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Modeling of Instabilities Induced by Cage Clearances in Ball Bearings©

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Generalized dynamic motion of balls and cage in a ball bearing are simulated by solving the differential equations of motion under prescribed operating conditions and bearing geometry. The general cage motion is parametrically evaluated as a function of clearances both in the ball pockets and at the guide lands. The design significance of the modeling approach is demonstrated by the prediction of critical clearances which trigger certain instabilities in the cage motion. In more practical terms, the correlation between cage clearances and instability defines a wear life for the bearing under the prescribed operating conditions.

INTRODUCTION

Modeling of frictional instabilities in rolling bearings has been a subject of considerable interest in a wide range of applications covering both liquid and solid lubrication conditions. Rolling element skid and instabilities in the cage motion have been known to be critical problems in many advanced applications. While minimizing cage interactions and the resulting torque variations is of interest in some precision oil lubricated applications, controlled cage interactions and the resulting wear are the crucial design parameters in solid lubricated bearings with sacrificial cages. In both cases parametric modeling of cage motion as a function of frictional behavior and internal geometrical clearances is essential for understanding the subtle coupling between internal bearing friction and the pertinent geometrical parameters. The development of such an understanding is the primary objective of the present investigation.

This paper represents the third part of an ongoing research project. In the first part (*1*) dynamic response of a high-speed ball bearing is correlated to the friction or traction behavior at the ball/race and cage contacts. The second part (2) presents similar modeling work of cylindrical roller bearings. The next step in the modeling process is to investigate the effect of cage clearances on overall cage motion and instabilities. Such a modeling effort for ball bearings is the subject of this paper, while a companion paper (*3*) is devoted to the modeling of cage clearances in cylindrical roller bearings.

Based on a number of investigations, of both experimental and analytical nature, carried out during the last two decades it is well established that both traction at the rolling element to race contacts and the friction at the cage interactions play a dominant role in determining the overall cage motion. A review of the available work on this subject is omitted here in view of a fairly extensive discussion presented in the first part of this investigation (*1*). In comparison to this available work on friction and traction effects, investigations centered around cage clearances are very few. For a cryogenic application, Merriman and Kannel (*4*) concluded that although the cage pocket clearance has little or no effect on stability, certain minimum clearance must be maintained to prevent excessive loads in cage pockets due to ball excursion. For a high-speed solid lubricated ball bearing, Meeks and Tran (*5*) have carried out dynamic simulation of cage motion to develop certain practical guidance for the design of cage pocket and guide land clearances. Experimental investigations to correlate cage motion to varying clearances and validate the predictions of the analytical models have yet to be performed. Due to the complexities associated with instrumenting cage motion in a rolling bearing, the required experimental work has been greatly restricted. Nypan (*6*) has used optical techniques to monitor ball/cage interactions to determine the pocket forces, while Gupta et al (7) have used displacement probes to measure the cage motion and thereby derive the whirl and radial velocity components. Indeed, a substantial further development for experimental investigations for both the localized cage contacts and the overall motion is necessary.

While the experimental work awaits advancement in instrumentation techniques, continued development in computing technology has greatly eased analytical modeling of complex interaction between bearing elements and real-time simulation of the dynamic performance of rolling bear-

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ings over extensive time domains. As demonstrated in the present investigation, bearing performance simulation over a number of shaft revolutions is now a rather routine task. Thus practical significance of complex analytical models and computer codes has been greatly strengthened.

Analytical modeling of cage motion in a solid-lubricated ball bearing as a function of cage clearances is the subject of the present investigation. After a brief review of the modeling approach and operating conditions, the simulated dynamic motion of the cage is used to develop correlations between the cage clearance and overall stability. In fact, under stable conditions the steady-state wear rates provide an estimate of time over which the cage pocket clearance may increase to a critical value which triggers serious instabilities in the cage motion. Thus the analytical correlations presented herein provide some practical guidance to the estimation of bearing wear life.

OPERATING CONDITIONS AND MODELING APPROACH

A 30 mm ball bearing used earlier in the first part of this project (*1*) is also used in the present investigation. The bearing employs a composite cage, ceramic balls and M50 steel races. Detailed bearing geometry and material properties are omitted here since they have already been documented earlier (*1*). Based on the parametric runs presented in Ref. (*1*) and some available experimental data on solid lubricant traction (*8*), a traction-slip curve shown in Fig. 1 is assumed to simulate the ball/race traction behavior. Note that unlike Fig. 1, the traction behavior is generally plotted as a function of slide-to-roll ratio with a prescribed rolling velocity and contact pressure. When the model is applied to an actual contact in the bearing, a relation corresponding to appropriate rolling velocity and contact pressure is used, if traction varies as a function of rolling velocity and contact pressure. This is particularly true for most liquid lubricants. In solid lubricants however, variations of traction with pressure and rolling speed are not yet known and therefore, traction is generally assumed to vary only with the slip velocity, which is defined as the relative sliding velocity between the ball and race at any point in the contact.

