient of friction when the temperature is raised beyond 150 deg. C. Extreme-pressure lubricant No. 4, however, shows a decreasing coefficient of friction when the temperature is raised beyond 150 deg. C.

(3) When lubricant No. 1 is tested in the Timken apparatus, above a certain pressure the scar produced decreases and the coefficient of friction diminishes with an increase in pressure as follows:—

Weight at end of lever, lb.	Scar width, mm.	Coefficient of friction
50	2·61	0·105
60	2·70	0·085
70	3·27	0·088
80	3·22	0·077
90	2·88	0·070
100	2·99	0·061
110	2·38	—

For a load between 70 and 80 lb. the character of the lubrication seems to change; this appears to be a change where extreme-pressure lubrication (*a*) comes in, on account of the release of extreme-pressure characteristics in the lubricant by the increased weight and generation of heat.

(4) The remark made above that oiliness may even be harmful under extreme-pressure conditions, seems to be substantiated by the behaviour of No. 7 oil on the S.A.E. machine (cf. Table 2).

From Table 1 it will be seen that, in general, all the testers (even if the S.A.E. tester is made to run at 1,000 r.p.m., as recently suggested by the S.A.E.) are less exacting, as far as pressure and sliding velocity are concerned, than the hypoid gear. At present there is no call to add to the collection of kindred testers, but there certainly is room for an apparatus subjecting the material to, if possible, even severer conditions than does the hypoid gear. An alternative is, however, to use the hypoid gear itself as tester and for the moment this would seem to be the best course for the examination of lubricants to be used in that kind of gear. It should be borne in mind that, since there are already several types of extreme-pressure lubricants on the market, also for milder conditions, an increasing number of uses under both less severe and more severe conditions will make their appearance. It will, however, be clear that one apparatus will not suffice at present to test lubricants for various purposes, but that different testers will have to be employed for different intended uses.

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OILINESS AS A RESULT OF THE HEATING OF LUBRICANTS.

By Dr.-Ing. G. Vogelpohl*

Definition of Oiliness. The term "oiliness" comprises all those properties of oils which, in presence of "mixed" friction, lead to different coefficients of friction under the same service conditions and equal service viscosity. Lower friction implies greater oiliness. Variations of this definition can be neglected. The meaning is always the same: differing friction, the same viscosity. The differing friction can be regarded as experimentally established, even if the measured results are not reproducible, but it is doubtful whether the viscosity was actually the same in those experiments. By "the same viscosity" is meant the value derived from the measurement of the temperature of the bearing or oil film and the corresponding value of the viscosity curve for the particular oil. Previous workers deduced that the viscosity was the same from the measurement of only one temperature for the whole bearing.

It is necessary, however, to show, not only whether the temperature can be measured with sufficient accuracy to serve for the determination of the actual temperature in the oil film, but also whether the statement of a single temperature is sufficient to define the actual viscosity in the film.

Considerations on Researches Relating to Oiliness. According to hydrodynamic theory, oils of equal viscosity used in the same bearing must show equal coefficients of friction (Ubbelohde 1911–12). Plant and animal oils have long been known to possess better lubricating properties than mineral oils. Gümbel (1914) tried to explain this on the basis of hydrodynamic theory.

Formerly, two oils were considered to be of equal value when they had the same viscosity at a given temperature. Later the viscosity curves were used (Voitländer 1929; Büche 1931) to find the temperatures at which the viscosity is the same (Fig. 1). When the whole region of the oil film in which an application of bearing load is followed by pressure development can be held at this one definite temperature, then will the hypothesis of equal viscosity be satisfied.

In practice, the temperature in the oil film is not constant, so that if equal viscosity is to be deduced from temperature measurements, not

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* Institut für Strömungsforschung, Technische Hochschule, Berlin.

only must the temperature distribution in the film formed by different oils be the same, but the oils must have the same viscosity curves. Alternatively, the temperature distribution must be such that the same distribution of viscosity must result from the viscosity curves. Then only the oil films are equivalent and show, in accordance with hydrodynamic theory, the same friction coefficients under the same external conditions.

It thus appears that equal viscosity can be proved experimentally only with considerable difficulty, and it is most doubtful whether such proof has yet been given. Up to the present, the cause of the difference in frictional values, and therefore the oiliness, has been sought especially in boundary lubrication. Although boundary lubrication has been closely investigated, it has not been shown that a lubricant in a bearing or a machine is subject to the same or even to similar conditions as in special testing apparatus (Marshall and Barton 1927). In "mixed" friction, most of the load is carried by an oil film; the remainder is subject to boundary friction. In the present paper, the attempt is made to explain the deviation of frictional values and oiliness, mainly by the

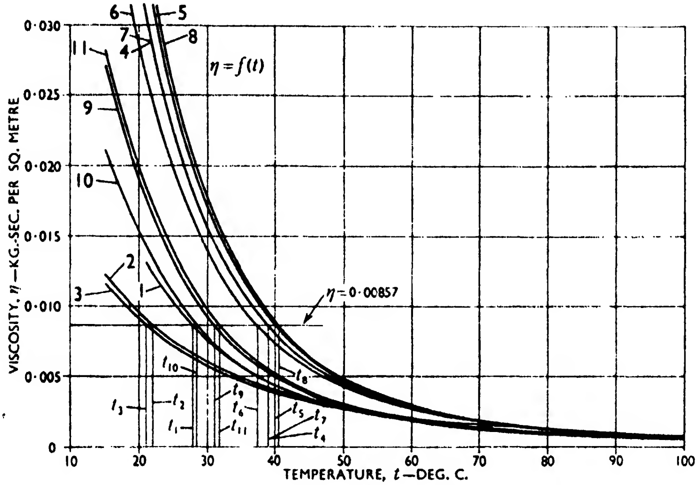

Fig. 1. Viscosity as a Function of the Temperature (Voitländer 1929)

1 Sugar solution.
2 Refined rape oil.
3 Crude ,, ,,
4 German machine oil.
5 Rumanian ,, ,,
6 Russian ,, ,,
7 Polish machine oil.
8 American ,, ,,
9 American ,, ,, (37 deg.).
10 German ,, ,, (14 deg.).
11 American ,, ,, (27 deg.).

varying load capacity of the oil film when different oils are used. For this a review of the experimental methods employed is necessary.

Temperature Measurement. There is no standard method for measuring the temperature of the oil film:—

(1) Kammerer (1920), Schneider (1930), and Gösser (1931) measured the temperature at the middle of the journal. The method is crude and unreliable (Fig. 2).

(2) Lasche (1902) and Biel (1920) employed a thermometer placed in the bearing.

(3) Stribeck (1902) and Nücker (1932) inserted several thermometers in the bearing brass, as close as possible to the oil film. Lasche (1902) also used this method. The accuracy is questionable.

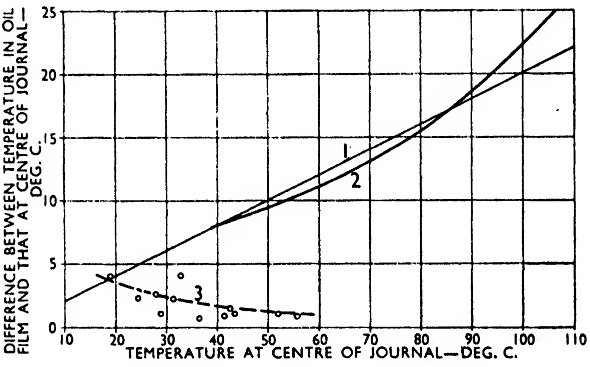

Fig. 2. Comparison of Temperatures in Oil Film and in Centre of Journal (Schneider 1920; Gösser 1931)

1 Temperature 20 per cent higher than in centre of journal.
2 Temperature according to Gösser } measured by method (5).
3 Measurements by Schneider }

(4) Von Freudenreich (1917) and Schneider (1930) used thermocouples inserted in the bearing metal close to the running surface; the results are somewhat better.

(5) The use of thermocouples in the oil film is common, but it is uncertain whether the temperatures recorded are really those of the oil.

(6) Nücker (1932) used insulated thermo-elements. In tests in a water bath, this arrangement showed, compared with the temperature of the brass, differences up to 30 deg. C., but only a difference of 3 deg. C. in the bearing itself. This proves that the readings obtained do not show the real temperature of the oil film. Even small differences

produce an effect in a region where the viscosity curve is steep. The temperatures must be measured *accurately.*

Results of Temperature Measurement. The investigations of von Freudenreich, Lasche, and Nücker have shown that, in the direction of movement, temperature differences as high as 30 or 40 deg. C. can occur. Considerable differences also occur in an axial direction, and this has led Nücker and Schneider to form different conclusions. For this reason, a comparison of the results of different investigators requires the greatest caution.

When the difference between the temperature of the bearing metal was measured as in method (4) above, and that of the oil film as in method (5) above, several temperatures were found at different points. This shows the influence of the heat conductivity of the bearing metal.

The distribution of the viscosity over the bearing surfaces cannot be followed from the readings either at a single measuring point, or at several points. Therefore an average value is required, and previous experiments offer no guarantee that reliable values have been obtained. Apart from the fact that it is difficult to define an exact "average value" for the load capacity and the friction, the deviations from it can be very large.

Doubts about the actual temperature in the oil film are confirmed when it is desired to estimate the temperature distribution across the oil film. This has not yet been accomplished experimentally and theoretical study is required.

Temperature Distribution Across the Film. Hummel (1926) has concluded that internal friction has no real influence on the temperature distribution across the film. But this conclusion is based on an error, as the heat conductivity of the oil is given per hour, whereas the other dimensions refer to seconds, so that on the calculation the heat removed is 3,600 times too great, and the temperature gradient across the layer is 3,600 times too small.

Kingsbury (1933) showed the great influence of frictional heat on the temperature gradient across the film; the difference in temperature was as much as 36 deg. C. and the viscosity changed accordingly in the ratio 1/3·2. Unpublished work by the author gave similar results. However, the reading given by a thermocouple in the oil film is rather indefinite, particularly as the dimensions of the thermocouple are usually greater than the film thickness. If the variations are not so large in this case as in the temperature distribution on the running surface, they lead to a further source of error which renders very difficult, if it does not prevent, the determination of the actual temperature and the actual viscosity. Despite this, the temperature distribution *across* the oil film must be taken into consideration in discussing the matter, and a plausible average value adopted which holds good for the viscosity in the region where the dependence on temperature is still considerable, as in the case discussed by Kingsbury.

Primary Conclusions. The conception of oiliness presupposes the accurate determination of the true viscosity in the oil film. The usual methods of temperature measurement are seen to be insufficiently accurate. In particular, a single reading is not sufficient for the viscosity in the whole bearing to be deduced, since the temperature and, therefore, the viscosity, are distributed too differently. Oiliness may depend essentially on thermal effects in the oil film which have not yet been investigated.

*Load Capacity of the Oil Wedge without Side Leakage in Relation to Heat Effects.** Reynolds (1886) and his successors investigated oil films of constant viscosity, i.e. the limiting case in which the heat developed momentarily in the bearing and shaft is removed, so that the temperature remains constant. The other limiting case, in which all the frictional heat remains in the oil, whose temperature thus increases, will be considered. For this, average values across the film will be considered. Neglect of the heat evolved is not completely justified by the facts, but calculation is easier and new data are obtained which were excluded by the assumption of constant viscosity. This case, however, may approach more nearly to reality than when constant viscosity is assumed.

Details of the calculation cannot be given, and mention will only be made of the equation for the dependence of viscosity on temperature. This equation, given by Reynolds for olive oil as

$$\eta=\eta(t)=\eta\,.\,e^{-\beta(t-t_1)}$$

can be used with good approximation for most oils within a certain temperature range. Further, the product γc of the specific gravity and specific heat of the oil influences the heating of the oil through internal friction. For a given geometry of the gliding surface and a given gliding velocity, then, on $\beta/\gamma c$ will depend the viscosity distribution and therefore the pressure developed.

From the curves of Fig. 3 (Büche 1931), relating to groundnut oil with a flat viscosity curve and tar oil with a steep viscosity curve, it is seen that with tar oil, a smaller pressure is developed than with ground-

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* This, and the following headings are extracted from papers by the author (Vogelpohl 1936, 1937).

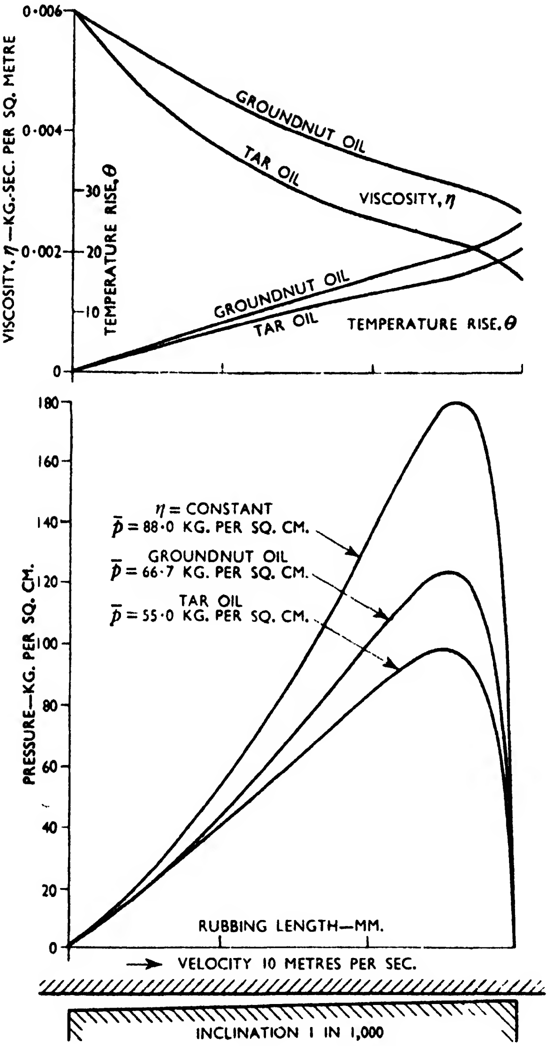

Fig. 3. Decrease in Viscosity, Increase in Temperature, and Pressure Distribution with a Constant Oil Wedge and Equal Viscosity at Admission

nut oil. The oil with a flat viscosity curve has a greater load capacity for the same shape of oil wedge. Here it is assumed that the product γc has roughly the same value. Large differences of γc can alter the load capacity in another way (see below). These results will now be utilized in the discussion of "mixed" friction.

"Mixed" Friction. With "mixed" friction, the load capacity of the oil film is insufficient to equal the bearing load P; the oil pressure will only suffice for the fraction P_f, the remainder of the load P_b being taken up directly by the brass under conditions of boundary friction. Therefore, $P=P_f+P_b$. The shearing stress in the oil leads to a frictional resistance S_f which, on introducing the coefficient of friction μ_f for the fluid friction, can be written as $S_f=\mu_f P_f$. With boundary friction the molecular forces exert a frictional resistance S_b which can be written as $S_b=\mu_b P_b$, where μ_b represents the boundary friction.

The value of μ_f is 0·001 to 0·01, and that of μ_b 0·1 to 0·3. The values of the boundary friction* are from 50 to 100 times greater than those for the fluid friction. Small differences in the load capacity of the oil film can cause large differences in the friction μ for the whole bearing, so that the frictional resistance in

$$S=S_f+S_b=\mu_f P_f+\mu_b P_b=\mu P.$$

The conclusions arrived at above concerning the load capacity of the oil film with an ideal geometrical shape can be applied to the complex shape of the oil film with mixed friction. If the pressure development is not to be worked out mathematically in detail, the effect can be taken fundamentally as that of an ideal flat surface. Rayleigh (1918) has drawn attention to the analogy between the latter and the cylindrical oil film in the journal bearing.

