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.

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## 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.

separation to mixed and boundary lubrication. In the present work the effects of the asperity contacts are modelled only via averaged surface tractions in which expected asperity contacts give a higher coefficient of friction, $\mu = 0.1$. A more detailed analysis in the spirit of references [A1] and [A2] is possible, but this was not undertaken in the present study.

It is true, and it is stated in the paper, that the cubic mean load is not, in general, adequate when a fatigue limit is introduced. This simplification is an acceptable approximation here because a few high loads dominate the whole load spectrum. At maximum load the projected life is approximately 0.3 of the life calculated with the cubic average load.

In the Weibull analysis the classical, two parameter, model was used in accordance with the standard life prediction methodology. We agree, however, as mentioned in the paper, that the early failures may belong to a "weak" population and that the main population has for practical purposes "infinite life."

### Part 2

The heat treatment of the parts in the component test and the bearing was the same. On the matter of the possibility of improved performance of smoother surfaces operating at the same $\lambda$ values as rougher ones ($\lambda \leq 1$) we would like to point out the higher risk of oil film collapse with higher roughness slopes (usually associated with higher RMS height values) because of the non-Newtonian oil behaviour, references [A3], [A4].

## Mr. W.E. Poole

We agree that modern bearings, when well and cleanly lubricated, exhibit an endurance limit and that enhancement of our prediction capabilities is necessary, not only to keep pace with bearing improvements with respect to material, design, manufacturing and quality assurance, but also to quantify the effects of important environmental influences, neglected or insufficiently treated hitherto in life predictions (like residual stresses, contamination, and roughness).

Again, on the point of the local surface effects, we would like to state that with the recent capabilities in the contact pressure and subsurface stress calculations of the Multigrid Method (reference [A1]) such work is underway.

## Mr. E. V. Zaretsky

### Part 1

(*a*) *and* (*b*) The ability to detect the presence of a material fatigue limit in a working mechanism like the bearing depends to a large extent on how well the theoretical Hertzian stresses derived from smooth surface contact calculation describe the actual stresses developing during the operation of the bearing. These are effected in the near surface region by the quality of manufacturing, lubrication, asperity contact, and contamination. In recent years, with all the improvements in bearings and their testing, tyhe fatigue limit has become more detectable by major bearing manufacturesr (e.g., reference [A5]). It is also expected that the high hydrostatic pressures that co-exist with the high shear stresses enhance the endurance of the through hardened steel, and a combined criterion which includes the local hydrostatic pressure has already been introduced in the predictions. This can account for the effects of residual stresses (reference [A6]).

(*c*) The assertion that $e$, the Weibull exponent, can dictate on its own whether $h$ is positive or negative, that is, if life is proportional to the depth of the stressed volume below the raceway or inversely proportional, is misleading. According to Lundberg/Palmgren, reference [3] in our paper, $c$, $h$, and $e$ are *material* exponents which are linearly related. $e$ is directly calculated from the life dispersion and $c$ and $h$ from two additional equations in which the load-life exponent $p$ and the ball diameter-bearing dynamic capacity exponent are *experimentally* determined. These equations have been derived with the assumption of zero fatigue limit and the corresponding values of $C = 31/3$ and $h = 7/3$ are valid for the straight lines that defined $p$ and the exponent of $D_a$ in the experiments of Lundberg and Palmgren prior to 1947.

In contrast with the observed curved load-life relation of modern bearings, that is, with a variable $p$ in the sense of the Lundberg and Palmgren theory (resulting from the fatigue limit), the exponents $c$, $h$, and $e$ have been introduced as independent material constants in the Ioannides/Harris model (with the same values as in the Lundberg/Palmgren theory). Thus the Ioannides/Harris model closely reproduces the Lundberg/Palmgren results only when the fatigue limit is set to zero.

(*d*)Recent experiments indicate that the equation (A1) in the discussion *does not* accurately represent the life increase with the load reduction, references [A5] and [A6]. Moreover, equation (A2) itself is not an accurate representation of the bearing life-stress relationship in the Ioannides/Harris model, [7] equation (10).

