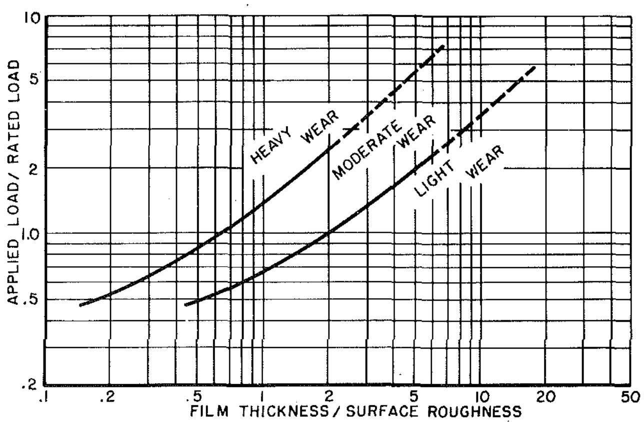

Fig. 10 Relation of gear performance to lube film thickness, surface roughness, and load

tant in predicting the onset of tooth surface distress. An exploratory study, Fig. 10, shows the treatment of a small amount of data that indicates this parameter might also be important in determining rating modifications or might determine a surface wear severity factor for gear teeth. This data is not yet fully developed.

Closure

This paper has shown how the EHD film thickness theory has been applied to the assessment of oil film thickness of industrial gearing. It is also shown that film thickness along with surface texture information and gear velocity can be used to assess the statistical probability of tooth surface distress.

From the studies of this paper it can be concluded that:

1 Gears operating in the full EHD regime, λ greater than two, do run virtually without surface distress.

2 Many industrial gear drives operate at λ values less than 0.7 and it is apparent that mixed or boundary lubrication is sufficient for many applications.

3 The probability of distress seems to be influenced by the lubricant type. As an illustration, it appears that it might be possible to operate with thinner ester type lube films than with petroleum oil films.

4 The reliability of the surface texture number could be improved by the incorporation of a modifier based on surface waviness.

5 It is apparent that further fundamental development is required before the effects of gear lubrication phenomena on gear performance can be accurately predicted.

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DISCUSSION

W. J. Bartz.[2]

In order to apply EHD oil film theory to gears using any film thickness equation, the temperature governing the effective viscosity within the gap between the meshing gears has to be known.

Using measured surface temperatures [18], we calculated the film thickness and compared these data with the onset of high wear and scuffing [19].

In studying the efficiency of the Dowson/Higginson equation for gear lubrication, the calculated film thickness depending on load of a given pair of gear wheels using this equation was compared with measured wear rate depending on load. The idea was to compare the calculated film thickness at the pitch point between two meshing gear wheels depending on load with the wear pattern measured during a test using the same gears and to find out any correlation between the calculated and the measured result. The background of this idea was the assumption according to which neither extreme wear nor scuffing and scoring would occur as long as an oil film of sufficient thickness would be available.

The experimental results have been achieved using the so-called FZGar Tester standardized in Germany and being standardized by the Coordinating European Council (CEC). For these tests, the toothing *A* with a mean surface roughness of the tooth surfaces about 0.2 μm before the test runs has been selected. The oil sump temperature has been maintained at 90°C. 30°C the circumferential speed of the gear wheels was 8.3 m/s, 16.6 m/s. Table 4 contains the description

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Table 4

Oil	Description	Viscosity at 50 °C, m^2/2	Density at 20 °C, kg/m^3
1	Mineral oil, mixture	$395{,}0 \cdot 10^{-4}$	$0{,}897 \cdot 10^3$
2	Mineral oil, white oil	$38{,}0 \cdot 10^{-4}$	$0{,}888 \cdot 10^3$
3	Synthetic oil, poly-alkyl-glycol-ether	$12{,}6 \cdot 10^{-4}$	$0{,}946 \cdot 10^3$
4	Synthetic oil, poly-alkyl-glycol-ether	$260{,}6 \cdot 10^{-4}$	$0{,}979 \cdot 10^3$
5	Mineral oil, white oil	$11{,}2 \cdot 10^{-4}$	$0{,}842 \cdot 10^3$
6	Mineral oil	$39{,}0 \cdot 10^{-4}$	$0{,}881 \cdot 10^3$
7	Mineral oil	$240{,}0 \cdot 10^{-4}$	$1{,}009 \cdot 10^3$
8	Synthetic oil, di-2-ethylhexyl-sebacinate	$8{,}2 \cdot 10^{-4}$	$0{,}912 \cdot 10^3$
9	Synthetic oil, mixture of chlorinated aromatic hydrocarbons and aromatic phosphate-esters	$32{,}7 \cdot 10^{-4}$	$0{,}368 \cdot 10^3$
10	Synthetic oil, silicone oil	$43{,}3 \cdot 10^{-4}$	$0{,}869 \cdot 10^3$

