Tables 6–8 also contain calculations of the peak Hertz contact stress at the pitch point and at its maximum point. Gear analysts often take the value of the pitch point as being representative of the system. These tables show that by doing so, for the gears examined here, may underestimate the peak contact stress by up to 17 percent. The tables also show no direct correspondence between contact stress and scoring.

Discussion and Conclusions

The EHD criterion was shown to be inadequate as a general design guide to prevent the scoring of gears. The inability to obtain a significant EHD film thickness was found to be a necessary but not sufficient condition for scoring to occur, hence designing gears to the EHD criterion may incur severe economic penalties through over design. However, for those applications where cost is not an important factor, the absolute certainty that scoring will not occur, the EHD criterion may be of some utility.

The general conclusions one can draw from this study are,

1. Gears scoring occurs under fully boundary-lubricated conditions and not even close to the breakdown of the EHD film.
2. Isothermal calculations of minimum EHD film thickness are in error both in magnitude and in trend with failure velocity. Analysis which takes thermal effects into account must be used.
3. Pitch point EHD film thickness does not reliably represent the minimum EHD film thickness, and therefore cannot be substituted for it.

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T. E. Tallian

The authors state that "lubrication analysts have come to believe that successful lubrication of concentrated contacts depends on obtaining a film thickness large in comparison to the combined roughness of the opposing surfaces." "Large" is from three to ten times the combined surface roughness. According to the authors, this concept has come to be widely accepted in the ball and roller bearing industry.

Having been intimately associated with the introduction of the film thickness/composite roughness ratio, as a criterion for the evaluation of the operating conditions of rolling bearings, this discussion notes a misunderstanding of the way in which the criterion is used in the rolling bearing industry, and also of its significance.

In rolling bearings, the typical life achieved prior to fatigue failure was shown to be related to the film thickness/roughness ratio. The longest lives are obtained if this ratio exceeds three. However, it would be preposterous to claim that fatigue life of rolling bearings is unacceptable short if the ratio is lower than three. The majority of all rolling bearings in service today operate with film thickness/roughness ratio less than three, and large groups of rolling bearings operate with ratios less than one, so that there is essentially no load carried by the elastohydrodynamic film. This discussor has found that one cannot predict fatigue life from the film thickness/roughness ratio alone, when this ratio is below unity. Under certain conditions, very drastic reductions in fatigue life occur in this regime, but under many other conditions, bearings operate very satisfactorily and for long periods of time.

Also, this discussor has found that the film thickness/roughness ratio is not a suitable criterion for scuffing (smearing) failure. One can thus readily agree with the authors that the film thickness/roughness ratio is of doubtful value as a gear scoring criterion. The authors' paper is of particular value in assembling a large number of film thickness/roughness ratios calculated for gears to document this position.

The authors state that overdesign would result from the use of a film thickness/roughness ratio criterion in gears. This conclusion appears doubtful since the only gear design parameter likely to influence the film thickness ratio is surface finish. Thus a high film thickness ratio will not be achieved by design changes.

Finally, it is not safe to assume, as the authors do, that a high film thickness/roughness ratio assures with absolute certainty that scoring will not occur (in applications where cost is less important). As pointed out in [15], film collapse can occur and permit scoring even in circumstances when the calculated or experimentally measured film thickness/roughness ratio is high, prior to the onset of the thermal excursions leading to scoring.

