Part 2

E. V. Zaretsky. The discusser would like to reiterate his comments made to Part I of the authors' paper. In addition, there is a wealth of controlled laboratory data which the authors' organization and others have generated together with rolling bearing fatigue data reported in the open technical literature. Accordingly, this discusser questions the authors relying on uncontrolled and questionably defined field data to attempt to substantiate their theory when well-defined laboratory data are available.

A question has been lingering for at least two decades as to whether a material life adjustment factor A_2 can be used in the low film regime. The result of the authors' data, if it is correct, would suggest that the factor A_2 cannot be used where lambda is less than one. Under the conditions the authors' bearings were operating, the mode of failure would be expected to be superficial surface pitting and/or wear.

The Lundberg-Palmgren analysis is based on subsurface pitting. Using an A_2 of 44 for VIM-VAR M-50 (Bamberger et al., 1976) and an A_3 of 0.3, the A_{23} factor would be 13.2. The A_2 factor for CVM AISI 52100 would be six. The A_{23} factor would be 1.8. Hence, it would be expected that the VIM-VAR M-50 would be approximately seven times that of CVM 52100. However, if the failure were surface originated, then the lives of both materials would be equal and approximately 30 percent of Lundberg-Palmgren's predictions. This appears to be the case. Homogeneity and cleanliness would be important only for subsurface fatigue.

An apparent conflict exists between the values of the maximum Hertz stresses, P_0, reported in Table 4 and those values reported in Figs. 2(*a*) and (*b*) and possibly Fig. 3. If the values of Table 4 are correct, then it would be expected that the predicted life of the high contact angle (HCA) bearings would be substantially higher than the low contact angle (LCA) bearings, as is shown in Table 3 and discussed by the authors. However, if the values are as shown in Figs. 2(*a*) and (*b*), then the results would be expected to be similar to the actual lives reported in Table 3. This can be illustrated as follows:

The life ratio (LR) based upon stress only is

$$LR=\frac{HCA}{LCA}=\left[\frac{P_{0LCA}}{P_{0HCA}}\right]^9=\left[\frac{1.37\ \text{GPa}}{1.61\ \text{GPa}}\right]^9=0.23$$

The life ratio based upon the material factor A_2 is

$$LR=\frac{HCA}{LCA}=\frac{44}{6}\approx 7$$

The resultant life ratio is

$$LR=\frac{HCA}{LCA}=0.23\times 7=1.61$$

Taking the 95 percent lower confidence limit for the LCA bearing of 6250 hours and multiplying it by LR of 1.61, a predicted life of 10063 hours for the HCA bearing is obtained. This value is the 95 percent upper confidence limit of the actual L_{10} life of the HCA bearings. Can the authors clarify this matter?

The life data presented in Fig. 1 appear to contradict established data already reported in the literature for AISI 52100 and AISI M-50. It is truly unfortunate that the authors have not reported these data in more detail nor shown the actual Weibull plots, the data points, the failure index and the confidence numbers nor the type of test rig used. Results of tests in the NASA five-ball fatigue tester with CVM material showed that the life of CVM AISI M-50 was 68 percent CVM AISI 52100 (Parker et al., 1971). Tests in the General Electric RC rig with VAR and VIM-VAR AISI M-50 showed that the VIM-VAR material produced average lives approximately twice that of the VAR material (Nahm, 1983). Both the NASA and G.E. data were run at approximately equivalent stress levels to that of the authors'. Hence, it would have been reasonably expected that the life of the VIM-VAR AISI M-50 for the authors' data would be approximately (0.68×2) 1.4 greater than the VIM AISI 52100 and not the reverse as shown in Fig. 1. Can the authors explain this anomaly?

Additional References (Part 2)

Bamberger, E. N., Zaretsky, E. V., and Signer, H., 1976, "Endurance and Failure Characteristics of Main-Shaft Jet Engine Bearings at 3×10^6 DN," ASME Journal of Lubrication Technology, Vol. 98, No. 4, pp. 580–585.

Nahm, A. H., 1983, "Impact of NASA-Sponsored Research on Aircraft Turbine Engine Bearing Specifications," *Advanced Power Transmission Technology*, G. K. Fischer, Ed., NASA CP-2210, pp. 173–184.

