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Ball Bearing Response to Cage Unbalance

Motion of the cage in a high-speed angular contact ball bearing is experimentally investigated as a function of prescribed unbalance, up to operating speeds corresponding to three million DN. The predictions of cage motion made by the recently developed computer model, ADORE, are validated in the light of the experimental data. It is shown the cage whirl velocity is essentially equal to its angular velocity at all levels of unbalance and over a wide range of operating conditions. For the inner race guided turbine engine bearing, the cage/race interaction takes place directly opposite to the location of the unbalance and the severity of the interaction increases with the level of unbalance and the operating speed. ADORE predictions, over the entire range of unbalance and bearing operating conditions, are in very good agreement with the experimental observations.

Introduction

Modeling of the general motion of bearing elements in a rolling bearing has been a subject of considerable interest over the past two decades. The feasibility of truly dynamic models, which integrate the differential equations of motion of the bearing elements to simulate the overall dynamic performance of the bearing, has been well demonstrated for a wide range of applications. In fact, for an acceptable design for critical applications, where the instability of the cage and rolling elements are primary factors, such dynamic modeling has proven to be, perhaps, the only analytical approach. The first analytical formulation of the time transient problem for a ball bearing is credited to Walters [1]. With holonomic constraints for ball motion, the motion of the cage was considered in a complete six-degrees-of-freedom system. A more generalized model for both ball and cylindrical rolling bearings, along with the computer program DREB, was presented by Gupta [2]. The work of Brown et al. [3] and Conry [4] has also been noted for the dynamic modeling of roller bearings.

With the proven significance of dynamic models, the computational effort required in solving the propagation problem was very quickly realized as a problem of significant practical relevance. Gupta [5], after correlating the time step size to very high frequency ball/race vibrations, introduced certain equilibrium constraints to suppress this high frequency vibration which resulted in substantially large step sizes and thus bearing performance simulation over several shaft revolutions became permissible; this led to the computer program RAPIDREB. With further emphasis on the reduction of computational effort Meeks and Ng [6] have proposed a simplified model for cage motion when the ball motion is fully constrained in a thrust loaded ball bearing. More recently, Gupta [7], while introducing the computer program ADORE, has extensively refined both the local geometric interactions between the bearing elements and the numerical algorithms which integrate the equations of motion. In addition to efficient numerical considerations, the refinement of geometric interactions in ADORE allows for a treatment of arbitrary variations in the geometry of bearing elements. Thus the influence of both geometrical and inertial imperfections, on the dynamic performance of the bearing can be modeled, perhaps, for the first time.

Along with a solution to the computer time problem, experimental validation of the dynamic models is essential before the models may be used as design tools with acceptable level of confidence. In view of the intricacies associated with the modeling of geometric interaction and the resulting motion of bearing elements, it is necessary that the model be validated in terms of the fundamental motion of bearing elements and validation on the gross level of overall bearing performance may not be sufficient. With the understanding of the required experimental validation at such subtle levels, Gupta et al. [8], carried out an experimental investigation to measure the motion of the cage in a high-speed ball bearings and validate the predictions of the computer models DREB and RAPIDREB. Later this work has been used to validate ADORE [7]. Validation of ADORE, on a somewhat gross level, has also been recently reported by Bandow et al. [9].

Validation of ADORE for some of the advanced modeling capabilities, such as the influence of inertial imperfections, is the subject of this investigation. The experimental facility used earlier by Gupta, Dill and Bandow [8] is used to measure

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Table 1 Cage Geometry

Outside diameter	= 0.146228	M
Inside diameter	= 0.129146	M
Width	= 36.2410	mm
Outer clearance	= 4.8000	mm
Inner clearance	= 0.520700	mm
Pocket clearance	= 0.685800	mm

Table 2 Operating conditions

Applied loads:		
	Set 1	Set 2
Thrust	4448 N	4448 N
Radial	0	2224 N
Operating speed:	5,000 to 30,000 rpm	
Lubricant:	MIL-L-7808	
Lubricant temperature:	92 C	
Cage unbalance:	3 to 20 gm-cm	

the components of cage motion as a function of prescribed unbalance. ADORE predictions are then compared with the experimental data. It is expected that such a validation will enhance the practical significance of sophisticated dynamic models for the design and performance diagnosis of rolling bearings in some of the most critical and advanced applications.