References


L. S. Akin

The discusser wishes to compliment the authors on a timely subject. The peak Hertz contact stress in a gear mesh may be significantly understated if calculated only at the pitch point.
discussion of some of the most recent aspects to be considered in
designing gears to resist the scoring mode of failure. Many com-
ments could be made regarding some of the data presented in
the paper, but it seems appropriate to reserve these comments to
those pertinent to the conclusions. The first conclusion states that
“gear scoring occurs under fully boundary-lubricated conditions
and not even close to the breakdown of EHD film.” This discus-
sion has never been convinced that Christensen’s breakdown theo-
ry is pertinent to the geometry and action of contacting gear
teeth. His experiments were done for bearings of a conformal-type
geometrical contact, not counterformal contact as is experienced
in gears. In the authors’ second conclusion they state that “iso-
thermal calculations of minimum EHD film thickness are in error
both in magnitude and in trend with failure velocity.” It is true
that for extremely high-speed gears, such as those produced by
the discussers company, Cheng’s shear inlet theory accounts for
a theoretical reduction in thermal film thicknesses from the iso-
thermal calculation by as much as 50 per cent. However, the ex-
amples used in Fig. 1 of the authors’ paper are not realistic ex-
amples for operating machinery. The samples used were for a test rig
where it was the expressed purpose to bring about the scoring
mode of failure. The load and velocity relationships were not like
any real transportation or industrial machine which might experience.
This discussor has never been able to find a real live case where
the increase in pinion speed can cause a decrease in film thick-
ness to the extent shown above 24,000 rpm in Fig. 1. The most se-
vere case that this discussor has been able to find is that increasing
the speed without reducing the viscosity has nearly a negligi-
ble effect in increasing the film thickness. It would be informative
if the authors could cite a practical case verifying the drastic re-
duction shown above 24,000 rpm in Fig. 1.

In the authors’ conclusions they state: “pitch point EHD film
thickness does not reliably represent the minimum EHD film
thickness, and therefore cannot be substituted for it.” The dis-
cussor would have to agree that this is correct when small gears
are used, and particularly if the pinion has a small number of
teeth, say below 25, or in some cases even below 35 teeth. How-
ever, it should be pointed out in large machinery that the invo-
lute profile is almost flat like a rack and that the roll angle is
only two or three degrees throughout the profile action, and in
this case pitch point analysis is satisfactory. However, in aircraft
and missile-type gears, more complicated formulas depicting the
most severe point of flash temperature on the line of action need
be investigated. The discussor will have to agree with the fourth
conclusion regarding Hertz contact stress. In closing the dis-
cussion, the discussor will have to point out that Figs. 1 and 3
cannot be compared directly, as was attempted in the paper, in
that the data from different test machines that have substan-
tially different operating characteristics. Also, the discussor was a
little confused in the authors’ use of T for temperature. Is this T
blank temperature or oil inlet? This is important in that tooth
blank or bulk temperature should be used in EHD calculations.

In closing it should be pointed out that gear scoring is far more
complicated than can be represented by a simple model. The analysis
of gear scoring probability is more fully discussed in a paper by
this discussor in Journal of Engineering for Industry, ASME Trans-

P. M. Ku

The authors have performed a noble service in dispelling some
misconceptions about gear scoring. I agree with their views in a
general way. However, inasmuch as their paper deals with a rather
fundamental issue, it is believed that the following remarks may
put the matter in proper perspective and help to strengthen their
conclusions.

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P. E. Fowles

There are two general topics in this paper that require discussion and clarification. The first concerns the so-called elastrohydrodynamic criterion for the scoring of gears. The authors define this criterion in terms of a belief in the ball and roller bearing industry that specific film thicknesses (A) of 3, 4, or even 10 are required for safe design and reliable operation, stating that this same criterion is also assumed to apply to gear scoring. This discussers question this definition. First, the specific film thicknesses quoted seem rather high even for the bearing industry. Most bearings operate at specific film thicknesses well below this value, and relative life versus A curves, such as those presented in [12], usually show that the L₁₀ life has risen to two or three times the book value at A = 2 or thereabouts.

Second, and more important, failure in the bearing industry is predominantly by a rolling-contact fatigue mechanism, while this criterion is not applicable to scoring in gears. In the former, the role of the lubricant is primarily the reduction of stress concentrations by minimizing asperity interactions, and the chemical effects of importance are those that govern the rate of fatigue crack initiation or propagation within the steel. In the latter, the lubricant not only minimizes sliding asperity interactions by separating the surfaces but must also maintain a boundary film to prevent metallic contact between asperities that do interact. As a result, the chemistry and composition of the oil are important factors governing its ability to prevent scoring.

Because of these considerations, the present controversy between those who favor a thermal scoring mechanism and those who favor a mechanism based on specific film thickness is not being waged on the basis of the criterion defined by the authors. Nor does this discussers believe that gears are being designed by this criterion. Instead, the criterion generally being considered is that scoring becomes imminent when the specific film thickness is reduced below some value that is nominally constant for a given lubricant and metallurgy, but that will vary among different lubricants and metallurgies. By its emphasis on gears, this criterion implicitly requires the presence of sliding and is therefore not at odds with observations made with disc machines that scoring becomes difficult to initiate, i.e., occurs at dramatically reduced specific film thicknesses, as the sliding speed is reduced to zero.

