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DISCUSSION

C. H. Danner[2]

This paper greatly increases our understanding of lubrication effect on fatigue life for roller bearings, particularly for lubricants with E.P. and antiwear additives.

In fatigue tests in mineral oils with only rust and oxidation inhibitors (R & O) conducted on tapered roller bearings for similar viscosity levels, as in Tests B1, B2, C2, and C4, the discusser has also found increasing fatigue life with increasing viscosity.[3] Based on our results, it is surprising that more significant differences were not found in the author's viscosity comparisons.

Comparison of Tests A1 and A2 indicate a very high surface finish effect on fatigue life. We feel caution should be used in extrapolating this result to other situations since the tests were conducted in Mil-L-7808 with E.P. and antiwear characteristics. It has been our experience with mineral oils that *type* of finish as well as relative magnitude of finish for mating components may override the composite surface roughness in determining fatigue life.

As reported in the paper by the discusser[3], we have found that with tapered roller bearings in mineral oil (plus R & O) there is no drastic life reduction such as given by the author in Fig. 8. We feel this difference in the two results is explained by the E.P. and antiwear additives used in the author's tests. It should be noted that using only the author's data for mineral oil (plus R & O and defoament), no drastic life reduction is indicated. For these reasons the lubricant life equation given should probably be limited to lubricants with E.P. and antiwear additives.

The discusser feels that it is premature to propose a lubricant fatigue life factor which would hold for any lubricant and any bearing design. Certainly the author's results, as well as those reported,[3] caution against an oversimplified view of the influence of lubrication on fatigue life.

[2] The Timken Roller Bearing Co., Canton, Ohio.
[3] Danner, C. H., "Fatigue Life of Tapered Roller Bearings Under Minimal Lubricant Films," ASLE Preprint 69LC-11.

J. W. Kannel[4]

Mr. Skurka's paper demonstrates that elastohydrodynamics plays a significant role in the life of rolling-element bearings. In this regard, the fact that he found some consistency in the relationship between predicted EHD films and bearing performance, from a large matrix of data with two types of bearings, is encouraging and hopefully will lead to further studies and evaluations. However, I do not believe that it is legitimate to extrapolate as heavily from these data as does the author.

The fact that Fig. 8 shows a fairly sharp transition between L_f values of 0.3 to values of 3.2 for $h/F_c = 1.5$ is largely due to the fact that most of his data were for conditions around one value of h/F_c. Further, most of the justification for the flat slope at the high values of h/F_c is based on two points around $h/F_c = 5$. However, while the largest percentage of his data points were obtained using SAE 30 oils, these two points were not. If these points were in error due, say, to errors in Grubin's equation or in the assumed values for α for his lubricants, then, the apparent flat slope at high values of h/F_c could well be nonexistent.

Grubin's equation, which was developed for well-behaved hypothetical Newtonian lubricants and rolling elements with very smooth surfaces, is becoming a widely accepted method for predicting film thickness for any type of lubricant and any type of surface. While some experiments such as those quoted by the author appear to substantiate this theory, others do not. X-ray film thickness data obtained at Battelle, for example [14, 15, 16],[5] have shown the lubricant films to be more sensitive to loading and less sensitive to α than predicted by Grubin's equation. Our current feeling is that lubricant parameters, in addition μ_0 and α, such as time delay effects on the pressure coefficient of viscosity, α, must be incorporated into film-thickness theories in order for them to be valid for high-speed rolling contacts. Also, experimental and theoretical studies conducted at Battelle, with rolling-contact surfaces with varying levels of roughness have indicated that, for many cases, the minimum separation between the asperities on the mating surfaces is not strongly affected by surface roughness. That is, the asperities do not necessarily puncture the lubricant film as is sometimes assumed, but rather they enhance the elastohydrodynamic process to the extent that they can literally ride on top of the film. It is quite possible that the shapes of the asperities influence the extent of this secondary form of lubrication. It is highly desirable, then, for fatigue test, which presumes to show the effect of varying levels of EHD films on fatigue, to be based on some type of measurement of the extent of the lubricant film, such as by the use of electrical continuity rather than simply estimated by film thickness theories.

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[4] Battelle Memorial Institute, Columbus Laboratories, Columbus, Ohio.
[5] Numbers in brackets designate Additional References at end of discussion.

T. E. Tallian[6]

This paper appears at a very opportune time. At the same session, a paper by C. H. Danner, entitled "Fatigue Life of Tapered Roller Bearings Under Minimal Lubricant Films" was also being presented. Both cover the subject of roller bearing lubrication and life. Considered together, they represent a very interesting contribution to the experimental evidence linking elastohydrodynamic lubrication conditions to bearing life.

