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Parametric Evaluation of a Solid-Lubricated Ball Bearing©

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Parametric evaluations, based on analytical simulations of the dynamic performance of a solid-lubricated ball bearing, indicate that, for prescribed operating conditions and lubricant traction behavior at the ball/race contact, a reduction in the ball/cage pocket and cage/race guiding land clearances result in an increased frequency of collision both at the ball/cage and cage/race interfaces. The actual magnitudes of the collision forces are relatively insensitive to these clearances. The reduced clearances also lead to a coning motion of the cage and tight clearances at both the cage pocket and guiding land result in adverse cage/race interaction, when the cage is outer-race guided for the typical turbine engine bearing considered in the paper.

INTRODUCTION

Analytical performance simulation of rolling bearings has been a subject of considerable interest over the past decade. The simple quasi-static models of Jones (*1*), (*2*) have been replaced by dynamic models which solve the differential equations of motion of the bearing elements and provide a real time simulation of the bearing performance. The models presented by Walters (*3*) and Gupta (*4*), (*5*) have been well known and it has been fairly well demonstrated that such models and simulation techniques not only provide valuable guidance to experimental investigations for critical applications but they have proven to be effective in the actual design of rolling bearings. A parametric evaluation of the design parameters can be very effectively carried out by such analytical models and the predesign experimental investigations may only be necessary over a greatly reduced range of parameters as identified by the real-time performance simulations obtained analytically. Therefore, to a certain extent, the analytical simulations can help replace some of the extensive experimental investigations. The objective of this paper is to present a case study where the strength of the analytical models in such parametric evaluation is demonstrated.

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It is true that any dynamic model which solves a propagation problem often demands an appreciable computational resource. However, with the advent of modern high-speed computers and vector processors, such limitations are being gradually relaxed. Also, with the objective of optimizing the computing effort, the general-purpose computer codes are being updated for efficient treatment of specialized applications. The recent computer code RAPIDREB by Gupta (*6*) is an enhancement of the original DREB program (*4*), (*5*). RAPIDREB provides performance simulations for ball bearings both with solid and liquid lubrication. With certain constraints, it has been shown that performance simulations over several shaft revolutions can be obtained with acceptable computational effort (*6*).

This present paper investigates the performance of a solid-lubricated bearing as a function of the various design parameters as simulated by the RAPIDREB code. For a solid-lubricated ball bearing, the traction behavior of the lubricant at the ball/race interaction has been shown to be the most significant parameter in the bearing design (*7*). Similarly, the lubricant friction at the ball/cage and cage/race interactions has proven to be an important consideration. When the solid lubricant is supplied in the form of a transfer film formed at the ball/race interface by the material released from the cage due to the ball/cage collisions, there is an intricate coupling between the ball/cage and ball/race interactions. For example, the absence of lubricant at the ball/race interface may produce high accelerations on the ball which, depending on the cage pocket clearance, lead to ball/cage collisions which, in turn, release the lubricant and provide the transfer film at the ball/race interface. Such a mechanism is often the most fundamental element determining the overall performance of the bearing. Similarly, for bearings with race-guided cages, the interaction at the cage/race interface is also quite complex because the contact between the cage and the guiding land is essentially determined by the ball/cage collisions in the cage pockets, the inertia of the cage, and the operating clearance between the cage and the guiding land. Thus, there is a definite coupling between the tractive behavior of the solid lubricant and the geometry of the bearing and a realistic modelling of the ball/race, ball/cage, and cage/race interactions is crucial to a successful bearing design.

For a prescribed traction behavior of a solid lubricant, the present investigation considers the bearing performance as a function of the ball/cage and cage/race clearances using the RAPIDREB computer program (*6*) as the basic model for the various interactions in the bearing. It is expected that the results presented herein will provide some insight into the design of solid-lubricated ball bearings.

BEARING SPECIFICATIONS

The bearing selected for study is an extra light series, 006-size angular contact bearing. This is one of the main bearings which carry the thrust load in a high-speed turbine engine. The bearing dimensions are listed in Table 1.

The balls and races of the bearing are constructed of AMS 6430 (AISI/M-50) tool steel. The properties of this material are also listed in Table 1. The cage is a three-dimensional-weave graphite/polyimide composite which is impregnated with the solid lubricant. Properties of this material have not been measured at the application temperatures and the properties listed in Table 1 are only rough estimates.

The solid lubricant with which the cage material is assumed to be impregnated is Galium-Indium-Tungsten-di-Selinide (Ga-In-W-Se_2). Although no friction data for these materials at elevated temperature are available, some data at room temperature, Hertz stresses to 1 GPa and rolling speed of 2 m/s are available (*8*).

In order to realistically model the behavior of rolling-element bearings, it is essential that the friction forces between bearing elements be realistically modeled. The contacts between the balls and races of a bearing are characterized by relatively large loads and rolling velocities; the sliding velocities, although small in magnitude compared to the rolling velocities, are often subject to large variations. The contacts between the balls and the cage and between the cage and the race are similar in nature. They are characterized by loads which are intermittently applied and which often vary greatly in magnitude while the sliding velocities are large and relatively constant. As a result of these differences, the friction behavior at the ball/race interaction is quite different from that at the ball/cage or cage/race interface.