According to the previous discussion, the load capacity becomes greater as $\beta/\gamma c$ becomes smaller under given external conditions. Small values of $\beta/\gamma c$ imply small coefficients of friction under conditions of "mixed" friction. This value is, at a first approximation, a criterion for those properties of a lubricant which have so far been included under "oiliness". †

Experimental Verification of the Results. The experimental confirmation of the above conclusions is only possible conditionally, as the statement of the specific heat of oil is not indicated in measurements of oiliness. In Büche's work only the viscosity curve is given, so that β can only be estimated. A comparison of the coefficient of friction and the

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* The researches of Büche with sliding velocities of 5 metres per second are most convincing. Such high values for μ are not found at low velocities alone.

† The heat conductivity of oils and bearing materials also affects the heat developed in the oil film and so must be taken into account in future work. From a purely quantitative point of view it can already be said that high heat conductivity ensures good lubrication properties.

values of β for Büche's oils shows that up to mineral oil E an increase in β is insufficient to characterize oiliness, as is shown by the good lubricating properties of a sugar solution (Voitländer 1929). In Fig. 4 measurements of the friction itself (*not* the average values) are given in relation to $\beta/\gamma c$ under loads of 5,000, 7,000, and 9,000 grammes, in order to show the dispersion of the results. The curves are to be considered as directrixes, indicating a connexion between $\beta/\gamma c$ and the measured frictional forces. The lubricating quality of the sugar solution, which has a high value of γc, is very good, whereas the value of β approaches that of mineral oil 10 (Fig. 1).

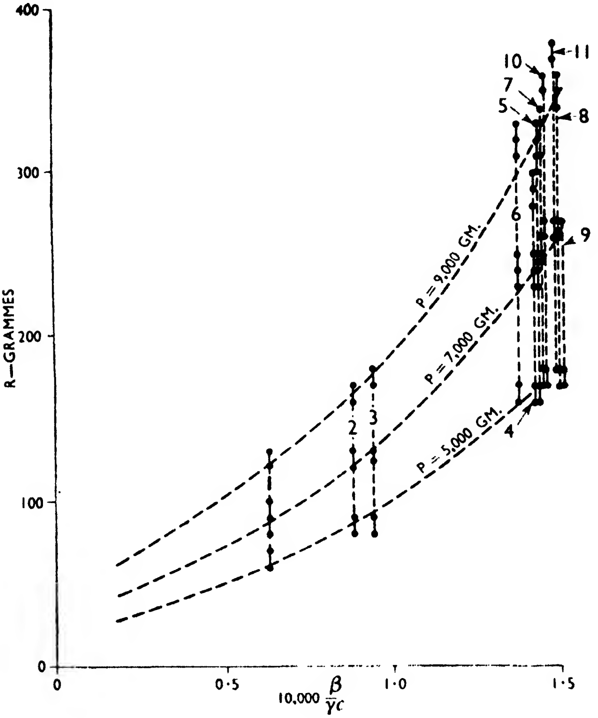

Fig. 4. Frictional Values measured on the Thoma Testing Machine in Relation to $\beta/\gamma c$ for Various Loads P

1 Sugar solution.	3 Crude rape oil.
2 Refined rape oil.	4–11 Various mineral oils.

It thus appears that the value of $\beta/\gamma c$ provides a criterion for the oiliness which is equivalent to the measurements on the Thoma testing machine.

For the investigation of mineral oil, in addition to the usual data concerning its physical behaviour in relation to the extended hydrodynamic theory, it is necessary to know, besides the viscosity, the specific gravity, the specific heat, and the heat conductivity, all in relation to the temperature. Thus, two oils are of equal value as lubricants when the above four values correspond in any particular range.

Summary. The idea of oiliness as the cause of variations in friction with equal viscosity is based:—

(1) On first approximations of the hydrodynamic theory, which neglect the influence of internal friction on the heating-up of the lubricant; and

(2) On experiments in which the viscosity is considered to be equal on the grounds of unsatisfactory temperature measurements.

The notion of oiliness has neither an experimental nor a theoretical basis. If the heat produced in the oil film through internal friction is taken into account in the Reynoldsian sense, then there result the phenomena of bearing friction which have previously been included under the term "oiliness."

A new criterion for lubricants consists of the relation $\beta/\gamma c$, which represents the measure of the steepness of the viscosity curve divided by the product of the specific gravity and the specific heat. The smaller this value is, the better are the lubricating properties of the oil. In addition, there is the influence, which is still undecided mathematically, of the heat conductivity, in the sense that a large value of λ has a favourable effect.

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THE PRINCIPLES OF THE TESTING OF LUBRICANTS IN THE LABORATORY AND ON THE TEST BED

By Dr. A. von Philippovich*

The pure chemical or physical investigation of materials by suitable scientific methods enables their chemical or physical properties to be ascertained. This is usually done by means of a single test condition, and not from the point of view of one of the many practical possibilities. Thanks to careful control of the experimental conditions the property ascertained in this way has a definite significance, in other words it is a real property of the material which can be considered objectively as established. Such objective characteristic values are few in number, and include the specific weight, refraction, dispersion, absolute viscosity, aniline point, ash content, and the content of carbon, hydrogen, oxygen, and foreign matter. On the other hand, numerous methods of test yield values which are largely determined by the choice of the experimental conditions. In such cases one can hardly talk of objective properties and it is better to express the result as the behaviour under certain conditions of test. Instances of these are the flash point, ignition point, cold test, asphalt content, oxidation behaviour, corrosion behaviour, and so on.

The "objective" properties of lubricants are of little significance for practical use. As the one practically important property which can be measured accurately there remains the viscosity, and its significance will be intrinsically reduced as soon as it is possible to define oiliness with accuracy. The reason for this is that the engine can hardly be conceived as a machine but rather as an organism, so that the individual purely chemical or physical properties reveal little or nothing about the behaviour of the oil under the manifold operating conditions. This is why attempts are made to simulate engine conditions in the laboratory, especially as regards the formation of deposits and stability. However, the more complex the processes in the engine become the more difficult does it become to choose the right condition of test from amongst the many possibilities. That this is true is shown by the many processes for the determination of particular properties of the oil, such as its behaviour at low temperature, its stability, and so on. The aim of laboratory investigation is in the last resort the complete prediction of the behaviour of the lubricant in the engine, i.e. the indication of its suitability or unsuitability for a particular application. To this end, the behaviour in the laboratory and in the engine must be

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* Deutsche Versuchsanstalt für Luftfahrt.

compared. In this respect, the question must be answered: to what extent can the behaviour of a lubricant in the engine be used at all to control laboratory tests?

Lubricant and Engine. It is necessary to know what is meant by behaviour in the engine. An oil must certainly be serviceable under all working conditions for it to show good behaviour in the engine. The resistance to cold, to the formation of deposit, to ring sticking, to use, to abrasion, in short, all its properties, must be determined and expressed numerically. Moreover, there are not only many types of internal combustion engine, but their working conditions are extremely variable. For each of the properties chosen each engine will have some other requirement. If a lubricant must be suitable under all the working conditions then the test in the engine must be devised accordingly. To do this there are two methods: to carry out a service test of the oil and to carry out a test in a single engine under different conditions.

Service Tests. In practice it is only with the greatest difficulty that the working conditions can be controlled accurately, as unexpected variations can occur, which affect the results but are often not reported. Therefore the accuracy of the evaluation of an oil in a single test remains uncertain, and can only be ensured by numerous tests on many engines of the same type. On account of the high cost of such tests it is only possible to use oils which do not harm the engine and are expected from experience to be suitable. Although service tests yield results which are highly reliable, such results are only valid for the engine in question, while the tests are expensive and can only be made with oils which are presumably appropriate.

Engine Tests. Engine tests require less time when they are carried out on the test bed expressly with the object of testing the oil. Naturally, various properties are neglected, such as the behaviour at low temperature, unless these are expressly investigated, though wear, consumption, stability, and ring sticking can be tested with a considerable degree of accuracy, if conditions are controlled with care. This is much more difficult than would appear at first sight. The distribution of the mixture, the specific consumption, the cooling, running time, oil supply, the clearance of the piston and piston rings must be always the same, otherwise the results from one and the same oil will vary more than those from different oils. The chief disadvantage of engine tests, however, is their extremely high cost, which makes it impossible to carry out large numbers of such tests. The question is, therefore, whether the same end can be attained by using a single-cylinder engine.

Testing in a Single-Cylinder Engine. If the test is restricted to a single-cylinder engine, then the results obtainable are still further limited. Whereas in fuel tests it appears quite possible to reproduce the conditions of an aero-engine in a single-cylinder engine, in testing oils the transition from one engine to the other implies a series of essential variations. In particular there is less change in the oil and in the consumption, while the temperature of the oil in the crankcase is much too low, so that it requires to be heated. In any case the oil circulation is not comparable with that in the aero-engine. The wear, too, is quite different, while it is impossible to make statements regarding behaviour at low temperature, abrasion, and stability which are based on comparable figures. The only part of the engine which works under the same conditions as in the aero-engine is the cylinder and the behaviour of the oil in the cylinder, i.e. its ring-sticking properties, is the sole object of such tests. As such tests cost less than those conducted with aero-engines and the conditions of test are more easy to control, they are really useful for the investigation of oils.

Causes for the Diversity of Results from Engine Tests. On considering the causes of differences in the results from engine tests, it would appear possible to modify the experimental conditions and so obtain a proportional evaluation of different oils. The behaviour of an oil is tested in a single-cylinder engine to find its ageing characteristics and particularly its tendency to ring sticking. Here, properties such as viscosity, volatility, resistance to oxidation and carbon formation, the reactivity of the carbon, the solubility of the asphalt, and lastly, to no small extent, the oiliness, come chiefly into question. But these properties change with pressure, temperature, and reaction time as well as with the dispersion, and all these must be known for different engines, in addition to mechanical factors such as piston velocity, the clearance of piston and piston rings, and the rate of heat transfer. Were all the relations to be explored, the number of researches required would be infinite, so that attention has to be concentrated on the most important factors.

Choice of Test Engine; Reproducibility of Results. It does not follow that results obtained with one engine will be applicable to other engines. Owing to variations in working conditions it is impossible to obtain results which are generally applicable, particularly if the test engine runs under a comparatively small load. Further, the results are not even applicable to other engines of the same type if the test conditions differ too much from service conditions. Thus if engine tests of an oil are to be made, not only must the test engine be either of the same type as, or very similar to, the service engine, but the conditions of test must correspond to service conditions. It can perhaps be said that an air-cooled aero-engine which is thermally highly stressed will yield results applicable to other aero-engines of similar construction which are also highly stressed thermally, or a water-cooled engine which is lightly stressed thermally will yield results applicable to another engine of the same characteristics; the results are not interchangeable to any great extent. If an oil is to be tested, e.g. for use in aero-engines, an engine or a cylinder must be chosen which is characteristic of such engines and makes the highest demands on the oil. It is then supposed —not wholly rightly—that an oil which behaves satisfactorily in an engine under high thermal stress will be suitable for an engine with less thermal stress. It must be remembered that the engine test has only restricted validity and that the results can be applied to engines of another type only when the working conditions are taken into account.

Tests in Small Engines. Even when a large engine is used for testing, the results obtained are not entirely applicable to service conditions. The results from a single-cylinder engine are still less conclusive, and even these are complex and expensive to obtain. Thus the question is whether a small engine can and may be used to test oils.

To shorten the running time, tests with a single cylinder from an aero-engine are often made more severe than service conditions. It is then supposed that the resulting grading of the oil is similar to that arrived at in practice. By careful choice of experimental conditions the same results can be obtained in a small single-cylinder engine as in large single-cylinder engines, but only when a definite characteristic, in this case the running time till efficiency decreases, is examined. If, however, such an agreement is obtained between a large and a small engine, then by altering the conditions of test in the small engine, the appropriate relations in other engines will probably be covered.

What are these alterations of conditions? Those which chiefly affect the oil are pressure, temperature, time (speed), dispersion, and catalytic effects due to metals. The most important condition appears to be the temperature, although the speed plays an important part, particularly as it, with the viscosity, influences the whole mechanical degree of distribution. These two conditions are sufficient to enable a small test engine to represent a very large range of engines, remembering that typical conditions in a small engine can be repeated only with difficulty: the play of the piston and piston rings, oil distribution and movement, the range of oil temperature will always be different. Nevertheless the behaviour of the oil in the engine in respect of ageing and, especially, ring sticking should be examined not under a single test condition but for a whole range of temperatures and at different speeds.

The engine performance found in this way must now be related to a series of large engines, and it should be ascertained whether the behaviour of the oil in the large engine can be deduced from the temperature or speed diagram of the small engine by taking the conditions in the large engine into consideration. But to do this the appropriate conditions must be selected.

Working Conditions in Aero-Engines. The speed and the piston velocity are simple characteristics which are as easily specified for the large engine as for the test engine. The temperature is a different matter, and though the temperature of the oil in the crankcase and the temperature at the piston can be utilized to test the tendency of the oil to age, to estimate its ring-sticking characteristics, the temperatures of the cylinder wall, cooling medium, cylinder head, piston head, and the piston ring belt should be taken, although the only interesting temperature, that of the oil on the piston, is not included. The essential point in comparing the behaviour of the oil in different engines is that too little is known about the actual effect of the working conditions on the oil as, for example, the temperature attained by the oil in the cylinder. For this reason, an important object of future work will be to ensure that working conditions in the engine are carefully measured by physico-chemical means.

Relation between Engine Tests and Laboratory Results. This discussion of engine tests shows that "engine performance" is a complex question which cannot be correlated on the basis of a laboratory experiment carried out under one test condition. Either the abrasion, the formation of carbon, ageing, or ring sticking must be selected as a criterion. For the comparison or evaluation of a laboratory test numerical data are required from the test bed with a complete aero-engine or a suitable single cylinder engine, or from the test bed with a small single-cylinder experimental engine, to show the behaviour of different oils under the same conditions. But the same difficulty appears as in the comparison between the test engine and other types of engine: results from a single method of test cannot be compared with the results from many different types of engine without arriving at wrong conclusions. The laboratory tests must therefore be modified properly. The factors of time and temperature can easily be varied, so as to obtain a diagram of the behaviour of the oil in the laboratory (asphalt formation, volatility, coke formation, etc.), which can be compared with the behaviour of the oil in various types of engine.

The relation obtained in this way would be justified if it was based on measurements of temperature, time, etc. in the engine. As only comparable things should be compared, laboratory results can hardly be compared with engine performance. For this reason the chief task awaiting research is to carry out a systematic determination of the conditions under which the lubricant works in the engine, and to develop methods by which the oils are exposed to the conditions thus ascertained or tested within the appropriate range of temperatures, so that a reliable basis can be obtained for the comparison of results obtained in the engine and in the laboratory.

The methods of testing oils can be classified as follows:—

Laboratory experiments	Small single-cylinder engine	Large single-cylinder engine	Aero-engine
(1) Oiliness . .	—	Comparative values	Gauging values in service and on test-bed
(2) Carbon formation	Comparative values	Comparative values	Gauging values in service and on test-bed
(3) Ageing . .	Comparative values	Comparative values	Gauging values
(4) Ring sticking .	Comparative values	Gauging values	,, ,,

To some extent laboratory methods have still to be worked out and modified in such a way that, instead of measuring a single characteristic, they cover several points of a temperature or time diagram.