Experimental

Extensive details of the test rig and the available instrumentation have been presented earlier by Gupta, Dill and Bandow [8]. The motion of the cage is essentially measured by radial displacement probes in two planes; thus, the whirl and any out of plane coning can be derived. The radial displacements are analyzed to determine the whirl velocities and the shape of whirl orbits.

A 100-mm angular contact ball bearing with an inner race guided cage, typical of high-speed turbine engine application is used as a test bearing. The geometrical details of the bearing are identical to those outlined earlier [8], except for the geometry of the cage, which is summarized in Table 1. On the inner side of the cage rims threaded holes for small set screws are symmetrically machined between the pockets; the actual position of these holes will become more clear while discussing the results. The cage is then balanced to approximately one gm-cm. In order to create any prescribed unbalance a definite number of set screws may be installed in the cage rims.

The operating conditions for the experimental program are summarized in Table 2. The indicated levels of unbalance are accomplished by symmetrically installing two to ten set screws in the cage rims. The output cage displacement data from the radial probes is first recorded on an FM tape recorder and later digitized for subsequent analysis. Appropriate calibrations are used to determine the various components of cage motion from the probe signals. The guide lands on the cage are examined after the tests for the extent of wear and the position of cage/race interaction relative to the unbalance.

The widely used military lubricant, with the MIL-L-7808 specification, was used to lubricate the bearing. The quantity of the lubricant was kept to a minimum in order to eliminate any excessive churning effects. The oil temperature was maintained at approximately 92°C.

The experimental results under the conditions of both a pure thrust load and a combined thrust and radial load on the bearing were closely identical. The cage invariably whirls in an almost circular orbit at all operating speeds above 10,000 rpm. The radius of the whirl orbit corresponds to the guide land clearance, which implies an almost continued contact of the cage/race interface. A typical whirl orbit is shown in Fig. 1. The spectral decomposition of the radial displacement data

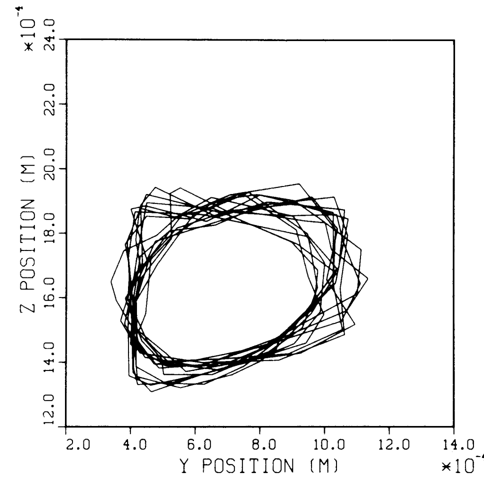

Fig. 1 Experimentally derived cage whirl orbit at 20,000 rpm shaft speed and with 17 gm-cm unbalance in cage

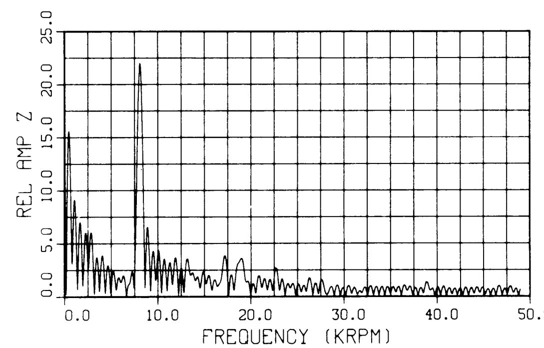

Fig. 2 Spectral decomposition of the radial probe data at a shaft speed of 20,000 rpm and with a cage unbalance of 17 gm-cm

clearly shows the dominant whirl frequency. Figure 2 shows such data at a shaft speed of 20,000 rpm. The high peak near 9000 rpm corresponds to the cage angular velocity. Thus the average whirl velocity is expected to be closely equal to its angular velocity. Such an observation holds under all operating conditions and at all levels of unbalance.

Examination of the cage at the end of the experiments revealed that at all levels of unbalance the contact at the cage/race interface took place directly opposite, or 180 degrees, from the position of the unbalance. The contact and the resulting damage on the cage surface became increasingly severe with the increase of both the operating speed and level of unbalance. Figure 3 shows typical damage on the cage surface after the test. This figure also shows the position of the machined holes for the set screws, which are used to create a prescribed unbalance, as discussed earlier.