Based on the available room temperature data (*8*), the traction coefficient for the ball/race contacts is assumed to increase linearly with sliding velocity to a value of 0.15 at a sliding velocity of 8 cm/s. For sliding velocities larger than 8 cm/s, the friction coefficient is assumed to remain constant at 0.15. The variation of ball/race traction with applied load and rolling speed, if any, is not considered. The friction coefficient at the ball/cage and cage/race interface was assumed to remain constant at a value of 0.075 for all conditions.

The objective of the study conducted is to evaluate the effect of changes in cage geometry on bearing performance. A total of 32 runs, as indicated in Table 2, is made to evaluate the performance. All runs are made with a stationary outer race, an inner-race velocity of 63 500 rpm, an axial load of 450 N, and a radial load of 225 N. The radial load is assumed to be rotating with the inner race, to simulate the effects of rotor imbalance. These operating conditions are typical of the actual application.

Both the race/cage and ball/cage clearances are assumed to vary in the range of 0.6 to 0.2 mm and both outer- and inner-race guidance are considered. For the present bearing design which has an outer-race guided cage, the nominal values for these parameters are 0.3302 mm for cage/race diametral clearance and 0.4572 mm for ball/cage diametral clearance. This nominal case is listed as Run #3.2 in Table 2.

TABLE 1—GEOMETRY OF THE TEST BEARING

Bore	=	29.542 mm
Outside Diameter	=	47.00 mm
Ball Diameter	=	5.556 mm
Number of Balls	=	17
Pitch Diameter	=	38.51 mm
Contact Angle	=	18.60 degrees
Outer-Race Curvature Factor	=	0.52
Inner-Race Curvature Factor	=	0.54
Cage Outside Diameter	=	41.224 mm
Cage Inside Diameter	=	37.389 mm
Cage Width	=	8.191 mm
Type of Guidance	=	Outer
Diametral Cage/Race Clearance	=	0.330 mm
Cage/Ball Pocket Clearance	=	0.457 mm
Elastic Modulus for Balls and Races	=	2.0×10^{11} N/m^2
Elastic Modulus for Cage	=	1.73×10^{9} N/m^2
Poisson's Ratio for Balls and Races	=	0.25
Poisson's Ratio for Cage	=	0.25
Material Density for Balls and Races	=	7.75×10^{3} kgm/m^3
Material Density for Cage	=	1.5×10^{3} kgm/m^3

BEARING PERFORMANCE SIMULATIONS

The variation in ball race/contact load as the ball travels along its orbit is shown in Fig. 1. Since the radial load is rotating with the inner race, the time between peak loads is somewhat less than that required for the ball to make one complete orbit. The corresponding variations in the contact angles and spin/roll ratios are also shown in Fig. 1. It is seen that although the spin velocity is relatively large at the inner race, some spin does appear at the outer race. Thus, the conventional race-control hypotheses (*1*), (*2*) are not generally valid for the present conditions.

The cyclic variations in the ball/race contact loads and contact angles result in corresponding variations in the ball/race slip velocities. Fig. 2(a) shows the variation in slip at the center of the ball/race contact. It is interesting to see that the slip at the outer- and inner-race contacts is out of phase, though the loads are in phase. Thus, an increasing contact load does not necessarily reduce the slip and the bearing kinematics has a strong influence. Although this general pattern of slip is insensitive to the cage geometry, the ball/cage collisions do tend to alter the slip pattern, as

TABLE 2—SUMMARY OF THE PARAMETRIC VARIATION IN BEARING GEOMETRY

RUN NO.	GUIDANCE TYPE	CAGE/RACE CLEARANCE (mm)	POCKET CLEARANCE (mm)
1.1	Outer	0.60	0.60
1.2	Outer	0.60	0.4572
1.3	Outer	0.60	0.30
1.4	Outer	0.60	0.20
2.1	Outer	0.40	0.60
2.2	Outer	0.40	0.4572
2.3	Outer	0.40	0.30
2.4	Outer	0.40	0.20
3.1	Outer	0.3302	0.60
3.2	Outer	0.3302	0.4572
3.3	Outer	0.3302	0.30
3.4	Outer	0.3302	0.20
4.1	Outer	0.20	0.60
4.2	Outer	0.20	0.4572
4.3	Outer	0.20	0.10
4.4	Outer	0.20	0.20
5.1	Inner	0.60	0.60
5.2	Inner	0.60	0.4572
5.3	Inner	0.60	0.30
5.4	Inner	0.60	0.20
6.1	Inner	0.40	0.60
6.2	Inner	0.40	0.4572
6.3	Inner	0.40	0.30
6.4	Inner	0.40	0.20
7.1	Inner	0.3302	0.60
7.2	Inner	0.3302	0.4572
7.3	Inner	0.3302	0.30
7.4	Inner	0.3302	0.20
8.1	Inner	0.20	0.60
8.2	Inner	0.20	0.4572
8.3	Inner	0.20	0.30
8.4	Inner	0.20	0.20

shown in Fig. 2(b). The influence of cage geometry on the overall performance of the bearing is best studied by a parametric evaluation of the ball/cage pocket clearance and the cage/race clearance at the guiding land.