The main point to be understood about the criterion described above is that the critical value of specific film thickness is lubricant-dependent. For straight mineral oils the critical value is almost constant among the oils, and a value in the range 1.5–2.5 would be reasonable. However, oils containing EP additives may allow satisfactory operation at much lower specific film thicknesses, often well below unity. This variation of the critical specific film thickness does not present a design problem if reasonable values can be identified for each class of lubricant.

A parenthetical comment seems appropriate here concerning the concept of boundary lubrication, which the authors have defined as anything non-EHL; that is, when the specific film thickness is less than about 4. On the basis of Tallians data [13] on the fraction of time there is metal-to-metal contact as a function of specific film thickness, the following three regions are generally defined: λ > 4: full EHL, with no penetration of the EHL film by surface asperities; 4 > λ > 1: partial EHL, in which the fraction of time there is asperity penetration of the film rises from essentially zero at λ = 4 to essentially unity at λ = 1; λ < 1: boundary, in which there is penetration all the time.

While these regions are confirmed theoretically by Johnson et al. [14], their work also shows that the consideration of asperity contact alone (which after all may be only between a single pair of asperities) provides only a limited picture of the situation. Thus, when the fraction of the total pressure at any point within the contact that results from asperity interactions is calculated for fairly typical steel surfaces with rms roughnesses of 10 μm, it amounts to only about 10 percent and 2 percent at total pressure levels of 200,000 psi and 50,000 psi, respectively, when λ = 1, and still only about 40 percent and 10 percent, respectively, when λ = 0.1. Even at λ = 0.1, therefore, the overall contribution of the asperity contacts to the load capacity of the contact as a whole is somewhat less than 40 percent under reasonable operating conditions. These considerations must be borne in mind when describing low specific film thickness operation as boundary, for although the lubrication between colliding asperities themselves may approach pure boundary conditions, the overall situation is very different from those contacts conventionally termed boundary in which EHL pressure generation is absent or negligible.

The question of scoring mechanisms, thermal or specific film thickness, is an important one. This discussers, having favored the latter, is beginning to feel that elements of both are involved. Any work that can shed light on the issue is therefore most welcome, and notwithstanding differences in the assumed EHL criterion, the numerical results of this paper might have accomplished this. Unfortunately, as a result of the considerations outlined below, which form the second general topic of discussion, the results in fact have little to offer.

Among the parameters that must be specified to calculate the specific film thickness in any EHL contact are the lubricant viscosity in the entrance region of the contact and the roughnesses of the two surfaces. It has been recognized for many years that since most of the lubricant in the entrance region is derived from fluid adhering to the upstream surfaces the temperature of this lubricant, and therefore the temperature at which the lubricant viscosity is to be taken, must be that of the surfaces entering the contact. In the case of gears, this is the temperature of the gear teeth during operation, which will usually differ considerably from that of the supply oil, especially at the higher speeds. Since viscosity is so heavily dependent on temperature, the authors use of the lubricant viscosity at the supply temperature necessarily leads to erroneous results.

To convert from film thickness to specific film thickness the authors have employed a single surface roughness for all the gears used by Borsoff, and another single value—this the mean of a quoted range—for those used by Ku and Baber. This again leads to erroneous results on two counts. First, the surface roughness of gears made to the same specification varies considerably, so that the composite surface roughness for any given pair of gears will also vary. This fact is implicit in the roughness range quoted by Ku and Baber, which allows a variation in the composite roughness by a factor of nearly two. Second, and much more important, the authors have completely neglected the effect of running-in on surface roughness. The surface roughnesses pertinent to any examination of the EHL scoring criterion are those equilibrium values that are attained after running-in and before scoring occurs. These may differ greatly from the original roughnesses of the unused gears, depending on the speed, load, and duration of the early operation and the activity of the lubricant. Thus on the basis of both temperature and surface roughness considerations, the authors numerical results include systematic errors that in addition to changing the absolute values of the data will very probably also change the trends, thereby severely reducing their usefulness in judging any EHL scoring criterion.
Additional References