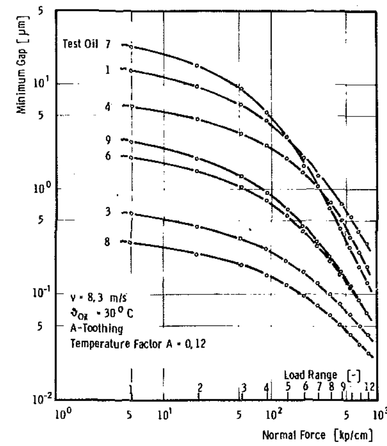

Fig. 11

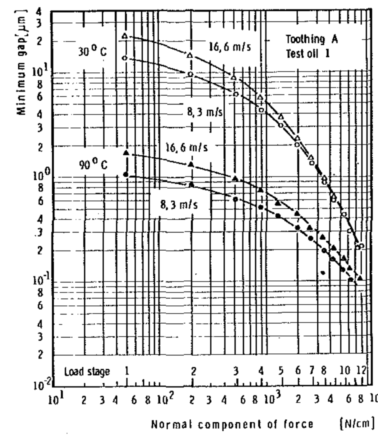

Fig. 13

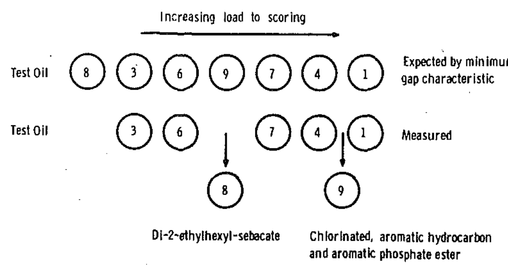

Fig. 12

of the test oils, which did not contain any extreme pressure additives. Viscosity temperature and viscosity pressure behavior have been determined too.

Fig. 11 shows the calculated film thickness using temperature, speed, and temperature pressure data over load.

Clearly, it can be recognized how the different viscosity temperature and the different viscosity pressure behavior of the test oils influence the film thickness at the pitch point. On the basis of this film thickness/load relationship, a specific order of the test oils can be deduced according to which a certain minimum gap depending on load will be reached. This could be equal with the classification of the test oils without any extreme pressure additives, concerning their load-carrying performance, that means their wear and scuffing behavior. This expected order of the test oils, evaluating the minimum gap/load dependencies, is shown in the upper line of Fig. 12. This order should be compared with the actual order of performance (lower line in Fig. 12) determined experimentally for the same conditions used for the calculation. Clearly, it can be recognized that with the exception of the two oils 8 and 9, there seems to be a rather good correlation between measured and calculated load-carrying capacity of these oils. Using the Four Ball Tester only with oil 9, some extreme pressure effect can be established to explain its behavior during the gear test runs.

Analyzing the results of Fig. 11 leads to the conclusion according to which, with the exception of the same two oils 8 and 9, scuffing and high wear occurred for calculated film thickness values between 0.2 and 0.3 μm, independent of the surface roughness of the gear wheels after the test runs. Comparing the scuffing load stage, the calculated film thickness, and the total of mean surface roughness determined in these tests, it could be recognized clearly that there did not exist any correlation between calculated film thickness and the total of the mean surface roughness. Completely unexpected much lower (calculated) film thickness values could be allowed before high wear and scuffing started than would have been expected regarding the surface roughness. On the contrary, often it will be assumed that no remarkable wear or scuffing and scoring will occur as long as the film thickness will be maintained in the order of magnitude of the total of surface roughness.

It would be expected that all measures that will result in higher effective viscosities and thicker oil film between the meshing gear wheels should increase load-carrying performance defined previously. This would be valid for lower oil sump temperatures as well as for higher circumferential speeds. Fig. 13 shows some results achieved with test oil 1. As expected, the lower temperature and the higher speed resulted in thicker oil films at the pitch point.