BALL/CAGE POCKET CLEARANCE

Typical ball/cage interactions are shown in Fig. 3, where the ball cage force is plotted as a function of time. The impact forces shown in Fig. 3 result from the retainer driving the rolling element. Generally, the retainer is driven by some rolling elements and, in turn, drives others. The magnitude of collision force may depend on the operating conditions, ball/race traction, and ball/cage pocket clearance. For the fixed operating conditions and the prescribed ball/race traction, it is found that although the magnitude of ball/cage impact force does increase with decreasing clearance, the effect is rather small over the range of clearances considered. However, a reduction in pocket clearance greatly increases the frequency of both the ball/cage and cage/race collisions. This is seen by comparing the two plots of cage/race force, in Fig. 4, which represent widely different pocket clearance but identical land clearance. Another interesting observation here is the fact that the cage/race forces at the two lands are not identical for the smaller cage pocket clearance. This essentially means that the reduced pocket clearance also leads to some coning or out of plane motion of the cage.

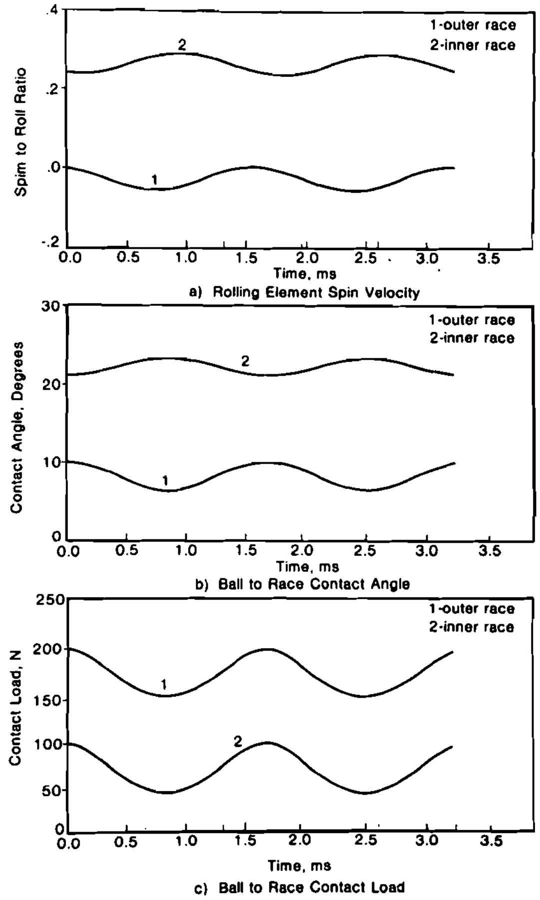
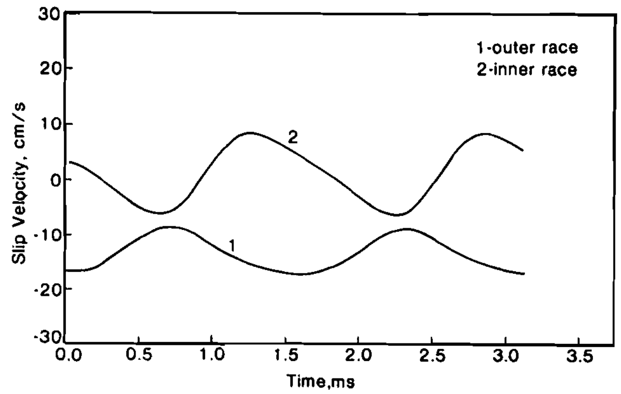


Fig. 1—Rolling-element motion, run 3.2 cage/race clearance = 0.3302 mm, pocket clearance = 0.4572 mm.

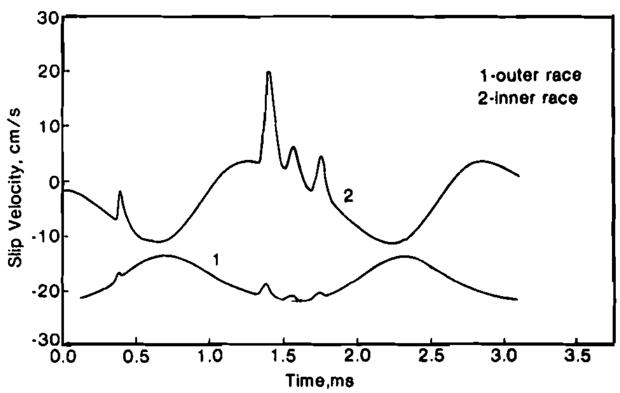
Figure 5 shows the variation in the total power loss of the bearing for the two pocket clearances. It is seen once again, that although the nominal power level is relatively unchanged, the reduced pocket clearance does show more frequent peaks corresponding to the ball/cage and cage/race collisions with the reduced clearance.

CAGE/RACE LAND CLEARANCE

The influence of cage/race land clearance is somewhat similar to that observed for the ball/cage pocket clearance. Figure 6 shows two plots for the cage/race variations with identical but small pocket clearance and widely different cage/race clearance. It is again seen that the frequency of collision increases with the reduced clearance but the dramatic effect is noted in the out of plane or coning motion of the cage, which is indicated by the difference in forces at the two lands in Fig. 6. Furthermore, the contact on one of the lands exists for a relatively large interval of time. This



a) Pocket Clearance = 0.4572mm



b) Pocket Clearance = 0.30mm

Fig. 2—Ball race slip velocity (a) run 3.2, (b) run 3.3 cage/race clearance = 0.3302 mm.

