Acknowledgments

The work reported in this paper evolved from the author's thesis submitted in partial fulfillment of the requirements for the Master of Science Degree at the Hartford Graduate Center of the Rensselaer Polytechnic Institute. The author wishes to than Dr. H. Kraus, Dean of the Graduate School, and Mr. J. Mauriello of AVCO/Lycoming Division for comments during the development of this model. Also, the author is grateful to Mr. A. B. Jones, Bearing Consultant; Mr. J. Lenski of the Boeing Company, Vertol Division, and Mr. J. Rumbarger of the Franklin Institute Research Laboratories for reviewing and commenting on the final draft of the thesis before its acceptance.

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DISCUSSION

P. F. Brown

This paper represents a contribution of considerable potential for advancing the design and development of roller bearings for optimum performance in high speed application.

The analysis embodied here contains the first comprehensive attempt to take into account the all important effects of interactions between the rolling elements, the retainer, and the oil/air medium in the context of both EHD and churning effects. The significance of this work will most likely be first felt in the application of advanced roller bearing design to aircraft gas turbines of the future which will require operation at DN levels of 3.0×10^6 DN and higher. Experimental test work already underway in this area has provided testimony to the need for an increased understanding of what causes cage distress and roller/raceway surface damage. Cage distress often shows up as severe wear in the pockets and on the surface contacting the guiding land. Raceway/roller surface damage normally associated with skidding produces a glazed surface or, in its advanced stages, a frosted appearance due to micro-pitting. Mr. Poplawski's analysis has real potential for development into a means that can provide sharp insight for solutions to these problems.

Due to the complex nature of the problem the author has attacked in his paper, certain assumptions had to be made. Some of these assumptions are well defined. Others, however, are apparently not defined, or at least were not obvious to this reader and an explanation or elaboration by the author would be helpful. Specifically, it was not defined where the cage-ball interaction force, F_{cage}, acts. The implication *is that it was assumed to act at the pitch circle.* For some bearing designs this would not necessarily be the case. Would the author expect any significant effect on his results with changes in the location of the point of action of this force? Also, in equations (10) and (11) which define the equilibrium of the inner ring, the hydrodynamic pressure forces, Qij, have been excluded. What is the rationale behind their exclusion? In this same vein, the Q forces are represented as acting through the roller center. This implies that the more generalized force vector acting at the roller-race contact also acts through the roller center. This force vector would be the resultant of the P and Q forces. What evidence exists in reality that these forces act in this manner, or was this an assumption needed to affect a solution to the equation system? The analysis also represents the friction coefficient at the roller-cage pocket interface and the cage-land interface by the same term, f_p. The parametric study included in the paper appears to use the same value of f_p for both cases. Can the author provide any justification for this apparent assumption? Can the two f_p's be varied independently in the computer analysis in order to evaluate the magnitude of the effects of variations on other parameters?

Other questions are prompted by the results of the author's parametric study of the 35 mm roller bearing. How does the author resolve the apparent contradiction between Figs. 1 and 5? These figures are plots of roller speed versus circumferential roller position for different levels of radial load. The test results shown in Fig. 1 imply that the maximum roller speed occurs in the plane of the applied radial load. In contrast, the results of the parametric study show quite the opposite effect with the roller decelerating as it enters the load zone and its minimum speed occurring in the plane of the applied radial load.

The author's analysis is limited to a roller bearing design with an outer land guided cage. Since most successful very high speed designs (2.5×10^6 DN) incorporate inner race land guided cages it would be of interest to alter the analysis for such a situation. Because the cage is now in contact with the driving race it is quite logical to expect a change in the bearing behavior as described by this analysis. Does the author have any feeling for how his parametric study results might differ for such a design?

