Dimensional Value of LUBRICANTS in GEAR DESIGN

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THERE have been considerable changes in recent years in the role of lubricants as engineering materials. In times past the designer of a machine had about the same regard for the lubricant that was to be used to lubricate its moving parts as he had for the paint that would be used to decorate its exterior. Both of these questions could be settled safely after the machine was built since neither had any bearing upon the design details. Today, however, as a result of the modern demand for more intensive use of structural materials, many design details are dependent upon specially compounded lubricants. These lubricants have thus become inseparable parts of the design and the designers of lubricants share with the engineer the responsibility for the success of the machine. In these modern high-output machines the added burdens that are placed upon the lubricants vary with the kind of machine part, with the nature of the increased load, and with the kind of service that is required. Since these requirements change from time to time, the oil technicians have had to set their pace to meet the changing demands.

While it is true that we live in a continuously changing world, it is also true that really new discoveries occur very infrequently. According to available records, the first use of the principles upon which our present powerful gear lubricants are based date from the middle of the last century, and they have been in continuous use since then in cutting oils. During this time many contributors have added odd bits of information that, in total, have proved very helpful in giving us a clearer understanding of our modern gear lubrication problem. Tomlinson showed that really clean surfaces of like material, such as glass, will weld in small areas when rubbed against one another even at light loads, but that they will not weld except at high loads if the surfaces are not freshly cleaned to remove contaminating surface films. Experiments had indicated that certain oils, such as lard oil, left strongly adherent films on the surfaces of metals. The film left by perspiring hands of workmen prevented successful scraping of iron surfaces such as lathe beds. Dry sulfur was often used to treat overheated bearings. Blacksmiths required special coal for use in the forge because they had found that they could not weld iron if the coal contained sulfur. Likewise a forge could be poisoned for welding by dropping a few bird shot in the fire. White lead was commonly used as a lubricant for lathe centers. Successful cutting oils contained lard or similar oils or high sulfur mineral oils or both. These and similar bits of information were the

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AUTOMOTIVE rear-axle gears present one of the most difficult gear-design and lubrication problems that are known, Mr. Almen emphasizes in this paper. The requirements are exacting, he explains, because (1) the weight must be held to a minimum; (2) the cost must be low; (3) the dimensions must be small; (4) the heat input is great; (5) the temperature is high, and (6) hypoid gears with their greater sliding velocity must be used.

Early in his paper Mr. Almen establishes the hypothesis that lubricant failure in gears and other highly loaded surfaces results in welding of small areas of the mating surface for the following reasons

1. The rubbing action removes weakly adhering films permitting contact between clean metal surfaces.
2. The temperature of a thin surface layer is very high from the friction of sliding under high load.
3. Welding of two surfaces can occur at temperatures considerably below the melting point of a metal if the pressure is great.
4. Welding will not occur if a sufficiently tenacious contaminating film is formed on the rubbing surfaces.

Explaining his title – "The Dimensional Value of Lubricants in Gear Design" – Mr. Almen points out that, as a result of the modern demand for more intensive use of structural materials, many design details are dependent upon specially compounded lubricants, and these lubricants have thus become inseparable parts of the design.

Considerable data, including three-dimensional charts, are presented to show the effect on the scoring limit of the lubricant, the pressure, the temperature, the sliding velocity, and the hardness.

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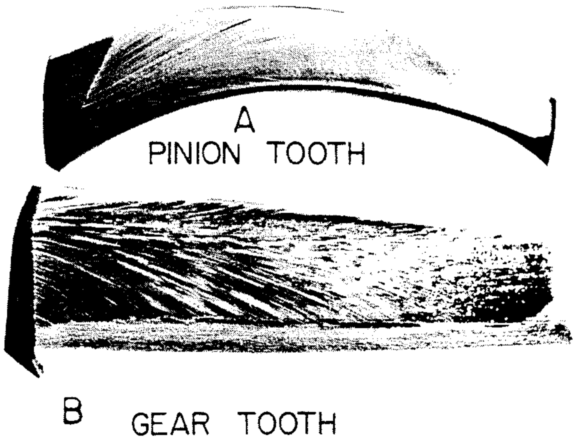
seemingly slender clues that were available to the lubrication detectives when the need arose to explain how a lubricant must function successfully to lubricate rapidly sliding steel surfaces under pressures in excess of 300,000 psi, such as occur between gear teeth.

When hypoid gears were first introduced by the Gleason Works for Packard cars, it was found that the ordinary gear lubricants were not satisfactory and only one lubricant made to a secret formula by a single manufacturer could be used. This limited source of supply was not a serious obstacle to the use of hypoid gears so long as they were limited to one make of car but, when it appeared probable that hypoid gears would some day be used by all automobile manufacturers, it seemed desirable to learn the secrets of hypoid gear lubrication.