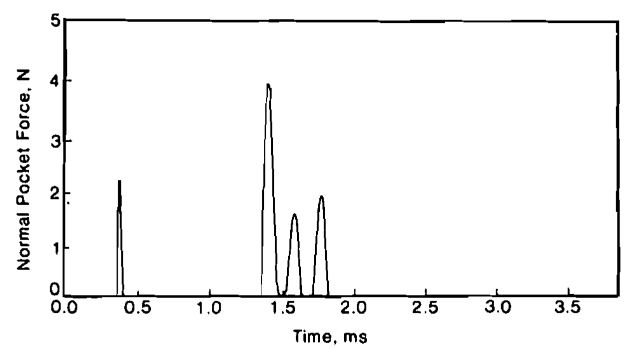


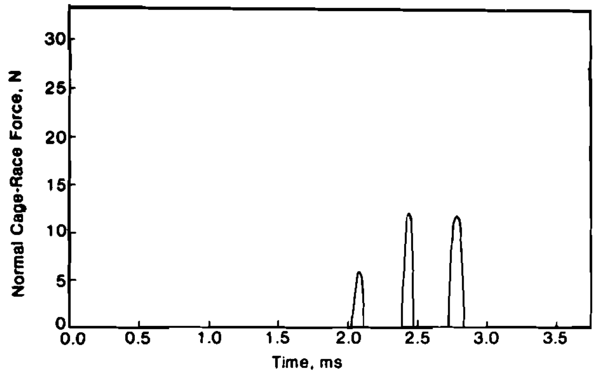
Fig. 3—Normal cage pocket impact force, run 3.3 cage/race clearance = 0.3302 mm, pocket clearance = 0.30 mm.

indicates a potential wear problem and certain instability in the motion of the cage.

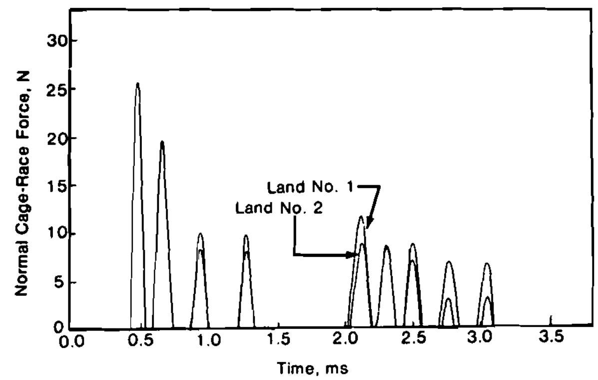
The cage mass center orbits corresponding to the cases shown in Fig. 6 are plotted in Figs. 7(a) and 7(b). It is also found that the mass center, angular whirl velocity of the cage increases with the reduced land clearance. This is roughly seen in Fig. 7, where the case with reduced clearance shows a larger number of orbits for the same amount of time.

OUTER- VS INNER-RACE GUIDANCE

From a dynamic standpoint, all the cases with outer- and inner-race guidance demonstrate similar results. Perhaps



a) Pocket Clearance = 0.60mm, Forces on the Two Lands are Identical



b) Pocket Clearance = 0.20mm, Forces on the Two Lands Tend to Deviate.

Fig. 4—Normal cage-race impact force (a) run 3.1 (b) run 3.4 cage/race clearance = 0.3302 mm.

the most difference in performance is seen in the case where both the ball/cage and cage/race clearances are small. Figure 8 shows the cage/race force variation for these tight clearances with the cage guided on inner race. When this is compared with the corresponding outer-race guided case in Fig. 6(b), it is seen that the steady contact with outer-race guidance is no longer present when the cage is guided on the inner race. However, the general magnitude of the collision forces is somewhat larger with the inner-race guidance. The cage coning seems to be somewhat reduced with inner-race guidance.

DISCUSSION

Based on the above results, it seems that, although the magnitude of the ball/cage and race/cage forces do not greatly change with the respective clearance variation in a practical range, the frequency of collision does demonstrate a notable change. This is an important finding because in order to maintain a certain thickness of the lubricant transfer film at the ball/race interface, a certain collision frequency may be required. This has to be further investigated by single-contact wear and transfer film experiments. Once the extent of required collisions is known, the present study indicates that an appropriate ball/cage pocket can be determined by a parametric evaluation similar to the one presented herein.

The influence of the ball/race traction on the ball/cage force has yet to be determined. Perhaps a stable configuration could be foreseen, where the reduction in transfer