The author has his hands on the tail of a technical tiger. This reader is left with the feeling that his grip is firm and that he could successfully cage it with some additional but concerted effort. Other factors that would seem to be of some concern that may not be too difficult to model mathematically are: (1) viscous drag effects at the roller end as it interfaces with the guide flange and (2) roller crowning effects on the magnitude of the EHD film pressure force. More difficult to model is the inertia effects introduced by the use of real and finite pocket clearances and, finally, the most complex of all; skewing effects as introduced by thermal or mechanical ring coning and misalignment in addition to roller coupled and uncoupled unbalance forces. Does the author plan any further work to expand his analysis to take these or other factors into account? This discusser is hopeful that the author does plan to continue this work and capitalize on the fine start he has made.

E. Kingsbury

The question of cage whirl is brought up in this interesting paper. It is assumed, apparently arbitrarily, that the cage in a large, high-speed roller bearing under radial load will whirl at half the cage rotational speed. Does the author know of any experimental evidence for cage whirl in such a bearing; and would he comment on his choice for whirl speed? Cage whirl at very high speed, or at shaft speed, or at nearly the same as cage rotational speed, or at exactly cage rotational speed (depending on running parameters), but never at half cage rotational speed, has been observed and explained in small, high-speed, axially loaded ball bearings.[2] If the force system acting on the cage is adequately described by the author's model, one might expect the whirl speed to be predictable.[3]

[2] Kingsbury, E. P., *ASLE Trans.*, Vol. 8, 1965.

[3] Walters, C. T., JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 93, No. 1, Jan. 1971, pp. 1–10.

J. W. Lenski, Jr.[4]

The discusser would like to commend the author for a very interesting and complete presentation dealing with slip and cage forces in high speed roller bearings. The topic covered by this paper is of great interest to me. At the Boeing Company—Vertol Division considerable time and money has been invested in experimentally investigating the effects and causes of high speed roller bearing skidding. Some of the results of these investigations were supplied to the author for use in correlating his analytical model. As shown in Fig 4 of his paper, the author's analytical model has demonstrated an improved capability of predicting skidding versus previous "simplified" models.

As a bearing designer involved in the development of high speed aircraft transmission systems, I appreciate the need for a mathematical technique for accurately accessing the percentage of skidding in lightly loaded, high speed roller bearings. With ever increasing operating speed requirements of advanced transmission systems, it is anticipated that bearing skidding may cause development problems. Without the aid of a rigorous mathematical analysis, bearing skidding problems would require time consuming and costly experimental test program. During the past few years, several papers have been presented which dealt with the subject; however, the methods presented were limited in the prediction ability. The author's improvement to previous complex mathematical models by the introduction of EHD effects, churning losses and other internal friction losses has provided a significant improvement in predicting the amount of cage and roller slippage and cage forces which were previously lacking.

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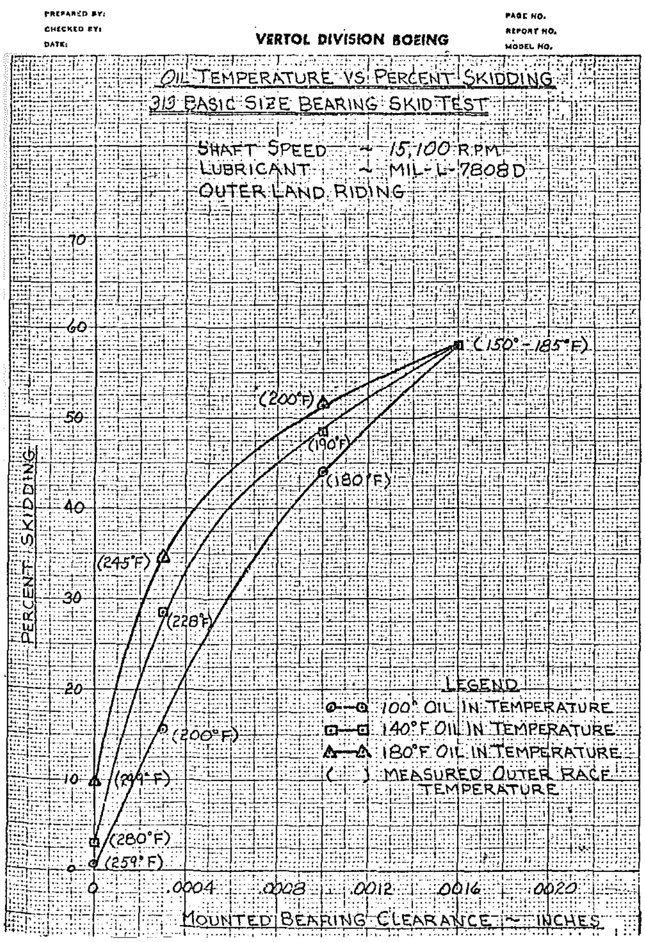