(*e*) On the inclusion of depth weight, $z_0^h$ or $z'^h$, we would like to offer the following comments:

Both the classical model of Lundberg/Palmgren and the earlier applications of the new Ioannides/Harris model are based on the stress field of an ideal, smooth contact and include a depth weighting factor, $z_0^{-h}$ or $z'^{-h}$. By its relative reduction in expected life of a bearing as the macro-Hertzian stresses approach the surface, this factor produces good agreement with experimental results. The additional micro-Hertzian stresses in the neighborhood of the surface arising from the rough contact (or in general from manufacturing defects, contamination dents, etc.) were clearly excluded from the stress field. Thus the depth factor represents the effect of such stresses to some extent. It follows that when such micro-Hertzian stresses are specifically included in life calculations or when they are absent under untypically smooth or clean conditions, the factor $z'^{-h}$ will no longer be required in the integrand of equation (10) and $h$ may be set equal to zero. This physical interpretation of the depth weighting factor is in fact consistent with the argument originally given in [3] which uses it simply to describe a depth-dependence of the material condition (not, as is often suggested, to refer specifically to crack propagation). A form of equation (10) with $h = 0$ has also been used with some success to fit test data taken under a variety of operating conditions [A5]. In this context it is significant that the data were obtained under conditions as close to ideally clean and smooth as possible.

Furthermore, the argument proposed by the discusser on the reduction of the volume term $V$ only as a result of a fatigue limit cannot account for stress concentrations, and, more important, the volume $V = az_0l$ in Lundberg and Palmgren is only a *measure* of the volume exposed to fatigue and cannot be compared to exact volume calculated from the new model [7]. In addition, it can be seen from Fig. 8 that the $V/l$* area exposed to fatigue (area enclosed by the contour drawn with a broken line) is far larger than 8 percent of $2az_0$, and it is estimated as 75 percent of $2az_0$ in this figure and therefore the accompanying calculations of life by the discusser should be reduced by almost an order of magnitude. Finally, it should be stated that the detailed stress fields, which can be obtained from fast Multigrid calculations, permit with the new model the assessment of the effects of many phenomena affecting

*$l$ is the length of the raceway.

the bearing life such as edge stresses, residual stresses, contamination, etc., as well as the asymptotic load-life behavior that cannot be accounted for by a constant $a_2$ factor, however large it is.

### Part 2

The authors would like to stress that the usefulness of a model lies primarily in its ability to describe the performance of machine elements in real applications. In this spirit the model was applied to field data *in addition* to laboratory test data.

The authors disagree with the discusser's statement that homogeneity and cleanliness of the material are important only for subsurface fatigue. The survival of the near surface region depends on the "strength" of the material and hence "surface life" is also influenced by it.

The authors are grateful to Mr. Zaretsky for pointing out the typographical mistakes in Fig. 2. In effect, HCA in the caption of Fig. 2(*a*) should be replaced by LCA and the reverse in Fig. 2(*b*). Moreover, Figs. 2 and 3 refer to cubic mean load values while Table 4 refers to the maximum load.

Unfortunately, we do not know the lubrication conditions under which Mr. Zaretsky's cited test results were obtained. What is exemplified in the paper by Fig. 1 is that, in the presence of marginal lubrication, i.e., where the lubricant film thickness is of the order or less than the composite roughness of the mating surfaces, the rolling contact fatigue endurance of M50 steel is somewhat less in the presence of a di-ester type lubricant as compared to 52100 steel. Additional information exists of a similar result confirming this apparent anomaly in the performance of what is otherwise recognized as a long-enduring bearing steel. It must be pointed out, however, that endurance of 52100 steels improved immensely during the 1980s. SKF has substantial evidence to this effect from endurance testing. It is probable that the 52100 steel referenced by Mr. Zaretsky did not perform at current levels.

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