Tests of single-cylinder engines must be carried out under various conditions so as to be applicable to engines of very different construction or working conditions. Aero-engines, however, must be tested under their usual working conditions, particularly in respect of temperature.

THE RELATIONS BETWEEN THE CHEMICAL CONSTITUTION AND THE PROPERTIES OF LUBRICANTS.

By Dr. Zorn*

On reading the titles of the papers to be presented at this Discussion, the chemist is astonished to see how much the engineer demands improvements in the quality of lubricants. In particular, the problems of the lubrication of internal combustion engines attract most attention. The service demands of aero- and automobile engines are continually increasing, but a limit is set by the quality of the available lubricating oil. The chief properties of lubricants which can be improved and the direction in which improvement could be made can be stated as follows:—

(1) *The temperature-viscosity relation.* The course of the temperature-viscosity curve, expressed as the viscosity index or the pole height, should be as flat as possible.

(2) *Resistance to ageing.* The oil must age as little as possible under use; that is, it must form a minimum of sludge and asphalt. In particular, it must, as far as possible, not deposit substances which cause ring sticking.

(3) *Carbon formation.* This should be as small as possible and any carbon formed should be soft and not adherent.

(4) *Oiliness.* This should be ensured under high pressures and at high speeds.

If these properties are to be improved, then the efforts of the chemist can be fruitful only when their dependence on the chemical constitution of lubricating oil of high molecular weight has been investigated. The more complete the purely scientific study of this question, the more successful will be the work of the refiner, as he will know the limits constituted by the chemical structure of the crude oil and will be able to utilize synthetic methods and products at the right place and in the right way.

Unfortunately, the amount of work on the lubricating properties of hydrocarbons of high molecular weight which are chemically homogeneous and of known structure is very limited. The first to show the way to a solution was Spilker (1891), who suggested that the concentration of methyl groups in the aromatic molecule resulted in increased viscosity. Recently, Hugel (1933), Miheska (1936), and Landa (1930–33) have studied the dependence of viscosity and the temperature-viscosity

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* I. G. Farbenindustrie A.-G., Berlin.

relation on the constitution of synthetic lubricating hydrocarbons. The results obtained may be summarized as follows:—

(1) Increasing the number of cyclic carbon atoms of a hydrocarbon molecule increases the viscosity. The degree of increase depends on the ring structure of the molecule.

(2) The number and order of the carbon bonds determine the temperature-viscosity relation of hydrocarbons.

(3) Hydrogenation of the aromatic ring increases the viscosity but impairs the temperature-viscosity relation.

(4) The concentration of aliphatic side chains on the aromatic ring increases the viscosity. The effect of the side chains increases with their length and their lack of branches. The straight unbranched carbon chain has the best effect on the temperature-viscosity relation.

(5) The viscosity of aliphatic hydrocarbons depends on the way in which the carbon chains branch.

(6) A double bond in the aliphatic chain improves the temperature-viscosity relation when it is conjugated with the double bonds of the aromatic ring. The function of the double bond in pure aliphatic hydrocarbons is not yet clear.

That is, on broad lines, the essence of present knowledge of the dependence of viscosity on chemical constitution. But it is not sufficient to provide an answer to all the questions which require solution. In particular, the action of the mutual influence of different hydrocarbons on viscosity requires investigation. In other words, the behaviour of synthetic mixtures requires close examination. Thus, the study of the behaviour at low temperatures of structurally different hydrocarbons is technically important.

Mention may be made of the study of the behaviour of hydrocarbon mixtures in relation to colloid phenomena. For instance, hysteresis was first found in natural lubricating oils by Schneider and Just (1905), confirmation being given by Glazebrook (1915). Dunstan and Thole (1917) describe heavy oils as isocolloids, i.e. systems in which the dispersed components are of the same chemical nature as the dispersing medium. Ward (1934), utilizing X-ray methods, found that a mixture of benzole and cyclohexane resembled an emulsion. Weisz (1931) designated the dependence of the time taken by cooled oils to flow from a capillary on the period of cooling and the suction as pseudoplasticity and assumed that it depended on the structure of the molecules and their fields of force.

Though a little is known about the relation between chemical constitution and viscosity, the knowledge of the behaviour of well-defined hydrocarbons of high molecular weight when attacked by oxygen is most unsatisfactory. Further research is urgently required on this important practical question. Cherneshukov and Krein (1933) have investigated the action of oxygen or air on numerous synthetic aromatic and naphthenic hydrocarbons in a Butkow bomb under a pressure of 15 atmos. at a temperature of 150–250 deg. C. for 3–6 hours. The following results were obtained with aromatic hydrocarbons:—

(1) Aromatic hydrocarbons with no side chains were either oxidized weakly or not at all. The resistance to oxygen was reduced by the introduction of an aliphatic carbon atom between the rings (diphenylmethane is less resistant than diphenyl). The more unsymmetrical and complex the structure, the less the stability. The oxidation products contained more resin-like polymers than acid products.

(2) Aromatic hydrocarbons with side chains are much less resistant to oxygen, their resistance being less, the larger the number and length of the side chains. Oxygen attacks the side chains first, forming acid bodies, including liquid acids. The formation of condensation or polymerization products decreases with increasing length of the side chains. Their chemical nature is different from that of the products from aromatic hydrocarbons without side chains. With the latter, the condensation and polymerization products are insoluble in petroleum ether and are of an asphaltenic nature, whereas, with the former, they are soluble in petroleum ether.

The results obtained with naphthenic hydrocarbons such as cyclohexane, methylcyclohexane, Detalin, and a few naphthenic hydrocarbons from petroleum were as follows:—

(1) The ease with which naphthenes oxidize increases with rising molecular weight. The introduction of side chains again increases the sensitivity to oxidation, but the oxygen does not attack the side chains (as in the aromatic compounds), but splits the ring at the point where the side chain is attacked. The oxidation products contain free and esterified acids and oxyacids.

(2) The sensitivity of the naphthenes to oxidation is reduced by adding aromatic hydrocarbons without side chains, which act as "anti-oxygens" and are themselves oxidized, though the conditions are such that, alone, they would not be oxidized. The naphthenes have to a certain degree an inductive action on aromatic hydrocarbons.

(3) The addition of aromatic compounds with side chains, to the extent of up to 10 per cent, also reduces the sensitivity of the naphthenes to oxidation. A further result of this addition is that asphaltic condensation products reappear amongst the oxidation products. Fig. 1 illustrates the "anti-oxygenic" action of various aromatic compounds on a naphthenic Grossny vaseline oil. Of interest is the strong anti-oxygenic

action of triphenylmethane and of the aromatic hydrocarbons isolated from a fraction of the Grossny oil boiling at 350–400 deg. C.

Thus these investigations shed some light upon the cause of the formation of asphalt and sludge. Under the action of oxygen, aromatic hydrocarbons form asphaltic bodies, whereas naphthenes form acid products which are soluble in oil. The removal of aromatic compounds from mineral lubricants by catalytic hydrogenation under pressure or by the use of selective solvent extraction may improve lubricants as regards their tendency to form sludge and asphaltic substances. But, as is shown by the "anti-oxygenic" effect of added aromatic hydrocarbons, too complete a removal of the aromatic bodies can be detrimental. This, perhaps, affords an explanation of the different behaviour of various oils refined by the selective solvent process, as was recorded by Born and Harper (1934). Further, the different "anti-oxygenic" behaviour of the various aromatic hydrocarbons indicates that the development of a

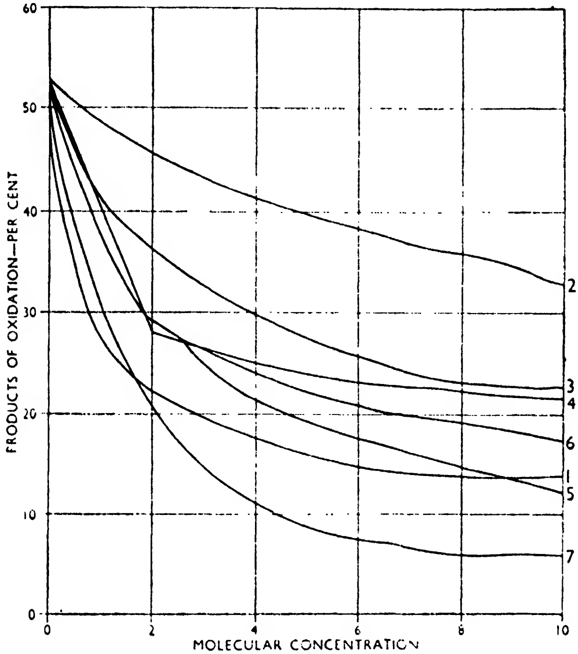

Fig. 1. Oxidation of the Hydrocarbons of Lubricating Oil Fractions from Crude Oil

1, $(C_6H_5)_3OH$; 2, $(C_6H_5)_2CH_2$; 3, acenapthene; 4, naphthalene; 5, anthracene; 6, phenanthrene; 7, aromatic compounds from Grossny asphalte-base crude oil from fraction boiling at 350—450 deg. C.

product with strong "anti-oxygenic" properties should be possible by means of systematic structural changes. This scientific study is thus a support for the rapidly developing subject of anti-oxidation compounds.

In conclusion, mention may be made of a few papers dealing with the relation between chemical constitution and oiliness. Dover (1935) determined the coefficients of friction of the following hydrocarbons with a Herschel machine: *n*-octodecane, $C_{18}H_{38}$, 0·096; *n*-octodecene, $C_{18}H_{36}$, 0·106; di-*m*-tolylethane, $C_{16}H_{38}$, 0·230; and tetratricontadiene, $C_{34}H_{66}$, 0·102. These figures show that the double carbon bond and the unsaturated nature of the hydrocarbons appear to have no significance for lubrication. Even the molecular dimensions appear meaningless according to these measurements when *n*-octodecane is compared with *n*-octodecene and tetratricontadiene. Further, the comparison of these compounds shows that aromatic compounds are the poorest lubricants. This observation is in agreement with the conclusion, due to Ubbelohde and Jaenisch (1935), that the aromatic hydrocarbons also behave most unfavourably in respect of their load capacity or resistance to pressure. These few results are the first contributions towards the explanation of the lubricating properties of hydrocarbon oils, but further research is most essential to ensure technical progress.

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REPORT ON PAPERS IN GROUP IV

PROPERTIES AND TESTING

By H. J. Gough, M.B.E., D.Sc., Ph.D., F.R.S., M.I.Mech.E.
(*Member of Council*)

The valuable papers contributed to this group may conveniently be discussed under the four important and independent subjects of boundary lubrication and wear; viscosity; oxidation tests; and the testing of ball bearing greases.

I. Boundary Lubrication and Wear

(*a*) *Boundary Lubrication.* Adam contributes an interesting review of surface phenomena, describing the mechanism of seizure of clean surfaces, and the function of the boundary layer in first cushioning the contact between the roughnesses of the surfaces and then providing easily-broken connexions when there is approach within the range of molecular attraction. Kyropoulos attributes the advantage derived from longer molecules, not so much to the greater elasticity of those molecules, as to the consequent increase of internal molecular mobility.

Both authors emphasize the small range of the surface forces and this is interesting in view of the known differences of friction obtained with different metals. On this point, Kyropoulos reasons that the lattice forces of the adsorbing surface ultimately control the lateral packing of the molecules of the boundary layer. This leaves the free ends of those molecules with residual fields of force, thus accounting for the surviving influence of the surface at a distance exceeding the range of the original surface forces.

Adam favours a single, or at most a few, molecular thicknesses to the boundary layer, while Finch and Zahoorbux show that the electron diffraction pattern showing orientation disappears with the third molecular layer of a long-chain hydrocarbon. Trillat, however, states that, both by calculation from the results of adsorption tests and by electron diffraction, stratification of ten or more layers has been found with saturated fatty acids.

Finch and Zahoorbux describe the effect of a polar group at one end of the adsorbed molecule in diminishing the angle at which the molecules are orientated with respect to the surface; depending on the size of the group, this effect may become so marked as to destroy the layer-line pattern, but the addition of a pure hydrocarbon restores the former orientation.

Adam refers to the part played by the reserve supply of polar bodies in the bulk of the oil in repairing the film. Kyropoulos states that the "accessibility" of these polar bodies to the surfaces depends on whether they do or do not form complexes by dipole association within the liquid. Trillat measures the rate of formation of the adsorbed layers by the rate of change of oil-water interfacial tension. Miss Nottage's measurements show that some change of the adsorbed layer must occur with temperature in the case of palmitic acid additions to B.P. paraffin to get a decrease of coefficient at first as the temperature is raised; there is no return to higher values on cooling. The influence of the addition of a non-polar wax in eliminating this effect of temperature is ascribed by Miss Nottage to an increase of the dispersion of the acid, thus improving the accessibility to the surfaces.

With regard to commercial oils with their very different components, Kyropoulos points out that the boundary layer would, to some extent, be expected to change continually in composition. In this connexion, he claims that the only direct method by which measurements can be obtained of molecular anisotropy *and* concentration of the desirable long molecules in coloured oils is his flow-orientation method—a hydrodynamic treatment of the oiliness problem. Trillat finds adsorption with commercial oils as with oleic acid-oil mixtures, and in fact puts forward his interfacial tension method as one very sensitive and suitable for the examination of commercial oils, expressing the activity in equivalent oleic acid content; he remarks that the French Air Ministry use an autographic form of this apparatus.

Trillat finds the adsorption at solid surfaces by testing the oil before and after passing over metal balls, through cotton wick, glass wool, and filter paper. The same effects are found for commercial oils as for oleic acid-oil mixtures, and Trillat states that this shows the disadvantages of the use of filtration in cleaning used oils. Burwell and Camelford, making kinetic friction tests with a fitted journal bearing, claim to have shown the adsorption of the active molecules by metallic powders by the adverse influence on the friction-temperature curves, and also by saponification tests; also by the decrease of the rate of flow of an oil through a tube tightly packed with strands of copper wire.

Adam, also Finch and Zahoorbux, refer to the orientation of graphite, with the cleavage planes parallel to the surface; this provides easy shear when the high points of the bearing surfaces come into contact, thus preventing local seizure of the actual metals.

Many references are made to surface finish, and reveal a very satisfactory appreciation of the relatively enormous size of the irregularities of even a finely finished surface in comparison with the thickness of the boundary layer. As Kyropoulos points out, the difficulties in the experimental study of boundary lubrication are largely due to the interference of surface irregularities in the tests. Both Heidebroek (in a paper in Group I) and Brillié point out that even in the hydrodynamic region of thin film lubrication of bearings, these irregularities have a local influence on pressure film formation; Brillié also suggests that swirls will be set up in the oil in the valleys, thus increasing the friction. Brillié also considers that the neutralization of the surface forces in the formation of the boundary layer leads to friction at the surface less than viscous friction, giving, in fact, boundary slip; the friction is thus decreased, but the safety of the bearing is lowered as the film thickness is diminished. Kyropoulos refers to his explanation of certain "oiliness" effects in hydrodynamic lubrication on his flow-orientation hypothesis; this flow-orientation is accompanied by a decrease of viscosity of 5–17 per cent. Both authors claim experimental proof. Fogg, however, explains variations of kinetic friction of an oscillating bearing from temperature and speed of oscillation as due to varying proportions of fluid and boundary friction; below 3 cycles per minute, plain boundary friction seems to be obtained. Thus, several explanations are offered to account for small differences in this transition stage from hydrodynamic to boundary lubrication; this affords a subject for discussion.