Based on the phase relationship between the radial probe data in two parallel planes located on either side of the rolling elements, it is concluded that the cage demonstrates no noticeable out of plane coning. This may be expected with a symmetrical unbalance.

Computer Modeling of Cage Motion

The computer program, ADORE [7], is used to obtain the analytical simulation of cage motion corresponding to the experimental operating conditions and levels of unbalance. A combined thrust (4448 N) and radial (2224 N) is used in analytical solutions discussed in this paper. Also, the lubricant traction model conforms to the behavior of the MIL-L-7808 type lubricant, as discussed elsewhere [8].

The greatly refined numerical algorithms in ADORE easily

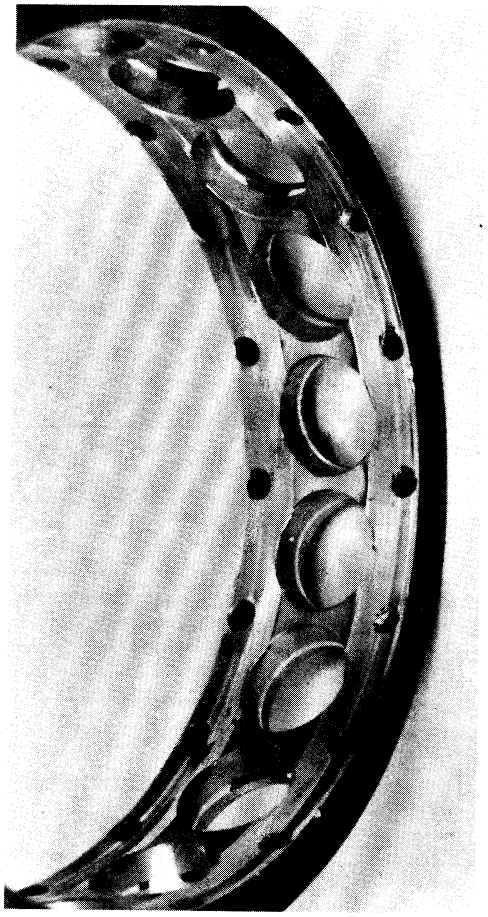

Fig. 3 Typical damage on cage guide lands after test. The damage is symmetrical about a point located directly opposite to the position of unbalance.

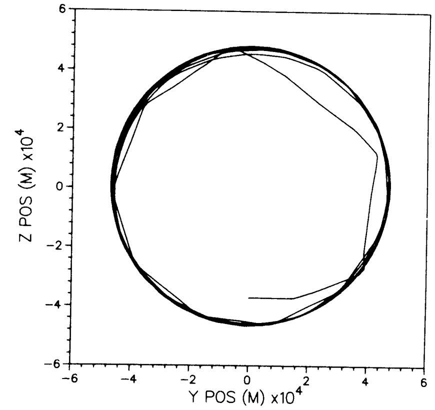

Fig. 4 Cage whirl orbit as simulated by ADORE at a shaft speed of 20,000 rpm and with a cage unbalance of 17 gm-cm

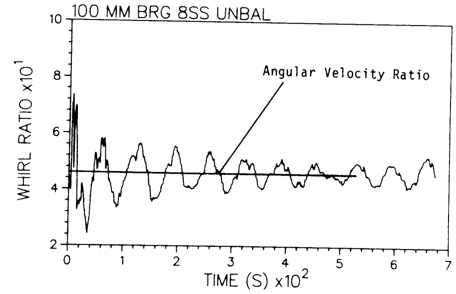

Fig. 5(*a*) Cage unbalance of 17 gm-cm

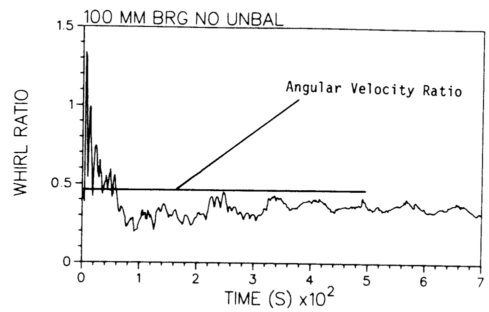

Fig. 5(*b*) No unbalance in cage

Fig. 5 Computer simulation of cage whirl at a shaft speed of 20,000 rpm; the whirl ratio is defined as a ratio of cage mass center whirl velocity to the shaft velocity

provide bearing performance simulation over a relatively large number of shaft revolutions within reasonable computing effort. In the present investigation a CDC Cyber 750 computer was used and a typical run, which corresponds to performance simulation over twenty shaft revolutions, required approximately one hour of computer time. Such computing efficiency permits a large number of runs over the range of operating conditions and a clear evaluation of the steady state motion as a function of the various parameters becomes possible.

