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# discussion

## Endurance and Failure Characteristic of Main-Shaft Jet Engine Bearing at $3 \times 10^6$ DN[1]

**J. A. Burnett.**[2] As you have vividly shown and stated, rapid crack propagation from a spall resulting in the breakup of the inner race can be a serious problem. I agree that part of the solution requires application of Fracture Mechanics principles to determine the level of fracture toughness required for such an application. The second part of the solution, as you mentioned, is in the material itself and naturally a steel of higher fracture toughness would be desirable. A possible solution may be the application of case carburizing steels which are capable of service at these higher bearing operating temperatures. Generally case carburized sections with appropriate case depths have 2 to 3 times the fracture toughness of a high carbon through hardening steel.[3] It would be worthwhile to test a suitable carburizing grade under similar conditions to determine whether the additional fracture toughness is sufficient to prevent rapid crack propagation.

**J. C. Clark.**[4] The authors have completed an outstanding program for the design and evaluation of bearings operating to 3.0 million DN. The fatigue life results are in good agreement to the calculated values. The failure modes will be discussed later. By examining the bearings in terms of a mission cycle, the authors have taken a realistic approach in presenting the data to the industry's designers. More researchers should employ this technique. This discusser suspects the authors also found a strong appreciation of the designers' requirements with this approach.

To really evaluate where we are in 3.0 million DN technology, it will be beneficial to evaluate current bearing life requirements. Bearing life is always calculated and quoted as 10 percent failures or in some cases one percent. The real goal is to have a failure rate which, even though finite, is extremely small. In current commercial engines, from all manufacturers, design goals might be to maintain failure rates below one per million flight hours. Military engines might operate at slightly higher rates, not because the designs are different, but due to mission severity. Assuming a Weibull distribution with a slope of 1.5 and an engine life of 10,000 hours, a failure rate of one per million flight hours implies an $L_{10}$ life of 65,000 hours. The life projected by the authors mission cycle analysis converts to a failure rate of approximately 20 per million flight hours. This shows, that while the current fatigue results are excellent, the task is not complete.

From the data in the paper it appears the bearing designs are identical for both test series. While the design is optimized for 3.0 million DN it is not for the slower speeds. A brief study was conducted on the same size bearing, but with curvatures, contact angle, and number of balls changed to agree with current designs operating near 1.44 million DN. By doing this, the life at the same mission cycle can be increased by three to one. Many of todays bearings are being manufactured from the VIM-VAR material and this is a credit to earlier phases of the program the authors are reporting. Therefore with current designs, using the latest materials, the 65,000 hour $L_{10}$ life is a reachable goal.

Returning to the authors' test data, the failure distribution shown for the $1.44 \times 10^6$ DN test group is based on one failure which was a single ball failure. Do the authors feel this is conservative or would the lower speed sample actually have longer lives if additional testing was accomplished? The failures experienced at three million DN were all on the inner race of the balls. The outer race has been considered the limiting race for high DN operation. In reviewing the design, it is apparent the race lives have been balanced by having the open curvature on the inner race. Would the authors comment on this, since the outer race is primarily the limiting item applying to the mission analysis?

The failure mode experienced was predicted in the discusser's paper (authors' reference [19]). At this particular time, it is a little disturbing to be correct, but gratifying to identify the problem prior to committing a design to 3.0 million DN. This will allow some period of time to find a solution to this problem.

In summary, the fatigue test results are outstanding, the failure mode very disturbing, and the utilization of a 3.0 million DN bearing still slightly in the future.

**G. Beverly.**[5] The authors are to be commended for this comprehensive fatigue test program. These results are very exciting and will eliminate most ball bearing fatigue life problems if they are repeatable.

Those who work with fatigue test results know of the wide variability from test to test. This very encouraging test is with one heat of VIM-VAR steel. Would other heats perform as well? How will other steel manufacturers' products perform?

The test bearings are reported to have had a very excellent race finish of $2\mu''$. This undoubtedly aided bearing life by improving lubricant film formation. How much did this contribute to excellent performance of this test?