Parker, R. J., Zaretsky, E. V., and Dietrich, M. W., 1971, "Rolling-Element Fatigue Lives of Four M-Series Steels and AISI 52100 at 150°F," NASA TN D-7033.

J. C. Clark[2]

This discusser agrees with the authors that the Lundberg and Palmgren method modified by ANSI and ISO is inadequate to predict the life of bearings in high performance aircraft engines. The analytical model described in the authors reference [7] brings a new and innovative approach to bearing life analysis. With this approach it may be possible to understand bearings that out performed, as well as those that under performed, the previous predictions.

The challenge will be to determine the stress field adequately. It is not clear from this paper that the analysis includes stresses due to the contact of asperities. This discusser believes that the near surface stresses due to asperity contact and the resulting shear stress due to asperity sliding will be the life limiting stresses as specific film thicknesses are reduced below the full film region. In this region, a major portion of the useful life may be crack propagation. Would the authors please comment on the effects of asperity contact, i.e., how are they treated in their analysis?

The efforts of the authors are greatly appreciated and they are encouraged to continue to validate the theory and to refine the stres analysis as deemed necessary. However, the use of data from the engines in Parts 1 and 2 seems to raise more issues than answers related to the new model. A more controlled test environment is required to establish the constants and limiting stress for various materials.

Part 1

The cubic mean load approach does not appear adequate when using the new method. The contact stress (Po) at the mean load is 1.8 GPa (261 KSI). The contact stress at the maximum thrust load point from Table 3 should be near 2.28 GPa (331 KSI). In the Lundberg and Palmgren analysis, the small percentage of operating time accounted for the limited damage accumulated at this condition. In the new analysis, many of the operating conditions may be below the limiting stress value and not contribute to the damage. In a limiting case, the mean load might predict an infinite life, while the maximum load could predict a very finite life. Miner's rule should be used and not the cubic mean load approach. Would the authors please comment on the projected life at the maximum load?

The fact that only 18 bearings failed out of 4173 exposed

[2]General Electric Co., Cincinnati, Ohio.

to service is impressive. The early nature of these failures causes concern with the way the data are used to compare to theory. The Weibull plot (the authors' Fig. 5) shows a slope of ~.7. In Weibull analysis, a slope of this magnitude supports an infantile failure mode with the failure rate improving with operating time. It is dangerous to predict a 10 percent life, projecting from these "early" failures. An alternate approach would be to assume a more classical slope for aircraft engine bearings (1.3–1.5) and not allow the few failure points to establish the slope. It is doubtful that sufficient time exists on the bearing population to project a 10 percent life of 204,000 hours. For example, what is the average time on the population?

Part 2

In reviewing Part 2 of the paper, this discusser was distressed to see the authors take data from a component test at Po = 4.825 GPa and apply this to full scale bearings at values of Po = 1.61 and 1.37 GPa. A contact stress of Po = 4.825 GPa probably causes subsurface stresses to be in a range where cyclic plasticity comes into play. In most component test on AISI 52100 operating at this stress level, the material would receive a different heat treat (higher hardness) that would be used on aircraft engine bearings requiring stabilization for higher operating temperatures. Would the authors comment on the heat treating of the parts from the quoted test results, as well as review other implied data that indicate AISI 52100 exceeds the life of M50 by two to one?

The conclusion reached, and expected, is that the new method gives far greater differentiation between bearings, of moderate loading, operating at low lambda ratios and high lambda ratios. This wide variation in performance has been recognized by the experienced bearing design engineers, although previous calculation techniques do not account for the extremes. Both of the bearings of Part 2 operate at lambda ratios near 1.0, and have surface finishes of .16 mirometers. Comparing the results from Part 1 and Part 2, it appears that a race surface finish change to 0.08 mirometers would result in an extremely long projected life for the LCA or the HCA. Would the authors comment on the benefits of such a surface finish change?

In summary, the new life prediction method being proposed may add significantly to bearing life technology. Controlled testing will be required to establish the constants and to determine the stress limits of various different bearing materials.