The consideration of high-point contact in relative motion leads logically to mention of the high temperatures produced at those contacts which have been demonstrated by Bowden and, later, by Blok for gear teeth. By simultaneous measurements of friction and temperature, Bowden and Leben have now shown that sliding is a discontinuous process and that when slip occurs there is a momentary jump in the temperature. The high temperatures lead to flow of the material, and, it is suggested, to formation of the amorphous Beilby layer, the importance of which layer is emphasized by Finch and Zahoorbux as giving more perfect orientation of the boundary layer. Blok works out the temperature rise for sliding contacts for a number of ideal cases.

(*b*) *Wear.* Turning now to the papers on wear by Neely, Brownsdon, Southcombe, Wells and Waters, Clayton, and van der Minne, these contain a large amount of interesting information, which is of a rather miscellaneous character and is thus difficult to summarize adequately; some of the salient points will be mentioned.

By means of a wear and friction testing machine, in which bronze buttons wear on a hard steel track, Neely shows that the effectiveness of addition-agents on the wear-reducing value of lubricants may be quite different in degree and sometimes in direction from the effect on friction; in that author's opinion it is therefore futile to attempt to predict wear-reducing value from friction tests. This is such an important matter that it should receive further attention.

Clayton describes tests in the four-ball apparatus under sliding contact with hard steel balls. A striking feature of these results is that not only are the breakdown load and wear in the expected order of increasing quality of mineral oil, fatty oil, and extreme-pressure oil, but the times to recovery from the high coefficient of friction of about 0·5 at seizure are differentiated in a similar way, mineral oils giving long ragged seizures and extreme-pressure oils recovering in from one to two seconds. Whereas additions of oleic acid and fatty oil to a mineral oil have only a slight beneficial effect, the unsaturated substances in an oil only moderately refined give it markedly good qualities.

Brownsdon has tested the effect of a large number of substances when added in small proportion to a mineral oil on the wear of brass by a hard steel wheel. As the author states, "no single characteristic can be put forward as the cause of their behaviour."

Southcombe, Wells, and Waters have carried out tests under a variety of conditions, using a Timken machine. These authors state that that property of certain lubricants which has been called "high film strength" does not appear to exist, and they conclude that an extreme-pressure lubricant must be defined as one which permits the metal to wear away smoothly when the load intensity approaches a certain limiting value—under their conditions of test. Their measurements of the coefficient of kinetic friction show that, at high intensity of pressure with an extreme-pressure lubricant, the friction falls when a temperature is reached at which decomposition of the lubricant occurs. This result is in agreement with one quoted by van der Minne. It may also be noted that Brownsdon regards the beneficial effects of some additive substances as due to decomposition at the contact and activity on the part of the resultant products. Van der Minne also describes results of tests on a series of oils with four testing machines. He explains the marked differences found as being due to differences of duty (load and speed): at low duty, "oiliness" characteristics are advantageous, while at high duty the high temperatures bring out the good features of the extreme-pressure lubricants. Ordinary "oiliness" characteristics may be a disadvantage under extreme pressure and temperature conditions. As a result of his work, this author suggests there exists a need for apparatus giving even severer conditions than the hypoid gear itself, and that different testing machines will have to be employed for different applications of the lubricant.

Reviewing this section it appears that there is a fairly satisfactory understanding of the boundary layer feature of "oiliness" from the physico-chemical aspect. More results are required, however, from static and kinetic friction tests to provide a firmer mechanical basis for these conceptions, and to show their practical significance. Neely's paper opens up an avenue for related wear and friction tests. In these fields, as well as in the severe seizure region, the necessity for comprehensive investigation of the effects of the different variables is apparent, and van der Minne's paper gives encouragement to expect reduction of apparently chaotic results to some order when sufficient information is available.

II. Viscosity

Groff describes the use of a chart for the graphical solution of a number of problems relating to the viscosity-temperature characteristics of oils.

When the need for more accurate viscosity measurements arose five years ago, the American Society for Testing Materials investigated the use of capillary tubes and has now adopted this method as a tentative standard. As in England, a resolution of the Petroleum Congress of 1933 has thus been implemented. Geniesse describes the suspended-level and modified Ostwald viscometers used, and the methods of calibration. An accuracy of ±0·2 per cent is claimed. A table for conversion between Saybolt seconds and centistokes is given, and the new viscosity-temperature chart is described. The general use of kinematic equipment is advocated.

Barr describes viscometer bath arrangements he has used for high-temperature measurements. He then considers several formulæ which have been proposed for expressing the relation between viscosity and temperature, and suggests a new empirical formula which gives better agreement with experiment than Walther's.

Suge describes his methods and the results obtained in measurements of viscosity at high temperature and pressure. The pressure ranged up to 2,000 kg. per sq. cm. (nearly 13 tons per sq. in.), and the temperature to 150 deg. C. At ordinary temperatures, mineral oils are affected by pressure more than fatty oils, but they behave similarly at high temperatures. These results are interesting in connexion with papers by Needs, also Bradford and Vandegrift, in Group I, on the pressure effect on viscosity in bearings; reference may also be made to Dow's results for mineral oils quoted in the latter paper.

Suge has also determined the compressibility of oils by measuring supersonic wave velocity. The product of the surface tension and the coefficient of compressibility was found to be closely constant, so that compressibility can be found from surface tension measurements. He also found the thermal conductivity of mineral oils to be at least 15 per cent less than that of the vegetable oils tested.

Barnard describes work by Arveson with a constant-shear viscometer over a wide range of rates of shear. In all thickened products the apparent viscosity tends to approach that of the base oil at high rates of shear, while the base oil shows no change in apparent viscosity. Results are given for the relation between the viscosity and its two determining factors (temperature and rate of shear) of a wax-bearing oil; it is stated that this is the first time this relation has been determined for this kind of oil.

At high rates of shear, temperature rise becomes marked. Brillié makes some calculations for a bearing oil film, the temperature rise of the oil being very high compared with that of the bearing surfaces. He makes suggestions for the modification of plain bearing surfaces by grooves, oil pockets, etc., to reduce these increases of temperature.

Thus the papers by Groff and Barr simplify the use of viscosity data for the practical user, while Geniesse's paper is important from the point of view of international standardization and simplification. The other papers add to our knowledge of the phenomena of viscosity.

III. Oxidation Tests

Andrews finds, using turbine oils, that the property of forming an emulsion with water varies with the age of the oil; the rate of separation from the emulsion is used as a test of the condition of the oil in service. In testing new oils they are aged by heating in contact with air, using copper strip as an accelerator, and then subjected to the emulsion test. Details of the tests are given and the limits of acceptance. Good correlation is found between the tests and the behaviour in steam-driven turbo-alternators and air compressors.

Considerable attention is given by the other authors to the more difficult problem of the oxidation of internal combustion engine oils. Moerbeek points out the unsatisfactory character of the multitude of oxidation tests that have been and are being suggested; disagreement is shown between the results of different tests, while there is a lack of correlation between these results and the behaviour of the oils in actual engines. It is stated that the large oil producers have little or no confidence in oxidation tests, and are guided by the results of extensive tests made on engines of representative types. In Moerbeek's opinion, it seems impossible that an oxidation test will ever be devised that will show reasonable correlation with all classes of engines, and unlikely for even a reasonable proportion of one class.

Von Philippovich gives an able review of the many difficulties and necessary reservations in laboratory and engine tests; he refers to full-scale service and bench tests, single-cylinder engine tests, and small engine tests. He considers the chief task awaiting research is to carry out a systematic determination of the conditions under which the lubricant works in an engine, and to develop methods in which the oils are exposed to these conditions or tested within the appropriate range of temperatures, so that a reliable basis can be derived for the comparison of results obtained in the engine and in the laboratory. He emphasizes the necessity for covering a range of each important variable both in laboratory tests and in engine tests.

In the light of these comments it is interesting to consider the remainder of the papers in this section. Mardles and Ramsbottom give extensive results of the type favoured by von Philippovich; they cover a wide range of temperature, time of oxidation, and method of oxidation. Hanson and Egerton introduce what appears to be a new factor into the problem. They are interested in gumming as affecting ring sticking; as gumming sets in more rapidly when knocking occurs, and as nitrogen oxides are formed particularly under these conditions, they have investigated the possible connexion between the two. In a modified Air Ministry test they have found that nitrogen oxides, in a concentration similar to that in the engine, have a marked catalytic effect on oxidation. Nitrogen peroxide tends to remove the induction period of oil oxidation. and also increases the rate of subsequent oxidation; it has a very marked effect on an oil containing a large proportion of castor oil.

While these results may help to explain some of the discrepancies and inconsistencies that have been found, they add to the complications of test. Moutte, Dixmier, and Lion follow a practical line which shows promise of improved correlation between laboratory test and engine behaviour. On the basis of the solubility or insolubility in chloroform of the products of oxidation separated from the oil by extraction with petroleum ether, and of the oxygen content of the two proportions, oils oxidized at low temperatures ($\ngtr 140$ deg. C.) and at high temperatures (200–400 deg. C.) have been examined. The quantities and qualities of the product were not comparable with those in the engine, until the oil which had been aged at low temperatures was also aged at high temperatures. The results are not yet equivalent to those obtained in the engine, but show a distinct tendency to approximate to them. The authors are cautious in stating conclusions drawn from their work, yet it appears that the combination of two temperatures in the oxidation test is a logical and promising development.

Evans and Kelman, on the basis that oxidation occurs on the cylinder walls of an engine, have heated the oil at 180 deg. C. in a Petri dish in the presence of iron, and claim that the consequent viscosity modifications are in closer correlation with engine tests than when blowing and absorption tests are carried out. They have found that some metallic soaps (e.g. tin oleate) can inhibit the catalytic action of iron on oxidation. The laboratory work was checked by single-cylinder engine tests;

good general agreement was found, but the best proportions and the order of effectiveness of tin and chromium were rather different. The best results were obtained with a mixture of tin and chromium oleates.

In this section may be mentioned the paper by Norlin giving the specifications and notation for motor oils in Sweden, with the author's comments on the significance of the several laboratory tests.

Thus while Moerbeek is pessimistic regarding the value of oxidation tests for internal combustion engine lubricants, several other authors present encouraging results which raise the hope that the present concentration of attention will lead to the development of some laboratory test, or tests, which will give a satisfactory guide to the behaviour of an oil in an engine. Arising from the paper by Moutte and his colleagues the possibility might well be discussed of adopting a test in which the oil is subjected, say cyclically, to moderate and high temperatures to imitate engine conditions in this respect.

IV. The Testing of Ball Bearing Greases

Kjerrman, also Cooper and Richardson in papers in Group III, refer to various tests of ball bearing greases. The importance of the practical running test is emphasized. Laboratory tests mentioned include drop points, acidity or alkalinity, content of foreign particles, etc. The alkali greases can absorb a considerable amount of water to form an emulsion. Kjerrman considers that this emulsion can be a protection against rust, but, in the opinion of Richardson, there is a danger of rusting by the emulsion if the bearing is left standing for long periods.

Concluding Remarks

The variety and amount of interesting information presented in this series of papers affords striking evidence of the great attention now being given to research in lubrication. But it is apparent that yet more information by comprehensive testing is required, particularly in connexion with boundary lubrication, wear, and oxidation tests. There is at least one direction in which it seems probable that co-operation might very usefully be effected, namely, if an arrangement could be made by which a common stock of a basic series of lubricants could be made available to investigators of lubrication problems. Such an arrangement would result in a much greater degree of correlation between results from different machines and with varying technique than is at present possible. Usually, not only are different lubricants used, but they cannot be sufficiently closely specified to allow other workers to use similar oils. This point might well form one of the topics for discussion.

The assistance of Mr. D. Clayton in preparing this summary is acknowledged.

DISCUSSION ON PAPERS IN GROUP IV

15th October 1937

The CHAIRMAN, Sir William Reavell (*Past-President*), in opening the meeting on 15th October 1937, at 10 a.m., said that the proceedings that day would represent perhaps the most important of the discussions which had been held during the week, because, however interested engineers might be in the various types of bearings which they proposed to employ, they required to have a good working knowledge of the properties of lubricants if the results of the machines which they designed were to be first-class. They ought, therefore, to have a clear conception of the meaning of the physical tests applied to the lubricants which they desired to use. With regard to chemical tests, it was most desirable that engineers should have some knowledge of and realize the effects of oxidation in oil, of gumming, and of the question of residues, the nature of those residues, and the effect which they would have in use. He would like to put in a plea for simplicity as far as possible in laboratory testing, so that designers and engineers generally could more readily interpret the real meaning of the laboratory tests which were carried out and their practical applications. It was important, if proper use was to be made of the important investigations which chemists conducted and the conclusions at which they arrived, that they should, as far as possible, be expressed in terms which could be readily assimilated.

Before coming to the meeting, he had received such a concise and clear definition of the objects of the discussion that he would like to read it: "To endeavour to establish a correlation between theory and practice, to show how bearing design can be applied, to consider the relationships between static and kinetic friction, to relate academic research with engineering practice, to classify the views upon bearing metals themselves, and to review and appreciate the significance of laboratory tests." If those present succeeded in achieving all those objects and clarifying them in their minds, they would have done a great deal that morning. He then called upon Dr. Gough to read his report on the papers.

Dr. H. J. GOUGH, M.B.E., F.R.S., then read his report (p. 465) on the papers in Group IV.

Discussion

Professor E. NORLIN (Statens Provningsanstalt, Stockholm) said there was, as some speakers had remarked, a great deal of confusion and many contradictory opinions. It might reasonably be thought that, having so much information, it would be easy to select a suitable oil for a given purpose. In fact, however, it was not so easy. It had been stated in the papers and discussion that there were difficulties in deciding from laboratory tests the suitability of an oil for practical purposes, and that bench tests of engines did not give results comparable with those from actual running experience. Various authors showed how difficult it was to obtain reliable information in the form of comparable figures from practical experience, and it had been stated that the only way to select an oil was to consult the manufacturer of the machine or the oil. For a motor car, the maker indicated a certain S.A.E. number for a summer oil and another for a winter oil. That S.A.E. number indicated a certain viscosity, but it did not give any idea of the quality of the oil or of its behaviour in the engine. If one asked the representative of an oil firm, one would be told to buy first-grade oil. There was something to be said for that. A factory making certain kinds of engines would know that a certain oil gave satisfactory results, while the big oil concerns had collected a vast amount of knowledge about their products. It was, however, necessary for various reasons that the consumer should have the possibility of choosing the oils he required.

It was possible to obtain a great deal of information from laboratory tests and bench tests and reliable figures from statistics of practical experience and from engineering measurements. A single property would not indicate the suitability of an oil for a given purpose, and for several properties it could be shown that the testing methods now available could be improved, but if certain characteristic properties evaluated in the laboratory were correctly combined, it should be possible to select an oil which would prove satisfactory in practice. In combining a number of properties, it was not necessary to estimate each property by complicated methods which were lengthy and expensive. It might be possible to obtain sufficient knowledge by some more simple methods, because the selection of an oil for a given purpose was always a compromise. It was not possible to take one single property and define the oil to conform to that property within very narrow limits; it was necessary to think of all those properties which were used to characterize the oil. A well-chosen specification made it possible for consumers of oils, and even those who lacked knowledge of oils, to buy a suitable oil for a given purpose.