■ Fig. 1 – Scoring of gear teeth

As just stated, the available information on the general problem was a number of disconnected observations that had accumulated over a long period of time together with the fact that certain lubricants would operate satisfactorily under conditions where normal lubricants would fail. Failure of lubricants in gears and similar service was always characterized by surface scratches which could not be attributed to foreign matter in the oil, and occasionally, projecting points could be found on the rubbing surfaces. This result suggested that, under operating conditions of very high pressures, accompanied with rapid sliding and the generation of heat by friction, the rubbing surfaces were instantaneously welded together in small areas. The welded areas were then torn apart by continued sliding with material being pulled out of one surface and adhering to the mating surface. These projections produced the scratches and, in the process, were themselves worn away.

A long series of tests on the road and in the laboratory showed that lubricants containing various materials, among which were certain animal and vegetable oils, sulfur, lead and chlorine, were superior to normal mineral oils in their ability to carry high loads without scoring or scratching. This finding was in agreement with the welding theory and with the experience of the blacksmith with sulfur coal which prevented welding. The hypothesis was then formulated that lubricant failure in gears and other highly loaded surfaces results in welding of small areas of the mating surfaces because: (1) the rubbing action removes weakly adhering films permitting contact between clean metal surfaces; (2) the temperature of a thin surface layer is very high from the friction of sliding under high load; (3) welding of two surfaces can occur at temperatures considerably below the melting point of a metal if the pressure is great and; (4) welding will not occur if a sufficiently tenacious contaminating film is formed on the rubbing surfaces.

With this hypothesis as a guide, many lubricants were tried that were specially compounded to produce films on the metal surfaces that were strongly adherent and that had anti-welding properties. It was found that, in such lubricants, under high load, the strongly adhering chemical film was the actual lubricant and that the oil acted principally as a coolant. The only difference between these experiments and the experience of the blacksmith was that, in the case of hypoid lubricants, the anti-welding film was formed under the heat of friction while the blacksmith produced the anti-welding film directly from sulfur by the heat of the forge.

Of course, the ability of a lubricant to form an anti-welding film is only one of many criterions for a satisfactory hypoid gear lubricant. Other requirements, such as stability, corrosion, activation temperature, compatibility with other lubricants, and so on, have kept the oil technicians busy for many years and promise to keep them busy for many years to come.

Meanwhile the engineers also have been busy with the same problem but have attacked it from a different angle. If gear scoring is a function of the unit pressure, the sliding velocity, the temperature and the surface hardness of the mating surfaces, as has been indicated, it should be possible to vary the magnitude of these factors by design and thus possibly reduce the burden on the lubricant or alternatively permit the use of smaller, lighter, and cheaper gears. It is probable that whatever design progress is made will be utilized in the latter manner because of the economic necessity of taking more work out of each ounce of structural material. The partnership between the lubricant designer and the machine designer is, therefore, likely to draw closer with each progressive step taken by either partner.

Progress in mechanical design has been slow because of the many difficulties that have prevented full understanding of the factors that are involved. Before such a problem can be attacked intelligently, it is necessary to assemble accurate data on the effect of each of the many gear de-

■ Fig. 2 – Instantaneous contact area between teeth sign variables. Since gears may fail to give satisfactory service for other reasons than scoring, such as tooth breakage, tooth pitting and wear, the designer must not favor the factors that influence scoring to the detriment of these other causes of failure. The only source of dependable data on gears, as on any other machine part, is actual service experience. The most useful service data come from failures and not from successes because only failures define the capacity limits of the strength of machine parts or of a lubricant. Service failures of any kind must, of course, be infrequent and, since a large number of factors are involved, the accumulation of sufficient data on any one factor, to be significant, requires a great deal of time and effort.

Many attempts have been made to devise laboratory apparatus from which basic quantitative information may be assembled more easily than from service experience, but such apparatus, no matter how elaborate, has never been successful in yielding reliable data. In general, laboratory test machines can be useful for some qualitative measurements only, since it is not possible to reproduce in the laboratory all or even a substantial part of the operating conditions that occur in service.

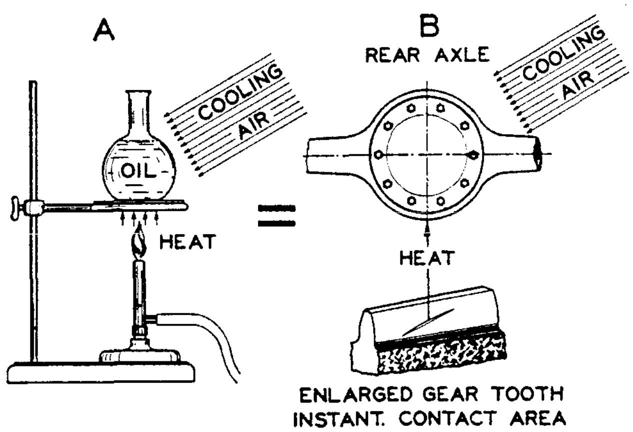
There are now in use in many laboratories a large variety of oil testing machines that not only do not agree with service experience, but fail just as badly to agree with one another. This condition is not surprising when the complexity of the problem is understood. Any testing machine that properly proportions each of the many dimensional and operating variables will have to be constructed and operated just like the machine in which the lubricant is to be used. For automobiles or trucks, this means that the testing machines will have to be complete, full-scale automobiles or trucks operated on the road. But even this is not enough because a test driver will not encounter all of the driving conditions that occur in service, nor will one or a dozen automobiles or trucks be representative of all of the automobiles and trucks that must be lubricated.