These questions must be resolved before the design engineer can comfortably rely on a life calculation.

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[1] By E. N. Bamberger, E. V. Zaretsky, and H. Signer, published in the October, 1976, issue of the JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 98, pp. 580–585.

[2] Research Metallurgist, The Timken Co., Canton, Ohio.

[3] Jatczak, C. F., "Materials for Elevated Temperature Service in Transmission Systems," Presented at SAE Off-Highway Vehicle Meeting, September 8–11, 1975, Mecca, Milwaukee, Wis.

[4] Manager, Advanced Bearing and Seal Design Technology, Aircraft Engine Group, General Electric Co., Evendale, Ohio.

[5] Pratt & Whitney Aircraft, West Palm Beach, Fla.

[6] Designate references in original text and additional references.

There is some confusion as to the source of the "processing variable factor-E." First the authors quote a 7.6 life ratio between the ASME design guide life and actual measured life. Later the authors refer to a factor of 22. Apparently this was arrived at by multiplying the observed 7.6 ratio by the ASME material processing factor of 3, but this isn't clear.

The authors state "This life ratio can be attributed to the use of the double-vacuum melted (VIM-VAR) AISI M-50 steel." It could also be attributed to fatigue test result variability due to material or processing random variation. Also, the influence of surface finish and lubrication on this life result needs to be better understood. It would be interesting to know the level of oil filtration in this test rig.

The results of a recently published report [22][6] on ausformed bearings illustrates the variability of fatigue test results. This program compares ausformed bearings to conventionally formed bearings. The baseline bearing life result was approximately 3 times theoretical life which is less than the ASME design guide life. This report doesn't state the baseline bearing material but it is assumed to be VIM-VAR since material melting procedure was not the variable being tested.

It was interesting to see that underrace cooling and lubrication was required for satisfactory operation. It does indeed provide lubricant where needed and does provide good control of axial and radial thermal gradients with a minimum of fuss. It would be interesting to know why cage bore oiling was considered necessary on this "thrust load only" application. Do the authors feel the 3 gm-cm. unbalance in the cage would have resulted in cage bore wear without direct land lubrication? Was more precise cage balancing considered?

Most interesting, was the finding that high speed induced stress can lead to race fracture with continued operation after race spalling. Are the authors able to estimate rate of crack progression from initial spall to race fracture? Can the authors suggest a race stress level below which cracks will not propagate to fracture? What influence did the through-race oil holes have on crack propagation?

This test represents a significant accomplishment. If the life improvement mechanism can be thoroughly understood and can be shown to be repeatable, then a major bearing design improvement will be available to the designer. Roller bearings also need the same life improvement demonstration. We hope this work will continue.

**P. E. Cowley.**[7] The authors are to be congratulated for such an impressive work. The excellent test results reported do indicate that a significant life improvement for AISI M50 is possible by advanced processing techniques, and is a welcome addition to the literature. However, the suggestion of the extraordinary life adjustment factor of 44 should be tempered with caution, and the data base fully qualified, so as to promote judicious interpretation.

To what material characterization do the authors attribute these excellent test lives? Specifically, what was the cleanliness; what was the heat treatment, was it conventional, or if specialized, what was the carbide morphology and will the heat treatment consistently produce a nominal hardness of Rc 63? Also, what was the actual geometric quality of the bearing components and assemblies, i.e., did they significantly exceed ABEC-5 requirements?

One of the problems associated with operating above 2 million DN described in the Introduction is the predicted decrease in life due to the increased stress in the outer race caused by centrifugal loading effects. Perhaps the authors would comment on the fact that none of the reported failures occurred in the outer race, contrary to the stated hypothesis.