Mr. D. **Clayton** (Engineering Department, National Physical Laboratory) said that, in view of the importance of obtaining more information on boundary friction, a good deal of attention was being devoted to that problem at the National Physical Laboratory. The outstanding difficulty was, of course, to clean the surfaces. Thus, numerous workers had tried to repeat Hardy's results, but had failed, even when trying to use his technique. It was, therefore, necessary to start on that problem at the beginning. Under kinetic conditions cleaning might not be quite such a difficulty in some cases, but it certainly was in others.

He did not find the paper by Messrs. Burwell and Camelford at all helpful, though apparently their results came out in the right direction. With regard to Professor Trillat's remarks on filters removing polar bodies (p. 418), it was necessary to make prolonged tests to see how long it took to saturate the filter surfaces; the abstraction of polar bodies might be only an initial phenomenon, and not a source of trouble during the remainder of the life of the filter. It would also be interesting to see results for used oils compared with the same new oils. It would appear from the work in question that abstraction of polar bodies might be added to the list of troubles due to wear particles and solid deposits.

With regard to wear, Dr. Brownsdon's results were interesting, but disturbing, because they showed how many factors might affect wear in practice. It was desirable to repeat Dr. Brownsdon's tests under other conditions and with different wearing materials.

Mr. Neely's results were certainly interesting, but he measured friction under a condition of considerable wear, which was certainly not the condition usually chosen for boundary friction tests. It was not what was thought of when the disagreement between wear and friction results was mentioned; in fact, it was necessary to apply the qualification of the *particular conditions of experiment* to Mr. Neely's results. With that experimental arrangement there might be a danger of the edges of the hard steel rails acting at low speed to some extent as blunt cutting tools for the bronze buttons. Alternatively the surface irregularities might gouge minute scores in the bronze, possibly on a scale so small as not to affect the apparent smoothness visually; for it seemed that the surface condition of the hard steel track had an important effect. Ideally, one wanted wear from surface cohesion to relate with friction; the stage at which some cutting action started would not be well defined, but it was desirable to keep to the cohesion end of the scale.

Messrs. Southcombe, Wells, and Waters did not merely mark the Hertzian impression on the test block of the Timken machine when no seizure occurred, as happened in the four-ball apparatus. Their wear apparently continued. That seemed to be a curious difference between the two machines. The interesting results which those authors and Mr. van der Minne revealed by doing unconventional tests should encourage those using other testing machines also to find the effect of varying the test conditions. He would not be surprised if the results were illuminating.

Mr. Southcombe did not like the term "film strength." Personally, he did not like it because it was often talked of as though it were of the nature of a puncture strength of something like a layer of celluloid on the surface. He presumed that the "film" was originally the boundary layer, but that was an extremely thin layer, quite small in thickness compared with surface roughnesses. Moreover, breakdown might occur at the high points, with slight wear, without anything happening which was like the so-called "film rupture" in, for instance, the Timken machine.

Ideally, with the conception of the boundary layers described in some of the papers, it should be possible to have parting of the two contacting boundary layers without removing metal. That would give friction without wear. He was inclined to agree, however, with Dr. Bowden that actually it would seem to be inevitable that the high pressures on the high points would lead to some metallic contact in relative motion. It must be concluded that it was an open question whether friction and cohesive wear would be related in the same way under different test conditions and with different lubricants. Actually, the marking out of the Hertzian areas in the four-ball tests was probably due initially to some "picking up" of the high points of the surfaces. It seemed that extreme-pressure lubricants played a part there as well as in severe seizure; the impressions were always smoother with extreme-pressure lubricants, and that was probably related with running-in quality.

Professor Dr. L. **Ubbelohde** (Technisch-Chemisches Institut, Berlin) (speaking in German, interpreted) said that one of the main problems confronting technologists was that of inadequate lubrication, which amounted to discussing the behaviour of the outermost boundary layer. He was strongly impressed by the difficulty of expressing the reason why such lubrication should in fact be inadequate, and suggested that the reasons adduced often amounted to nothing more than a re-statement of the problem in different terms. Of the various explanations given for inadequacy of lubrication, some attributed it to surface irregularities, some to molecular forces and some to other causes, but the true cause was not yet known. It was therefore essential that the question should be attacked by true scientific methods, and for that reason, he welcomed the attack which had been summarized in Dr. Gough's introduction (p. 465). He would like to draw an analogy with the way in which the question of viscosity had for a very long time exercised the minds of technologists, and to point out that, until quite recently, the question of viscosity was regarded from a purely conventional and empirical standpoint. Only in recent years had they succeeded in interpreting the empirical data in absolute terms, but great harm had been done to technical progress by that lack of co-ordination of the interpretation of the empirical data in absolute terms, and in that one might see a warning against the same thing happening with the other problems now under consideration. It was therefore essential that those problems should be attacked at once from a truly scientific standpoint.

He was in entire agreement with Professor Norlin's paper (p. 386) except in its advocacy of viscosity index, which was based not on any true physical foundation, but on arbitrary measurements. While at first sight that might not appear to be a disadvantage, and was in fact more convenient, yet that lack of disadvantage held good only so long as no attempt was made to base exact physical calculations on the use of the viscosity index; as soon as that was attempted, difficulties were encountered, difficulties expressed in the fact that straight lines would probably be curved lines in a diagram, so that there was no true scientific correlation between the two sets of data. If the use of the viscosity index was continued, he feared that it would lead to the same difficulties as those which had for thirty years attended the attempt to utilize conventional viscometers for scientific work. If the viscosity index was to be used at all, it must always be remembered that the data which it gave were inexact and were not truly related to the fundamental physical data.

The foundation for recent standardization in the consideration of viscosity and temperature was Walther's formula, which gave a connexion between viscosity and temperature. That formula assumed that the viscosity-temperature relationship was a straight line, whence the concept of the viscosity pole; but if use was made of the viscosity index, the line obtained was not straight but curved, and no correlation existed between it and the arguments based upon Walther's formula.

One principal reason why research in lubrication had made such slow progress was that it was interpenetrated on all sides by so many other fields of science—physics, mechanical engineering, rock-drilling, mining, and so on—and, for a very long time, workers in each of those fields had tended to work quite independently of one another so far as they were concerned with petroleum. In each field they used their own language, which, to a great extent, was not understood by workers in adjoining fields. Hence the great value of the present Discussion, which was attended not by representatives of any one branch alone, but by all branches in common, able to discuss their problems together.

Mr. E. **Malschaert** (Ghent, Belgium) was glad that Dr. Ubbelohde had raised the question of the viscosity index. In Fig. 1 of Mr. Groff's paper (p. 318), in the lower right-hand corner, one found those two old friends, viscosity index and pole height. Both those concepts were supposed to represent a characteristic property of a family of lubricating oils. Recently he had shown* that there could be no possible compromise between the viscosity index and pole height, and Mr. Groff's diagram brought out that point still more clearly, for in his diagram the family of viscosity index curves should be a network of straight lines, converging on the point P_1, if viscosity index and pole height were the same thing.

Since at least one of the viscosity index curves represented a known oil-family, namely, the 100 viscosity index oils, it would be useful if, in future work, the concept could be based on that curve or on something which represented the common property of a family of oils. If there was to be some alteration of the viscosity index, that alteration should not be such as to wreck the whole concept of viscosity index, but should aim at retaining its good points and discarding the others.

The viscosity index as revised by Dean and Davis had three properties to which objection could be made. First, it was dependent on the temperature of measurement, and viscosities must be determined at 100 and 210 deg. F. in seconds Saybolt. Later, Dean and Davis devised the kinematic viscosity index, where the temperatures were 100 and 210 deg. F. and the viscosities were measured in centistokes. That was a big advance. Second, the viscosity index could not be extrapolated beyond 100, though, especially for compound oils, extrapolation seemed to be necessary. It would be seen from Fig. 1*a* that for a viscosity index of 100, the curve went down to the right, which was, of course, absurd. Another objectionable property of the viscosity index was that it did not apply to mixtures. If an oil of low viscosity index was mixed with one of high viscosity index, a mixture might be obtained which had a viscosity index higher than that of the oil of high viscosity index in question. That was shown in Fig. 1*c*.

As it stood, the viscosity index would probably never be adopted on the Continent as a measure of a quality of an oil. The International Viscosity Committee formed as a result of the Paris Congress, should try to define the viscosity index in a way which would obviate the inconveniences to which he had referred. The formula appended to Fig. 1 gave the "independent viscosity index" and the original kinematic viscosity index. The advantage of the small alteration which resulted in the independent viscosity index was shown by Fig. 1*b*. First, it was practically independent of the temperatures at which the viscosity deter-

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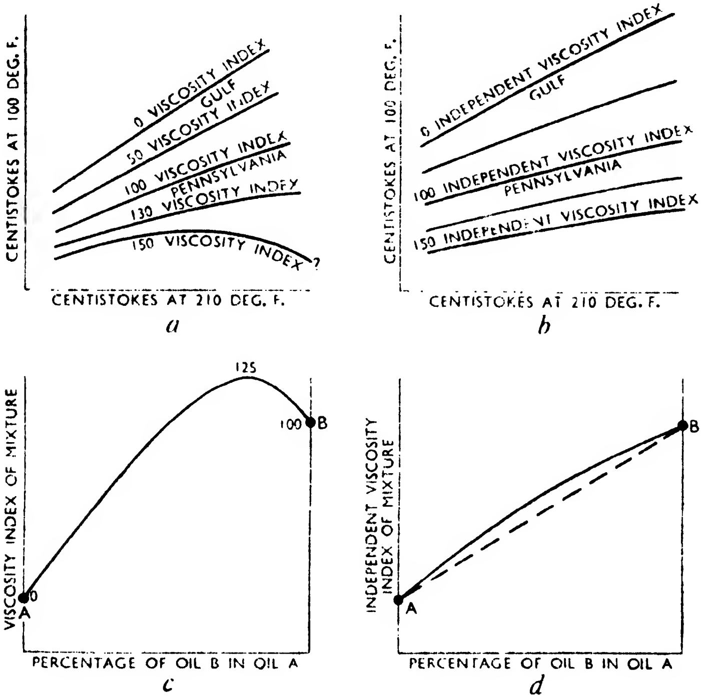
* Second World Petroleum Congress, Paris.

minations were made; if the viscosity was measured at, say, 50 and 100 deg. C., 37·78 and 98·89 deg. C., or 100 and 210 deg. F., and the independent viscosity index derived therefrom, roughly the same figure would be obtained. Fig. 1*d* showed the independent viscosity index



a *b*

c *d*

Fig. 1. Comparison of the Ordinary Viscosity Index and the Independent Viscosity Index and their Application to Mixtures

Kinematic viscosity index $=100\times\frac{\nu_0-\nu_x}{\nu_0-\nu_{100}}$, where ν_0 is the viscosity in centistokes at 100 deg. F. for zero viscosity index; ν_x is the viscosity in centistokes at 100 deg. F. of the oil under test; and ν_{100} is the viscosity in centistokes at 100 deg. F. for a viscosity index of 100.

Independent viscosity index $=100\times\frac{W_0-W}{W_0-W_{1000}}$ where $W_0=\log\log(\nu_0+0{\cdot}8)$, $W_x=\log\log(\nu_x+0{\cdot}8)$, and $W_{100}=\log\log(\nu_{100}+0{\cdot}8)$.

of mixtures of oils, and it would be noticed that the lines were not straight lines. Mixture diagrams, however, rarely were straight lines. The change proposed was quite simple, and in the Dean and Davis and Lapeyrouse tables it would suffice to print two columns of functions next to the normal viscosity figures, and then the calculation of the independent viscosity index would be just as simple as the calculation of the normal viscosity index.

Professor A. C. G. EGERTON, F.R.S. (Imperial College of Science and Technology) said that the discussion that morning ranged round a diverse series of communications of extraordinary interest, which had been admirably summarized by Dr. Gough. Professor Adam put things in their right perspective to start with by drawing attention to the fact that the irregularities in the plainest possible surface might be 200 times the distance over which a perceptible intermolecular force could make itself felt, so that the points of contact were few and far between. Dr. Bowden's researches showed that in spite of the presence of an oil film, very high local temperatures might be produced at those points of contact. One wanted to know what happened to the oil molecules or the lubricant molecules in the neighbourhood of those points, and whether they had time to decompose. There was still a great deal to learn about the kinetics of decomposition at high temperatures of the many complicated molecules which went to make a lubricating oil. It was desirable to start, however, with comparatively simple substances.

For complete lubrication the viscous properties of the oil mattered most, but from the point of view of boundary lubrication or those properties that the oil had in preventing the troubles due to the touching of those little high points, it was their stability or decomposability that mattered. In that connexion, many of the papers—the Chairman himself had drawn attention to the matter—brought out the importance of the presence of oxygen. Not only was oxygen needed to cause oxidation, but a little was generally necessary to bring about polymerization of the hydrocarbons.

He would like to follow up his remarks about oxidation by a further word on the paper by Mr. Hanson and himself (p. 334). The amount of nitrogen oxides which could be formed in the cylinder of a petrol engine working under knocking conditions rose throughout the expansion stroke. A value of nearly 0·5 per cent was reached, so that the amount of nitrogen oxides present was quite considerable. Mr. Hanson had devised methods whereby it was possible to distinguish between the nitric oxide and the nitrogen peroxide formed. The nitrogen peroxide was the main constituent which did harm. It was interesting, in view of the fact that the flame front itself was comparatively narrow, that the gases continued to form. Knowing that oxides of nitrogen were produced, and particularly under knocking conditions, and knowing that gumming tended to occur particularly under knocking conditions, and that nitrogen oxides catalysed oxidation of the olefines, they tried to find whether the addition of nitrogen peroxide to oxygen in the oxidation test would increase the rate of oxidation of a lubricating oil. That was found to happen and their paper described those results. That also had a bearing on the lack of correlation between the oxidation test and engine tests on lubricating oils. Nitrogen peroxide was present in the engine, and therefore the behaviour of the oil would vary. As Dr. Gough pointed out, that made it more difficult to get regularity of behaviour in engine tests for oils, but at the same time he hoped that it might lead to the possibility of arriving at a better oxidation test which was more consistent with engine conditions. Further experiments on the effect of wear in the presence of nitrogen oxide showed that there was a considerable effect, and he hoped that that would help to explain the difference between corrosion wear and mechanical wear.

Mr. J. E. SOUTHCOMBE (Germ Lubricants, Ltd.) remarked that Mr. Neely considered that it was a simple matter to find an addition agent which improved oils. During the last few years scores of patents had been taken out, chiefly in the United States of America, for so-called addition agents purporting to improve the oiliness characteristic. He had examined many of those organic compounds, and the results seemed to indicate that in no case was it possible to obtain a satisfactory improvement in the oiliness characteristic unless a long chain with a carboxyl group at its end was present.

Certain authors had put forward unnecessarily complicated theories to explain the phenomenon of oiliness and the influence of oiliness, but he felt that most of the phenomena could be explained quite simply as a kind of mixture in varying amounts of fluid film and boundary lubrication. Mr. van der Minne said that there was some indication that oiliness might be a disadvantage under conditions of extreme pressure, and he agreed with that statement. Oiliness had the effect rather of increasing the lubricating efficiency under moderate loading, and tended to prevent metallic wear, and, as extreme-pressure lubricants depended on making wear easy, the oiliness agent would naturally work the opposite way round.