In commenting on the present paper, it will be assumed that the roller bearings tested were made of carburizing grade steels. The author does not appear to so state explicitly, but in view of the type of bearing discussed, and the author's association with Federal-Mogul, it seems to be a reasonable assumption. On this basis, then, the author's results and those of C. H. Danner, should be directly comparable. Also, this discusser has, with coauthors, submitted a discussion for the Danner paper, and much of what has been said there equally applies to the present paper, such as the risk incurred by anyone basing conclusions on calculated film thickness only, as well as the problem of run-in of relatively rough surfaces during load cycling, which leaves the h/σ (in the author's notation h/Fc) film thickness to roughness ratio somewhat ill defined.

The present author has undertaken a very ambitious program with four variables, viz., surface finish, lubricant, temperature, and rotational speed. There is also the variable of bearing sizes, and heats of material used to make all the sizes tested. Clearly, a full separation of all variables is not possible. Nonetheless, much interesting information has been obtained.

The good correlation between a life improvement factor and film thickness ratio, shown in Fig. 3, is very impressive. This discussor is apprehensive about the justification of fitting as complicated a functional form to the data as is given in (Equation (5)) since it is questionable whether the numerous constants involved can be determined to sufficient levels of significance. However, it is manifest from inspection of the plot that the curve fits well, and one should, of course, not argue with success.

The present author's data and those of the paper by Danner are in some ways complementary in that the latter paper covers in more detail the regime of very low h/σ values, whereas the present author has most of his test points located at $h/\sigma = 1$ and above. Nonetheless, there is enough overlap to notice that the two papers seem to be at variance with each other. A much sharper drop in life with reduced h/σ is observed by the present author than has been found by Danner.

In our discussion given to the paper by Danner, we showed some ball bearing data using carburized steel rings, which yield only a very modest life decrease for a drastic reduction of h/σ values. Taking these data as one extreme, and the ones of the present paper as the other extreme, considerable differences are apparent.

This discusser does not believe that it is possible at the present time to be certain which of the relationships between h/σ and life found by different authors is more general, or whether any can claim generality. Observation of the behavior of rolling contacts in the partial elastodyhrodynamic regime offers convincing evidence that there are many factors other than h/σ influencing life. It is felt, that while for truly high h/σ values (as determined from measurement rather than from calculation) lives will be consistently good, lives for low h/σ will vary as a result of a multitude of factors such as carburized versus through hardened bearing material, details of the surface roughness, the chemistry of the lubricant and probably others which are at the present time insufficiently understood. The only way to gain more understanding of these complex matters is by tests such as those published in the present paper, and they are therefore most welcome.

[6] Vice-President, Research, SKF Industries, Inc., Engineering and Research Center, King of Prussia, Pa.

D. P. Townsend[7] and E. V. Zaretsky[8]

The author should be congratulated for undertaking an ambitious research project and for the first time experimentally confirming a phenomenon speculated to exist by many. SKF Industries, Inc. first showed a curve of similar shape (Fig. 9) [17] to that of the author's. However, the SKF curve showed the amount of surface separation (percent film) as a function of film parameter (Λ) where $\Lambda = \frac{h}{Fc}$, using the author's symbolism. Tallian [18] presents a curve of life versus Λ. This curve, shown in Fig. 10, is primarily based upon field experience and some laboratory. These data show an exponential relation of life with Λ which may not be realistic. Tallian, in essence, states in [18] that at Λ values above 4, the life range is less well explored than in the lower Λ values. It is the discussers' opinion that the curve of the author showing a finite rather than an infinite life trend is more realistic. Hence maximum life potential can probably be achieved at a $\Lambda \geq 4$. However, where does all this data leave the user and designer of rolling-element bearings?

Using the data of Figs. 9 and 10, curves of life potential expressed as a percent against percent film can be plotted as shown in Fig. 11. The curves can be extrapolated to give a linear rela-

[7] NASA-Lewis Research Center, Cleveland, Ohio.

[8] NASA-Lewis Research Center, Cleveland, Ohio. Mem. ASME.