Typical cage whirl orbit, as simulated by ADORE over the range of experimental conditions, is shown in Fig. 4. The orbit is circular and the radius corresponds to the operating guide clearance, which implies steady state cage/race contact, as observed experimentally. While comparing both the shape and size of the whirl orbit with experimental data, possible uncertainties in the probe calibration must be recognized. These uncertainties, in particular the expected influence of thermal effects, will possibly account for the differences between the experimental and predicted whirl orbits.

The predicted average whirl velocity at all levels of unbalance is closely equal to the cage angular velocity. Figure 5(*a*) shows the simulation of cage whirl velocity, plotted as a ratio to the shaft speed, at 20,000 rpm with a cage unbalance of 17 gm-cm. For comparison purposes, Fig. 5(*b*) shows the corresponding solution with no unbalance. It is seen that, although with ideal balance of the cage the steady state whirl velocity is slightly less than the angular velocity, it is equal to the angular velocity with an unbalanced cage, as observed experimentally. This finding holds at all levels of unbalance and Fig. 5(*a*) is representative of the large number of solutions

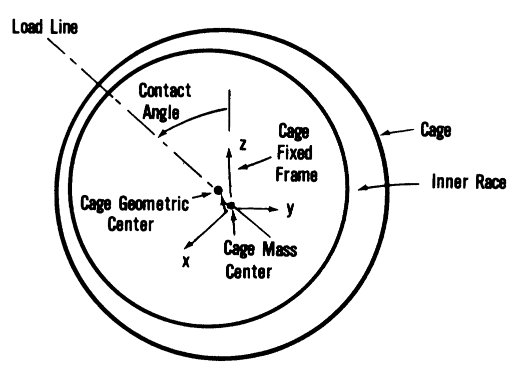

Fig. 6 The definition of cage/race contact angle

obtained over the range of experimental conditions. The frequency of the cyclic variation in the whirl, as seen in Fig. 5(*a*), also corresponds to the cage angular velocity. The very high frequency and small amplitude peaks in the whirl velocity correspond to the ball/cage and cage/race collisions.

In order to precisely locate the position of cage/race

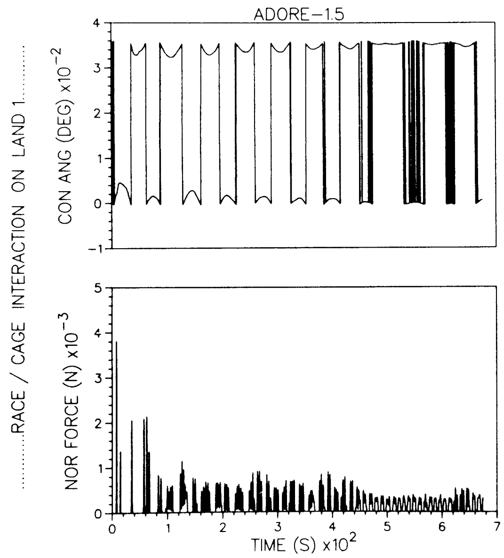

Fig. 7 Cage/race interaction as simulated by ADORE at a shaft speed of 20,000 rpm and with a cage unbalance of 17 gm-cm

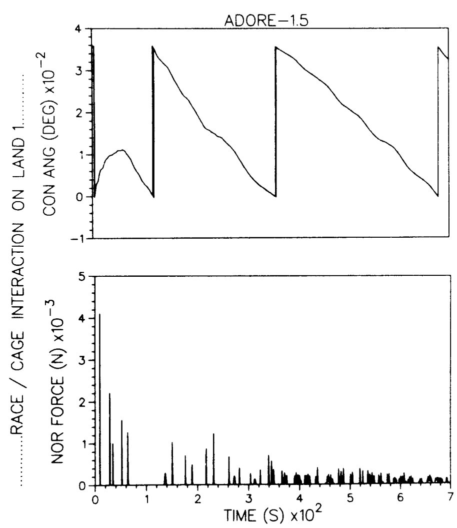

Fig. 8 Cage/race interaction at a shaft speed of 20,000 rpm with no cage unbalance