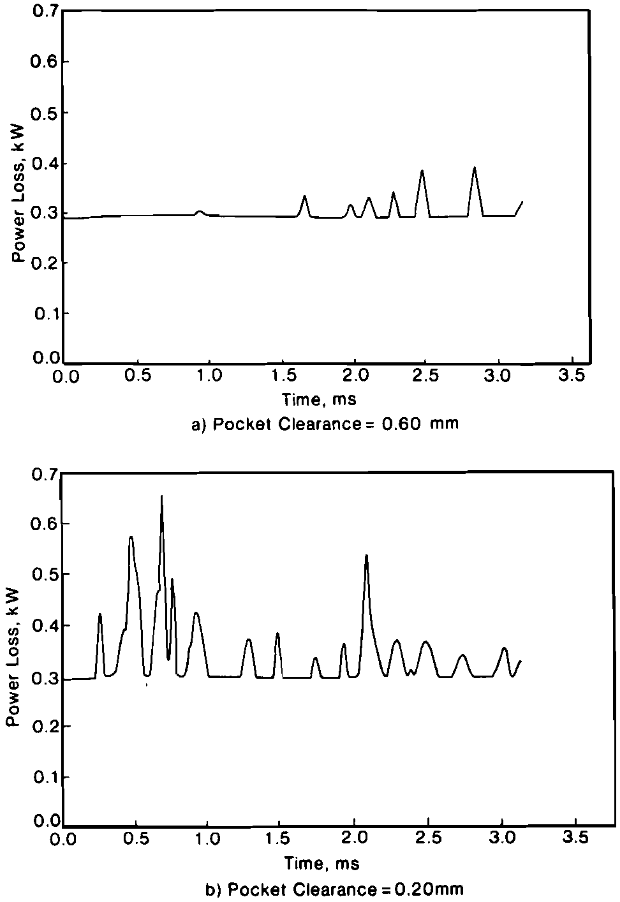


Fig. 5—Total bearing power loss (a) run 3.1 (b) run 3.4 cage/race clearance = 0.3302 mm.

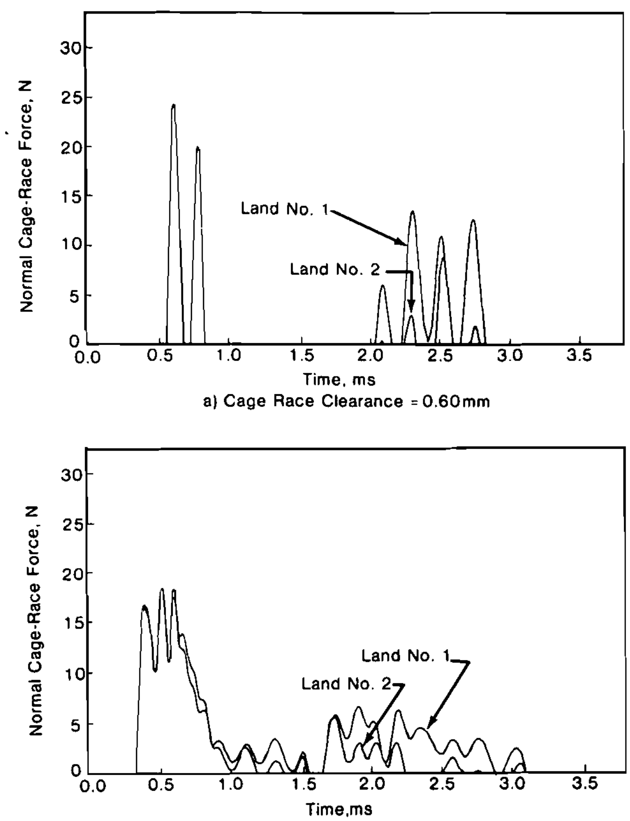


Fig. 6—Cage-race impact force (a) run 1.4 (b) run 4.4 pocket clearance = 0.20 mm.

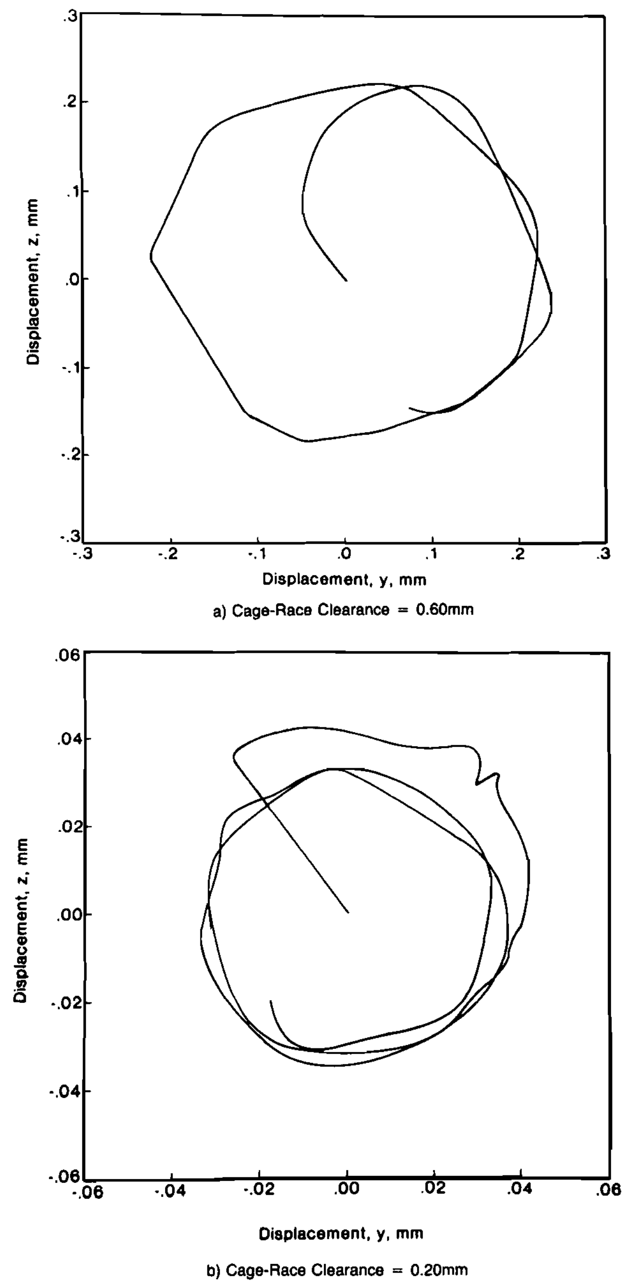


Fig. 7—Cage mass center orbit (a) run 1.4 (b) run 4.4 pocket clearance = 0.20 mm.