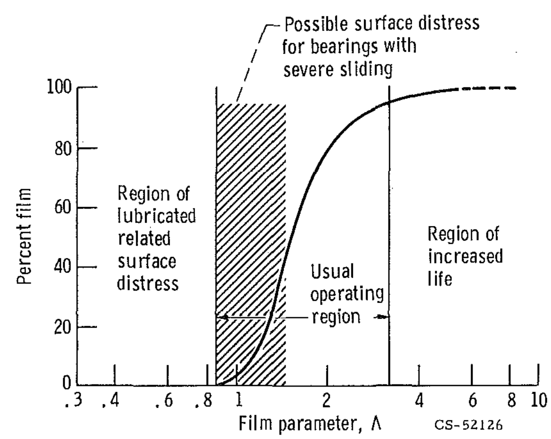

Fig. 9 Percent film as a function of film parameter

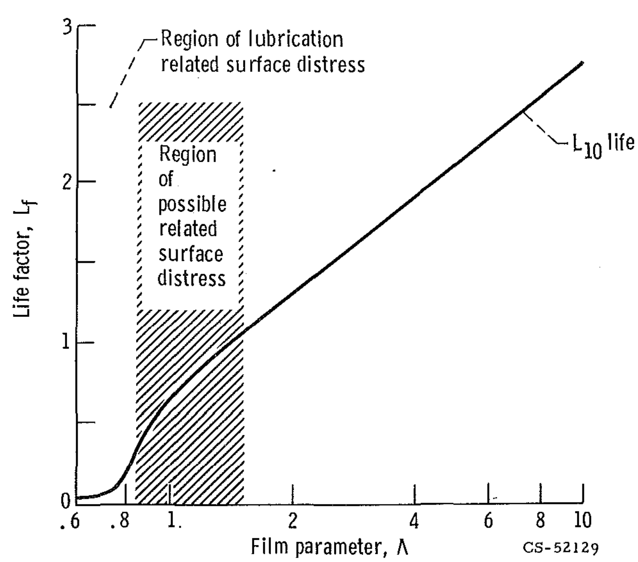

Fig. 10 Relative life as function of film parameter Λ [18]

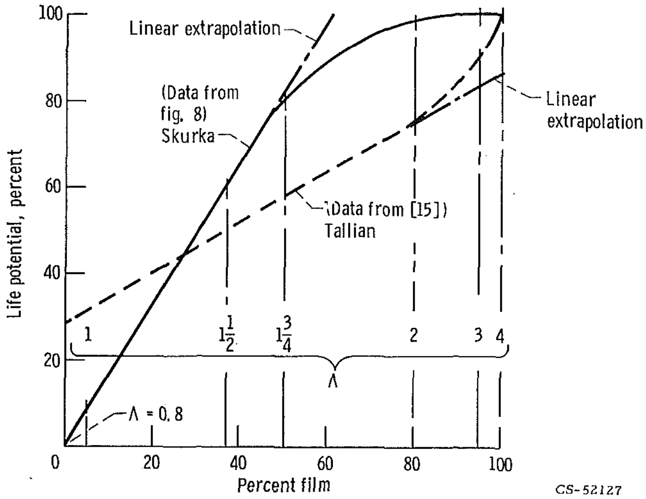

Fig. 11 Life potential based on a fatigue criteria as a function of surface separation

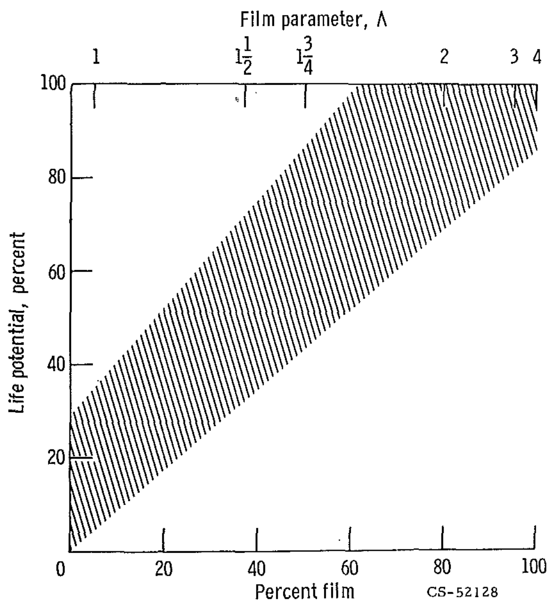

Fig. 12 Range of life potential as a function of surface separation based on data from Fig. 11

tion of life with percent film. From a practical viewpoint, these curves then define a range within which life can vary with EHD film thickness. This range is shown in Fig. 12.

Using Fig. 12, the engineer can determine a range within which a certain life potential can be achieved based upon EHD film thickness. It must be remembered, however, that no specific number can realistically be placed upon this life potential without extensive testing. AFBMA life is a rather conservative rating based upon technology which is over 25 years old. Steel melting techniques, material batch control, material chemistry, heat treatment, and material cleanliness are among some of the variables that affect rolling-element bearing life. Hence, the author's equation 5, which may be applicable to the bearings from which the author's data were obtained, cannot be generalized to other bearings made from other materials and manufacturing techniques. Along this line, could the author indicate which groups of bearings were tapered or straight roller, the bearing steel used, the melt technique, the heat treatment, and the steel inclusion content?