The many different oil testing machines that are used often serve to confuse rather than to clarify the lubrication problem, since the operating conditions of almost any such machine can be altered so as to show any one of several different lubricants to be superior to any other of the group. In comparing lubricants, any oil testing machine should be used as a limited grading device only. Suppose that lubricants *A*, *B* and *C* have been found, through service experience, to rate in that order of excellence in preventing the scoring of gears. It is then usually possible to adjust the operating conditions of any machine to rate these lubricants in the same order, but it is not safe to assume that a fourth lubricant, *D*, on which no service data are available, will be properly graded by this machine.

An increasing number of laboratories are installing dynamometer equipment for laboratory testing of full-scale axles in the hope of obtaining more reliable data than are afforded by testing machines. Such tests are perhaps more reliable than arbitrarily designed machines, but it must be remembered that an axle in the laboratory is still only another form of testing machine for which the operating conditions can be varied so as to grade a lubricant to agree or to disagree with normal road service. However, when the operating conditions are adjusted so as to produce the same scoring characteristics as previously have been determined by service data, a dynamometer axle test becomes a reasonably accurate measure of scoring, but not necessarily a reliable device for measuring other lubricant characteristics. All of which brings us back to statistical data from actual service experience as the only source of reliable information for the designer of lubricants, as well as for the designer of gears.

The result of a dynamometer lubrication test on a hypoid rear axle that is in good agreement with service experience is shown in Fig. 1 in which *A* is a pinion tooth and *B* is a mating ring gear tooth. The lubricant used in this test was a satisfactory hypoid lubricant under ordinary operating conditions, but the gears were severely scored, as is shown in the photograph, when the bulk oil temperature was increased above normal operating temperature, as might occur in ascending a long steep hill on a warm day. Note that the score lines are in the direction of sliding.

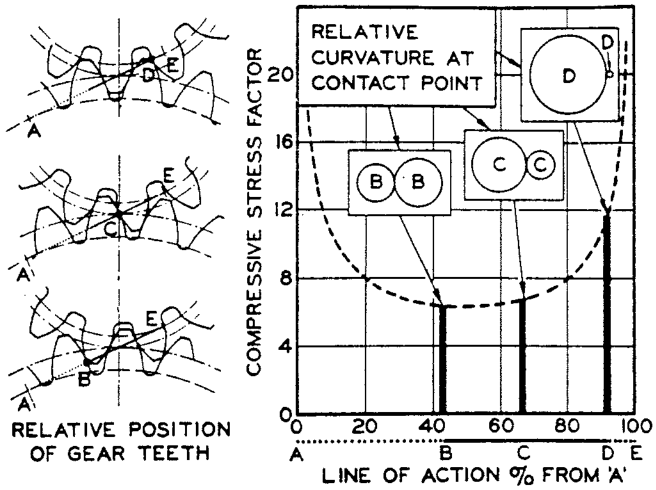
Although many serious attempts have been made, there are no means yet available for measuring the temperature of gear tooth surfaces but, that the temperature of a thin surface layer is very great, can be inferred from indirect evidence. A good mental picture of the instantaneous temperature may be had from the illustration, Fig. 2,



■ Fig. 3 – High temperatures on gear teeth due to small area of contact

which shows the size of the instantaneous contact area between mating gear teeth. All of the driving force of the gear is transmitted through this small area and, since we know the total load and the degree of curvature of each of the mating teeth, we can calculate the unit pressure. In highly loaded gears, the unit pressure may be 350,000 psi to 400,000 psi and, since these pressures occur while the teeth are sliding upon one another at high velocity, it is not so difficult to accept the welding theory of scoring. The wonder is that welding can be avoided.

A graphical representation of the rate of heat generation is illustrated in Fig. 3-*A*, in which a flask containing the oil from a rear axle is maintained at axle operating temperature by a gas flame, while air is blown over the flask surface, equivalent to the air movement over the axle while moving along the road. Obviously a considerable flame will be required to keep the oil in the flask at the axle operating temperature. In an axle the flame is



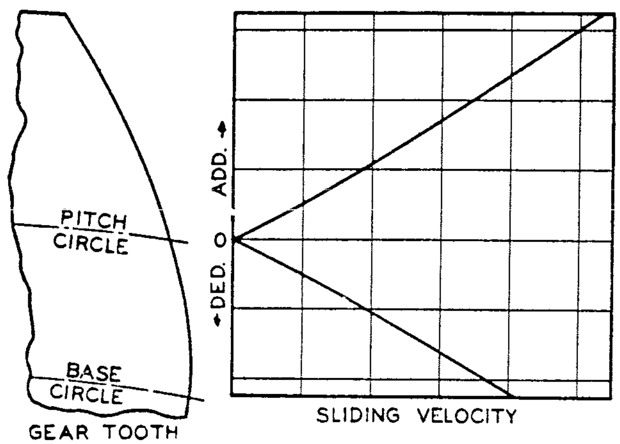
■ Fig. 4 – Influence of gear design on compressive stress

replaced by the hot instantaneous contact area as was shown in Fig. 2, which area is reproduced to a ten-times magnified scale in Fig. 3-*B*. The axle, Fig. 3-*B*, must now be maintained at axle operating temperature by the heat flow from this small area except for the additional friction heat from the ball and roller bearings. These illustrations may convey some idea of the conditions under which gears and their lubricants must perform their jobs.

