Gear Lubrication: Relating Theory to Practice

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The review begins with consideration of gear geometry, the fundamentals of wear and of boundary lubricant film formation as they affect the performance of gears.

The influence of the hydrodynamic oil film, of EP additives, and of mechanical and metallurgical factors on fatigue pitting are then outlined. Next, methods for estimating the onset of the little-understood form of thermal failure of lubrication, leading to the kind of severe wear known as scuffing and the associated problem of heat dissipation from gears, are described and discussed.

The influence of gear material on scuffing tendency, and of lubricant type on friction, together with the important process of running-in are considered and lead to recommendations for the selection of oils for parallel shaft gears and for worm gears. Finally, some comments are made on the significance of bench tests for load-carrying capacity.

GEAR CONTACT CONDITIONS

Gear teeth of whatever type (spur, helical, bevel, hypoid or worm) transfer power and motion through relatively tiny areas of mutual contact in the form of very narrow bands or ellipses, so narrow as to be spoken of as the "line of contact". This is because the curved surfaces of the mutually contacting teeth are opposed to one another, as they also are between the rolling elements and the inner ring of rolling bearings; whereas, in contrast, in plain bearings the curvature of the shaft and bearing conform to one another. Thus, contact conditions in gear teeth and in rolling element bearings are said to be counterformal, while those in plain bearings are said to be conformal. The tiny contact areas in counterformal contacts are thus subject to very high stresses and contrast with the low stresses in conformal contacts. Gear teeth thus have to be made from strong, hard materials, such as steel and the harder bronzes. As a further consequence, because they have very limited plasticity, they need to be very accurately made and aligned if excessive local overloading is to be avoided.

Opposing gear teeth move over one another with a combination of sliding and rolling motion. Rolling motion does not promote rapid wear of surfaces, but sliding does. Gear teeth cannot work without some sliding action and pure rolling action only occurs in spur, helical, and bevel gears, and then only momentarily at the pitch point. In all other types of gears, there is some sliding combined with the rolling action at all positions of the point of contact. The faster the rate of sliding, the more difficult the conditions of lubrication become. The sliding speed depends on the peripheral or pitch-line speed of the gears and the distance of the point of contact from the pitch point. The maximum rate occurs at the tips of the teeth and the larger the tooth, i.e. the smaller the ratio:

$$\frac{\text{Pitch circle diameter}}{\text{Addendum height}}$$

which is usually close to the number of teeth in the gear, the greater the degree of sliding. High speed gears are, therefore, conveniently made with relatively large numbers of teeth.

In the case of spur, helical, and bevel gears, there is no sliding along the line or band of contact; and, because the band is so narrow, any contact between asperities is of comparatively short duration so that the probability of damage is relatively small. In worm and hypoid gears in contrast, there is some sliding along the line of contact which increases the severity of the conditions of lubrication and makes for different approaches in the selection of lubricants.

As a final point on conditions of contact, the important difference in tooth action between spur and helical gears and between straight bevel and spiral bevel gears must be mentioned. In the case of spur and straight bevel gears, there are alternately one pair, then two pairs of teeth in contact. Any wear that occurs during single pair contact does not result in relief by transferring the load to another pair of teeth so that, once started, wear can continue. In contrast, in helical and spiral bevel gears, there are usually two or more pairs of teeth in contact. Thus, if any wear occurs between one pair of teeth, more load is transferred to the other pairs of teeth in contact, and the load on the pair undergoing wear is reduced. The conditions of lubrication are thus much easier in helical and spiral bevel gears, and the truth of the saying "helical gears run-in, but spur gears run out" will be appreciated.

WEAR

Under normal conditions, nominally unlubricated or "dry" gear teeth are covered by a film of oxide and adsorbed gases, water vapor, etc., from the atmosphere. This film may properly be considered as a primary lubricant film, for if it is removed, by severe rubbing, by operation in inert gases, or under very high vacuum, wear rates and friction increase enormously.

Leaving aside, for the moment, surface fatigue for separate consideration, there are basically three types of wear:

(1) Mild wear of the oxide film
(2) Adhesive or severe wear of the metal (scuffing)
(3) Plowing

When asperities on opposing surfaces contact one another during sliding, there must be some mutual deformation which may be either elastic or plastic. If the asperities have a slope of about 1 degree or less, they deform elastically and the oxide film remains intact. Some wear of the oxide film only occurs, but at a rate which, in practice, little affects the performance of the gears within their working life.

If, however, the asperities have greater slope than about 1 degree, the deformation is plastic and the relatively brittle oxide layer on the surface is unable to follow adequately the movement of the underlying metal. It, therefore, breaks up exposing the bare metal which can then readily adhere or weld to similarly exposed metal on the opposing surface. As sliding continues, the area of the welded junction at first increases until, eventually, it is broken. This adhesion thus greatly increases sliding friction. Moreover, the welded junction is liable to work-harden considerably so that it may become stronger than one or both of the parent metals. Then, as the surfaces slide apart, fracture occurs, not at the weld, but in the weaker of the two parent surfaces so that particles of metal become transferred from one surface to the other, eventually becoming dislodged to form a discrete metallic wear particle. Under these conditions, wear can be very rapid, and since the associated rise in temperature makes welding easier and conditions generally more severe, the rate of wear usually considerably increases in rapidity. The prevention of adhesive wear, especially the severe form known in the U. K. as "scuffing", and in the U. S. as "scoring", is the main purpose of lubrication. But, before this aspect is considered, it will be convenient to discuss the third form of wear: plowing or abrasion.

What is called plowing in fundamental wear studies is often known as "abrasion" when it occurs between gear teeth. Alternatively, it is sometimes called "scoring" in the U. K. or, when minor, "scratching". One form of plowing occurs when a very hard, rough, surface is mated with a softer one. In the general case, the asperities on one surface wear down the asperities on the other, but, in this case, the asperities on the harder surface are not worn down by rubbing of the softer one but, instead, continually cut into it.

In this case, therefore, it is important that the harder rubbing surfaces are properly smoothed or polished. The most common example of this is the worm gear in which the case-hardened steel worm should be highly polished so as not to wear the phosphor bronze wheel. The problem can also arise, however, when a case-hardened steel pinion is mated with a soft steel wheel, unless the pinion teeth are particularly finely finished.

A second form of plowing occurs when hard, sharp-edged particles, such as sand, or grinding dust, get caught between gear teeth. Such particles are liable to stick temporarily to gear teeth surfaces and plow into the surface of teeth which mate with them. Where two gears have teeth with markedly different surface hardnesses, the abrasive particles tend to embed themselves better in the softer teeth, consequently the harder teeth tend to suffer the greater amount of wear. Plowing from hard particles or dirt is particularly liable to occur in grease lubricated gears, since the particles cannot escape from the vicinity of the mesh as easily as they can with oil. According to a recent study, (1) only particles greater than about 20 microns are important for this form of wear in gears.

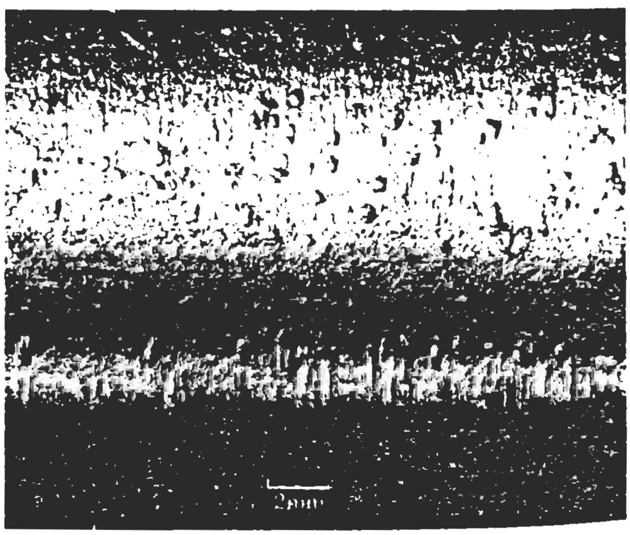

Fig. 1—Abraded surface of oil pump gear. The abrasion was caused by sand left in the engine block casting.

Because of the rolling action between gear teeth, the plow marks from hard asperities or hard particles are relatively short in length (*2*), as shown in Fig. 1.