Mr. Clayton had pointed out that the conditions in his experiments were not quite the same as in those described in his (Mr. Southcombe's) paper (p. 400), in that Mr. Clayton's wear figures referred to the condition after seizure, whereas those in the paper were measured before seizure. In both cases, however, it was probable that a bedding had occurred, either with seizure or without seizure.

Finally, he would like to point out that Mr. Evans had mentioned on the previous day (p. 190) that a sulphide did not appear to form when extreme-pressure bodies had been used, but work he had carried out with Messrs. Wells and Waters showed that the debris formed always contained ferrous sulphide if the oil contained sulphur or ferrous chloride if the oil contained chlorine.

Professor H. A. Everett (Pennsylvania State College, United States of America) pleaded for further information concerning viscosity. If it were possible to give an explanation by means of simple concepts, a great advance could be made. In pamphlets issued by one of the oil companies in the United States to filling station apprentices, viscosity was defined as "the time it takes oil to flow out of a hole." That would be funny if it were not pitiable. When one found a fundamental concept such as viscosity, shearing force, obscured by the process by which it was measured to such an extent as that, it was unfortunate, and clarification of our basic ideas was necessary.

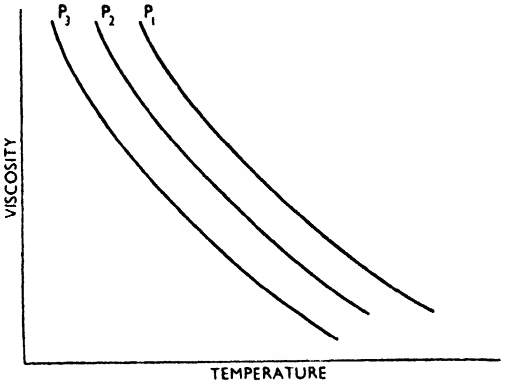


Fig. 2.

The emphasis laid by previous speakers on the viscosity index was admirable, but it was necessary to go a step further. Some recent work which had been referred to earlier, suggested that it was necessary to take in the pressure effect as well as the temperature. Viscosity and temperature relations were well known; as the temperature rose, the viscosity decreased. With pressure the reverse effect was found; with increase of pressure the viscosity—speaking, of course, of absolute viscosity—increased. The temperature effect was easy to obtain, but the pressure effect, though obtainable too, was not so easily determined. With complete data, it was perfectly easy to combine the two into a single diagram. They were both exponential curves, and the diagram could be plotted as curves or logarithmically, and in either case there would be a curve of viscosity against temperature for a series of pressures.

He felt that with a curve of that character one had considerably more comprehensive data on viscosity.

Those high-pressure effects were readily obtained, and bearings were working at higher and higher pressures. Smaller bearings were operated with mean bearing pressures as high as 10,000 lb. with good surfaces, which meant that spot pressures must be higher still. In any journal bearing, from the time of entrance to the time of exit, there was a continual change in temperature and in pressure. The viscosity was affected by both those factors. It was desirable to have the complete picture of the viscosity characteristics of the oil, and attempt to explain the behaviour of the oil film in terms of viscosity rather than in some of the newer terms.

It must be remembered that the viscosity obtained as the result of an average pressure and average temperature was by no means the average viscosity which existed in the bearing, and, if it were necessary to deal with averages, the average viscosity had to be taken rather than the viscosity corresponding to the average temperature or pressure; but, until the possibilities of correlating performance with truly representative viscosity data had been exhausted, it was unwise to accept new concepts which might give rise to other ills while curing one.

It was very rare to have a gathering of such diverse interests from so many widely separated countries, and yet which had a common thread of interest. Lubrication was perhaps unique in the extent to which that common bond pervaded all. He suggested that it might be well to utilize the opportunity thus provided to foster some things by joint action which must be done jointly if they were to be done effectively, such things as the possible appointment of subordinate groups to consider international action along lines which would be of permanent benefit. Such suggestions were perhaps more valuable if they were concrete, and he could think of nothing finer which could result from the General Discussion, as a concrete accomplishment, apart from the wonderful papers which had been presented, than that the suggestion made by Mr. Guy should be taken up and an attempt made at an international unification of nomenclature. If such a project could be fostered, it would be very helpful and a grand idea. (Applause.)

Mr. G. L. J. **Bailey** (Imperial College of Science and Technology, London) said that although the fundamental fact of electron diffraction was discovered ten years ago, it was only in the last two or three years that some applications to technical problems such as the problem of wear in lubrication had been made. The examination of any surface of a convenient size was possible; specimens could be taken from an actual bearing, or be prepared in the laboratory, using a suitable test machine.

He had been interested in the remarks made by Mr. Alcock (vol. 1, p. 608) with reference to running-in. The essential point was that a well run-in bearing had surfaces of an amorphous nature. They were not crystalline; in fact they were more like glass, and absolutely devoid of crystallinity. The first reference to the glazed surfaces which he had been able to find in connexion with bearings was by Mr. J. E. Hurst about 1915, using micrographic methods, and several contributors to the Discussion had emphasized the importance of those glazed surfaces. Experience at the Imperial College indicated that the strength of the boundary oil film was governed very largely by the texture of the surface on which the film existed, and an amorphous surface gave a very coherent type of film. Any reference to oiliness must therefore make some statement of the condition, not only microscopic but also submicroscopic, of the adsorbing surfaces.

The question had been raised (vol. 1, p. 633) of the effect of amorphous carbon, such as was formed by combustion in the internal combustion engine. It appeared to be thought by some that the carbon deposit formed in that way in a cylinder and circulated in the oil might act as a kind of graphite lubricant. The identity of the so-called amorphous carbon deposit had been in doubt since it was first investigated in 1912–13 by Debye and Scherrer, using X-rays. Its crystal size was so small that it was very difficult to say whether it was or was not graphite, but it was certain from electron diffraction photographs that it did not behave as a lubricant; i.e. the crystals did not orientate themselves with their slip planes parallel to the surface, as did true graphite.

Contrary to the view of one contributor to the Discussion, electron diffraction had indicated that a cast iron with a coarse graphite structure was one which, by reason of the graphite content and distribution, might be expected to wear well.

Mr. H. HIGINBOTHAM (Messrs. E. G. Acheson, Ltd.) remarked that Professor Adam (vol. 2, p. 197) referred to the value of graphite, particularly in boundary film conditions of lubrication, and pointed out the ease of orientation and attachment of the graphite particles to surfaces and also emphasized that, when contact occurred between two bearing faces, the contact was between outsides of sheets of graphite and not between two metals. Professor Adam also referred (p. 200) to the importance of chemical stability in lubricants, and personally he would suggest that in that connexion graphite showed to considerable advantage.

Mr. Bailey had referred (p. 483) to work done by means of electron diffraction. Professor Finch had shown by means of that method the value of graphite in preventing pick-up and allowing the metal, where conditions of metal-to-metal contact usually occurred, to flow and form a smooth Beilby layer. That constituted the value of the colloidal form of graphite of suitable particle size and purity, in enabling running-in of new parts to be achieved and wear to be prevented. One author in another group had referred to the value of graphite in the colloidal form in the drawing of mild steel wire at high temperatures. It was safe to say that most engineers were aware that it was *the* lubricant used in the drawing of tungsten and molybdenum wire down to extreme fineness. Another author said that colloidal graphite was the only satisfactory lubricant for needle roller bearings at high temperatures.

A contributor to the Discussion in another group had claimed that wear on automobile engines had been increased when using colloidal graphite. He could only suggest that the road tests carried out were erratic, as had been suggested by the Group Reporter when commenting on the paper, and that possibly the author in question was using graphite which did not have the qualities which the speaker had outlined in his earlier remarks. There was a great deal of difference between road tests such as those in question and carefully controlled laboratory tests such as had been carried out by various bodies in Great Britain and notably by the Institution of Automobile Engineers.

Dr. Kjerrman dealt with the addition of solid compounds to grease used for the lubrication of ball bearings. In speaking of solid constituents, presumably Dr. Kjerrman had in mind, when dealing with graphite, the addition of graphite powder and not graphite in the colloidal form, which, as was well known, had a particle size of the order of 0·001 mm. and less. It had been shown both by tests and by experience that the use of the colloidal form of graphite did not interfere with the clearances in the ball bearings, nor did it have any other detrimental effect. Tests carried out at the National Physical Laboratory had shown that a lubricant containing colloidal graphite of suitable purity and particle size was quite satisfactory for ball bearings.

He had made those remarks in an endeavour to correlate the work of Professor Adam and Professor Finch on the properties with laboratory testing and with practical applications. Several speakers had indicated how difficult it was to make viscosity measurements, or said that viscosity measurements made to-day were unsatisfactory. He wondered what the effect on making viscosity measurements would be when one was dealing with additions to lubricants, especially as additions to lubricants of various kinds were being generally accepted to-day.

Dr. F. P. **Bowden** (Laboratory of Physical Chemistry, Cambridge) said that an aspect of friction which interested him was the question of the surface temperature of rubbing metals. That could be measured in a very simple way by using two different metals and sliding them together. It was no use embedding a thermocouple near the surface, because the temperature effects were very local indeed; but if a slider made of one metal, such as copper, was run on a piece of steel, it was possible, by using that junction as a thermocouple, to measure the temperature of the surface itself. When constantan sliding on steel was used without lubricant, temperatures of the order of 1,000 deg. C. were attained. Even if a lubricant was employed, those local temperatures still occurred. With a good commercial lubricant, the surface temperature was of the order of 500–600 deg. C. at the points of contact, which, of course, would cause volatilization and decomposition of the lubricant molecules in the way suggested by Professor Egerton. When oleic acid was used, the result was a little better than for the lubricating oil, but very high temperatures were still obtained. There was a limit to the temperature which could be obtained, and that limit was the melting point of one of the metals. Thus, when lead was slid on steel, the temperature reached the melting point of lead, and, of course, could not go any higher. When the metals were slid together intense local heating and melting of the points of contact were obtained even when they were well lubricated. That must have an important bearing on the working of bearings. Surface irregularities were not so much worn away as melted away. From a practical point of view, the properties of bearing metals should probably be studied not at room temperature but at the high temperature of sliding, and particularly the relative properties of the two metals. If the metals had a similar melting point, then at the metallic bridges they would come up to the melting point and fuse together, and tearing of both surfaces would result; but if a bearing was used which melted at a lower temperature than the shaft, then the shaft could not be damaged, because the bearing metal would flow instead.

An aspect of the question which was perhaps more fundamental still, was the nature of sliding itself. The usual way in which friction was measured was to pull a specimen along, and measure the average force necessary to keep it sliding. It seemed probable, however, on theoretical grounds, that that force might not be constant, but might be fluctuating violently. Most methods of measuring friction would not record those fluctuations if they occurred, but would measure only the average force necessary to maintain the sliding. When, however, a high frequency apparatus was constructed which was capable of recording fluctuations, should they occur, it showed that the friction did fluctuate violently. The friction rose to a certain value, and then a sudden and very violent slip occurred, and that was repeated indefinitely. The question was what should be taken as the value of friction? Most measurements, of course, merely recorded an average value over a comparatively long period of time. The fact that the friction might not be constant was very important, both theoretically and practically. In his opinion, many of the so-called fundamental laws of friction, such as that the friction was independent of load, could be regarded only as the crudest approximation.

An analysis of part of the curve obtained when both friction and surface temperature were measured, showed what happened very clearly. The surface temperature before the slip remained roughly constant, but at the instant of slip there was a sudden and very violent rise and fall of the temperature. The whole temperature flash, the whole impulse, was less than 0·001 second in duration. The limit there was the limit of the period of the instrument, but measurements were now being made with a very much higher period. There could be no doubt that the frictional and temperature fluctuations occurred very rapidly indeed. It meant, of course, that if two surfaces were put together and pulled along they did not slide along in a continuous way but stuck together; the surfaces clutched one another, there was no motion, there was a strain, and suddenly the bridges broke and there was a violent slip. The temperature rose in that way and the bridges were formed again, and that process was repeated indefinitely. Even if the surfaces were lubricated, then with most mineral oils under boundary conditions the effect was essentially the same. Large and very rapid fluctuations in the friction were observed. In machines, of course, it was true that the surfaces were not able to halt so readily, but were driven remorselessly on; nevertheless, the interaction—clutching and breaking away—of the surfaces was occurring, and the area of contact, the friction, and the surface temperature were undergoing violent fluctuations.

Dr. H. VOGEL (Estoril, Portugal) said that he wished to discuss the question of wear tests on lubricants under controlled physical conditions. Some years ago, he had carried out wear tests together with Count von Soden of the Zahnradfabrik, Friedrichshafen, on a machine* especially constructed for testing gear oils. The method was to press an unhardened gear wheel against a hardened wheel and measure the wear of the soft wheel after a run of some hours at 80 deg. C. Count von Soden defined the lubricating value of a lubricant as the reciprocal value of the wear. The results obtained with some oils and greases could be reproduced with sufficient accuracy and showed that an extreme-pressure lubricant gave only 1/25 of the wear due to a lime-base grease.

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* VON SODEN. *Automobiltechnische Zeitschrift*, 1934, vol. 19.

Personally, he had been much impressed both by the great differences in wear and by the definition of the lubricating value as the reciprocal value of the wear, and so he repeated the tests under better controlled conditions of load, speed, temperature, and so on. It might be necessary to introduce a new term in the science of lubrication, namely, the wear resistance of an oil. It could be expressed in C.G.S. units. A new type of wear tester was constructed, according to his suggestions, by the firm of Sommer and Runge, and he had received some results which led to the following conclusions:—

(1) While it was not difficult to obtain exact physical conditions so far as load, speed, temperature, and the wear resistance of the test material were concerned, it seemed nearly impossible to obtain a physically defined surface. He was very happy to have the information about surfaces which Dr. Bowden had given. The test metals were unhardened steel and Böhler steel and the surfaces were of the roughness of an ordinary treated journal. Within the first 200,000 revolutions, at 200 r.p.m., the wear dropped from 700 to 1. Between 200,000 and 400,000 revolutions, the decrease in wear could be calculated by a logarithmic function, which tended towards zero at an infinite number of revolutions.

(2) No wear of the unhardened test piece could be observed at a load of 300 kg. per sq. cm. after 12,000 revolutions if the surface of the Böhler steel was polished like a mirror, which was in accordance with the observation that at an infinite number of revolutions, i.e. when the surface was polished, the wear tended towards zero. The only lubricant used in that test was air.

(3) At a load of 1,200 kg. per sq. cm., the wear of the test piece when lubricated with a spindle oil (viscosity, 50 centistokes at 20 deg. C.) was, under the same physical conditions, six times as large as with castor oil and 36 times as large as with a sulphurized extreme-pressure lubricant.

(4) The wear was nearly proportional to the pressure for spindle oil between loads of 100 and 300 kg. per sq. cm., but showed a great increase for castor oil. The values (Table 1, p. 489) obtained for wear under various loads per revolution expressed in milligrammes $\times 10^{12}$, and reduced to the same surface roughness, were accurate within 10 per cent.

(5) The results seemed to confirm an idea which he put forward at the Second World Petroleum Congress, namely, that at certain pressures, the lubricating value of the oil was substituted by the lubricating value of the two boundary layers metal I-lubricant and metal II-lubricant. Much confusion in treating the oiliness problem could be avoided if the fact was always considered that in any lubricating

TABLE 1. WEAR UNDER VARIOUS LOADS.