The statements "Bearing lives at speeds of $3 \times 10^6$ DN with the VIM-VAR AISI M50 steel were nearly equivalent to those obtained at lower speeds." (presumably $1.44 \times 10^6$ DN), and, "There is good correlation between the theoretical analysis and the experimental results showing the effect of speed on bearing life." are very speculative considering they are based on only one failure at the lower speed. What confidence can be assigned to the $L_{10}$ values for the 3 million DN, the 1.44 million DN and the relationship between the two?

The comparison ". . . with similar bearings made from CVM AISI M50 steel ran under the same conditions" is confusing. (The comparison is apparently with work reported in [10].) The bearing tests reported in [10] were actually run at 1.44 million DN rather than 3 million, with a greater load and a different lubrication system. In addition, the bearings had a different contact angle. It is also notable that the experimental $L_{10}$ life reported in [10] was 64 million inner race revolutions which was "adjusted" to 105 million in Fig. 4. It would be informative to list the various factors involved in this adjustment, especially the factor for the change in contact angle. In regard to the two different lubrication systems employed, namely under race and jet, if it was considered necessary to change the lubrication system for the lower speed group to ". . . compare the life of these test bearings with that previously obtained with oil jet lubrication [10]," then does not that same rationale compromise the basis for the speed comparison in this work?

Certainly this singular program with the VIM-VAR process does show great promise. Hopefully, future work involving a multiplicity of material heats, processing and test programs will be as successful.

## Authors' Closure

The authors would like to thank the discussers for their timely and thought-provoking comments. Mr. J. A. Burnett's suggestion to use carburized steels, at least for the inner races of a high-speed angular-contact ball bearing, has considerable merit and has been considered by the authors. It is probable that the advanced carburized materials now becoming available can have long term rolling-element fatigue life as well as the fracture toughness required at operating speeds of 3 million DN. However, at the present time, it would appear that the race fracture problem will limit bearing operation to speeds less than 2.7 million DN. Obviously, much additional material research is still required.

Mr. Cowley questions the comparisons made between the CVM AISI M-50 bearings run at 1.44 million DN and the VIM-VAR AISI M-50 bearings tested at 1.44 and 3 million DN, inasmuch as there was variance between the thrust load, contact angle and, at 3 million DN, the lubrication method used. It will be recalled that at 3 million DN underrace lubrication was utilized as opposed to jet lubrication at the 1.44 million DN speed. The means of injecting the lubricant should not affect fatigue life although it would affect bearing temperature [6].[8] In our tests, the bearing temperature for both jet and underrace lubrication were controlled, and were nearly identical for all the fatigue data. Based upon the data of [23], any difference in temperature which may have existed would not be expected to be reflected in the life results.

The effect of thrust load and resultant contact stress was adjusted using an inverse 9th power relation between life and stress. This approach was verified in [24]. A study of the effect of contact angle on life was reported in [25]. The study confirmed the relation used by Lundberg and Palmgren [12, 13], and was the same as that used to correct for contact angle in the comparison of the two bearings with different contact angles. Elastohydrodynamic film thickness effect on life was also considered and factored into the comparison. Using

[7] Federal Mogul, Ann Arbor, Mich.

[8] Number in brackets designates References in original text and additional references.

the methods of Harris [15], the $L_{10}$ life reported in [10] was adjusted to reflect the bearing design and operating conditions of the instant research. Had these specific adjustments not been made, the reported life improvement would have been even more pronounced.

Mr. Cowley also asks about the metallurgy of the VIM-VAR AISI M-50 steel. The steel was indeed extremely clean, although this is consistent with the premium-quality VIM-VAR M-50 available today. It was bought to an existing premium-quality bearing material specification, and no attempt was made to procure a "special" heat. Using normal inclusion rating methods, no inclusions were noted. Again, this is as expected. It is in line with the authors opinion that existing inclusion ratings are not realistic for today's double vacuum melted steels. It should not be construed, however, that inclusions do not exist in the material. What is important is that their frequency and number is insignificantly small.

Mr. Cowley's asks whether the heat treatment would consistently produce a Rockwell c hardness of 63. The answer is in the affirmative. This is primarily attributable to the fact that the previously cited premium-quality specification requires the carbon content of the M-50 to be toward the upper limit (0.80–0.85) of the standard AMS 6490 specification. This assures more consistent and higher hardenability.