BOUNDARY LUBRICATION

Returning to the conditions of adhesion, to reduce friction and wear the normal film of oxide, adsorbed gases and vapors may be protected, supplemented, or even completely replaced by other, more suitable, solid films. For example, the surfaces may be given a prior phosphating treatment, or impregnated with graphite or molybdenum disulphide powders. The most common way is, however, to incorporate so-called "boundary lubricants" in the oil so that they may produce, by physical adsorption or chemical reaction, the desired film which will be soft and easily sheared, but difficult to penetrate or remove from the surfaces.

Typical of these "boundary lubricants" are long chain fatty acids such as stearic acid, which form closely packed films, either by adsorption of their acid end groups into the surface oxide, or by reaction with the oxide to form the soap of the gear metal (*3*). This type of film is indicated schematically in Fig. 2a. These films become ineffective at the desorption temperature or the melting point of the metal soap, which are generally around 100 C only, but, within their range of applicability, they give low friction. In gear lubricants, these boundary additives are chiefly used in connection with worm gears where the bronze wheel forms a chemically reactive partner and where low friction is especially desirable.

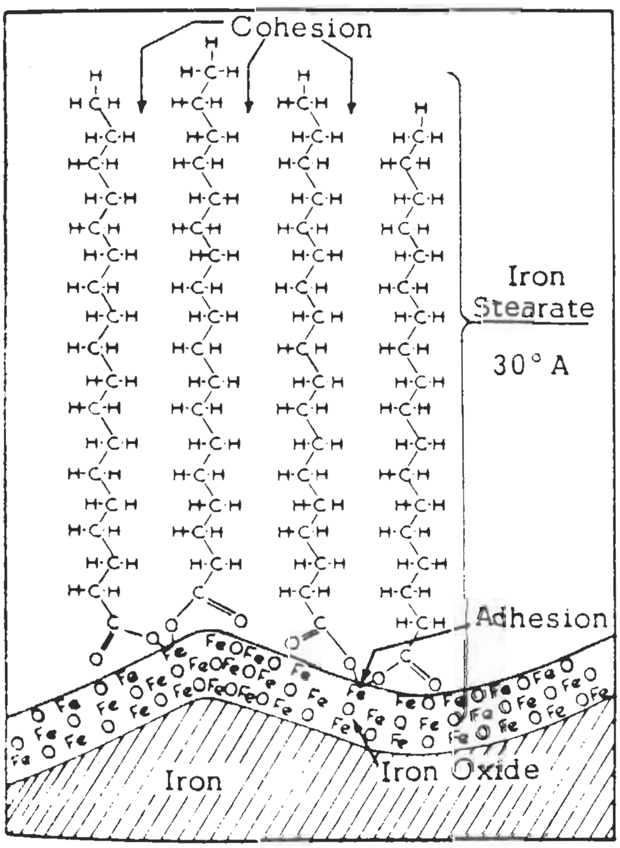

Fig. 2a—Schematic diagram representing chemisorption of stearic acid on an iron surface to form a monolayer of iron stearate with an essentially organic outer surface Ref. (3).

For the more severe conditions in steel-steel gears, expecially hypoid gears, this type of boundary lubricant is normally inadequate and additives, which will form films having higher melting points and greater adherence to the metal substrate, are required. More chemically reactive oil-soluble materials involving one more of the elements chlorine, phosphorus, and sulfur (often also lead and zinc) are thus used. These materials are generally known as extreme pressure or EP agents, though "extreme temperature" would have been a better description, and were originally designed for lubricants for the hypoid gear axles of passenger cars and trucks. The principle has, however, been extended, to lubricants for industrial gears.

The active elements; i.e., the chlorine, phosphorus and sulfur may be incorporated individually into organic, oil-soluble compounds, several of which may be added to an oil, or, alternatively, two or more of the elements may be present in one organic compound. Examples of the first kind are chlorinated paraffin wax, tritolyl phosphate, dibenzyl disulphide and, though an extreme case, free sulfur. Examples of the second kind are sulfo-chlorinated fatty oils, chlorophosphonates, and thiophosphates.

Basically, these additions undergo decompositions at the pressures, temperatures, and other conditions of the contact, the decomposition products producing on the teeth surface a film which is softer, less brittle and more adherent and therefore more effective than the oxide. The detailed mechanisms involved are seldom known, and even then usually in relation to performance in laboratory rigs. Essentially it seems, however, that those additives which are effective under mild conditions produce a mainly hydrocarbon film firmly anchored to the surface by their reactive groups (*4*). Under more severe conditions, however, these groups react with the surfaces to produce layers which are largely inorganic; e.g., iron phosphates, chlorides, sulphides, probably all of rather complex types (*3*) (*5*). This type of film is indicated schematically in Fig. 2b. Broadly speaking, the phosphorus containing additives are generally effective under relatively mild, steady running conditions, but those containing chlorine, and especially those containing sulfur, are required for the severer conditions of service, including shock loads. The melting points of the iron chlorides are, of course, considerably lower than those of the iron sulfides. The reacted layers tend to have greatest thickness where sliding speed and pressure, and consequently surface temperature, are greatest. Thus, Fig. 3, taken from the work of Borsoff (*6*), shows, in the case of spur gears, a minimum EP film thickness at the pitch line.

The catalytic effect of the bare metal of the tooth surface may be needed in some of these additives, particularly where phosphorus is the principal element in-

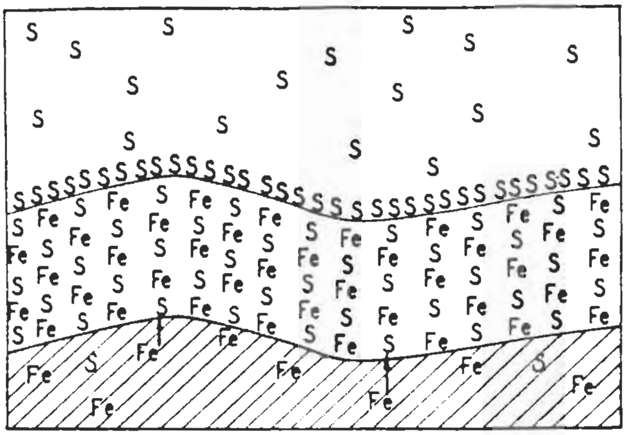

Fig. 2b—Schematic diagram representing inorganic film of iron sulfide on iron formed by sulfur in the oil.

volved. For the metal to be catalytically active, the oxide layer has to be removed from the surface by the rubbing action of its partner. Consequently, such additives are not very effective outside the zone of contact and are not very effective under conditions of suddenly applied loads where distortion of the teeth under load may bring parts of the surface into contact which have otherwise not been subject to sliding (*7*). Accordingly, sulfur or chlorine containing EP additives are preferred to those containing phosphorus for shock load conditions where parts of the teeth not previously run together may be brought into sudden contact.

To be effective rapidly and economically, it is desirable that EP additives should be able to concentrate on the surfaces needing their protection. That is, desirably, the additive should be able to adsorb onto the surface and highly polar substances are thus advantageous. It follows, too, that the oil should preferably not contain other additives, or itself contain components which are more polar than the EP additive, otherwise the EP activity may be insufficient or the EP additive may have to be used at a higher concentration than otherwise necessary (*8*). It is, for example, a common finding that EP additives are less responsive in the more aromatic base oils.

If the EP film formed is readily wiped away by the rubbing action, for example if it is working at a temperature close to its melting point or if the film formed does not adhere well to the base metal, scuffing may be prevented at the expense of an excessive rate of wear. Thus, the EP additives intended for use with new hypoid gears often allow excessive wear of bronze worm gears at high oil temperatures. Such oils are, therefore, not generally advisable for bronze worm gears if oil temperatures are consistently above about 70 C.

Compared with automotive gear oils, industrial gear oils have to work under a wider range of conditions. They have, for example, to be able to lubricate steel-on-bronze gears as well as those of steel on steel, they have to be compatible with the materials of construction of more complicated systems and they have to be compatible with water. In the latter respect, they should be able to separate easily from any water that might get into the lubrication system, while the EP additives, and the EP films formed by them, should not be decomposed by water. It is for the latter reason that chlorine-containing additives have never been popular for industrial gear oils, since iron chloride readily hydrolyses and it is not always possible to prevent this occurring by using a second protective film. Lead soap, inactive sulfur type oils have, therefore, generally been preferred. But, recently, because of fears of environmental pollution by lead and certain limitations in thermal stability, there is a steady trend towards the use of sulfur-phosphorus type additives.