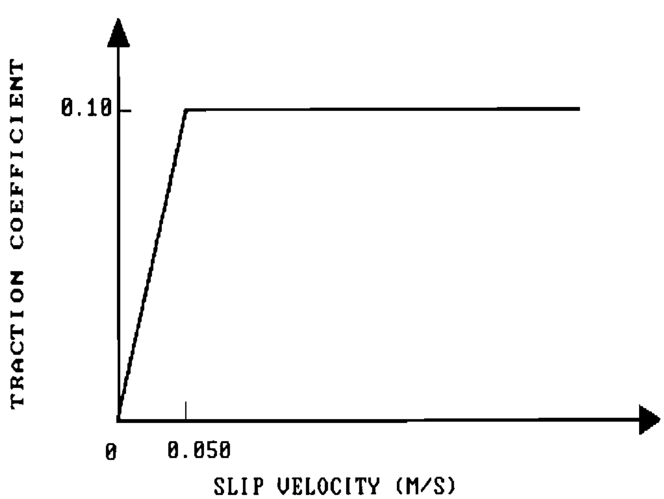

Fig. 1—Ball/race traction-slip relation.

Thus Fig. 1 shows the assumed traction variation with slip velocity and this relationship is assumed to hold at all rolling velocities and contact pressures. It should also be noted the flat part of the traction relation simply prescribes an upper bound on a linearly varying traction coefficient and such a behavior does not induce any numerical problems while solving the equations of motion. In fact, in most cases the bearing operates on the increasing part of the traction relation where there is a positive coupling between slip and traction. The friction coefficient at all cage contacts is somewhat arbitrarily assumed to be 0.050. Similar to the earlier work (*1*), the bearing is assumed to operate at a shaft speed of 70,000 rpm with a combined thrust load of 1,000 N and a rotating radial load of 500 N. It must be emphasized that neither the bearing geometry nor the operating conditions represent any specific practical application. All parameters are selected quite arbitrarily to demonstrate the practical significance of the modeling approach.

With the above bearing geometry and operating conditions the bearing dynamics computer code ADORE (*9*) is used to obtain parametric simulation of dynamic bearing performance as a function of cage pocket and guide land clearances. The clearances are varied in the range of 0.05 to 0.50 mm and both outer and inner race guidance are considered. The numerical performance simulations are evaluated in terms of gross mechanical interactions of the cage, which provide an estimate of "time-averaged wear rate" (*10*), and overall dynamic stability. Based on such evaluations critical values of clearances which trigger instabilities are identified.

RESULTS

Typical variations in ball/race loads, contact angles and spin/roll ratios under the combined thrust and rotating radial load are shown in Fig. 2. The simulated solutions are plotted over approximately 25 shaft revolutions, which is the time domain for most performance simulations in the present investigation. It may be noted that in the present case, under the traction behavior shown in Fig. 1, the classical race control hypothesis does not hold and in steady-state the ball develops relative spin on both inner and outer races; although the average spin on the inner race is appreciably smaller than that at the outer race. Also, the steady-state behavior is well simulated within two ball orbits or approximately four shaft revolutions. The simulation time required to reaching steady-state cage motion is somewhat dependent on the cage clearances. After a number of trial runs it is found that a real time corresponding to about 25 shaft revolutions is more than adequate for establishing steady-state cage motion under the range of operating conditions considered in the present investigation. Such a computer run on ADORE results in a time integration over about 10,000 steps, which takes about one hour of computer time on a mainframe super computer system.

With an outer race guided cage and with a cage/race clearance of 0.20 mm, cage instability as a function of increasing pocket clearance is demonstrated in Fig. 3(a)–3(d). Based on the critical parameters associated with stability, as outlined earlier (*1*), the whirl ratio (mass center whirl ve-