W. E. Poole[3]

Rolling contact bearing research during the last decade has shown that with modern steel manufacturing processes, bearings have an endurance limit, below which subsurface fatigue failure doesn't occur, [D1, D2]. In [D2] the authors defined the theoretical basis for these laboratory observations and how it relates to traditional life predictions. In this paper, the authors apply their theoretical technique to actual applications to help explain obvious deviations from accepted life prediction techniques.

This is most welcome. Bearing application engineers have long recognized that many applications deviated substantially from Lundberg-Palmgren type life predictions with performance routinely exceeding predictions. Bearing life prediction capability hasn't kept up with advances in bearing performance, now perhaps our prediction capability will come of age.

This improved understanding of bearing performance provides a basis for improved reliability. As designers become aware of the importance of surface effects on bearing performance [D3] they will alter the emphasis of bearing design from a calculated fatigue life to improved contact dynamics. Bearing life improvements will result from better lubricant cleanliness, better surface finish and geometry at the contact interface and reduced contact shear stresses from reduced friction and contact traction.

In Part 2 of this paper, the authors present data showing improved performance of 52100 steel compared with M50 when operated with thin EHD films. No explanation is offered, however insight into marginal EHD film lubrication is available in the literature. Surface peeling and microspalling occur as a result of operating with marginal EHD film thickness and has been related to large carbides in high alloy steel, [D4]. It follows that material with a fine microstructure free of large carbides, such as 52100, should perform better than M50, a high alloy tool steel, in marginal EHD film applications. This is further evidence that bearing life is sensitive to conditions at the contact interface.

It is hoped the authors will extend their work to include local surface effects, including contact slip and traction. Much work has already been accomplished to show the stress concentrations due to surface roughness and the mitigating effects of soft metal coatings on these contact stresses, [D5]. We need a theoretical basis for the observed life improvements with thin surface films, [D6] and an organized method for accounting for this benefit during design.

Broad acceptance will be required to maximize the benefit from this refined approach for bearing life prediction. We routinely accept that other structural members may have infinite life under some stress conditions, its reasonable to believe the same physics applies to rolling contact fatigue. Aircraft gas turbine overhaul shops report that very few bearing rejects are due to fatigue, [D7]. With the understanding we are getting from work such as this, designers will be freed from needless worry about fatigue and can address the actual causes of bearing malfunction. The industry will be the beneficiary.

Additional References

D1 Zweirlein and Schlicht, "Rolling Contact Fatigue Mechanisms—Accelerated Testing Versus Field Performance," *Rolling Contact Fatigue Testing of Bearing Steels*, ASTM STP 771, J. J. C. Hoo, Ed., American Society for Testing and Materials, 1982.

D2 Ioannides and Harris, "A New Fatigue Life Model for Rolling Bearings," ASME Journal of Tribology, Vol. 107, 1985.

D3 Bamberger, et al.,"Improved Fatigue Life Bearing Development,"Interim Report AFWAL TR-87-2059, 1987.

D4 Pearson and Dickinson, "The Role of Carbides in Performance of High Alloy Bearing Steels," presented at the International Symposium on Effect of Steel Manufacturing Processes on The Quality of Bearing Steels, ASTM Committee A1 4–6 Nov. 1986.

D5 Merriman and Kannel, "Analyses of the Role of Surface Roughness on Contact Stresses Between Elastic Cylinders With and Without Soft Surface Coating," ASME Journal of Tribology, Vol. 111, Jan. 1989.

D6 Hochman, R. F., et al.,"Rolling Contact Fatigue of Cu and TiN Coating on Bearing Steel Substrates," *Journal of Vacuum Science and Technology*, A3(6) Nov./Dec. 1985.

D7 Cunningham and Morgan, "Review of Aircraft Bearing Rejection Criteria and Causes," *Lubrication Engineering*,Vol. 35, No. 8, Aug. 1979.

Authors' Closure

The authors wish to thank all the discussers for their interest in the paper and the comments. With respect to the specific points raised in their respective discussions, we offer the following replies:

Dr. J. C. Clark

Part 1

We agree that asperity contact induced stresses are progressively important as the lubrication regime moves from full

[3]Pratt & Whitney, West Palm Beach, Florida.