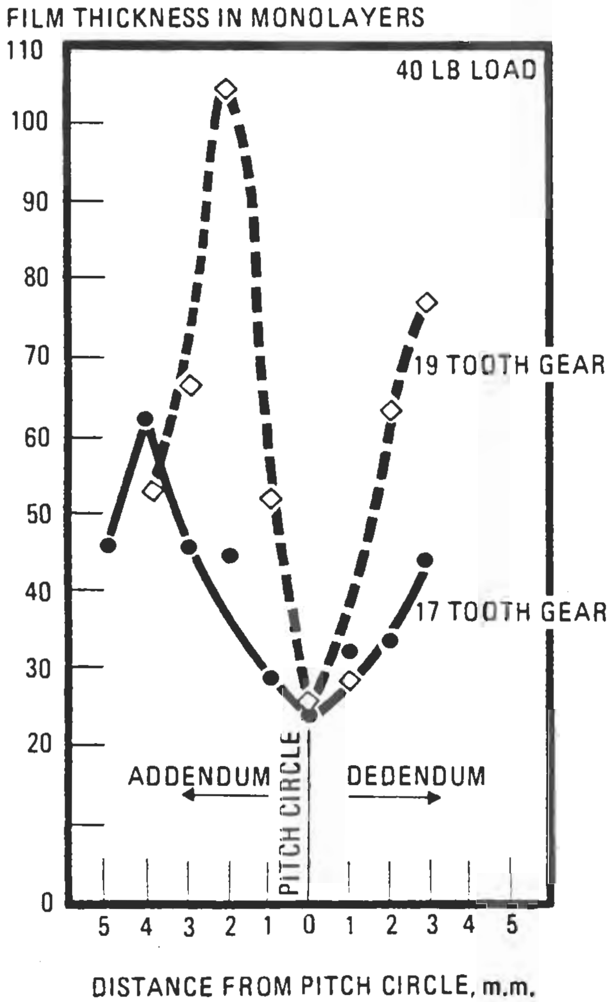

Fig. 3—Intensity of EP film along a tooth profile

HYDRODYNAMIC LUBRICATION

Most gears run at loads and speeds at which continuous operation under boundary or EP conditions would result in excessive rates of wear. Except during starting and stopping, most gear teeth run with only intermittent and transient contact between opposing asperities. For the majority of their life, most of the load is carried by a hydrodynamically generated film of oil, typically averaging a thickness of one to two microns. For many years however, calculations based on the theory of hydrodynamic lubrication published by Osborne Reynolds in 1886, and shown to be valid for plain bearings, gave film thicknesses which were far too small to be considered true. It was, therefore, believed that most gears ran under mainly boundary lubrication conditions, but nevertheless many instances were known of gears which had run for very long times and yet still showed original machining marks over the majority of the working surface.

These inconsistencies have been resolved over the last 20 years by the derivation of more accurate mathematical analyses and their confirmation by accurate and sophisticated laboratory measurements. The classical theory of Osborne Reynolds and his followers assumed that the solid surfaces were perfectly rigid and that the viscosity of the oil remained constant. These assumptions were reasonably close to the truth in the case of plain bearings where the contact surfaces were conformal and the pressure low, around 15 kg/cm^2, but, in the case of gear teeth with highly counterformal contact and very high contact pressures, up to 100 kg/mm^2, they were rather far from the truth. Thus, the elastic deflection of the surface in contact could be shown to be many times the oil film thickness being sought, while at pressures of 30 kg/mm^2 the oil viscosity could be raised 100,000 times.

Separate allowance for these two effects in the Reynolds theory produced insufficient increase in the calculated film thicknesses, but, when the two effects were taken together, film thickness, film profiles, and pressures distribution within the film could be calculated which agreed with practical experience and with laboratory measurements made at about the same time and subsequently *(9) (10)*. Lubrication under these conditions has been given the name "elastohydrodynamic" or EHL.

The theory and results of EHL are by now well known. One of the most interesting results, apart from the realization that oil films of about one micron thickness can, after all, exist between gear teeth, is that the film thickness, under constant temperature conditions, is hardly affected by load at all. According to the widely used formula due to Dowson and Higginson, film thickness varies only as load raised to the power minus 0.13 and can, therefore, be ignored in approximate calculations. For parallel-shaft gears; i.e., spur and helical, an approximate expression for film thickness in terms of gear dimensions is:

$$h_{min} = 0.0017\ (\mu_o \quad V_p)^{2/3}\ (D)^{1/3}$$

where

h_{min} = oil film thickness at the pitch line in microns

μ_o = oil viscosity at atmospheric pressure and the temperature of the gear tooth surface in centipoise

V_p = pitch line speed m/s

D = center distance mm.

It may seem extraordinary that film thickness is virtually independent of load, but the point is that the very thin oil film under high pressure actually becomes more rigid than the metal. Thus, an increase in the load results in a flattening of the metal and an expansion of the area of contact rather than a decreased film thickness.

It must be appreciated, however, that this expression assumes that the viscosity remains constant; that is, the temperature remains constant. In actual fact, an increase in load generally raises the temperature and decreases the viscosity, thus indirectly decreasing film thickness.

FATIGUE PITTING OF GEAR TEETH

Knowledge of elastohydrodynamic film thickness between gear teeth has been of great benefit in increasing our understanding of the phenomenon of fatigue pitting of the working surface of gear teeth. Typical fatigue pitting is shown in Fig. 4. Note that the pits mainly arise below the pitch line; i.e., in the dedendum of the tooth flank, the point of the typical "oyster shell" shape being pointed in the direction of sliding. Surface fatigue is the result of repeated surface or sub-surface stresses beyond the endurance limit of the material. The stresses may be excessive due to local overloading, from misalignment or undulations in the surface and from dynamic loads, aided by stress-raisers, either in the surface in the form of dents or asperities, or below the surface at hard non-metallic inclusions in the alloy. Moreover, as suggested by Broersma *(11)*, pitting may be initiated by the sub-surface being weakened by too rapid hobbing.

The extent of contact between asperities on the opposing surfaces seems, however, to play a major part in the phenomenon. A very close correlation between tendency to pitting and what may be called the specific roughness ratio D:

$$D = \frac{\text{Sum of peak to valley roughnesses}}{\text{Theoretical elastohydrodynamic film thickness}}$$

has been obtained in laboratory tests *(12)* as shown in Fig. 5, and in surveys of practice. Thus, when D was 0.1 or just over no pitting occurred even after 10^7 cycles. There is evidence that when the surfaces are of unequal hardness the roughness of the harder is the more important *(13)*. However this may be, the correlation suggests the advisability of having, sharp, smooth, well-finished cutter blades, correct cutting technique, cutting oil, etc.,

Fig. 4—Fatigue pitting

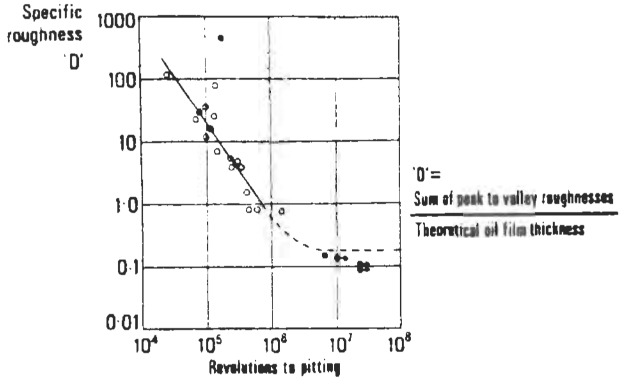

Fig. 5—Influence of specific surface roughness "D" on fatigue pitting.

since no tooth surface can have a better surface finish than that of the cutter which produces it.

Another way of restraining pitting in practice would be to improve the "D ratio" by using more viscous oils, but, in practice, an effective increase in the viscosity of the oil on the teeth is not always possible: An increase of oil viscosity grade is often limited by considerations of low temperature starting, and, where high speed plain bearings are lubricated by the same oil, the bearings heat the oil almost irrespective of its viscosity grade until it has more or less the same operating viscosity.