film will result in increased ball/race traction which, in turn, may result in an increase in the magnitude of ball/cage force and thereby ascertain increased flow of the lubricating material to maintain a required level of transfer film. This simply emphasizes a very close coupling between the material behavior, the operating conditions, and the optimum geometry of the bearing.

Although the results of outer- versus inner-race guidance did not show any dramatic effects, there are some practical considerations which should be carefully studied. For example, it is seen that for a tight clearance, outer-race guidance tends to result in a somewhat steady contact at the cage/race interface. This implies that increased heat will be generated at this interface. This heat may further reduce the operating clearance if the thermal coefficient of expansion of the cage material is larger than that of the outer race. Also, with the outer race stationary, the centrifugal

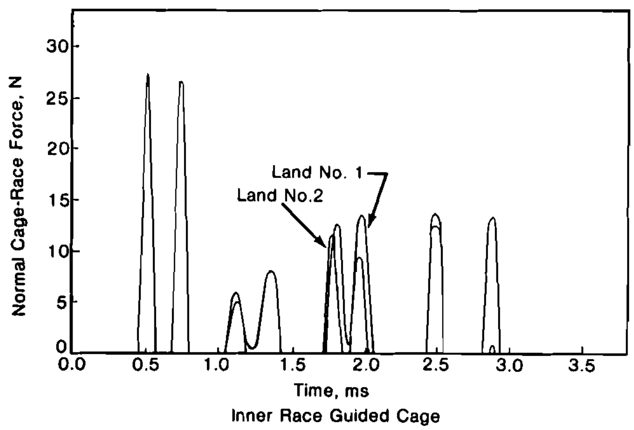


Fig. 8—Cage-race impact force run 8.4 cage/race clearance = 0.20 mm, pocket clearance = 0.20 mm.

expansion of the cage may further worsen the situation. In all, a catastrophic failure of the cage may soon be reached. However, if an acceptable operating clearance is assured, guidance on the stationary outer race may tend to wear the cage in such a manner so that the unbalance introduced by the ball pocket wear is compensated. This may be desirable if the balance of the cage is, indeed, a problem. On the other hand, the inner-race guidance, although possibly desirable for the conditions discussed herein, may produce rather erratic cage/race interactions when the two races are misaligned relative to each other. This will reemphasize the close connection between the material behavior, operating conditions, and optimum geometrical configuration of the bearing. A parametric evaluation of the type presented herein may, therefore, have a substantial design significance.

CONCLUSIONS

Based on strictly analytical simulations of the performance of a solid-lubricated ball bearing, the parametric evaluations lead to the following findings:

1. For prescribed operating conditions and traction behavior at the ball/race, ball/cage, and cage/race interfaces, the reduction in ball/cage pocket and the cage/race guiding land clearances lead to an increased frequency of collisions both in the cage pocket and at the guiding land. However, the magnitude of the collision forces do not vary greatly with the clearance change in the range considered.
2. Increased coning or out-of-plane motion of the cage is simulated with a reduction in both the ball/cage and cage/race clearances. For the case where both clearances are small, the interaction between the cage and the guiding land is adverse and a catastrophic cage failure is indicated, especially for the outer-race guided cage.
3. If outer-race guidance is desirable from certain practical considerations, the present studies indicate that a safe operating clearance will have to be ascertained after allowing for relative thermal growth and centrifugal expansion of the cage.

ACKNOWLEDGMENTS

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DISCUSSION

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The value of an efficient ball bearing dynamic analysis is well stated by the authors. Practical applications of a full dynamic analysis, with complex time-dependent interactions between each bearing element, have so far been limited in scope, largely because of excessive computer time requirements. The development of RAPIDREB addresses this aspect of the problem. A number of potential time-saving improvements have been identified by this discusser following review of the original DREB program [authors' Ref. (5)]. They include interpolation between tabulated Hertzian contact solutions, rather than calculating elliptic integrals iteratively at each time step, a predictor-corrector method to reduce the number of derivatives required in integrating the equations of motion and approximate correlation formulae for the traction integrals, rather than multiple evaluation at each contact surface. These suggested improvements may be of interest to the authors insofar as they compare with the constraints imposed on RAPIDREB. The need for improved data correlations still remains for both oil-lubricated and solid-lubricated bearings.

