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

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