The "D factor" is, of course, not the only factor involved in the phenomenon of pitting. Other important factors are the hardness and microstructure of the two gear materials, in particular a large amount of free ferrite in steels is believed to promote pitting. Accuracy of alignment and profile, and a low range of undulations along the width of the teeth, are also important. Actually, Fig. 4 is a clear example of the profile error known as "split marking". Thus, there is a band of heavily deformed metal on the addendum surface and heavy pitting on the dedendum. In between these two protuberances, from the profile, there is a band showing original machining marks which carried no load. This "split marking" is due to the hob not having been mounted concentric to its axis of rotation and is also characterized by a corresponding single protuberance in the middle of the profile of the reverse flank.

Associated with the effect of tooth errors is the effect of EP additives. When the effect of EP additives on fatigue pitting is investigated in laboratories using very accurate surfaces such as in disc machines, it is almost always found that EP additives promote pitting. This effect is still discernible in laboratory gear rig tests, where although alignment, profile accuracy and surface finish are very good, they are not as good as in disk machines. But experience, in practice with several sets of gears for marine steam turbines having rather pronounced undulations, repeatedly showed that whereas a standard turbine oil allowed heavy pitting on the crests of the undulations, a chlorine-containing EP turbine oil prevented pitting from occurring (*14*). There is good reason to believe that the effect is due to the EP additive preventing the crests of the undulations from work hardening so that they can continue to deform plastically, thus spreading the load and reducing surface stresses.

Finally, perhaps most important of all, is the effect of shock loading on pitting. Recent work by Onions and Archard (*13*) has shown that, under otherwise comparable conditions, case-hardened steels suffered pitting in a gear rig at only 100th of the number of cycles that they did in a disk machine. Their conclusion was that this difference was due to the dynamic effect of load from the intermittent contact of the gear teeth. The effect of shock loads in practical gears must be even greater.

FILM FAILURE: SCUFFING AND INCIPIENT SCUFFING

If the load is increased on a pair of running gears, lubrication eventually fails. The power absorbed and the noise suddenly increase and smoke may be produced. The tooth surfaces are found to be worn and damaged as shown in Fig. 6a. This condition is known in the U. K. as "scuffing", and in the U S A, as "scoring" and is often preceded by a somewhat milder form of damage shown in Fig. 6b.

The causes of this latter form of rapid adhesive wear are even less certain than for scuffing, but it may possibly be due to the boundary film failing on one side before it does on the other. It is, however, recognized as a precursor of scuffing (*15*), and seems to be identified with the phenomenon of "ridging" which occurs in hypoid gears under some conditions of high torque—low speed operation (*16*). At the risk of making a confusing terminology even worse, it seems preferable for the purpose of this paper to call it "incipient scuffing" and thus avoid any suggestion that it is caused by plowing of hard

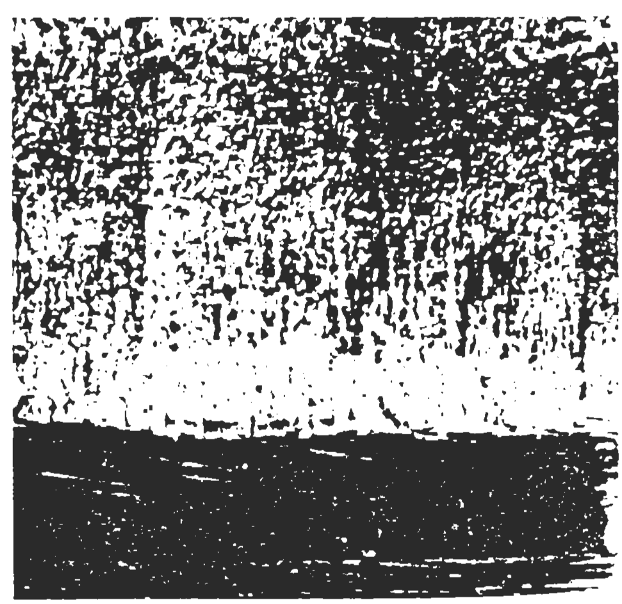

Fig. 6a—Scuffing (FZG Rig)

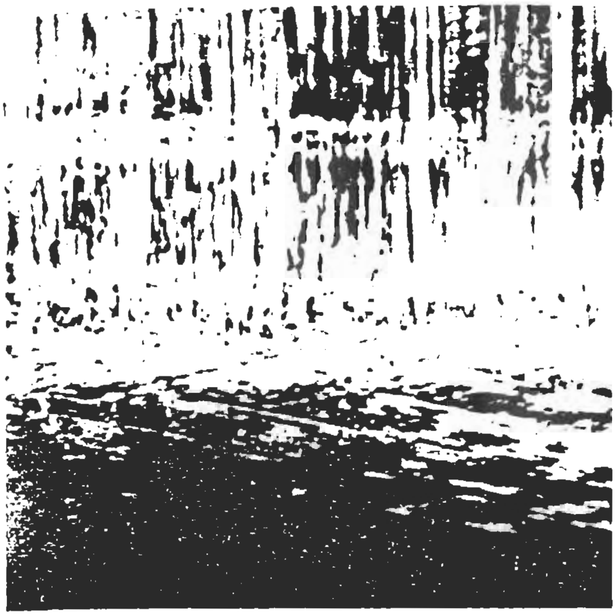

Fig. 6b—Incipient scuffing (FZG Rig)

particles or asperities which its other names "scoring" and "abrasion" may do.

Scuffing is a particularly severe form of adhesive wear. Metallic junctions grow in areas during sliding and it is believed that when these junctions grow to the extent that they coalesce with one another, or cannot be prevented from coalescing by the lubricant, the sudden increase in strength of the weld brings about the special condition of scuffing (*17*). In practice, however, there is often uncertainty as to whether or not scuffing has occurred and various descriptions such as "high speed", "low speed", "self-healing", and "self-aggravating scuffing" are sometimes used.

What seems to be an invariable concomitant of scuffing with steels, whose presence is used to decide cases of doubt, is the appearance in the microstructure of the rubbing surfaces and sub-surfaces of white etching layers. Although there is not yet unanimity on the nature and method of formation of these white-etching-layers, a large section of informed opinion considers them to be supersaturated, untempered martensite. They are believed to be formed under conditions of rolling and sliding contact by the combined hydrostatic pressure and shear producing such high stresses that micro cracks are formed whose flanks are heated up to the melting point by the rapid deformation. The surrounding carbides are then dissolved into the deformed γ iron lattice (*18*). Whether the white layers form immediately before or immediately after scuffing is not clear, but one can easily imagine that surface films would be easily disrupted by the local contraction and softening of the steel.

It has been suggested that such failures may occur by some thermal breakdown of a previously complete EHL film. Whether this is so or not, such damage must certainly result if boundary lubrication breaks down over a substantial area and, having broken down, the increased roughness of the surfaces must reduce the amount of load that could be carried by the EHL film. Recent work (*19*) in fact suggests that just before scuffing occurs around 10 percent of the load may be carried by boundary contacts, though these contribute as much as one-half to two-thirds of the total friction.

EHL theory cannot as yet account for failure of the film, but there seems little doubt that thermal effects, possibly including desorption, melting, etc., of surface films is involved. Though there is some doubt as to its absolute validity, the "Flash Temperature Theory" of H. Blok (*20*) has proved very useful in practice for gears running at moderate speeds. According to this theory, when the transient temperature of the tooth surfaces of gears lubricated by a straight mineral oil exceeds a value which is dependent only on the nature of the oil and the gear materials, the lubrication film breaks down and the tooth surfaces scuff. The transient surface temperature, T_c, is made up from two components, the steady surface temperature of the gear blank, T_b, and the momentary temperature flash, T_f, due to the heat developed by the friction of the contact. Thus,

$$T_c = T_b + T_f$$

For spur gears, an expression for T_f, also due to Prof. Blok, is

$$T_f = 1.11 \left(\frac{f}{b}\right) \frac{w}{\sqrt{Z}} \sqrt{V} \left| \frac{\sqrt{U_1} - \sqrt{U_2}}{\sqrt{V}} \right|$$

where

f = instantaneous coefficient of friction for the contact area

$b = (k\rho c)^{0.5}$, the thermal contact coefficient for the tooth face material in which k = thermal conductivity, ρ = density, c = specific heat/unit mass

w = tooth normal load/unit length of contact in the meshing position considered

Z = Hertzian contact band width

U_1 U_2 = tangential speeds of teeth 1, 2 perpendicular to the line of action for the meshing position considered

V = pitch line speed of the gears.