The analytical results presented in this paper are impressive but some of the interpretations are not immediately obvious. The authors' Fig. 1(c) shows a period of 1.7 ms between peak ball/race contact loads. This is the same result that would be obtained from simple kinematics, assuming pure rolling with no cage slip, no ball/cage or cage/land interactions and no traction. Furthermore, Fig. 1(a) and

2(a), respectively, show that the outer-race contact spin velocity and both inner- and outer-race contact slip velocities are very small in comparison to their rolling velocities. These dynamic results tend to support the conventional simplified quasistatic raceway control theory [authors' Ref. (1), (2)].

The significance of the dynamic analysis lies in its coupling of the bearing element interactions. The resulting out-of-phase slip velocity phenomenon shown in Fig. 2(a) deserves further investigation, particularly for operating conditions with higher magnitudes of slip than occur here. Transferring the solid lubricant from the cage to the balls is clearly the most important mechanism in the subject design. As the authors have noted, the absence of lubricant at the ball/race contact may produce high ball acceleration into the cage pocket. Was this effect confirmed in the model, for example by modifying the traction coefficient? It is stated that a reduction in ball/cage pocket clearance increases the frequency of collisions between the ball and cage, thereby enhancing the lubricant transfer mechanism. Comparisons showing this effect would be of interest, including any differences between outer- and inner-ring guidance of the cage.

It is not immediately apparent how the cage motion depicted in the authors' Fig. 7 corresponds to the cage/land collision forces shown in Fig. 6. The cage mass center orbits, which exhibit typical behavior due to cage or shaft imbalance, are at eccentricities of approximately 0.8 for the higher clearance and 0.4 for the lower clearance. Collision would seem to require eccentricities approaching 1.0. While oil lubrication between the cage and land would generate a hydrodynamic force corresponding to the minimum oil film thickness, no such mechanism is apparent with solid lubrication. Is there an additional force, besides the ball/cage pocket friction, which resists cage centrifugal motion?

In their discussion on the cage/land interface, the authors identify a steadier contact with lower clearance, based solely on Fig. 6. The cumulative wear effect of the contact force depends also on its circumferential location on the cage surface which depends, in turn, on the cage rotational and whirl speeds. The potential wear problem and cage motion instability quoted in the paper are not reflected in Fig. 7. The authors' comments on this would be welcome.

The authors are encouraged to expand their worthwhile efforts on ball bearing dynamic analysis. Further correlation of the model is essential and it is suggested that the effects of operating clearance, fits, temperatures, heat balance, and wear be included to further enhance the program as a valuable and practical design tool.

DISCUSSION

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The authors discuss a model, proposed for ball bearing design, applicable to one method of solid lubrication. This method is dependent on the cage or separator as a source of transfer films to critical wear zones in the bearing. The cage material described in this paper is a graphite/polyimide composite and assumed to contain a mixture of Ga-In-W-Se_2. Several comments and questions concerning the proposed model are outlined below:

Since the cage is the sole source of lubrication in the bearing design being considered by the authors, it is extremely important to develop precise information for the cage composition and the mechanical/physical properties. Mechanisms of film transfer will depend not only on the collision dynamics but the composition being transferred and subsequent attachment of that composition to the metal surfaces. Based on the cage components, being assumed, the composition of lubricating film transferred could vary significantly as a function of temperature.

1. What operating interval/performance time and temperature range is being represented by the proposed model?
2. Since no solid lubricant exists at the ball-race contact zone during bearing start-up, would some surface damage be anticipated? Does the model include the impact of any initial metal-to-metal contact?
3. What factors are represented in a complete description of the model?
4. In preparing the model, did the authors consider design factors identified in previously reported research, e.g., Kroll and Devine reported data, for solid-lubricated ball bearings equipped with graphite/polyimide cages? The latter research focused (1) inner and outer land guided cage configurations, (2) one-piece and two-piece cages, and (3) the importance of the cage/land contact zones.

Clearances involving the ball/cage pocket and cage/land components are recognized as important in determining satisfactory dimensions for the bearing assembly, however, there are a number of other significant considerations.

Additional questions on some of these considerations are as follows:

5. Are impact strength and deformation under load being examined for the cage?
6. Is bearing run-in being conducted under conditions of no-load and low speed?
7. Did the authors select a one-piece or two-piece cage? If two-piece, what was the method of attachment?
8. What is the minimum thickness for transfer films on raceway, ball and land areas to achieve effective bearing lubrication? What operating time interval is required to produce the minimum film thickness for the solid lubricant selected for this study?

Solid film transfer from cage to race is a two-stage process compared to a single-stage process for the cage/land area. Further, one process involves sliding and the other a combination of sliding and rolling.

9. Are simultaneous and equivalent solid film transfer rates basic requirements for achieving the friction coefficients selected by the authors? If so, based on the differences in transfer processes, what degree of validity exists in the assumption?
10. What assessment is provided by the authors for surface chemistry effects including chemical species formed on the surface in the *fundamental mechanism* cited on page 2 of the paper?

The authors are to be congratulated for their study of an extremely complex and important problem. It should be recognized that only one approach to solid lubrication of ball bearings has been examined by this study. Several other techniques have been successfully demonstrated for long-term ball bearing lubrication based on solids. Further modeling studies, to include such techniques, would provide a more comprehensive design data base.