The compressive stress between operating gear teeth is a function of the driving load and the relative curvature of the contacting teeth. Since the curvature varies from point to point on a gear tooth, the compressive stress also varies. Fig. 4, left, shows mating gear teeth in three different positions. In the diagram at the right are shown, by means of contacting cylinders, the relative curvatures of the mating teeth for each of the three tooth positions shown at the left. Also shown by a graph is the relative compressive stress at constant load for any point on the tooth. Note that the compressive stress, for the two-to-one gear ratio here considered, is greatest when the contact occurs near the tip of the gear tooth as at *D*. If the height of the pinion tooth above the pitch line is made somewhat longer, and the depth below the pitch line correspondingly shorter, the maximum compressive stress can be reduced without loss of tooth action. The compressive stress can also be reduced if the radii of curvature of the teeth are increased, as would result by the use of a greater pressure angle. The use of finer pitch and, therefore, shorter teeth confines the tooth action to a narrower range of compressive stress; that is, the tooth action may be designed to occupy only the lower portion of the compressive stress graph of Fig. 4. This assumes, of course, that the reduced strength of the finer teeth is permissible.

The velocity of rubbing or sliding of a gear tooth upon its mate varies with speed of rotation of the gear and the distance of the point of contact from the pitch point. The sliding velocity of spur and helical gears of the same proportion as shown in Fig. 4, is qualitatively shown in the graph Fig. 5, in which the sliding velocity is plotted against the height of the tooth. It will be seen that, from a velocity of zero at the pitch line, the sliding velocity increases as the contact moves toward the tip or the root of the tooth. It will also be seen that a decrease of the height above the pitch line and a corresponding increase of the depth below the pitch line will reduce the maximum sliding velocity without changing the tooth action. The use of finer pitch teeth is the most effective way to reduce the sliding velocity, since, the shorter the teeth, the less the sliding. We thus see that the same design change, in the particular gears shown in Fig. 4, will be effective in reducing both the compressive stress and the sliding velocity, from which it follows that the temperature also will be reduced.

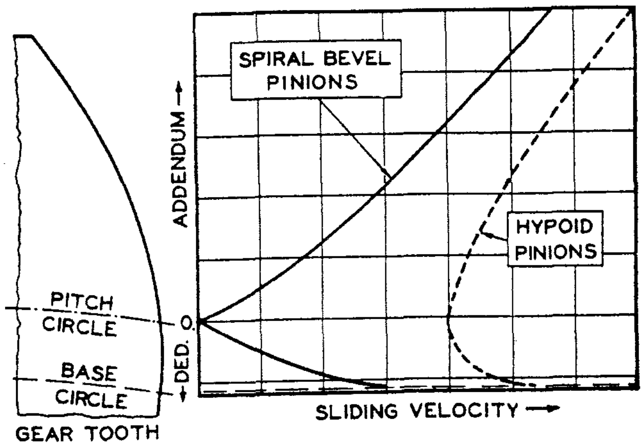
For gears of constant surface hardness, the tendency to score is measured by the temperature, the compressive stress, and the sliding velocity. The temperature of any area of a gear tooth is roughly proportional to the compressive stress multiplied by the sliding velocity (*PV*) since the coefficient of friction probably does not vary greatly for similar gears in similar service. We should, therefore, expect that scoring will appear first at the tips and at the roots of the teeth because we have seen from Figs. 4 and 5 that both the compressive stress and the sliding velocity are greatest in these regions. Fig. 6 is a spur gear tooth that shows this expectation to be true. It will be seen that the scored areas are limited to narrow bands at the tip and at the root. The portion of the tooth where the sliding velocity and the compressive stress are low is not affected.



■ Fig. 5 – Tooth height versus sliding velocity of spur and helical gears

The selection of gear tooth proportions must consider the strength of the teeth as well as the compressive stress and the sliding velocity. In automotive spur and helical gears, of the sizes now being used in transmissions, fine pitches are permissible without exceeding safe bending stresses and the occurrence of scoring in transmission gears is, therefore, infrequent. In spiral bevel and hypoid rear axles, however, the loads on the gears are relatively greater than in transmissions because of their smaller relative size and greater torque. It is, therefore, necessary to use coarse pitch teeth to avoid dangerous bending stresses. With the coarser pitches come increased compressive stress and greater sliding velocities as was shown in Figs. 4 and 5.