The value of $|\sqrt{U_1} - \sqrt{U_2}| / \sqrt{V}$ is dependent on the proportions of the teeth, as indicated in Fig. 7, which relates to conditions at the tip of the driving pinion tooth. This figure shows that the value, and, therefore, the severity of conditions, increases with small ratios of Pinion diameter/Addendum height; i.e., the severity increases with the size of the tooth. It also shows that the higher the reduction ratio the easier the conditions. Furthermore, when the wheel drives the pinion; i.e., the reduction ratio is less than one, conditions are more severe than when the pinion drives the wheel.

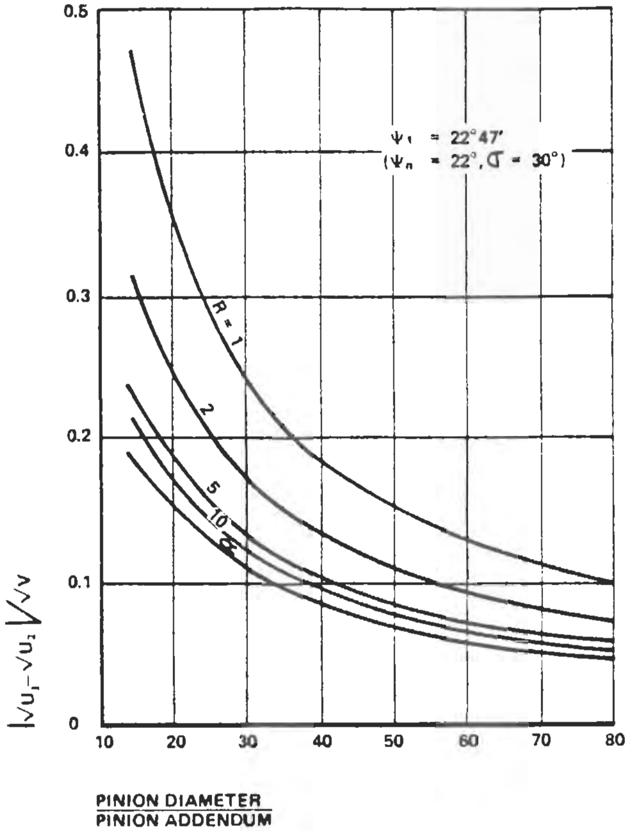

Fig. 7—Effect of addendum proportions and gear ratio on flash temperature.

Most companies using the flash temperature theory have developed typical values for the critical temperatures for the oils and gear steels that they use and published data is rather scanty. However, typical, values for paraffinic straight mineral oils and En 34 and 20 Mn Cr 5 (the steels for the IAE Gear Rig and FZG Rig respectively) are:

Oil Viscosity cSt at 60 C	Critical Scuffing Temperature, °C
20	120-140
23	130-160
34	150-200
50	170-210
81	190-230
147	210-260

In order to improve the accuracy of the flash temperature, various modifications have been suggested from time to time. In particular, Carper and Ku *(21)* have found in laboratory tests that the critical temperature varies with a dimensionless parameter consisting of the absolute viscosity of the oil at the conjunction temperature, sliding speed, and sum velocity divided by the product of radius of curvature and the average Hertz stress.

These methods require a fairly accurate estimate for the blank temperature T_b. In some cases this may not be very difficult, but, in other cases, elaborate calculation by means of "thermal network theory" *(22)* may be necessary.

Indeed, according to recent work by Niemann and co-workers *(23)*, the blank surface temperature is, in fact, all-important. According to these workers, scuffing occurs not at a constant critical value of the transient surface temperature, but at a critical value of the steady surface temperature, T_b. And the critical value is not constant but increases with speed. This theory takes EP oils into account, which the Blok theory does not, the slope of the critical surface temperature versus speed lines increasing with EP activity, as shown in Fig. 8.

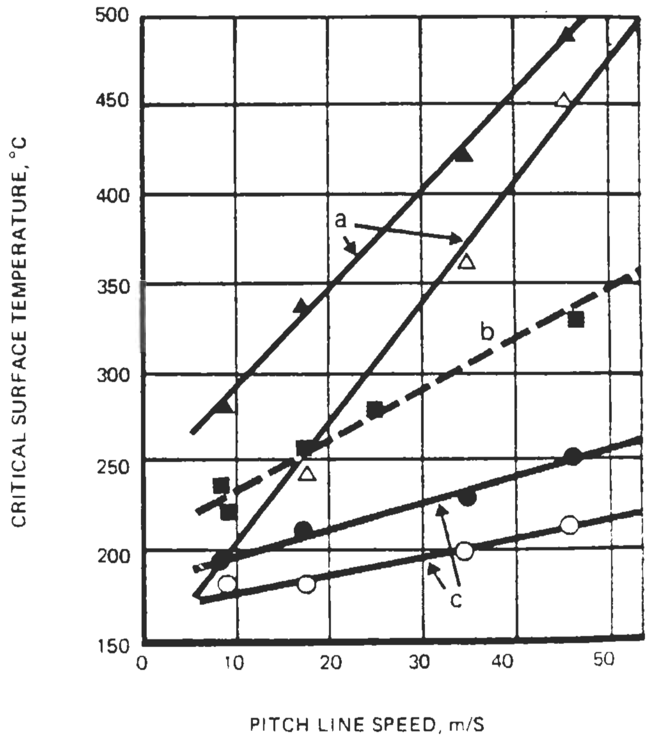

Fig. 8—Critical surface temperatures for scuffing according to Niemann/Seitzinger.

Another type of criterion is the "Frictional Power Intensity" originally suggested by Matveevsky *(24)* and defined as the rate of production of frictional heat per unit area of Hertzian contact. Bell and Dyson *(25)* have shown that this gave the best results in their disk machine tests and Moorhouse *(26)* found that frictional power intensity divided by sliding speed raised to the power 0.8 correlated best with scuffing failures in service.

As shown in Fig. 6a and b, when scuffing occurs there is usually an area along the pitch line where the sliding conditions are not severe enough to disrupt the oil film. But when, by accident, there is no oil there at all, scuffing may occur over the whole of the tooth flank, as shown in Fig. 9 *(27, 28)*.

Fig. 9—Scuffing in the absence of a lubricant

HEAT DISSIPATION

The vital importance of the surface temperature of gear teeth is apparent from the previous discussion and underlines the importance of the cooling function of the oil. Oil has to be supplied to gear teeth at a greater rate than necessary for lubrication in order to remove frictional heat as rapidly as possible, thus to keep the blank temperature as low as possible and to minimize the risk of thermal failure of the oil film.

The process of heat removal by this means has been analyzed in detail by de Winter and Blok (*29*) who conclude that there is an upper limit to the amount of heat which can be withdrawn, formulated as follows:

$$Q_{tot} = 5.6\, mb\, \theta_s \omega^{-1/2}$$

where

Q_{tot} = the maximum amount of heat that can be withdrawn per unit width per meshing cycle,

m = the tooth module,

b = the thermal contact coefficient of the oil, approximately 500 SI units,

θ_s = the difference between oil supply temperature and gear surface temperature,

ω = the angular speed.

All the above factors expressed in consistent units.

Such rates are fortunately within the rate of supply used in practice, the lower limit of which is formed from the practical consideration that the sprayer nozzles should have a minimum bore of about 2.5 mm in order to avoid becoming accidentally choked by dirt in the oil.

Heat is best removed from the gear teeth by spraying the oil onto them when they are hottest; i.e., as they come out of mesh. Only in the case of very high speed gears is there a danger that the amount of oil left on the teeth on reentry into mesh will be insufficient to form a lubricating film and, therefore, to need a supplementary supply at the ingoing side of mesh. And, only in the case of heavily loaded gears with large numbers of teeth and running at high speeds, do these authors conclude that better cooling than by spraying onto the gear teeth is needed. In such cases, it is suggested that cooling can be significantly improved by supplying oil to the inside of the gear by means of a banjo, the oil then passing through holes in the rim into the tooth spaces, as already is the practice in the sun gears of the Stoekicht design of epicyclic gear.

EFFECT OF GEAR MATERIALS ON RESISTANCE TO SCUFFING

It has long been known that hardness and strength of gear materials are not reliable indicators of resistance to adhesive wear. To quote Merritt: (*30*) "Of two carbon steels, one containing 0.4% carbon and the other 0.55% carbon and so treated to give similar physical properties (i.e. Brinell hardness, tensile strength etc.) the steel with the higher carbon content will be found in all but exceptional circumstances to resist wear to an appreciably higher degree. Similarly, a chromium steel containing 0.3% carbon, 3% nickel and 0.8% chromium and heat treated to a strength of 55 tons/inch^2 will usually be found to have a resistance to wear inferior to that of a 0.60% carbon steel of similar hardness."