The heat treatment for the VIM-VAR AISI M-50 steel was conventional and the same as that for the CVM AISI M-50. The carbide morphology was normal for the size of the rings and the diameter of the balls.

As regards the manufacturing tolerances of the test bearings, they met, but did not exceed, the geometric qualities of an ABEC Grade 5 bearing.

The design of the 3 million DN bearing was such as to make the bearing primarily inner race life dependent. This would not necessarily be so in all bearing designs. The design for the instant investigation was based on two considerations. These were: (*a*) optimizing fatigue life, and (*b*) minimizing heat generation. The bearing geometry and contact angle selected were based upon a compromise of these two considerations. The life prediction methods of Harris [15] indicated that even at high speeds, the bearing life was inner race dependent. This, of course, does not necessarily preclude the probability of an outer race failure.

With the VIM-VAR AISI M-50 at 3 million and 1.44 million DN, there was no significant difference in life. However, between the VIM-VAR AISI M-50 at both speeds and the CVM AISI M-50 at 1.44 million DN, the confidence number was approximately 99 percent, which is rather significant.

With regard to Mr. Beverley's question pertaining to whether other VIM-VAR M-50 heats would perform as well, it is the authors opinion that properly melted and heat treated other heats will statistically perform as well. As a practical matter, experience has shown that over a long period of time, most test results of the same material of different heats can perform within ±33 percent of an average life value for the material tested. Whether the authors' data are on the high or low side of what might eventually be a mean life value can only be speculated upon at this time.

In order to clear up an apparent confusion regarding the life adjustment factors, the *Processing Factor E*, from the ASME Design Guide [14] is 3. The *Material Factor D* for CVM AISI M-50 is 2. The resultant life modifying factor is ($E_1 \times D$) or ($3 \times 2$) or 6. The *Lubrication Factor F* for these data was approximately 2.4. Hence, for the ASME prediction at 1.44 million DN, the life adjustment factor would be ($E_1 \times D \times F$) or ($3 \times 2 \times 2.4$) equals 14.4. The predicted life at 1.44 million DN obtained by the methods of Lundberg and Palmgren [12, 13] was $25.2 \times 10^6$ inner race revolutions. As a result, the ASME prediction for this speed is ($E_1 \times D \times F$) or $14.4 \times 25.2 \times 10^6$ which equals $363 \times 10^6$ inner race revolutions. For the VIM-VAR AISI M-50 material, factors D and F remain unchanged. The Processing Factor E for CVM is replaced by Processing Factor $E_3$ for VIM-VAR. The experimental life at 1.44 million DN is estimated to be $2700 \times 10^6$ inner race revolutions. Therefore,

$$E_3 = \frac{2700 \times 10^6}{2 \times 2.4 \times 25.2} = 22$$

If the above exercise is repeated for the 3 million DN life results, $E_3$ would equal 24. The lower, or more conservative, value of $E_3$ for VIM-VAR was chosen. Hence, the life modifying factor would be $2 \times 22$ or 44 for VIM-VAR AISI-M-50.

Surface finishes for the bearings of [10] and that of the instant investigation were, for all practical purposes, identical. As a result, surface finish effects were not a variable in the reported comparisons. The effect of surface finish, however, is factored into the life predictions through the Lubricant Factor F [14].

Mr. Beverley references work performed on ausformed bearings [22] to illustrate what he terms the variability of fatigue test results. Unfortunately, the baseline bearings bearings cited in [22] were not made of VIM-VAR and even more importantly, were not specifically designed for the conditions under which they operated. As reported in [22], these bearings exhibited cage wear and resultant debris damage. Consequently, the experimental life of 3 times theoretical is not unexpected nor does it contradict the results of the instant research.