DISCUSSION

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We would like to congratulate the authors for the interesting work. Some of our thoughts after reading the paper are:

1. We feel it would have been better if the model was checked with experimental data first and evaluated the validity of assumptions made in the model. If quantitative data are not available, a qualitative evaluation should be made. The model neglects aerodynamic, thermal effects, as well as wear of the dry lubricants. At such a speed, the aerodynamic effect may be important. With some of these effects incorporated, a different conclusion may have been drawn—instead of Conclusion No.2.
   We have heard of a case where a solid-lubricated ball bearing with small clearances with the outer race of guided cage was successfully run for a relatively long time.
2. From the figures shown, especially Figs. 4, 5, and 6, it is not clear that the calculations were carried out long enough to draw any conclusions. In Fig. 4, the contact force for the case of a larger clearance is increasing and for the case of a smaller clearance is decreasing. The same thing can be said of the power loss from Fig. 5.
3. Authors concluded from Fig. 7 that the mass center, angular whirl velocity of the cage increases with reduced land clearance. The reason is that the case shows a large number of orbits for the same amount of time. However, the figure also shows much less displacements for that case. By calculating the total amount of distance, the mass center travels during the time period, its whirl velocity of the cage with the reduced land clearance is much smaller (less than one-third) than the other case.

In conclusion, we strongly feel the danger of using a model, not verified with experimental data, as a design tool and performing parametric evaluations.

AUTHORS' CLOSURE

The authors are grateful to the discussers for their interesting comments and questions.

The required computational effort often restricts the use of sophisticated computer programs. A solution to this problem has been attempted in RAPIDREB and many of the simplifications, discussed by Mr. Barnsby, including a Predictor-Corrector algorithm, have been incorporated in this program. The required computer time when using RAPIDREB is significantly less than that required by the original DREB program. Some of the current ongoing work in this area is expected to further increase the computational efficiency and, hopefully, the practical utility of computer tools, such as DREB and RAPIDREB, will be significantly enhanced in the near future.

When the traction coefficients at the ball/race interface are high, the conventional race control hypotheses are quite reasonable as observed by Mr. Barnsby. Furthermore, the overall kinematics of the bearing is quite simple due to very low sliding velocities at the ball/race contacts. It is for this reason that the results shown in Fig. 1 can be easily interpreted in terms of simple kinematics.

Coupling between the bearing geometry, such as ball/cage and cage/race clearances, the solid-lubricant behavior is an important consideration and it requires extensive work both in the areas of materials development and computer modelling of bearing performance. As pointed out by Mr. Devine, materials research addressing the question of lubricant transfer film thickness is, indeed, a key element in solid-lubricated bearing development and the need for such research cannot be underestimated. The present investigation simply assumes a prescribed traction-slip relationship and it only emphasizes computer modelling of bearing performance. In response to some of the specific questions raised by Messrs. Devine and Barnsby, it should be noted that the discussion in the paper on the influence of lubricant traction and resulting ball motion is only a speculation at this point and extensive parametric evaluation is necessary before such a finding can be thoroughly substantiated.

In a solid-lubricated bearing, the motion of the cage is only influenced by the ball/cage and cage/race interactions and any hydrodynamic effects present in the case of an oil-lubricated bearing are absent. However, at very high speeds, the relative centrifugal expansion of the cage and race is a significant factor in determining the overall cage/race interaction. After a proper account for the change in operating clearance, it is found that the eccentricity, indeed, approaches 1.0, as suggested by Mr. Barnsby, when the cage/race collision forces of Fig. 6 are generated. The problem of cage wear in the case of excessive cage/race interaction can be significant when the contact forces at cage/race interface are large and they are exerted over large time intervals. Also, as pointed out by Mr. Barnsby, the circumferential location of the contact on the cage surface is important. Although the results obtained in the present investigation are not sufficient to provide a detailed wear distribution on the cage surface, they do suggest a continued cage/race contact and excessive wear under certain conditions.

The authors completely agree with Dr. Kubo to the fact that experimental validation of all analytical models is necessary before they can be comfortably used for design purposes. Recently, the authors have carried out such validations for an oil-lubricated engine bearing and RAPIDREB predictions are shown to be in fair agreement with experimental observations. This work will soon be published.

It is true that the extraction of a steady-state behavior is often difficult from the limited transient solutions, particularly for the ball/cage and cage/race interactions, as observed by Dr. Kubo. The authors have found that a change in time-averaged ball/cage or cage/race interaction provides a better indication for steady-state conditions rather than the examination of individual peaks in the collision forces. Such a rationale is used in deriving the conclusions, presented in the paper, from the computer simulations.

The increased number of cage mass center oribits, in a fixed time, with reduced land clearance do indicate an increased angular velocity of cage whirl. Perhaps, Dr. Kubo is confusing this observation with the reduction in the linear velocity of cage mass center over the smaller orbit size.

Finally, in response to the questions raised by Mr. Devine, the authors would like to reiterate that the present investigation was aimed only at modelling the behavior of a solid-lubricated bearing with the use of the computer models developed earlier (4), (5) and (6). A one-piece cage is used and it is assumed that the cage shall serve as a reservoir for the solid lubricant and this is, indeed, only one possible mode of solid lubrication. None of the materials factors and the impact of various thermal and chemical phenomena are considered in this paper; however, the authors agree that such factors are significant and considerable materials research is necessary for the development of solid-lubricated rolling bearings.