Load, kg. per sq. cm.	Wear, milligrammes $\times 10^{12}$ per kg. load	
	Spindle oil	Castor oil
100	14·4	1·72
300	18·9	4·05

problem a part was played not only by the quality of the lubricating oil but also by the quality of the five materials: metal I, boundary layer between metal I and oil, oil, boundary layer between oil and metal II, metal II. There were five materials, and it was essential to know the surface qualities of the metals and the qualities of the boundary layers and the oil. He thought that the new method which he had described might throw more light on the complex term "lubricating value ".

Mr. H. SHAW (Whitworth, Lancashire) said that he had been impressed during the Discussion with the extent to which reference was made to surface roughness. Earlier in the Discussion, Professor Karelitz said that there was no means of specifying surface roughness. Personally, he ventured to differ, because the instrument which he had devised would record the shape of surface roughness to within about 2 per cent, and he had also presented a method for specifying the roughness.

In studying certain of Professor Finch's remarks, there was a likelihood of forgetting that they related to submicroscopic conditions and phenomena, although Professor Finch, referring to the speaker's researches on surface contours, said that the observations made were at most microscopic, whereas his own were submicroscopic. When Professor Finch spoke of the depressions being filled with amorphous material and of the whole bearing surface being covered with an amorphous layer of "flowed" metal, he was referring to the submicroscopic depressions, and the bearing surface was really what the speaker called the contact surface. Professor Finch was also referring to those submicroscopic depressions in stating that colloidal graphite assisted the formation of a polished surface by preventing the pick-up of particles of metal into the oil. Those remarks also applied in a modified form to the microscopic or comparatively microscopic depressions and surfaces.

To observe the microscopic phenomena, the bearing surface at a chosen spot was examined with a "Contorograph" at various stages during running. The records showed how the roughnesses of the machine process were being reduced by the removal of material from the top of the peaks of the surface. The oil utilized was thoroughly filtered, so that the scoring effect had no influence. A similar bearing was run with a similar oil containing, however, 2 per cent of colloidal graphite and the wear was found to be of a different nature, being more of a plastic deformation or a flowing of the surface roughness, and a pressing down of the hills into the valleys. In one case, there was the actual opening up of an incipient crack in the whitemetal, due to defective bonding. From that, it appeared that ultimately the whole of the microscopic hills would flow into the valleys and form a flat surface, covered with the amorphous layer produced by the flowing of the submicroscopic roughnesses. The change from reduction by removal of material to reduction by plastic flow meant that the surface was smoothed off without any great increase in clearance, and that was of great importance.

Mr. Brillié's "oil roller" theory rather bore out certain theories which the speaker developed as a result of experiments on bearing surfaces finished by different processes. In the first place, he concluded that the roughness influenced the wear, and that the greater the roughness the greater would be the initial wear. As a result, in the published report of the tests he made the statement—rather rashly; as he had since realized—that the best surface was the smoothest, and that statement had since been reiterated by other authors. Since then, he had found that, though those remarks were probably correct for typical machined surfaces, they did not altogether apply to mirror-finished surfaces and matt finishes. If a mirror-finished surface was run at a high speed under a light load, it assumed a matt appearance, and investigation with the "Contorograph" showed that the matt surface consisted in effect of a number of saucer-shaped depressions. A typical size of the saucer was 0·000,050 inch long in the direction at right angles to the axis, 0·000,030 inch wide in the direction of the axis, and about 0·000,010 inch deep. The network or honeycomb of saucer-shaped depressions seemed to spread out from a minute surface defect, and, in order to investigate the effect of such surface defects, an apparatus was constructed which consisted of a glass tube containing a specimen of soft metal in which a comparatively large defect was created. Oil was admitted from one end, and the flow was controlled by means of two perforated sheets which were slid upon each other to control the flow. That was used instead of using a single orifice, because a single orifice would alter the streamline flow of the oil. A dye was admitted through a fine capillary tube to enable the flow or variation in flow caused by the defect to be examined. It was found that if the oil flow was as described, then the surface defect appeared to create a little local oil swirl, which led to the formation of a small saucer-shaped depression and caused the building up on the edge of a little mound of material which in turn appeared to act as a surface defect and created a second oil swirl. With that action spreading, ultimately the whole surface, or a fair area of the surface, became covered with a number of small saucer-shaped depressions. From the action observed, it would appear that those little local oil swirls would act as oil rollers, as mentioned by Mr. Brillié, creating resistance to the flow of the oil. With a mirror-finished surface, the friction over a period of time dropped rapidly to a lower value, then gradually to a still lower value, and then remained almost constant. If the load was increased considerably, and then the lower load was again returned to, it was found that the lowest value did not apply, but that a higher value was returned to. Continued running under the lighter load, however, again reduced the coefficient of friction, and the return to the lowest value took place when the matt finish was again formed on the surface. The conclusion was that there was a lower friction with matt finish than with mirror finish.

With regard to Mr. Clayton's paper, it would be interesting to know whether Mr. Clayton had investigated the effect of using bronze against steel in the 4-ball apparatus instead of steel against steel. Tests which he himself had made, using a rather similar machine, seemed to indicate that the readings of wear obtained with bronze balls against steel balls gave a different order for the oils from that obtained with steel against steel.

Mr. C. I. KELLY (Ilford, Essex) drew further attention to the paper by Messrs. Hanson and Egerton on the influence of catalysts on the oxidation of oils. His interest in the possibility that nitrogen oxides had an influence on the oxidation of oils dated back to the years 1927–8, when, with the assistance of Mr. C. Chilvers, he examined many used oils from numerous automobile engines. The dilution was determined by a steam distillation method, and in many cases copious red fumes of nitrogen oxide were ejected from the end of the water-cooled condenser. That chance observation led him to pay attention to the formation and occurrence of nitrogen oxides as products of combustion (or rapid oxidation) of hydrocarbons in air. He stressed that point, because he felt that it was essential, in discussing oxidation, to consider oxidation in the engine as a whole, and not to confine attention to oxidation in any one part, such as the crankcase and its content of lubricating oil.

Several years later, Graefe published his observations* on used oils from Diesel engines. Graefe especially emphasized the existence of nitrogen oxides in used oils, and made the very significant statement

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* GRAEFE. *Petroleum Zeitschrift*, 1932, 8th June.

that the quantities of nitrogen oxides found in Diesel engine lubricating oils after use were much greater than in used oils from petrol engines.

More recently, another German chemist made a similar observation. He was interested in protecting gasoline tanks from the danger arising out of the formation of explosive mixtures, and his idea was to fill the free space above the oil surface with the inert exhaust gas produced by Diesel engines. He had to abandon the idea, because the liquid products in the tank deteriorated in colour. The colouring bodies were identified as yellow nitro-compounds of organic compounds present in the engine gasoline stored in the tanks.

It was clear, therefore, that the cooled products of combustion, whether they were exhaust gas from petrol and Diesel engines, or the flue gases from a boiler, whether operating on fuel oil, coal, or coke, contained proportions of nitrogen oxides, which formed nitrous and nitric acids with any small quantities of free oxygen and water present in the liquid state when any of the so-called inert gases were cooled below their dewpoint. Mr. Haslam, Vice-President of the Standard Oil Development Company, had informed him in reply to an inquiry on the matter that they had had similar trouble, which caused them not inconsiderable expense when they started up the new hydrogenation plant at Bayway, New Jersey. They had to remove all the air from the hydrogenation unit before starting up the plant in order to avoid the formation of explosive mixtures between the hydrogen and oxygen (air). The method adopted was to burn available hydrocarbon refinery gases in air in order to consume the oxygen and convert it into the harmless oxides of carbon. The products of combustion, considered to be inert, were then passed through the hydrogenation plant to displace the air from the pipe lines and vessels. When parts of the unit were examined later, considerable corrosion was found. Mr. Haslam stated that the first cause of the corrosion was the nitrogen oxides formed during the combustion of the refinery gases in air for the production of the inert gas. He had the permission of Mr. Haslam to publish that information.

Another very interesting observation was made by Dr. Mucklow, who, in a paper* on experiments with a high-duty aero-engine, described a pressure gauge used for the estimation of maximum explosion pressures. The instrument had a non-return valve between the gauge and the combustion chamber, and between readings, when Dr. Mucklow released the gas pressure in the body of the gauge to the atmosphere, he found that some fluid was emitted which, on analysis,

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* Proc. I.Mech.E., 1932, vol. 123, p. 373.

proved to consist chiefly of nitric acid, some sulphuric acid and some nitrobenzene (the fuel contained benzol).

In the circumstances, and in view of the fact that other products of oxidation arose when a hydrocarbon fuel was burned in air, it seemed almost impossible, as indicated by Moerbeek, for a laboratory test to be devised to take all those factors into account. That should not deter a study of the probable effects of each suspected combustion product, singly, as Messrs. Hanson and Egerton had done in the case of nitrogen oxides.

Dr.-Ing. B. **Kjerrman** (Göteborg, Sweden), referring to Mr. Higinbotham's remarks on the use of a lubricant containing graphite for ball bearings, wished to warn all those who used ball bearings not to mix in the grease any solid compounds. Even if their particle size was as small as that referred to by Mr. Higinbotham, it was dangerous, and the smaller the bearings the greater the danger. With a very large roller bearing it might not be so dangerous if colloidal graphite was present in the grease. In the laboratory at Göteborg with which he was connected, it had been shown that the formation of dents could result from the presence of small solid compounds. Those dents might be microscopic, but they would lead to scaling which would spoil the bearing.

Mr. H. G. **Catterall** (Vacuum Oil Company, Ltd.) said that on the previous day, some very contradictory information had been given. He had read the papers in Group III, and he had gone carefully through the paper by Dr. Kjerrman (p. 341) with reference to ball and roller bearing lubrication and the testing of lubricants for those bearings. An enormous amount of work was behind that paper, but he felt that possibly Dr. Kjerrman had based his conclusions on an examination of products which were obtained locally. Dr. Kjerrman stated that lime-base greases had very limited use in the lubrication of ball and roller bearings, and two of the papers presented the previous day showed a tendency to subscribe to that view. Another speaker stated that lime-base greases should not be used. The basis of that opinion, he thought, was the fact that lime-base greases tended to separate into their constituent parts at certain temperatures, depending upon the rate at which the water, which usually kept the plastic mass stabilized, was driven off. A speaker on the previous day showed that it was now possible to take a lime-base grease almost up to the flashpoint of the oil, after which the grease would set again. Personally, he had never conducted such a test, but he had examined more than thirty lime-base greases which could be heated very much above their melting points and would then re-set to a grease which was quite serviceable. A standard test used by a ball-bearing manufacturer in this country was to heat a grease at 240 deg. F. for one hour, and at the end of that time, the grease must set again without a separation of its constituent parts. That firm had on its approved list over thirty different grades of grease. The fact must be accepted, therefore, that to-day a lime-base grease could be made which would not separate into its constituent parts. That was a vital point, because it might be thought that the only type of grease to be used for ball or roller bearing grease lubrication was a soda-base grease, and that was not correct, as almost all the motor car manufacturers recommended a lime-base grease for the lubrication of the wheel bearings. That fact should not be overlooked, and he wanted to emphasize the point that to-day lime-base greases had improved so much that, although the observations made in the discussion and in the three papers to which he had referred were undoubtedly true ten years ago, to-day it was not correct to say that all lime-base greases would separate when heated above their melting point.

Mr. H. L. GUY, F.R.S. (Metropolitan-Vickers Electrical Company, Ltd.) said his purpose in speaking was not to attempt to add anything to the discussion of the papers in Group IV, because he was afraid that if he gave his own definition of viscosity, for instance, it would be even more shocking than that given to the garage attendants, to whom Professor Everett referred; it would be something like this, that it was the property of the oil which determined the amount of coal which had to be burned under the boilers in order to turn the bearings round, coupled with the fact that it was also the property which allowed the bearings to be turned round without running them out.

It was on another point raised by Professor Everett that he wished to suggest some action should be taken. Professor Everett had suggested that the amount of friction in the study of the literature of the subject might be reduced and the amount of work involved lessened if there could be some simple standardization of symbols and nomenclature. Personally, he thought that it would be a good thing if, before those attending the General Discussion dispersed, contact was made with a few representative engineers from other countries who might assist in making liaison in regard to such matters with the institutions which were co-operating with the Institution of Mechanical Engineers in the Discussion. He would therefore ask whether the following gentlemen would be willing to meet for a short time that afternoon at the Institution of Mechanical Engineers:—

Professor Everett (U.S.A.).
Dr. Kjerrman (Sweden).
Professor Dr. Föttinger (Germany).

Mr. Brillié (France).

Professor Swift (Great Britain).

The gentlemen named signified their willingness to meet that afternoon.

The CHAIRMAN said that while there were many words of wisdom which would have come from those who still desired to speak, the time had come when, unfortunately, it was necessary to close the meeting. Before doing so, he would propose a cordial vote of thanks to the authors who had contributed papers, to Dr. Gough the Group Reporter, whose synopsis of the papers had been greatly admired, and to all those who had taken part in the discussion of Group IV.

The vote of thanks was carried by acclamation, and the proceedings then terminated.

Communications

Dr. GUY BARR (Metallurgy Department, National Physical Laboratory) wrote that the occurrence of increased amounts of combined oxygen in oils after service in internal combustion engines had led to the institution of various ageing tests in which samples were heated in more or less intimate contact with air or oxygen. Mr. Moerbeek (p. 367) deplored the multiplication of such tests and even their sponsors admitted that they were liable to grade oils incorrectly. It was shown by Messrs. Moutte, Dixmier, and Lion (p. 372) that the discrepancies were due to the fact that no investigator had yet found the physical and chemical treatment necessary to ensure that the reactions induced in the laboratory were even of the same type as those that occurred in service. Their work supported the suggestion of Messrs. Evans and Kelman (p. 285) and of others that an important stage of the practical deterioration took place on the surfaces of the cylinder bores and piston rings. The highest temperatures occurred on those surfaces during that half of the cycle when the oil film was either chilling a flame or was in contact with exhaust gas containing a very low proportion of oxygen. Though Messrs. Hanson and Egerton (p. 334) had studied the effect of adding to the air used in oxidation experiments traces of oxides of nitrogen (0·02 per cent) such as are produced by explosion, no one appeared to have investigated the changes occurring when oil was exposed to an atmosphere more similar to exhaust gas, i.e. containing $\frac{1}{7}$ to $\frac{1}{8}$ of its volume of steam with not more than a few thousandths of its volume of oxygen. Messrs. Moutte, Dixmier, and Lion obtained their most encouraging results by the heating in sealed tubes at

392 deg. C. of samples that had already absorbed most of the required oxygen at 140 deg. C. If combined—rather than free—oxygen was significant for the high-temperature treatment, experiments should be made in which that oxygen was provided, at least mainly, in the form of water vapour.

Mr. H. N. Bassett (Cairo, Egypt) wrote, with reference to the paper by Messrs. Mardles and Ramsbottom, that he had had occasion to study various oils used for Diesel engine lubrication, two of them (A and E in the appended Table) for high-powered engines, running at moderate speeds, and the others for high-speed engines of lower horsepower. Determinations were made of the static coefficient of friction by means of the Deeley machine after heating in glass dishes in an oven at 140 deg. F. with normal access of air. The characteristics of the oils and the results obtained are shown in Table 1.