The key to long term high-speed bearing operation is effective cooling and assuring adequate lubrication at locations of rolling and/or sliding contact. The cage/land area is one of these locations. The results of [3–5] for short running times without direct lubrication to the cage/land surfaces did not result in gross wear. However, it was the authors decision that good engineering practice called for assurance of a continued supply of lubricant at the cage/land area for continuous long term operation at 3 million DN. Whether or not the lack of direct lubrication to this location would have resulted in cage bore wear is open to speculation. More precise cage balancing was not considered.

In regard to Mr. Beverley's question regarding the influences of throughrace lubrication holes on crack propagation, it is the authors opinion that if there is any effect, it is secondary to the stress concentration of a fatigue spall. In the nearly 75,000 hours of $3 \times 10^6$ DN bearing operation, there was no indication of any crack initiating at a lubrication hole.

Based upon the experimental data, the time from the initial fatigue spall to a total race fracture was 7-½ minutes. Inasmuch as the fatigue spall acts as a stress raiser and fracture becomes, in part, a function of the size of crack generated by a typical fatigue spall, it is not possible to estimate with our current level of knowledge a safe hoop stress level below which the race will not fracture after a spall has occurred.

The lubrication system for the test bearings incorporated a 10 micron filter. No surface debris damage nor surface initiated fatigue spalling was noted.

Mr. Clark suggests a novel approach to bearing selection and design wherein bearing life is based upon an acceptable bearing failure rate. Mr. Clark's approach has considerable merit. The approach implies that at the 10,000th hour of engine life, the hazard rate for the bearing is one per million flight hours. At engine times prior to 10,000 hours, the hazard rate is considerably less. The cumulative failure probability for bearings with a 65,000 hour $L_{10}$ life between time zero and 10,000 hours is one failure in 1-½ million engine hours.

The failure probability and hazard rate have different physical meanings which are discussed in [26]. The hazard function may dictate lives which may not be attainable with current designs and material technology if the hazard function is used. This approach to bearing selection deserves additional consideration and study.

## Additional References

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23 Zaretsky, E. V., Anderson, W. J., and Bamberger, E. N., "Rolling-Element Bearing Life from 400° to 600°F," NASA TND-5002, 1969.

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BRICATION TECHNOLOGY, TRANS. ASME, Series F. Vol. 96, No. 3, 1974, pp 391–397.
25 Zaretsky, E. V., Anderson, W. J., and Parker, R. J., "The Effect of Contact Angle on Rolling-Contact Fatigue and Bearing Load Capability," *Trans ASLE*, Vol. 5, 1962, pp 210–219.
26 McCall, J. C., Renewal Theory-Predicting Product Failure and Replacement, *Machine Design*, Vol. 48, No. 7, Mar. 25, 1976, pp 149 154.

# Analysis of Tapered Roller Bearings Considering High Speed and Combined Loading[1]

**C. A. Moyer.**[2] Dr. Liu has presented an interesting analysis of a single tapered roller bearing attempting to show the effects of misalignment and speed on such an isolated, single bearing. It appears because of some of the beginning assumptions Dr. Liu uses, however, that the calculations made and the results presented may not provide realistic insight into the actual performance of such a bearing. Several assumptions are in question.

First, the author states that the raceway and roller is assumed in line contact although a significant crown radius of 2680mm (105.5 inches) is given for the roller body. The load considered is sufficiently low that the elliptical contact would be just truncated for these conditions. Therefore, point contact deformation should be considered at both the inner and outer raceway contact.

Second, the general statement is made that the total load at roller and outer raceway contact is independent of speed. This can only be true if sufficient thrust load is being applied to the bearing so that centrifugal force effects are overcome. Also, whether Figs. 9 and 10 are correct in the author's paper depend on the order of application of the thrust load and centrifugal force.

In considering load and speed changes, does the author consider that the total axial deformations remain constant, that is, does the distance in the '$Z$' direction in the author's Figure 1 between the cone backface to cup backface change or remain the same as speed increases centrifugal force and external applied loads stay constant? Such problems in interpreting the single bearing internal conditions illustrate the difficulty in relating the single bearing analysis to real life situations.