High carbon steels would thus be preferred if the only consideration were wear resistance, but strength and resistance to tooth breakage are possibly even more important properties, and alloy steels are, therefore, commonly used for the manufacture of highly rated gears. Tungsten and molybdenum, as in high speed steels, are generally found to increase the resistance to scuffing, the effect being attributed to the presence in the microstructure of massive carbide crystals having very high hardness at high temperatures, as is also the case with white cast irons. In contrast, the most popular alloying elements, chromium and nickel, either singly or in combination, generally seem to impair resistance to scuffing. Recently, Matveevsky and co-workers, (*31*) using a slow-speed bench rig, were able to correlate the surface energy of adhesion of oil to steel with the tendency to scuffing. Figure 10 shows the results obtained by these authors for additions of either chromium, nickel, or tungsten to high carbon steel.

In an earlier approach to this problem, Niemann and Lechner (*32*) obtained, as shown in Fig. 11, a significant correlation between the retained austenite content of a number of different steels and their tendency to scuff in the FZG test. These two approaches may have much in common, for certainly nickel increases the tendency to retain austenite, while chromium and tungsten do not.

When the lubricant film fails, the nature of the underlying oxide film is, of course, of first importance for the prevention of adhesion. And the nature of the oxide layer on steel may depend on its alloying elements. Chromium, particularly, has a greater affinity for oxygen than iron

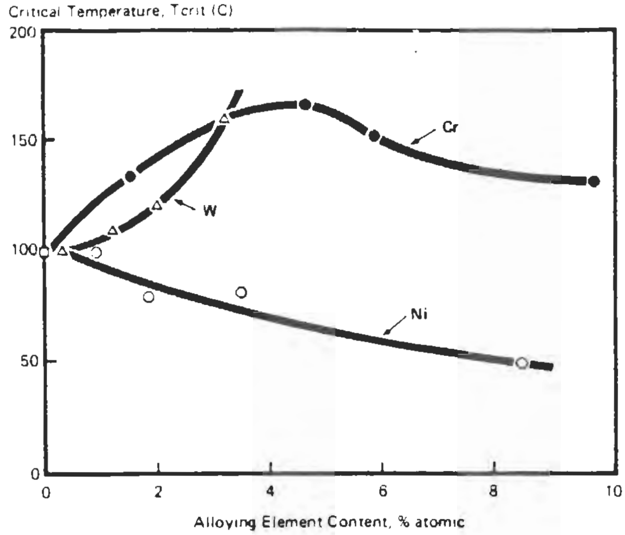

Fig. 10—Dependence of critical temperature on alloy content

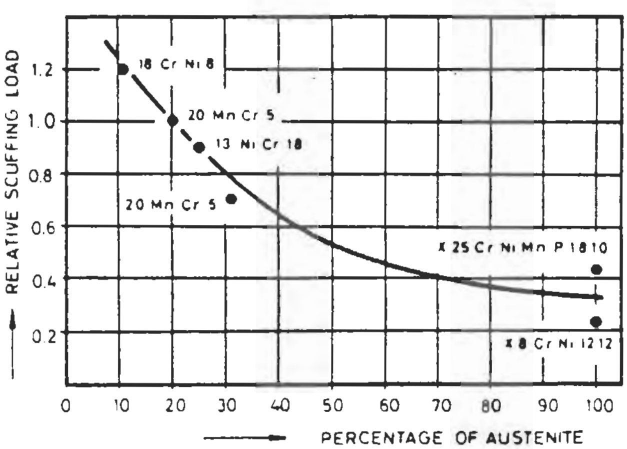

Fig. 11—Decrease of relative scuffing load with increasing austenite content of gear material.

has, and, with 12 percent Cr, there is an almost complete layer of chromic oxide rendering such steel stainless and resistant to further oxidation; e.g. scaling. This chromic oxide layer, which will tend in low alloy steels to occur particularly on high spots where there is rubbing, is very hard and brittle and, therefore, very liable to break up under sliding load to expose bare metal. Moreover, the hard oxide fragments may abrade the oxide film on the mating surface. Adhesion would thus be promoted. This may be the cause of the drop in load-carrying capacity at higher concentrations of chromium shown in Fig. 10. Thus, the difficulty of avoiding scuffing with 18/8 austenitic stainless steel may be due more to its thin, brittle oxide layer than to its austenitic structure. But experiments by Grew and Cameron (*8*) showed that the nature of the oxide film is not the only factor involved. They used the same 4.25 percent Ni 1.25 percent Cr case-carburized steel; i.e., having the same type of oxide layer, but with different quenching treatments to give in one case a structure with 25 percent retained austenite, in the other case only 5 percent austenite. It was found that, in the former case, there was a breakdown of the lubricant film at a temperature of about 150 C, whereas in the latter case there was no failure up to 190 C. Thus, the question as to why austenite should be more prone to scuffing than martensite once the oxide film has been removed still needs an answer.

However, when in turn the oxide film fails, the final barrier to adhesion is the monolayer film of oxygen bound to the steel, or monolayer film formed from the lubricant, or a combination of the two. These films can be very tightly bound to metal surfaces, as has been clearly shown by Keller and co-workers (*33*), and need vacua of 10^{-9} Torr. and argon ion bombardment for their removal in the laboratory or, in practical situations, high temperatures or gross plastic deformation. Work in the USA has shown that films from light saturated organic vapors do not form a very effective film on pure iron, whereas oxygen and some oxygen and sulfur compounds do (*34, 35, 36, 37*). There is also evidence that small concentrations of carbon in iron greatly reduce adhesion. The picture is incomplete, but it would seem reasonable to suppose either that the presence of carbon atoms in the metal surface by themselves reduce adhesion, or that hydrocarbon films bond more tenaciously to the carbon atoms than to the iron atoms in steel. Thus, another factor in the relative resistance of steel to adhesion or scuffing might be the relative concentration of carbon atoms in their surfaces. This depends on the degree of strain necessary in the metal lattice to accommodate the carbon atoms, and it is certain that there is much less strain, and accordingly less surface segregation, in austenite than in martensite.

The availability of carbon atoms in the surface thus offers a possible explanation for the differing resistance to scuffing shown by austenitic and martensitic structures. The reasoning also extends to the case of unhardened pearlitic steels, although in this case it is likely that the interruption of the ferrous surface by the carbides restrains scuffing by preventing the junctions from growing in size. But, it is of interest that this explanation would not apply in the case of the bronzes where the function of tin seems analogous to that of carbon in steel.

With the bronzes, for worm wheels a 12 percent tin phosphor bronze has been found by experience to give optimum results, but though this material is exceptionally resistant to scuffing, it is rather apt to permit pitting and somewhat deficient in bending strength. Aluminium bronzes are, therefore, used for slow-speed high-torque worm gears, a further advantage being their lower cost. Unfortunately, however, aluminium bronze is prone to scuffing, even after prolonged running-in (*38*). This is partly attributable to the thin, hard, and brittle nature of the aluminium oxide which covers the surface and also gives this material its considerable resistance to corrosion, partly perhaps also to the fact that aluminium is inherently a more difficult metal to lubricate than tin. But it is significant that recently developed processes for improving the wear-resistance of yellow metals are basically the diffusion of high tin alloys into the surface (*39*). This may be because, like carbon in steel, tin, which is in the same group of the Periodic Table, is able to form stable organoderivaties from the oil, which aluminium, being in a different group, does not do so readily or so well. At all events, the best policy with aluminium bronze gears seems to be to depend as little as possible on boundary films and to maximize hydrodynamic lubrication by using as viscous an oil as possible.

RELATIONSHIP BETWEEN PITTING AND SCUFFING

Pitting and scuffing are usually treated as being distinct and separate phenomena, the former generally occurring at low speeds, the latter generally occurring at high speeds. There is however, much evidence (*40, 41*) that, at least sometimes, the two are interconnected, so that scuffing is sometimes found to promote subsequent pitting which, especially the form known as micropitting, is held to cause subsequent scuffing. Although EP oils may or may not delay pitting, once it has appeared there is reason for their use on such occasions since they may prevent scuffing from occurring subsequently.