Coarse pitch and meshing characteristics of rear-axle gears require that the addendum of the pinion teeth be made long with consequent increase in the sliding velocity as compared with symmetrically formed teeth. Fig. 7 is a graph showing the sliding velocity in relation to the tooth height for an automotive spiral-bevel pinion and for an automotive hypoid pinion. Note that the sliding velocity is greater at the tip of the tooth than at the root for both types of pinions because of the long addenda. The velocity of sliding of the spiral-bevel pinion is zero at the pitch line but, for the hypoid pinion, the sliding does not reach zero velocity at any point and is everywhere greater than in the spiral-bevel pinion. The increased sliding velocity of the hypoid is due to the offset of the pinion by which it becomes something of a cross between a bevel gear and a worm gear. This increased sliding accounts for the greater difficulty in lubricating a hypoid axle than in lubricating a spiral-bevel axle.



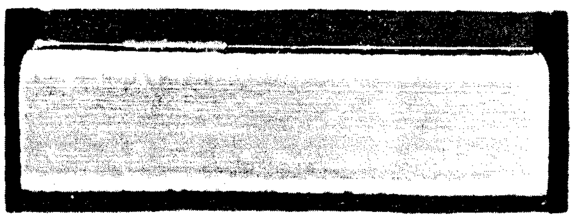
■ Fig. 7 – Tooth height versus sliding velocity of spiral-bevel and hypoid pinions

Automotive rear-axle gears present one of the most difficult gear design and lubrication problems that are known. The requirements are exacting because: (1) the weight must be held to a minimum since the axle is carried as unsprung weight; (2) the cost must be low for economic reasons; (3) the dimensions must be small for road clearance; (4) the heat input is great due to high sliding velocity and high compressive stress; (5) the temperature is high due to poor heat dissipation from the housing, because of its small size, and because it is swept by hot air from the engine and hot air from the road; and (6) hypoid gears with their greater sliding velocity must be used to accommodate modern low suspension and to provide additional gear tooth strength.

Notwithstanding all of these adverse requirements, the size of rear axles is constantly decreasing in spite of the increasing torque and speed. Specifically, a few years ago one manufacturer of large cars required ring gears 14 in. in diameter, whereas today his heavier, faster and more powerful car is equipped with ring gears of 9⅜-in. diameter and he is experiencing less field trouble with the small gears than when the gears were large. The same order of progress has been made by almost all other car manufacturers for which much of the credit must go to compounded lubricants.

The downward trend in rear-axle size is still in progress and will probably continue until an economic balance is attained between the decreasing cost of the gears and the increasing cost of the bearings. The bearing cost will probably increase due to greater loads as the gears become smaller. With the decreasing gear size will come higher tooth loads and possibly increased temperature which will have to be met by improved lubricants and by designing the gears to reduce, wherever possible, the factors that are responsible for scoring.

The job ahead will become easier as our understanding of the problem is improved. Very little data of a quantitative nature are available, but we occasionally find instances of borderline tooth scoring in which the operating gear loads and speeds are known such, for example, as was shown for a spur gear in Fig. 6. The product of compressive stress and sliding capacity (PV) to produce borderline scoring of this gear varied with the kind of lubricant that was used. For mineral oil the PV value at borderline scoring was about 3,000,000; for a mild EP lubricant the PV value equalled approximately 4,000,000; and, for the powerful hypoid-type lubricant, the PV value was approximately 5,500,000.