Third, the author implies that a high speed bearing can operate with sufficient misalignment that the rollers will actually lift off the raceway allowing *no* contact over about 30 40 percent of the raceway (Fig. 8). In a realistic application, either the bearing would need to be redesigned or the application modified in order to maximize expected performance.

The author mentions the sliding at the inner ring flange or cone rib and calculates only the sliding performance. In terms of rib roller end performance, this may be misleading since both the rib surface and roller end surface are moving and the mean surface velocity $[\frac{1}{2}(u_1 + u_2)]$ is about $2\frac{1}{2}$ times the sliding velocity $[u_1 - u_2]$ for this bearing. For proper evaluation, it is better to include both, perhaps in terms of the slide/roll ratio.

[1] By J. Y. Liu, published in the October 1976, issue of the JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 98, pp. 564–574.

[2] Assistant Chief Engineer, Physical Laboratories, The Timken Co., Canton, Ohio.

## Author's Closure

The author welcomes the discussion by Mr. Moyer, who has certainly brought up some valuable and interesting points on the subject paper.

The roller and raceway are designated to be in line contact because the so-called slicing technique is used to define the load-deformation relationship. The slicing technique considers a contact to be composed of a number of slices of constant width, each of which is taken as a line contact. According to reference [1],[3] it can accurately predict the contact characteristics for either line or point contact, provided that the contact length to width ratio is large and/or the stress level is high. These requirements, of course, cannot be strictly fulfilled in the numerical example of the paper, where, when misalignment is present, a roller raceway contact may neither be classified as line contact nor as point contact. It is probable that the numerical accuracy can be improved if a larger number of slices was chosen for each contact at the expense of more computer time.

The author cannot follow how the roller centrifugal force can be overcome by sufficient thrust load since the centrifugal force is needed in the roller equilibrium conditions. Nor can he understand the effect of the order of application of the thrust load and centrifugal force on the reliability of Figs. 9 and 10 in the paper, since the paper considers only the dynamic equilibrium of the bearing. The reasons that the total contact load at the roller and outer raceway contact is independent of speed are: (1) the outer ring is assumed to be rigid, and (2) the sum of the horizontal components of the outer raceway contact loads must balance the applied thrust load. However, as pointed out by Mr. Parker in his discussion on the paper, when a pair of tapered roller bearings are axially set up with a given clearance, an induced thrust load due to speed will come into being in addition to the applied thrust load. This induced thrust load will then increase the total load at the outer contact.

The bearing inner ring displacement in the $Z$-direction will change with load and/or speed. In the analysis of the paper, all the inner ring displacements will change as the bearing operating conditions change, unless some restrictions be applied to them. If the displacement in a certain direction is specified, then the number of bearing equilibrium equations will be reduced by one and the bearing load distributions will also be different. For instance, in order to avoid that 30–40% of the roller be out of contact with the outer raceway at $\psi = 0$ degrees as shown in Fig. 8 of the paper, one must be sure that the misalignment angle, $\gamma_y$, be kept very small, probably less than 0.1 degrees.

It is true that the slide/roll ratio at the flange contact is meaningful as it has a direct relationship with the EHD friction coefficient [2]. However, the heat generation and smearing occurrence are associated with, among other things, sliding. In the analysis, it was assumed that pure rolling exists between the roller and the raceways. Therefore, the exposition of the flange sliding will also serve to disclose the kinematical difference between the roller flange contact and the roller raceway contacts.

### Additional References

1 Crecelius, W. J., Jr., "Evaluation of a Slicing Technique Used to Calculate the Characteristics of Concentrated Contacts," MS thesis, Penn. State University, 1975.
2 McCool, J. I., and Chiu, Y. P., et al., "Influence of EHD lubrication on the Life and Operation of Turbine Engine Ball Bearings," Air Force Contract No. F33615-72-C-1467, July 1974.

[3] Numbers in brackets designate Additional References at end of closure.