Fig. 10

Although the analysis presented by the author provides good correlation with cage slip test data, the effect of oil inlet temperature versus cage slip does not agree with Boeing-Vertol test data. In Fig. 9 of the paper, the author shows that slip decreases with increase in oil inlet temperature. Fig. 10 of this discussion shows the results of testing conducted at Boeing-Vertol on a 313 roller bearing indicates that higher oil temperatures increased cage slip. Although the test data is for a larger size bearing, the differences in results should be investigated to determine why the authors model does not agree with test data. Other than this one lack of correlation, the discusser believes that the approach taken by the author is sound and he should be congratulated for making a significant improvement in analyzing roller bearing skidding.

J. H. Rumbarger[5] and E. G. Filetti[5]

The author is to be complimented on a good systems analysis of the high speed main shaft roller bearing problem. This is an especially ambitious project at the Master of Science level. The present paper adds additional information to the growing literature in this field which was initiated by Harris reference [3] and Boness reference [5]. This solution differs in that it computes equilibrium conditions for each roller in the bearing including the unloaded rollers. The model also includes efforts at additional sophistication regarding dissipative losses. The basic EHD theory is restricted to the isothermal Dowson, Higginson.

The analytical treatment of "churning loss" as shown in equation (38) is a first step toward an attempt to describe dissipative fluid losses in the bearing. As stated by the author, this is intended to stimulate thinking in this area since churning effects are a significant problem in today's high speed applications. An examination of the kinematics of the high speed roller bearing shows that dissipative fluid forces acting on the cage can be either driving or retarding depending on the bearing configuration. As an example, an inner race land riding cage will experience driving forces from the inner race to cage contact. The relative velocity of fluids with respect to the cage surface is in opposite directions for inner and outer cage surfaces. This effect is not included in the present model. The use of an effective density of the air-oil mixture per equation (39) is a practical approach. *The author's comments regarding the sensitivity of the solution or percentage cage slip to variations in the percentage density of the mixture would be welcomed.*

The individual roller solutions for both loaded and unloaded rollers are very sensitive to the valve of the coulomb coefficient of friction between the roller and cage pocket. This is a high speed rubbing condition under light load between a well lubricated hardened steel and a softer silver. Curves presented in the paper use a coefficient varying from 0.05 to 0.1. Professor Dowson commented on the sensitivity of the individual roller solution to very small changes in torque about the center line of the roller. This comment was also made by Boness [5]. Recent work at The Franklin Institute indicates that cage slip can change as much as 50 percent with a 20 to 30 percent change in the coefficient of pocket friction. *The author's comments regarding the sensitivity of his solution to changes in this variable would be welcomed.* Also any comments regarding his selection of the 0.05 to 0.1 coefficients for this material combination would be appreciated.

Increased sophistication is required in high speed rolling element bearing analysis. An improved dimensional model, fluid model, and thermal model are greatly needed. The dimensional model should include centrifugal and mounting effects upon bearing dimensions, especially in the operating internal diametral clearance and the operating cage-to-land clearance areas. Roller profile, or crowning effects, must also be included.