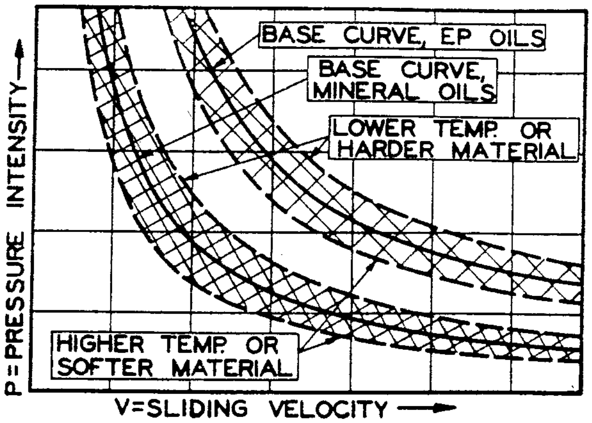


■ Fig. 6 – Scored gear tooth

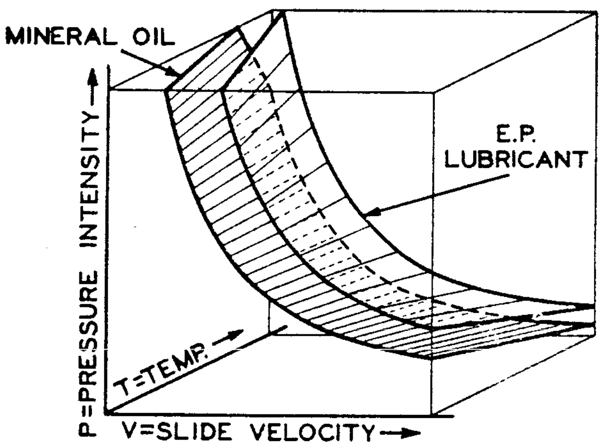
In contrast to these figures, data on spiral-bevel gears indicate permissible PV values of 1,500,000 for mineral oil and about 1,900,000 for mild EP lubricants. No PV figures are available for the more powerful types of hypoid lubricants in rear-axle gears because we do not know the relative radii of curvature of hypoid gear teeth and, therefore, we cannot calculate the unit pressures. It is probable, on the basis of other data that, in rear axles at high speeds, PV values in excess of 2,500,000 may be tolerated when hypoid lubricants are used. The discrepancy between PV values for the spur gear and spiral-bevel gears just noted will be discussed later in this paper.

The product (PV) of unit pressure (compressive stress) and sliding velocity as a measure of scoring is reasonable so long as the coefficient of friction is constant, but the constancy of the friction coefficient is questionable, especially for large changes in the magnitude of P and V. It is probable that a more general expression would take account of a decreasing coefficient of friction with an increase in velocity, and might be approximated by the expression $PV^{1/n}$. This would assume constant tooth surface temperature and constant tooth hardness.

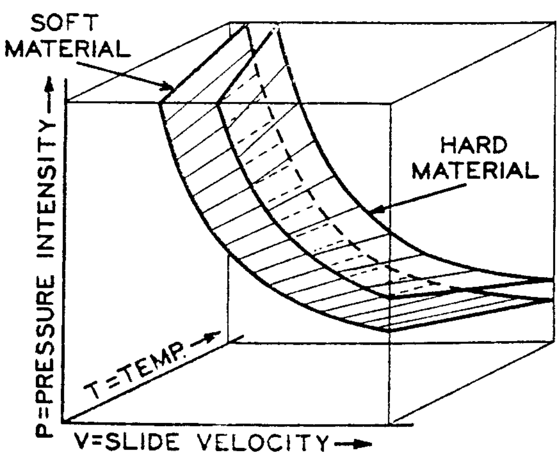
The chart, Fig. 8, shows two qualitative $PV^{1/n}$ curves which indicate the limiting values that are permissible in gears lubricated with mineral oils and with EP lubricants; that is, scoring would occur for any $PV^{1/n}$ value lying above the curve for the lubricant being considered. The position of the curves for the same lubricant would be elevated as the temperature is reduced and would be depressed as the temperature is increased as is shown by the shaded areas for each of the curves. These shaded areas will also serve to indicate that the scoring limit is elevated as the hardness of the tooth surfaces i ncreased and is



■ Fig. 8 – Effect of hardness, temperature, and lubricant on scoring limit



■ Fig. 9 – Effect of pressure, temperature, sliding velocity and lubricant on scoring limit



■ Fig. 10 – Effect of pressure, temperature, sliding velocity and hardness on scoring limit

depressed as the tooth surface hardness is decreased as is indicated on the chart.

Perhaps the relationships between pressure, velocity, and temperature can best be illustrated in a three-dimensional chart such as Fig. 9 in which a neutral mineral oil and an EP lubricant are plotted against these three variables. The scoring limit for each lubricant is a warped surface and the distance between the two surfaces indicates in a sense the "structural strength" of the compounded lubricant in relation to the neutral lubricant. As in Fig. 8 the position of the scoring limit surfaces will be depressed as the hardness of the tooth surface is reduced and elevated as the tooth hardness is increased.

Fig. 10 is a *PVT* chart for one lubricant only in which the scoring limit surfaces indicate the effect of hard and of soft gear tooth surfaces. This hardness effect may be an important consideration in compounding lubricants for industrial gears that are cut after heat-treatment, and it also indicates the importance of heat-treating to the maximum practical hardness.

A rather frequent occurrence of late years is the appearance of rippled hypoid gear tooth surfaces as is shown in Fig. 11. This rippling, which strikingly resembles the rippled surface of wind-blown or water-washed sand occurs during gear operation at high loads and appears to be due to actual slipping or smearing of a thin surface layer. It is perhaps the result of a relatively soft and very thin surface layer overlaying a hard base material. The shearing stress of friction causes this thin, soft layer to be displaced in the direction of slide thus forming rippled ridges at right angles to the direction of slide. The soft layer may, conceivably, be the result of operating temperatures so high as to soften the tooth surface to a depth sufficient to account for the ripples. There are some indications that rippling occurs more frequently with some hypoid lubricants than with others, which condition may be due to differences in the friction characteristics.

The surface temperature, and therefore the scoring limit of the contacting areas of gear teeth, is influenced by the degree to which the surfaces can be cooled in the interval between successive periods of contact. The heat that can be dissipated from the teeth depends upon the temperature of the lubricant, the viscosity of the lubricant, and the manner in which the lubricant is flushed over the gear teeth. The viscosity should be as low as is practical from considerations of leakage. The lower the viscosity, the more efficient it becomes in transferring heat from the tooth surfaces to the bulk oil and also from the bulk oil to the housing and thence to the atmosphere. The quantity of oil in the housing should not be greater than is required to flush the gear teeth since excess oil increases the temperature by churning and by trapping between the gear teeth. Wherever possible the lubricant should be kept away from the teeth as they go into mesh. The only effective lubricant at the point of contact is the thin film adhering to the surfaces of the teeth. More lubricant than is required to wet the surfaces merely increases the temperature.