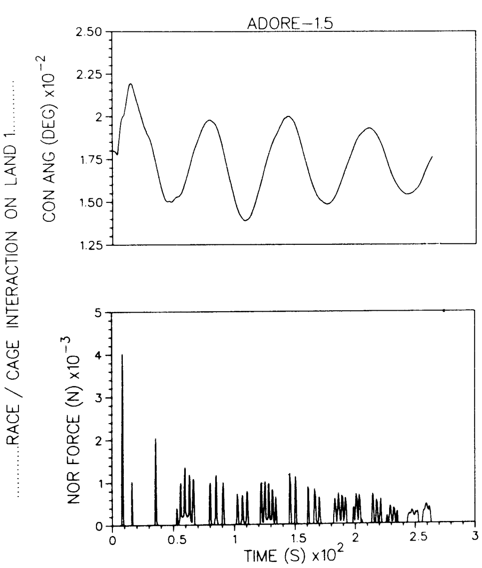

Fig. 9 Cage/race interaction at a shaft speed of 20,000 rpm, cage unbalance of 17 gm-cm, and with an outer race guided cage

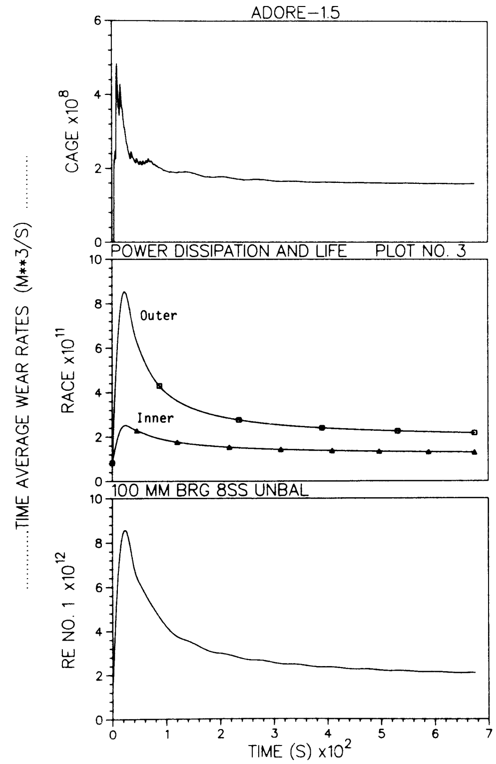

Fig. 10 The time-averaged wear rates as simulated by ADORE at a shaft speed of 20,000 rpm and with a cage unbalance of 17 gm-cm

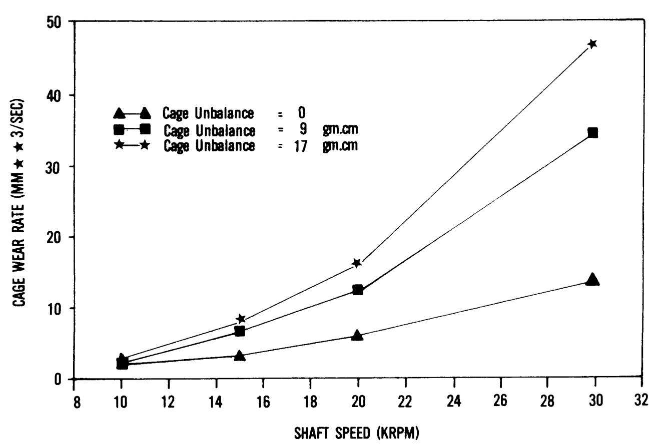

Fig. 11 ADORE predictions of time-averaged cage wear rates as a function of operating speed and cage unbalance

contact, ADORE output is slightly modified to define the contact angle in a cage fixed reference frame, as shown in Fig. 6. In the simulations discussed later in the paper, the cage geometric center is located on the positive z axis; thus a contact angle of zero degrees implies a contact directly opposite to the position of unbalance.

Figure 7 is typical of the cage/race interaction with an unbalanced cage. The contact load, plotted in the lower part of the figure, generally increases with increasing speed and unbalance. The contact angle, clearly, varies slightly around the zero degree mean value, as observed experimentally; the jump in contact angle from 0 to 360 degree in Fig. 7 should be ignored because this is just a result of the scale used in the plot and 0 and 360 degree imply same contact point on the cage. For comparison, Fig. 8 shows the cage/race interaction with a balanced cage. Note the two differences, the contact angle now varies continuously between 0 and 360 degrees, which implies contact on the entire cage surface, and the cage/race contact forces are somewhat lower in steady state, as may be expected.