An improved fluid model should use boundary layer theory to predict roller compartment losses, including rotating disc effects on the roller ends. In addition, fluid dissipation force acting on

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the cage surface should be predicted by similar techniques. It should be noted that operation is in a highly vortex-turbulent flow regime for high speed roller bearings. As noted earlier, benefits in the form of driving effects, resulting in less cage slip, can be obtained with an inner land riding cage. Cross-flow effects on roller end drag due to pumping lubricating oil through the bearing should also be considered. The use of a coulomb coefficient of friction for the cage land contact can be improved by considering the governing fluid mechanics in more detail.

Most important is the need for a thermal model which will include thermal effects in the EHD contact and will define oil temperature at the inlets to the contact. Such a solution should also reflect effects of heat removal by means of an overall heat balance analysis of the entire bearing. The results would be additional definition of the dimensional model by including thermal effects on operating internal diametral clearance and cage-to-land clearance areas. Ultimately, the thermal model should become sufficiently sophisticated to predict regenerative thermal lock-up in high speed bearings.

Some slip occurs in all high speed bearings. The problem is to determine when "skid damage" is likely to occur. *The author's comments regarding use of Blok flash temperature calculations or similar techniques for predicting unacceptable slip would be appreciated.*

Author's Closure

Mr. Brown's implication is correct, the cage force was assumed to act at the pitch circle. A more rigorous analysis would be to account for each element pushing against the cage at slightly different radii and reflecting this difference into a residual force and torque acting on the cage.

The hydrodynamic pressure forces Q_{ij} were excluded from the inner ring equilibrium and roller load vector since they are small when compared to the inner race contact forces. For example, a 300 lb radial load applied to the 65-mm test bearing resulted in 42 percent slippage. At this condition the lightly loaded roller had 60 lb inner ring roller load and a hydrodynamic pressure force of 0.46 lb.

The computer program has provision for inputting different values for the pocket and pilot friction coefficient. A graph showing the effects of these friction coefficients is available upon request.

The author apologizes for not specifically explaining how the data shown in Fig. 1 were obtained by Boness [5]. The measured roll speeds are for a *stationary cage* arrangement gotten by rotating the inner and outer races. The kinematics of this configuration are different from the inner ring rotation situation. Fig. 1 was merely used to establish the fact that roller speed variation does exist.

The computer program can be used to analyze both inner and outer race land guided cages. For the inner ring guided cage, the tractive force developed between the cage land and inner race is calculated as a drive rather than a drag effect. I have seen successful high speed operation of both inner and outer ring guided cages during my employment in the gas turbine industry.

The cage whirl speed was *assumed* as stated in the paper. I would not be too quick about applying results obtained on ball bearings to roller bearing operation. The whirl speed could be calculated using an additional iterative loop. The treaties on ball bearing dynamics by Walters [5] is an excellent paper on the transient behavior of the elements and cage. I cannot understand why it has received little attention from those active in high speed bearing design and analysis.

The temperature data presented by Mr. Lenski shows outer race temperature measurements. Whether skidding increases or decreases as you change oil inlet temperature (oil supply temperature) depends upon the overall system heat balance and how this affects the operating temperature gradient within the bearing. For example, if a decrease in oil inlet temperature causes a larger ΔT (hot inner race to cold outer race temperature gradient) than at the reference point, the operating internal clearance decreases and results in less cage slippage. However, if the ΔT across the bearing decreases the operating clearance increases and skidding can increase. Since we have no inner ring temperature measurements, it is extremely difficult to assess what is happening.

Messrs. Runbarger and Filetti inquire about the sensitivity of the solution to certain variables. I have not had the opportunity to perform a sensitivity analysis on every variable in the program and therefore cannot make a definite statement at this time.

The author wishes to thank these gentlemen for taking the time to comment on his presentation. I hope that those of you who are developing the next generation of roller bearing slip programs can incorporate many of the refinements that have been suggested.