Ideal gear lubrication would consist of a dry housing and the provision of jets to spray the lubricant over the teeth just after they come out of mesh. In this manner the surface heat can be flushed off the teeth before it has time to soak deeply into the metal and it assures that the excess lubricant is thrown off the teeth by centrifugal force by the time they again come into mesh. The cost of such a system would, of course, be prohibitive in most automotive installations, but can be afforded in large industrial and marine gear sets.

Gear lubricants should be regarded: (1) as coolants; (2) as chemical agents for building anti-welding films; (3) as structural materials in the sense that they are important factors in determining the size of the gears and; (4) as lubricants in the usually accepted sense of a liquid film separating the two rubbing surfaces. The many volumes that have been written on the subject of lubrication make only slight reference to the lubricating conditions that prevail in gears, probably for the very good reasons that have so frequently been indicated in this paper – namely, that there are so very few data upon which to build.

When gear teeth that have seen considerable service are examined, we find little reason to suspect that the teeth have not been uniformly loaded over their entire working surfaces. Of course, we really mean that such narrow instantaneous contact bands as were shown in Figs. 2 and 3 have uniformly utilized the entire width of each tooth and have moved over the height of each tooth during each engagement. This action, however, does not actually occur because all of our structural materials are elastic and they, therefore, bend or deflect when any load is applied.

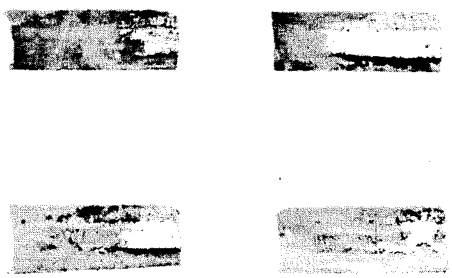
If any pair of mating teeth is exactly parallel at one load it cannot be parallel at any other load because, as the load is changed, the amount of deflection of the housing, the bearings, the gears and the shafts will change. These mating errors cannot be compensated for in cutting the gears and we must, therefore, accept as inevitable the partial use of the tooth width provided by the designer. The reason that a used gear appears to have uniformly utilized the entire width of the teeth is because the gear has experienced all manner of loads from light to heavy and it, therefore, will show the composite results of all load conditions.

When the swept area of contact for one load only is shown, as in Fig. 12, in which Scotch-tape transfers taken from transmission gear teeth reveal the actual swept area for that load, we see that the tooth width is only partially used. At very high loads the contact may extend over the whole width, but the pressure will rarely be uniform.

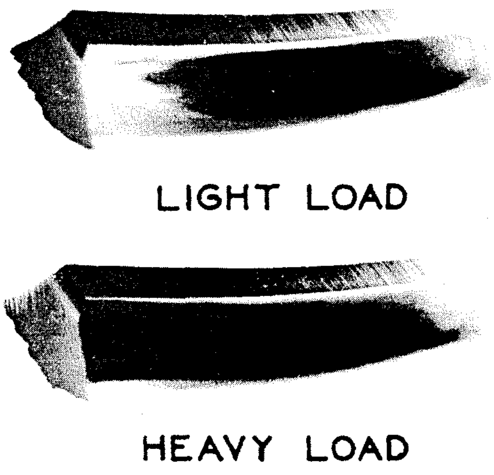
The same deflections occur also in spiral-bevel and hypoid gears but to a greater degree. In such gears, however, the teeth are cut so as to mismatch and thus provide a considerable compensation for the mating errors that would otherwise be intolerable. By mismatch is meant that the concave surfaces of the gear teeth are cut to slightly greater radii than the convex surfaces of the mating pinion teeth, and the mating teeth may, therefore, rock upon one another as loads and deflections change, like the rockers of a rocking chair upon the floor. This action is shown in the swept-area photographs, Fig. 13, which show the contact area for light loads and for heavy loads. Note that a portion of the large end of the tooth at light load is not making contact and that, at the high load, a portion of the small end of the tooth is not making contact.