The cage/race contact directly opposite to the position of unbalance, implies that the unbalance in the cage will get worse as the cage wears out, which will further increase the severity of cage/race interaction; thus a form of instability may be eminent for the conditions under consideration. Such a nature of the cage/race interaction for the inner race guided cage considered in the present investigation, may be explained in terms of the centrifugal force exerted on the cage as a result of unbalance; due to the centrifugal force as the cage tends to move out radially, it contacts the guiding inner race at a point directly opposite to the location of unbalance. Based on such an argument, it may be expected that the cage/race contact, for an outer race guided cage, will be in phase with the unbalance. This is confirmed by ADORE predictions obtained with the cage/race clearances at the outer and inner races switched to simulate outer race guidance. The result is shown in Fig. 9, where the contact angle varies around an average value close to 180 degrees. Thus with outer race guidance the cage wear as a result of unbalance leads to a reduction in unbalance. In other words, the cage is "self-balancing."

In order to correlate the overall cage interactions with the operating speed and level of unbalance, the time-averaged wear rates, as simulated by ADORE are quite useful. Since the forces acting on the cage in the various ball pockets and at the guiding lands are highly dynamic in nature, a time-averaged wear rate, $W(T)$, is defined at any time, T, as

$$W(T) = \frac{K}{TH} \int_0^T Q(t)\, V(t)\, dt$$

where K is the Archard-type wear coefficient, H is the hardness of the material, $Q(t)$ and $V(t)$ are, respectively, the contact load and sliding speed at any interaction at time, t. Similar definition applies also to the wear of the balls and the races in the bearing.

For the present study where the balls and races are made of conventional bearing steel and the cage is fabricated of a relatively softer steel, the wear coefficients for the balls, races and cage are arbitrarily assumed as 0.0000050, 0.0000050, and 0.00050, respectively. It should be noted, however, that the wear coefficient appears simply as a scale factor and all the predicted trends of the wear rates are independent of the actual values of the wear coefficients.

Typical simulations of the wear rates are shown in Fig. 10; the cage wear rate shown includes the wear at both guide lands and in all ball pockets. From the results obtained over the range of the experimental operating conditions and the level of cage unbalance, ADORE predictions demonstrate that the time-averaged cage wear rate increases with both the increasing operating speed and level of cage unbalance. The results derived from a number of computer runs are summarized in Fig. 11. Such variations in cage wear rates are in good qualitative agreement with the experimentally observed variations in cage damage as a function of operating speed and unbalance in the cage.

The predicted ball and race wear rates are rather insensitive to the level of cage unbalance. This may be due to the fact that the ball pocket clearance in the bearing is adequate to accommodate all the irregularities in the cage motion resulting from unbalance. Perhaps, with a tighter clearance the cage unbalance may affect the cage interactions and the resulting ball and race wear rate.

Conclusions

Based on both the experimental results and analytical predictions made by the computer program, ADORE, the present investigation leads to the following three conclusions:

1. The cage whirl speed is essentially equal to the cage angular velocity at all levels of cage unbalance and throughout the investigated range of operating conditions, which include speeds corresponding to three million DN.
2. For the inner race guided cage, as used in the test bearing, the cage/race contact takes place directly oppposite to the location of the unbalance. Thus the unbalance gets worse as the wear on the guide surface progresses. For an outer race guided cage, however, the analytical results show that the cage/race contact is in phase with the location of unbalance. This leads to a reducing unbalance with the increasing wear of the cage.
3. The time-averaged cage wear rates increase with both increasing operating speed and unbalance in the cage.

Over the entire range of operating conditions and the cage unbalance level, the analytical predictions of cage motion, provided by the computer program ADORE, are in very good agreement with the experimental observations.

Acknowledgments

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References

1 Walter, C. T., "The Dynamics of Ball Bearings," ASME Journal of Lubrication Technology, Vol. 93, 1971, pp. 1-10.

2 Gupta, P. K., "Dynamics of Rolling Element Bearings, Parts I to IV," ASME Journal of Lubrication Technology, Vol. 101, 1979, pp. 293-326.