FRICTION IN EHL FILMS

EHL theory has been very successful in accurately predicting the thickness of lubricant films in counterformal contacts. But no simple theory has yet been developed to predict correctly the friction arising in such contacts over the whole range of speeds, though some complex analyses have come fairly close (*42*). It seems that, within the contact and under the influence of the very high shear stresses (*43*), the lubricant viscosity becomes abnormally low. That is, lubricants which are Newtonian under other conditions are non-Newtonian in EHL contacts. This is a highly desirable effect because otherwise frictional resistance and power loss would be very much higher. But, if the non-Newtonian effects are too pronounced and occur in the inlet region of the film, then an insufficient amount of oil is forced into the load-carrying region and the film thickness is reduced. Ideally, therefore, a lubricant for EHL should be Newtonian at the conditions occurring at the inlet to the film, but non-Newtonian when fully within the load-carrying film.

Naturally, some types of lubricants are further from this ideal than others. The silicones are too non-Newtonian to be able to form sufficiently thick films, while mineral oils, especially the LVI types, are insufficiently non-Newtonian in the load-carrying film. Fatty oils such as castor oil, the synthetic diesters, and the polyglycols are pretty close to the ideal.

These properties are important, even though in practice EHL films may not be complete. They find their fullest importance in the lubrication of worm gears, since their power capacities are usually limited by temperature rise due to friction. Of these three types of lubricant, the polyglycols have proved themselves most suited for heavily loaded, hot-running worm gears, since they are available in high viscosity ranges and have excellent oxidation stability. In contrast, the diesters are available only in comparatively low viscosity ranges and castor oil has relatively very poor oxidation stability.

While this paper mainly deals with the lubrication of toothed gearing, it is appropriate at this point to mention the oil lubricated friction drives. In such transmissions, one surface drives another by tangential forces transmitted through EHL oil films, and high friction in the film is, therefore, required. Accordingly, the opposite types of oil are often preferred for friction drives compared with those for toothed gears. Thus, while worm gears are best lubricated by oils having predominantly linear structure, like fatty oils, polyglycols and paraffinic mineral oils, the friction drives develop less slip with lubricants having a greater amount of cyclic groups: e.g., aromatics and naphthenes, as found in LVI mineral oils and certain synthetic oils such as polyphenyl ethers, alkylated naphthalene, chlorinated biphenyls. Although slip can be reduced by using such oils in preference to paraffinic mineral oils, the power loss is not necessarily reduced because it is the product of friction and slip. Since a reduction in slip is only the result of an increase in friction, the product may in fact increase.

RUNNING-IN

When gear units are first assembled, the contact between the teeth never forms a full and continuous line because of the unavoidable inaccuracy in tooth cutting and in assembly. Initially, therefore, the areas of tooth contact are very limited and the local pressures tend to be very high. If full load and speed is applied immediately to such a gear there is, therefore, a great danger that the surfaces will be damaged by abrasion or by scuffing. By running, however, under easy conditions such that abrasion and scuffing do not occur, the high spots on the surface are smoothly increased in length and area until contact is obtained over 80–90 percent of the face width. The gears may then take their proper designed loads and speeds without fear of damage.

During running-in, adjustment of the surfaces occurs partly by local wear at high rates with production of metallic debris and partly by plastic flow. At the same time, the asperities on the surfaces are smoothed down and a thick oxide film develops on the surfaces so that wear increasingly becomes of the mild type with oxide debris. Eventually, the surfaces may become smooth enough, and of large enough area for EHL films to form. At that stage, of course, wear virtually ceases, but, if the original surface undulations were rather pronounced, there may still be considerable areas of the teeth which do not bear load. Consequently, those parts of the teeth which do carry the load may be subject to excessive stress, especially under shock loads, and may eventually fail by pitting. Thus, while running-in is important, it cannot entirely eliminate the effects of poor machining.

Occasionally, it is possible to run-in gears using a lower viscosity oil than is afterwards used in service, so that the area of contact is rapidly increased. EP oils are also used during running-in. Initially, the idea was to protect against scuffing while promoting mild wear, but it now appears that at least certain EP additives promote running-in by permitting plastic flow of the prominent areas of the surfaces (*14*).

Unhardened or through hardened chromium steels are particularly difficult to run-in because the surface oxide is not wholly iron oxide, but, especially where load and surface distortion are heaviest, contains large amounts of chromic oxide. Chromic oxide does not develop thick layers readily, and, because it is also very hard and brittle, it breaks up easily under sliding loads exposing the bare metal thus promoting scuffing (*44*).

Some gears, such as hypoid and spiral bevel gears of automotive rear axles, are particularly difficult to run-in. This is because, owing to mutual distortion of the gears and their mountings, the full width of tooth contact only occurs when very heavy loads are applied. Thus, certain parts of the teeth cannot be run-in gradually under light load. It is to overcome this difficulty that EP additives for hypoid gear lubricants have to be rather chemically active, so that they can rapidly form thick scuff-resisting films on steel surfaces even before they have ever come into contact. Then, when sudden overloads are applied and those surfaces come into contact, no damage occurs.

SELECTION OF GEAR LUBRICANTS

It will be apparent that about the most important property is viscosity. High VI paraffinic oils (80 VI and over) are generally preferred for gear lubrication because of better oxidation stability, better response to antioxidants and to EP additives, as well as a smaller change of viscosity with temperature than naphthenic oils. Where low pour-points are required, however, naphthenic oils are necessarily and quite happily used. For hot-running worm gears, however, because it is so important to minimize tooth friction, paraffinic oils are definitely preferred to naphthenic oils, and, under the severest conditions, polyglycol types have proved extraordinarily effective. Similarly, where gears have to start up at very low temperatures, it may be necessary to use synthetic oils, such as the diesters and the polyglycols, which change relatively little in viscosity with temperature and have very low pour points.

Generally speaking, providing the oil can be properly fed to the tooth surfaces, the higher the viscosity grade of the oil the greater is the protection against the various forms of surface damage. Oil viscosity grade is, however, limited in practice either by excessively high temperatures arising in fast running bearings or by difficulties in starting up from cold, or both. Thus, low viscosity grades are used for high speed gears and vice versa. Accordingly, there tends to be a greater risk of scuffing at high speeds than at low, but this is offset to a considerable extent by the practice of designing high speed gears to have smaller teeth and, therefore, a lower degree of sliding, as well as to have lower loads.

Some systems of lubricant recommendation, therefore, simply use the pitch line speed of the gears to determine the viscosity grade required; e.g., $\nu_g = 7{,}000/\sqrt{V}$ where ν_g is the viscosity grade required in cSt at 100 F and V is the pitch line speed in ft/minute. But load can be usefully taken into consideration as shown in Fig. 12 developed a few years ago (*45*). This relates the viscosity grade ν_g to the value K/V where K is Lloyd's load criterion related to Hertz pressure and defined as follows:

$$K = \frac{W_t}{d}\left(\frac{\rho + 1}{\rho}\right)$$

where

W_t = tangential load in pounds per inch of face width

d = pinion pitch diameter in inches (taken at the large end in bevel gears)

ρ = gear ratio

Figure 12 applies where ambient temperatures are in the range 10–24 C (50–75 F).

Higher values of ν_g are required under the following conditions:

1. Ambient temperatures normally exceed 24 C (75 F). The appropriate correction will be 10 percent in viscosity grade per 3 C (5 F) increase over 24 C (75 F).
2. The gears are subject to shock loads; e.g., rolling mill gears. An indication of the appropriate increase in viscosity grade can be made by multiplying the K factor by 1.5 for conditions of moderate shock or 2.0 for severe shock.
3. The mating gears have (a) similar chemical analyses or (b) include nickel chrome steel (except case hardened or nitrided steel). Increase the indicated value of ν_g by 35 percent where this condition applies.
4. The indicated value is less than that required for the bearings.

Lower values of ν_g than those indicated may be used:

1) When the ambient temperature is normally below 10 C (50 F)

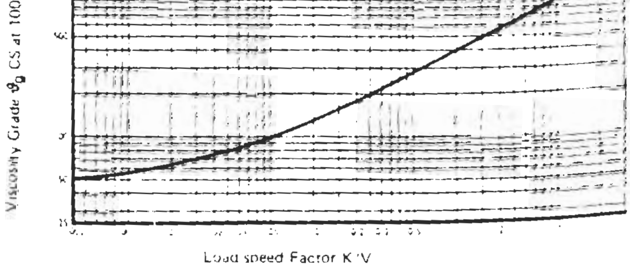

Fig. 12—Oil viscosity grade recommendation chart for parallel shaft gears.