While this method of cutting does not permit the utilization of the whole tooth width, it is greatly superior to teeth designed to make contact over the entire width because, in the latter case, under changing loads and there-

■ Fig. 11 – Rippled hypoid pinion

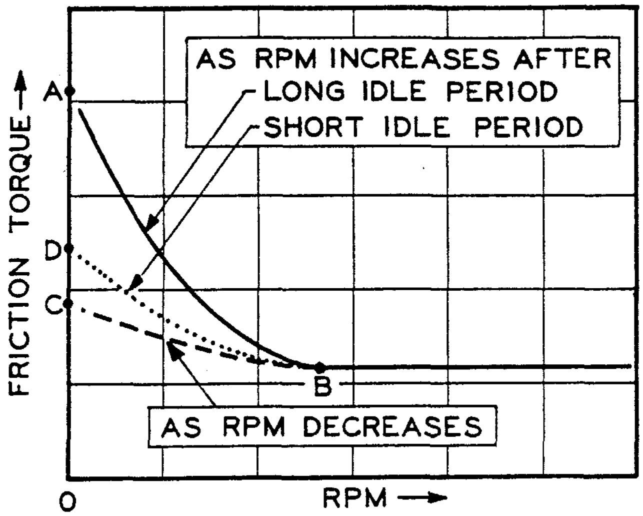


■ Fig. 12 – Gear contact impressions



■ Fig. 13 – Contact impression of hypoid gear fore deflections, the contact could occur only at the extreme ends of the teeth. The amount of mismatch that is needed depends upon the rigidity of the design or upon the compensating deflections of the design. Since the lubrication difficulty varies with the unit pressure and, therefore, with the degree of mismatch, the elastic characteristics of the housing, shafts, bearings, and gears become part of the lubrication problem.

That elasticity can be useful as well as harmful is shown in Fig. 6 in which uniform load distribution over the width of the tooth is indicated by the uniform width of the scored bands. In this gear, the toothed rim was attached to the hub through a thin central web or diaphragm, whereby lack of parallelism between mating teeth was adjusted. A load applied at one end of the tooth would cause the diaphragm to deflect and closely approach parallel relationship between the mating teeth.



■ Fig. 14 – Friction torque versus rpm between bearing balls and races

It will be recalled that the permissible *PV* values for the gear tooth shown in Fig. 6 were about twice as great as the permissible *PV* values of spiral bevels. This discrepancy is at least partially explained by the more uniform load distribution over the tooth width as has just been discussed.

The persistence of an oil film under high load is illustrated strikingly by an experiment in which a ball bearing was used as a friction-measuring device. The bearing contained only three balls and was constructed so accurately that it could be run for considerable periods without the usual ball separator. Since the separator was omitted, the readings were the friction between the balls and races only in contrast to the usual ball-bearing friction tests which include the friction between the balls and the separator. The bearing was loaded by means of a calibrated spring to a maximum pressure of 240,000 psi between the balls and the races.

The friction torque was measured at small speed increments including the starting torque with the results shown in Fig. 14. From the starting torque at *A* the friction diminished rapidly with increasing speed until it reached a constant value as at *B*. With decreasing speed the friction torque followed the line *BC* and, if immediately started again, the friction retraced the line from *C* to *B*. If after stopping, the apparatus was not again started for several seconds, the friction would follow the line *DB* as the speed was increased and then retrace the line *BC* with decreasing speed. The original starting torque *A* would not be re-established until the apparatus was permitted to stand for a considerable period before being started again, whereupon the friction would follow the original line *AB*. This result seemed to indicate that a relatively long time was required to squeeze the oil from between the balls and the races with pressures up to 240,000 psi.

This persistence of the oil film at high pressures seemingly contradicts statements previously made in this paper that the lubricant plays but a small part in separating gear tooth surfaces. However, the thickness of an oil film under the foregoing conditions is extremely small in comparison with the surface irregularities of the most highly finished mating surfaces. Metallic contact probably occurred under all of the test conditions enumerated; but the oil trapped in the surface irregularities served to reduce the intensity of the pressure on the projecting points and thus reduced the friction. Note the surface roughness of the gear tooth shown in Fig. 6 and the absence of wear on the unscored portion of the tooth after approximately 25,000,000 meshing cycles at unit pressures on the order of 200,000 psi.

The viscosity of the lubricant used on this gear conformed to SAE 10 specification. However, the actual viscosity of the lubricant trapped in the contact area was probably quite different. The high temperatures of the contact area would tend to reduce the viscosity, whereas the high pressure would cause the viscosity to increase. We do not know the temperature or the pressure-viscosity characteristics of the lubricant and we can, therefore, only speculate on the actual viscosity. Since the *PV* values and, therefore, the instantaneous temperature increase range from zero at the pitch line to a maximum at the tip and root of the tooth, it may well be that the viscosity was high in the central portion of the tooth and relatively low at the tip and the root. This condition would be in substantial agreement with the markings on the tooth.

■ Product of Many Industries

Automobiles, like most other things that we regard as necessities, would not be available to so large a percentage of our population but for the cooperative efforts of many industries and their technicians. Through industrial team work the cost of the finished product comes within the financial means of the many. The modern automobile is the joint product of the steel manufacturers, the machine-tool builders, the paint makers, and so on through a long list of industries. In no case, however, is the partnership so close as between the petroleum industry and the automobile manufacturer because both come in direct contact with the consuming public and both are directly responsible to the consumer for the products which they sell. The size, cost, and performance of the automobile engine is directly related to the knock characteristics of the fuel supplied by the petroleum industry. Similarly, the size, cost, and performance of hypoid gears is measured by the available lubricants.