One of the most important aspects of fatigue testing is proper control of the test variables. This is the weak link in many of the conclusions derived by many investigators. For the author's data only tests B-1, B-2, C-1, C-2, and C-4 can be classified as tests with controlled variables. Here, one heat of the same material and three different viscosity grades of the same lubricant were used. Other variables such as material and lubricant type were eliminated. Hence the range of variables are somewhat reduced from those listed by the author.

The discussors believe that the test data for groups A-1 and A-2, showing the effect of surface finish, are good data when compared to each other. However, the results cannot be easily compared to the other data because of too many other variables.

Test group C-2 with a Λ of 1.5 shows a ten-percent bearing life of 31×10^6 inner-race revolutions, while test group B-1 with a Λ of 1.45 shows a fatigue life of 13×10^6 cycles. The test group B-1 has twice the bore diameter and should operate with a much higher Λ value. Therefore test group B-1 should give a better life than test group C-2 since all other conditions were nearly equal. Could the author comment upon this?

Based upon experience [17], the type of failure obtained in the bearing operating range of Λ less than 1 should be from gross surface distress. At a Λ value between 1 and $1^1/_2$, failure should be from superficial surface pitting which propagates into a spall. At Λ values greater than $1^1/_2$, failure should be from subsurface classical rolling-element fatigue. Could the author comment upon his observations regarding the failure modes in the bearings for the tests reported?

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Author's Closure

The author wishes to thank the discussers for their valuable comment and welcomes the opportunity to answer the questions raised.

The risk incurred by using calculated versus measured values of film thickness based on Grubin's theory, such as suggested by both J. W. Kannel and T. E. Tallian, is considered to be small in view of the supporting experimental data that have been published. The author's primary purpose was to augment the considerable experimental data on lubricant film thickness with bearing fatigue test data under controlled test conditions. To this end, the author feels the risk and limitations of using calculated versus measured lubricant film values are justified.

The run-in of test bearing surfaces mentioned by T. E. Tallian did produce changes in roughness, particularly for $h/F_c < 0.9$. However, based on measurements after short duration of operation, the change in roughness was not considered appreciable being less than 2 rms in all cases.

The cylindrical and tapered roller bearing test data are covered in Tables 3, 4, and 7 and 5 and 6, respectively. All bearings tested were manufactured in accordance with ASTM A 534-65 from carburizing grades of vacuum degassed steel.

To properly evaluate and compare the author's results to those of C. H. Danner would, in the author's opinion, require equating the size as well as mode of fatigue that occurred. For the results presented, the modes of fatigue that predominated and the corresponding range of h/F_c are shown in Figs. 13, 14, and 15. These results correlate well with available field experience, especially for the low speed, light oil (SAE 10W) applications where $h/F_c < 0.9$.

The difference in L_{10} lives between tests B1 and C2 is primarily due to the slope of the Weibull plot. The C2 test bearings were unusually uniform in dimensional characteristics which may account for the steeper Weibull slope.

The purpose of the author and his associates was, first, to provide experimental bearing endurance test data under controlled lubricant conditions and, second, to relate this data to the controlled variables to provide the designer and user with a method (curve Fig. 8 or equation (5)) to improve reliability through better estimates of actual field performance. We are pleased to see the data used by Messrs. Townsend and Zaretsky in their discussion and hope to provide additional test data in the future.

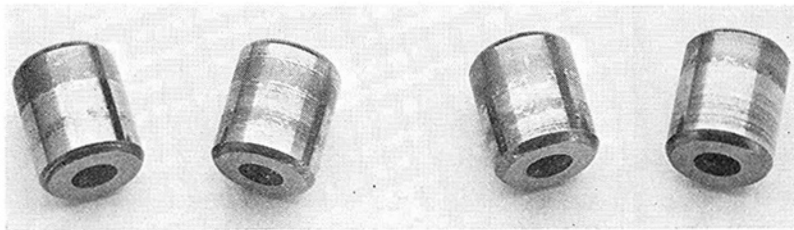

Fig. 13 Roller body frosting and smearing. $h/F_c \leqq 0.9$

Fig. 14 Roller path frosting and smearing with superficial spalling. $h/F_c < 1.25$

Fig. 15 Conventional subsurface fatigue. $h/F_c \geqq 1.75$ (spall is shaded for emphasis only)