TABLE 1. CHARACTERISTICS OF THE OILS TESTED AND THE RESULTS OBTAINED IN OXIDATION EXPERIMENTS

Oils	A	B	C	D	E
Flashpoint (closed), deg. F. .	410	380	380	400	400
Specific gravity at 60 deg. F. .	0·880	0·934	0·913	0·878	0·906
Viscosity, sec. Redwood at—					
100 deg. F.	637	751	—	615·5	708·5
140 deg. F.	223	213·5	211·5	218·5	219·5
200 deg. F.	—	67·5	67·5	75·5	71·5
Saponifiable matter, per cent .	0·12	0·27	0·18	0·12	2·00
Conradson carbon, per cent .	0·80	0·66	0·36	0·31	0·38
Acid value, mg. KOH per gr. oil after heating 30 hr. at 140 deg. F.	0·084	0·196	0·112	0·084	0·168
Static coefficient of friction:—					
New oil	0·145	0·166	0·189	0·193	0·124
After 250 hr. at 140 deg. F. .	0·183	0·189	0·222	0·228	0·147
After 500 hr. at 140 deg. F. .	0·200	0·196	0·234	0·237	0·160

The Conradson figures were the mean of two closely concordant determinations in each case, and the coefficients of friction were the mean of six or eight concordant readings. Only oil E was a compounded oil, the saponifiable matter in the other oils being that normal for most mineral oils. There was no change in the acid value after heating for 500 hours at 140 deg. F. and there were no signs of sludge formation, the results differing in that respect from those obtained by Messrs. Mardles and Ramsbottom. On the change in the coefficient of friction, oil B was the best, having the lowest change, but on the actual coefficient, oil E was better, owing to its content of fatty oil. The viscosity after heating was not measured, but from the figures given in Table 7 of the paper (p. 361) the alteration would have been slight with the temperature and time of heating involved.

The effect of the added water on lubricating oils was very striking, and one would like to know whether it was present in a smooth homogeneous emulsion, or merely mixed with the oil. The experiments mentioned by Mr. Samuelson (vol. 1, p. 269) indicated that an emulsion of oil and water had satisfactory lubricating properties, but did not suggest that it was so much of an improvement on plain oil as might be inferred from the paper by Dr. Brownsdon. The writer recognized that the results referred only to 70/30 brass, but previous work by Dr. Brownsdon suggested that somewhat similar results would be obtained with other alloys.

Experiments with cylinder oil emulsified with lime water* which contained 50 per cent of oil were not entirely conclusive, as in some cases the wear of the cylinders and rings was greater, and in others less, than that of cylinders and rings lubricated in the normal way. On the whole, the results were unfavourable. On the other hand, the results by Professor Nordmann and Herr Robrade suggested that a very similar emulsion gave satisfactory results and did not lead to excessive cylinder and, or alternatively, piston ring wear. If Dr. Brownsdon would continue his work, using cast irons and steels of varying compositions, and emulsified oils such as had been discussed above, the results would be extremely interesting.

The work by Messrs. Burwell and Camelford on the removal of the "oily" constituents by adsorption was partly a repetition of that done by Hardy and Nottage,† and confirmed the results obtained by them. It differed, however, in the use of metal powders instead of glass beads as the media for adsorption of the active (polar) constituents. The differences between the results obtained with the various metals suggested the possibility of chemical as well as of physical action, i.e. that adsorption was not the only result of the admixture of the metallic powder and the oil. Copper powder was obviously the most reactive, and the use of copper pipes for lubricating oils, especially when of

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* VINCENT. Congrès de Graissage, Strasbourg, 1931.

† Lubrication Research Paper No. 1, Department of Scientific and Industrial Research.

small diameter, might be suspect for that reason. Steel pipes would be better, presumably. Rhodes and Lewis* did similar work using finely divided Wood's metal. They found it impossible to remove all the adsorbed polar constituents from the metal by benzene, which supported the suggestion that there was chemical as well as physical action between the polar bodies and the metal. It would have been interesting if the oil which was passed through the capillary apparatus (p. 270) had been passed again through another similar apparatus. It would then have been seen whether (*a*) the initial rate of flow was the same and (*b*) the rate of change of flow was as found in the first instance. Presumably, as the first passing removed much of the polar constituents, the rate of change of flow in the second case would be slower.

The inference from the reference to oxygen-containing compounds on p. 268 was that an increase in the content of such compounds should improve the lubricating qualities of an oil. The experiments of Rhodes and Lewis, in which the static coefficient of friction of oil was measured before and after heating in a current of dry air, showed an increase in the coefficient at higher temperatures, with a slight decrease only when the temperature was lowered again. The results indicated a loss of lubricating value on heating, but did not show whether the deterioration was due to the destruction of the polar constituents or to the formation of some substance which had a marked effect upon the frictional coefficient.

Some work by the writer on the static coefficient of friction (measured by means of the Deeley machine) of oils before and after heating in glass dishes for periods of 250 and 500 hours in an oven with normal air circulation at 140 deg. F. indicated in every case an increase in the coefficient. Some degree of oxidation had taken place, but the coefficient was not improved. It was submitted that the inference that oxidation improved a lubricant was not justified.

Mr. H. BRILLIÉ (Bourg-la-Reine) wrote that there appeared to be general agreement between the theory and experiments relating to surface conditions as expressed in the contributions of Messrs. Trillat and Shaw and the writer. Woog showed that metallic surfaces possessed a field of force whose effect could cover several molecular layers (perhaps one hundred, according to Trillat), i.e. 10^{-5} mm. as a maximum, and whose intensity was much greater than that corresponding to the coefficient of viscosity. There was therefore a tendency for the formation of a layer of molecules of lubricant almost entirely attached to the surface. It was assumed, for the sake of simplicity, that

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* *Industrial and Eng. Chemistry*, 1934, vol. 26, p. 1011.

the first layer of molecules was fixed on the surface, but it was not certain that that condition (when there were no effects of oiliness) was fulfilled. With ordinary oil, there might and *should* be, if Woog's theory were correct, an effect in opposition to that of oiliness. A corrective coefficient was thus arrived at which affected the expression of the speed V in the formula for viscosity and took the form $(1+f/e)$, f being a dimension (length) related to the height of the intensity of the field of force of the surface, e being the film thickness. If there was a continuous series of "oil rollers" with a depth of 23×10^{-5} mm., the oil rollers led to the result that the thin film was no longer within the field of force of the metallic surface, and the "sliding" was in agreement with the ordinary formula for viscosity. The corrective coefficient mentioned above then disappeared. For the oil rollers, the effect previously indicated by the writer (Congrès de Métrologie), with $[1-\frac{1}{2}1(h''/e)]$ as denominator, was negligible here. It was a question of secondary irregularities of much smaller height than the primary irregularities considered by the writer.

Dr. H. W. BROWNSDON (I.C.I. Metals, Ltd.) wrote that, as remarked by Messrs. Auld and Evans, it was surprising that so little discussion had centred around extreme-pressure lubricants, perhaps the most important and certainly the most interesting of recent developments for the mitigation of wear. In the light of concepts put forward by Professor Adams, the experimental work of Dr. Bowden, and personal observations, the writer suggested that the behaviour of such lubricants was explicable on simple chemical grounds, if the addition agents used in extreme-pressure lubricants were conceived as reacting with the free metal atoms within the sphere of atomic attraction, converting them into much less harmful metals or chemical compounds. In short, the metal atoms, which might attach themselves to or alloy with the opposite surface, and thus lead to increased surface roughness and wear, became neutralized at birth.

Sulphur in one form or another had been cited as one of the most commonly used addition agents, and Colonel Auld suggested that its action was possibly due to the formation of long-chain polysulphide polar molecules. Was such a conception necessary when a trace of hydrogen sulphide gas or solution acted in a similar manner to much larger quantities of sulphur or sulphur compounds ? In both cases it was fairly certain that any free metal atoms present at the seat of friction were converted into molecules of metallic sulphide, and the more reactive the addition agent, the less was the quantity required for that reaction to take place. The more stable sulphur compounds only became reactive when decomposed under the temperature and pressure conditions which existed at the seat of friction. Mr. Evans stated that there was no evidence that sulphide films were formed, but the writer would agree with Mr. Southcombe that films of metallic sulphides were produced.

The chlorinated hydrocarbons, an entirely different class of addition agents, were undoubtedly decomposed at the seat of friction, giving rise to metallic chlorides. Their effect was enhanced by the addition of small quantities of water, which by hydrolysis facilitated their decomposition. An exactly similar effect was produced by the addition of a trace of hydrochloric acid, which, the writer believed, was the active constituent which gave extreme-pressure properties to oils containing organic chlorinated substances, its action being to convert what might become accumulations of abrasive metal atoms into less harmful chlorides.

Experimental evidence could be adduced to explain the value of tin salts of the higher fatty acids as additions to form extreme-pressure lubricants, but inorganic tin salts gave a similar result, suggesting in that case that the metal radicle produced a lubricating metallic film and was the reactive component in inhibiting wear.

The writer believed that plain straightforward chemical reasons could be put forward for the behaviour of extreme-pressure lubricants, and that there was no need to associate their properties with long open-chain polar molecules. The writer thought that their future development would profit from a very close study of the chemical reactions which took place under the temperature and pressure conditions existing at the seat of friction between different materials and different lubricants and under heavy loads.

Mr. D. Clayton (Engineering Department, National Physical Laboratory) wrote in reply to Mr. Shaw (p. 491) that work using bronze balls both alone and in conjunction with steel balls was being carried out, but had not yet gone far enough to enable useful comment to be made on the results.

Mr. W. E. Gooday (Brettenham House, W.C.2) wrote that judging from the written and verbal contributions on the subject of lubricating oil specifications, a few further comments from the standpoints of supplier and user might not be inappropriate.

It was obvious that an indiscriminate growth of specifications—good or bad—would lead to an unnecessary profusion of grades of oil in the distributors' lists and from that point of view was to be deprecated. There had developed therefore a strong opposition to specifications. For nearly all practical purposes, a suitable product was to be found amongst the wide range of lubricants now marketed. It was only natural that where suitable and well-tried lubricants were already available, there should exist little inclination to meet fastidious whims and idiosyncrasies. To the suppliers that seemed to be a reasonable argument against the use of specifications.

On the other hand, where the large user might not wish to maintain the trade-name identity of products and within his own organization had systematized the buying, storage, distribution, and use of lubricants, there was a strong disposition to buy to specification and, in fact, that method prevailed and was economical. Study of the technical factors involved suggested—as Professor Norlin pointed out and Professor Ubbelohde confirmed—that for most purposes, protective and, for many purposes, selective specifications could be prepared effectively.

Bearing in mind developments such as solvent refining and extreme-pressure lubricants, was it worth while to the medium and small user to bother with specifications ? The answer was in the negative. The technical equipment and accumulated knowledge and experience of the suppliers coupled with their technical service to customers had reached such a level that it was quite impossible for the outsider to keep pace in a field of ever-changing values.

Another aspect of specifications should be mentioned. Many of the compilations that paraded as specifications must have been constructed with the most fragmentary acquaintance with the methods of test and the meaning of the results obtained. When the British Standards Institution's work on lubricants began in 1923, the writer was responsible for suggesting the substitution of "Classification" for "Specification," and he still thought that the less specific term was more appropriate. The situation from both standpoints was that specifications would be and were useful to the larger consumers for all general purposes of lubrication and that they were not worth while to the smaller user.

But since they had come to stay, it would be well if they were expressed always in terms generally accepted by the industry to whose products they applied. It was not out of place, although late in the day, to call attention once again to the "Standard Methods for Testing Petroleum and its Products", published by the Institution of Petroleum Technologists. Although those methods had been adopted generally by many official and industrial organizations, one still found from time to time ambiguous, if not impossible, demands in so-called specifications. It had always been the writer's view that they should have in this country an authoritative guide to the significance of tests on petroleum products comparable in its field with the relevant publication of the American Society for Testing Materials.

Mr. D. F. **Pilkington** (Transport Officer, Lancashire Associated Collieries) wrote that he did not agree with Mr. Moerbeek's conclusion when he quoted the President of the Studebaker Company (p. 370). He had known car manufacturers to recommend three oils on a filler cap; surely, that was asking for sludge formation.

Messrs. Moutte, Dixmier, and Lion (p. 372) summed up the consumers' point of view: "The real test of a lubricant is one carried out on the engine which it should lubricate, but engine tests are long and costly and should be as few as possible, while it is desirable to avoid the damage to the engine which might be caused by a test of an unknown lubricant."

With regard to the Air Ministry oxidation test, useful results were obtained for mineral oils in comparison with the carbon deposit found in practice on the top of the piston, but tests made at 150 deg. C., instead of the 200 deg. prescribed officially, gave no useful results. Even if 200 deg. was considered excessive, it was still necessary, as would be seen from the following results:—

Oil	Conradson number of new oil	Air Ministry at 150 deg. C.	Air Ministry at 200 deg. C.
A. . . .	0·18	0·30	2·71
B. . . .	0·78	1·08	2·85
C. . . .	0·29	0·50	5·64

Oil B was used in a motor car, but, provided its other characteristics were satisfactory, the writer would accept that oil in respect of its carbon-forming properties without any further trial. The writer would like to know of any similar test which could be used for compounded oils.

Attention might be drawn to the useful Jung viscometer, as it was apparently not mentioned in the papers. The instrument was used for viscosities at 60 deg. and 40 deg. F.; it was an extremely reliable instrument, as tests could be duplicated and the same result obtained within ±1 second at 40 deg. F. in a run of ten minutes.

It had been rumoured that solvent-treated oils were unsatisfactory. As there were several solvent-extraction processes, it would be useful to know of any test that the ordinary consumer could use to find whether the oil producer had carried his process too far.

Some oil producers in the past had been rather opposed to consumers using tests, whereas, if they tried to make the different tests really commercially useful, they would be of great help to the user.

With reference to the paper by Herr Ostwald, the writer would be glad if there was in this country a Bureau such as that mentioned by him on p. 533, vol. 1.

Mr. T. E. ROONEY (Metallurgy Department, National Physical Laboratory) wrote that the discussion on the papers of Group IV seemed to centre round the difference between the results of laboratory tests and the behaviour of the lubricants in service. In that connexion it was noted that in the "R.L. emulsion test" described by Mr. W. O. Andrews (p. 202) the test was carried out with distilled water. Some years ago, the writer had occasion to test two crank-chamber oils. The engineer reported that the more expensive oil gave the better results. The usual laboratory tests indicated that there was very little difference between the two oils. In service, one oil tended to become "gummy" more readily than the other. A chemical examination of both oils was made after they had been in service and small amounts of calcium carbonate and some water were found. Synthetic mixtures of the oils with similar amounts of water and calcium carbonate were made and vigorously shaken. An egg-like emulsion was formed in each case which tended to be rather "gummy." It was noted, however, after prolonged shaking that the emulsion formed with the more expensive oil suddenly thinned out. The engineer's report was, therefore, confirmed. The trouble was really due to contamination of the oil in service by hard water (containing calcium carbonate), and one oil had much less tendency to gum under those conditions. It would, therefore, appear advisable to use for the emulsion test the water with which the oil was liable to come into contact under service conditions.

A point in the discussion on which there was a divergence of opinion was the value of graphite as a lubricant. In the paper by Dr. Bowden (p. 236) the high temperature which could be attained at points on the surface of lubricated metals suggested that particles of hard carbide might be formed when ferrous materials were used with a graphite lubricant. That point could be tested experimentally by a search for the presence of carbide particles in such a lubricant after a certain period of service.

INDEX OF AUTHORS AND PARTICIPANTS IN THE DISCUSSIONS

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