2) In some cases where the teeth have been phosphated or sulfur impregnated, the risk of scuffing during running-in being reduced. A reduction of v_g of up to 25 percent may be possible.

Worm gears generally require a rather higher viscosity grade of oil than spur, helical, and bevel gears do for the same input speed because the reduction of tooth friction is of greater importance. A recommendation system based on experience for units with case hardened steel worms and phosphor bronze wheels is given below:

RECOMMENDED OIL VISCOSITY GRADES (CST AT 100 C) FOR TOTALLY ENCLOSED WORM GEARS

Center Distance, mm	Worm Speed, rev/min				
	250	750	1,000	1,500	3,000
Up to 75	17	17	17	17	17
75 to 150	43	31	31	31	24
150 to 300	43	31	31	24	17
Over 300	31	24	24	17	14

Where worm gears frequently start and stop under high load, oils compounded with 5 percent fatty oil or its equivalent are recommended.

When information on output torque is available, a more accurate selection may be made from Fig. 13 using the parameter M_w/C^3N_w in which M_w is the output torque in lb. in., C the center distance in inches, and N_w the worm wheel rpm. In this parameter, an average relationship has been assumed between center distance and worm wheel diameter, and the result is an empirical measure of the quotient of surface stress in the conjunction between the rubbing surfaces and the tendency to generate an oil film (*45*).

The formation of a load carrying oil film in worm gears is specially liable to impairment by deflection of the wheel and worm under heavy loads (*46*). It appears that tooth friction can be an important part of the distorting forces since Kara and Wirtz (*47*) found that for each lubricant there is a maximum torque which, independent of speed, worm gears can carry. These authors have, therefore, suggested replacing the speed factor in Dudley's expression for maximum torque (*48*) by a factor which increases with increasing viscosity grade of the oil.

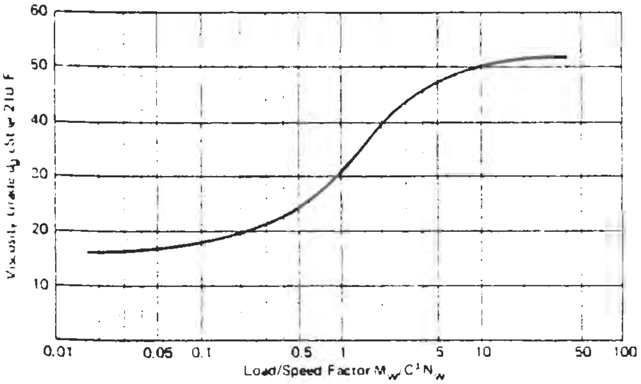

Fig. 13—Oil viscosity grade recommendation chart for worm gears.

TESTS FOR LOAD CARRYING CAPACITY

Straight mineral oils are perfectly adequate for the majority of gear units, but, where excessive loads may arise even if for only short periods; where the accuracy of the gear teeth or the compatibility of the steels may be in doubt; or where there is no time to run the gears in gently, EP oils are commonly used. EP properties are often specified by type, such as "lead-containing", "sulfur-phosphorus" etc. Often a particular level in a laboratory test rig is also specified; e.g., a Timken OK Value, (ASTM D2782), one of the many Four Ball criteria (IP239T), but most commonly the Weld Load (10 second or 60 second), an FZG test (DIN 51 354), Ryder Test (ASTM D1947), or an IAE test (IP166/68).

All such tests are "steel on steel" and can clearly distinguish between a straight mineral oil and an EP oil, but there is little confidence in their ability to otherwise rate load carrying capacity. Experience with hypoid oils shows that there are basically two types of load carrying additives: those that protect against scuffing under steady load conditions; and those which can protect against shock loading. Phosphorus containing additives such as the triaryl phosphate esters and especially the zinc dithiophosphates, give protection under steady load conditions but not under shock load conditions. In contrast, highly chlorinated hydrocarbons and highly sulfurized materials, as in hypoid or full EP oils, are good under shock conditions. The standard gear rig tests represent the steady load conditions very well and the zinc dithiophosphates give good results in these tests, but the hypoid oils usually cannot be failed by them. It has been pointed out that the steady way in which the load is raised in these tests helps to run-in the test gears by gradually relieving the tips and roots of the teeth. Such tests, therefore, favor zinc dithiophosphates with which wear is very slow (*49*). Greater definition between the zinc dithiophosphate type of additive and the full EP type has been obtained by starting the IAE test at a high load (*50*), but this procedure has not been widely accepted. In developing a modified FZG spur gear test rig equivalent to the CRC L-42 High Speed Shock Test for hypoid oils, the FZG Institute have increased the center distance from 91.5 mm to 140 mm with 8 mm module teeth instead of 4.5 so that the diameter/addendum ratio for the pinion reduces from 8.7 to 7.9. Also, to increase the severity the 19 tooth wheel drives the 15 tooth pinion instead of the 16 tooth pinion driving the 24 tooth wheel (*51*). (Refer Fig. 7.)

Oils containing phosphate esters or zinc dithiophosphates are often called antiwear oils because in the Four Ball Wear Test at low loads and long duration; e.g., 30 kg for 100 minutes, they allow only very small wear scars on the lower balls, whereas by other criteria, including the Weld Load, and in the Timken test they are only a little better than the base oil. Oils which allow relatively large Four Ball wear scars under these conditions, do not, however, necessarily allow rapid wear of gear teeth; e.g., the lead soap active sulfur hypoid oils, so that the size of the wear scars on the lower balls is of limited significance. The difficulties of interpreting Timken OK Values in the range 35 lb to 80 plus have been well reviewed by Culp and Lieser (*52*), but no recent discussion of the significance of the Four Ball Weld Load appears to have been made.

Preference has usually been given to Four Ball criteria based on the conditions which first produce severe wear; e.g., two and one-half second seizure delay and the initial seizure load, on the grounds that they relate best to conditions in gears. In contrast, the Four Ball Weld Load has not been highly regarded for two reasons: because its value depends not only on the quality of the lubricant but also on the maximum torque that the machine can supply; i.e., because the results depend on the size of the motor and on the supply voltage; and because the rate of wear of the lower balls is so colossal that it cannot relate to events between gear teeth. Nevertheless, it is a quick and popular test for specification purposes; e.g., U. S. Steel specifications 220, 221, 222, 223, and general experience is that no hypoid oil with any pretensions to protection against shock load has a 10 sec Four Ball Weld Load lower than about 300 kg. In fact, the Royal Navy recently added a minimum Four Ball Weld Load (60 sec.) of 200 kg to the IAE Scuffing Load of 90 lbs minimum for their EP Turbine Oil specification in order to ensure a degree of protection against scuffing under maneuvering conditions. Furthermore, the oils with the best shock load characteristics in hypoid gears, (the lead soap active sulfur type) have very high Welds Loads—over 800 kg.

It seems possible that the essential characteristic of scuffing under shock loading is the sudden, simultaneous removal of the protective film over large areas on both sets of gear teeth. Up to the weld load in the Four Ball Machine, the wear is concentrated on the lower balls, but at the weld load the film must also disappear within the time period on both moving surfaces. Thus, the weld load may reflect the ability of lubricants to form and maintain an effective film rapidly on the top ball even though an inordinate amount of wear has to take place on the lower balls to provide the necessary degree of severity (*53*).

PRACTICAL FACTORS

The detail gone into above must not be allowed to obscure the importance of practical engineering factors: The teeth must be accurately cut as regards pitch, profile, helix angle, and concentricity; they must be accurately and rigidly mounted and they must be supplied with an adequate amount of clean, cool lubricant. (*54, 55*). The more complex the lubrication system is, the easier it is for it to become contaminated, and the more difficult it is to clean and inspect it. And not only is it more difficult, but more important that these complex systems should be free from all debris.

CONCLUSION

This paper attempts to review the most important aspects of gear lubrication from an enormous mass of literature that continually grows. But even so, important questions like the true nature of scuffing, how it comes about, and how it can accurately be taken account of in design, are not completely settled. The subject not only concerns engineers but brings in metallurgists, chemists, physicists, and petroleum technologists.

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