The film thickness increasing influence decreases with increasing loads according to the simultaneously occurring viscosity decreasing influence of temperature rise. From the minimum gap/load relationship of Fig. 13, the following order for changing the wear rate from low level to high level or for the onset of scuffing and scoring could be deduced:

1 *temperature* 90°C, *speed* 8.3 m/s
2 temperature 90°C, speed 16.6 m/s
3 temperature 30°C, speed 8.3 and 16.6 m/s; no difference.

Fig. 14 shows the wear pattern depending on load actually measured during gear test runs with oil 1 applying the combinations of operation condition mentioned before. As expected at the higher oil temperature, the higher speed resulted in higher load carrying performance. That means higher loads before scuffing occurred, whereas at the lower oil temperature no influence of speed could be found. This means a very good correlation between the conclusions on the basis of the elastohydrodynamic theories and the wear behavior actually measured. But again, no strong relationship between surface roughness, minimum film thickness, and the onset of high wear was established.

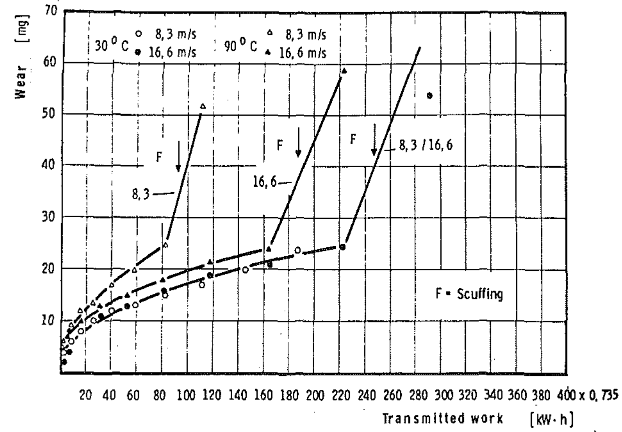

Fig. 14

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R. Errichello[3]

The authors are to be congratulated for their valuable contribution of a large amount of experimental data which will provide a much-needed design guide. It is hoped that the authors will continue their experiments, especially with surface-hardened gearing. The nomograph will be appreciated by gear designers who often need a rapid assessment of film thickness. An important result of the authors' work is the quantifying of the influence of pitch line velocity on the probability of surface distress.

As indicated by the authors, several different schemes have been proposed for defining the composite surface roughness; however, in this discusser's opinion, the authors' adoption of the average value is less desirable than the "square root of the square of the RMS values." It is true that any definition of composite roughness can be used with equal success as long as the designer who uses the data is aware of the particular definition. However, the square root method has seen greater usage in the literature, and in the interest of achieving better conformity and correlation among various experimental data, it is suggested that the square root method is a better scheme.

Regarding the authors' definition of surface distress, it is not clear whether the definition includes classical contact fatigue as opposed to scoring or wear. Perhaps the authors would clarify whether the tests were of short enough duration to rule out the possibility of cyclic contact fatigue.

L. S. Akin[4]

Mr. Wellauer has written this paper on a subject of considerable interest to the discusser and provides some new data that may be useful to gear designers if we can get a better understanding of how the data were obtained. So the discusser will ask a few questions and make some pertinent comments that he hopes will provide some of the needed clarification.

First of all, the author in his paragraph on composite surface texture proposes equation (10) identical to the one asserted by this discusser [20] several years ago. I now disagree with this formulation and agree with Tallian [21] that we should use the RSS (λ_t) method proposed by him which is generally accepted by practitioners working with EHD phenomena. It is very confusing in the literature if we all use the term "specific film thickness" unless we all have the same definition for it. Tallian's definition has statistical roots and is directly relatable to the correlation coefficient and power spectral density functions which have recently been shown [22] to be significant in characterizing individual surfaces.

It should further be pointed out that characterizing individual surfaces is not enough in that we are really interested in surface asperital interaction in concentrated and lubricated contacts (simulating a gear mesh). Thus (λ) ratio alone is not really sufficient, but we also need the average asperital radius of curvature and peak or summit density for a contact interaction analysis to be made.