3 Brown, P. F., Carrano, M. J., Dobek, L. J., McFadden, R. J., Miner, J. R., and Robinson, J. D., "Main Shaft High-Speed Cylindrical Roller Bearings for Gas Turbine Engines, Parts I to IV," Report NAPC-PE-60C, prepared for Naval Air Propulsion Center, Trenton, New Jersey and Air Force Aero Propulsion Laboratory, Wright-Patterson Air Force Base, Ohio, 1980.

4 Conry, T. F., "Transient Dynamic Analysis of High-Speed Lightly Loaded Cylindrical Roller Bearings, Parts I & II," NASA Technical Reports 334 and 335, NASA-Lewis Research Center, Cleveland, Ohio, 1981.

5 Gupta, P. K., "Simulation of Low Frequency Components in the Dynamic Response of a Ball Bearing," Advances in Computer-Aided Bearing Design, *Proc. 1982 ASLE/ASME Lubrication Conference*, ASME Publication, 1982.

6 Meeks, C. R., and Ng, K. O., "The Dynamics of Ball Separators in Ball Bearings, Part I—Analysis," Presented at the 1984 ASLE Annual Meeting, Chicago, Illinois, 7-10 May 1984, to be published in *ASLE Transactions*.

7 Gupta, P. K., *Advanced Dynamics of Rolling Elements*, Springer-Verlag, 1984.

8 Gupta, P. K., Dill, J. F., and Bandow, H. E., "Dynamics of Rolling Element Bearings—Experimental Validation of the DREB and RAPIDREB Computer Programs," ASME Journal of Tribology, Vol. 107, 1985, pp. 132-137.

9 Bandow, H. E., Gray, S., and Gupta, P. K., "Performance Simulation of a Solid Lubricated Bearing," Presented at the 1985 ASLE Annual Meeting, Las Vegas, Nevada, May 1985, to be published in *ASLE Transactions*.

DISCUSSION

E. Atkinson

The application of a more computer efficient dynamic analysis to a potentially costly real problem area is a welcome step along the road towards the goal of making this technique a viable design and troubleshooting tool.

Cage and guide land wear has, at times, been a major irritant to several aero engine manufacturers and can prove, in the discussor's experience, to be less than consistent in its response to normal modifications. The ability to theoretically compare potential design solutions and to subsequently correlate with service experience over several million hours is therefore a goal worthy of sustained effort.

Within the general debate on cage wear the subject of cage balance periodically claims the focus of attention and hence the comments of the authors would be particularly welcome on the following observations.

Examination of ex-service cages from mainshaft applications, mostly in excess of 2 million DN, has shown frequent evidence of uniform light polish of the silver plate. The inference drawn was that the mass eccentricity was less than 50 percent of the diametral clearance and cage/race contact was essentially due to ball excursions and rotor dynamics.

The authors' suggestion that the cage eccentricity approached 1.0 for all levels of out-of-balance and that the contact was always opposite to the out-of-balance is therefore of specific interest. It would be particularly useful if the authors could indicate the calclated mass eccentricity at each level of out-of-balance and how it compares with the hot running diametral clearance between the cage and the locating lands.

In conclusion, the authors are strongly encouraged to expand this work to encompass variations in cage mass, flexibility, location land geometry, and lubrication.

Authors' Closure

The authors are thankful to Mr. Atkinson for his very practical and encouraging comments.

Actual values of mass eccentricity for the levels of unbalance investigated in the paper are as follows:

Unbalance (gm.cm)	Mass Eccentricity (mm)	$\frac{\text{Mass Eccentricity}}{\text{Radial Clearance}}$
0	0.	0.
9	0.133	0.511
17	0.249	0.958

The values of mass eccentricity, indicated above, represent actual shift of the cage mass center relative to its geometric center. The position of cage mass center relative to the center of the guiding race is equal to the radial clearance at all levels of unbalance, which results in a steady cage/race contact, as reported in the paper. A light uniform polish on the cage guide surfaces basically implies a very light cage/race contact force and a continued variation in cage/race contact angle form 0 to 360 degrees, as shown in figure 8 for the balanced cage. Thus, the observation of a uniform light polish on the cage surfaces, as reported by Mr. Atkinson, would indeed imply that the cage/race interaction was essentially a result of ball excursions within the cage pockets, and there was no significant unbalance on the cage.