This leads to my first question. Does the author believe that the isothermal solution is sufficient for use in determining the specific film thickness for "high speed industrial gears" (20,000–40,000 fpm)? This discusser uses Cheng's thermal solution [23] taking shear inlet heating into account in this speed range. Further, how does the author assure himself that the gear is being oil cooled so that starvation [24] or flooding does not exist?

The discusser agrees with the author that if the EHD film is thick enough ($\lambda > 3$ using Tallian's λ_t), scoring is a very remote possibility; however, when $\lambda_t < 1.5$, surface contact temperature becomes very important in the scoring mode of failure and $\lambda_t > 3$ does not necessarily preclude pitting if the surface compressive stress is sufficiently high. Also, when $\lambda_t < 1.0$ (the author's 0.7), the operation will be in keeping with one or more of the known wear regimes [25]. Perhaps this is a good time to clarify just how λ_t ratio should be used in gear design. It is a good indication of operating lubrication regime. For example, $\lambda_t > 3$ indicates that no scoring or any other kind of wear can take place—pitting is still a possibility if the surface stress and stress cycles are high enough. When $\lambda_t < 1.0$, some form of wear can take place but the wear regime cannot be determined by λ_t alone. Thus, no generalized wear formula such as the author's equation (12) can be relied upon in all applications. Further, the use of λ_t ratio to determine the wear coefficients (K) such as in the author's Table 3 is absurd in that K is sensitive to some 50 parameters [26] of which λ_t is far from the most influential. Thus, since the problem becomes so complex for $\lambda_t < 1.0$, how can the author justify the calculation of his λ ratios as low as 0.13 and even 0.011 provided in Table 1 of the paper? The EHD action would be negligible with the asperities carrying most of the load under boundary lubrication conditions. The author's words of explanation will be appreciated.

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Authors' Closure

The comments of the discussers are very much appreciated.

The authors are pleased to see that the detailed test work reported by Dr. Bartz is in general agreement with their findings—namely, that parameters that increase the calculated minimum oil film thickness tend to reduce the degree of surface distress.

Dr. Bartz's conclusion that no strong relationship exists among surface roughness, minimum oil film thickness, and the onset of high wear does not seem justified in view of the limited range of surface roughnesses reported on in his work. The authors' data does, however, suggest this possibility as noted by the intermingling of distressed and nondistressed data points in Fig. 9, particularly for specific film thickness values of less than 1.0. This intermingling of data was a major reason for treating it in a statistical manner.

With reference to Mr. Errichello's request for further explanation of the definition of the term "distress," it is to be noted that virtually all tests and field cases reported experienced lives in excess of 10 million cycles, and thus were well into the range where cyclic fatigue phenomena can occur.

Gears reported on were operated at "service loadings," and thus the contact loadings were at or below AGMA durability ratings; wear, rather than progressive pitting was the primary mode of distress. Messrs. Errichello and Akin suggested the use of the square root of the sum of the squares of the surface roughness of the conjunction surfaces as the surface texture characteristic number, rather than the

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simple averaging of the two roughness values. Dr. Akin and the authors both have referred to the complexities that go beyond the roughness attribute alone in arriving at a fully adequate conjunction surface texture characterization number. Recognizing the current state of the technology, it was the judgment of the authors that for a simplified design guide, sophistication beyond the simple arithmetical average is not warranted at this time.

The authors' high-speed (over 5000 fpm PLV) gear data were analyzed using both thermal and isothermal solutions for film thickness. While specific film values varied, no significant differences in the order of the data or in the distress probability curves were noted; thus the isothermal solution procedure was adopted for the entire range of gear speeds shown for the sake of simplicity.

The matter of flooding or starvation of the high-speed gears was not examined per se. Lubrication for the cases shown was deemed adequate based upon a broad background of successful experience with such gearing.

Dr. Akins' point that specific film thickness, λ, is a good indication of wear regime is quite true, as are his remarks concerning pitting at large λ values if high loadings prevail; so are his remarks concerning the complexities of wear prediction at low values of λ.

However, so that the important significance of the data and curves of Fig. 9 is not missed, the authors wish to emphasize that these data as portrayed represent actual long-life gear performance at loadings at or within AGMA allowable values at the λ values (or regimes) indicated, and thus summarize past experience and offer a gear designer or applier a guide for expected performance for other gears operating under similar conditions.