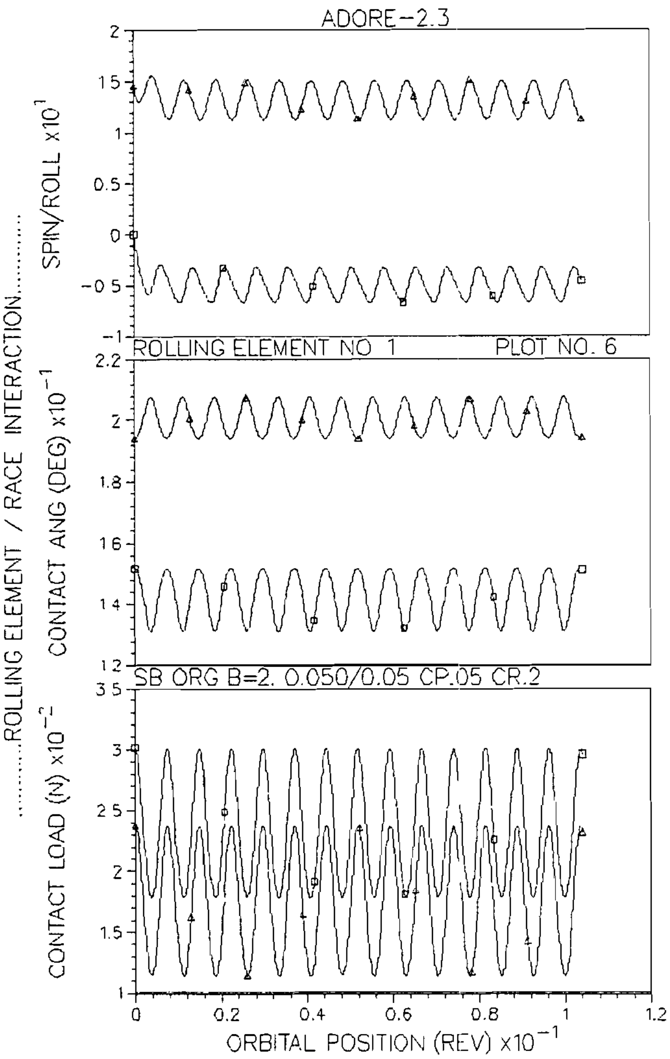

Fig. 2—Ball/race contact loads, angles and spin/roll ratios.

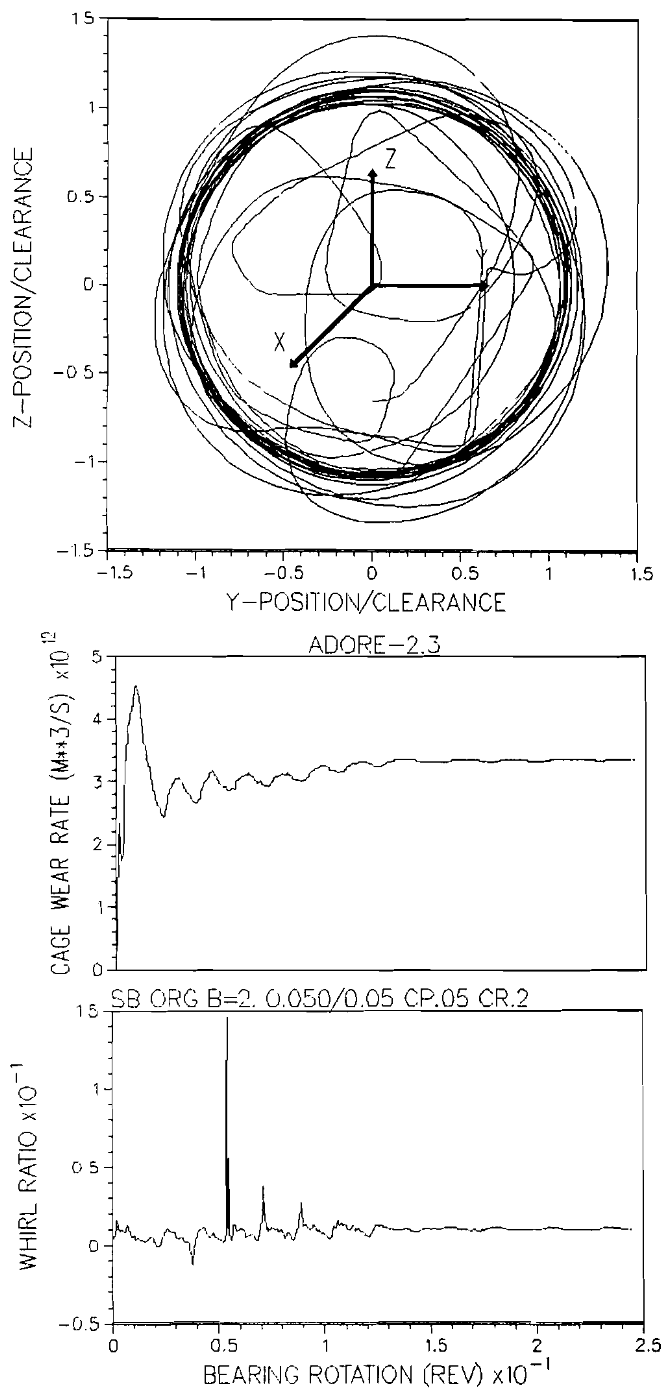

Fig. 3(a)—Bearing performance parameters with an outer race guided cage with a guide clearance of 0.20 mm and pocket clearance of 0.050 mm.

locity divided by the shaft velocity), the gross mechanical interaction in terms of the time-averaged wear rate (*10*) and the shape of the whirl orbit, are plotted in these figures for the pocket clearances of 0.050 to 0.50 mm. It is clearly seen that as the pocket clearance increases all of these three performance parameters become unstable. The orbit changes from circular to polygonal to a grossly erratic shape where the orbit radius keeps progressively increasing, which contributes to increasing mechanical interaction. This, in turn may lead to a rather catastrophic cage failure. The instability may be further elaborated in terms of the radial velocity of the cage mass center, which is plotted in Fig. 4. Note that the whirl velocity solution of Fig. 3(d) is shown again in this figure to establish the onset of instability in the radial motion with reference to cage whirl. Also note that the cage mass center actually develops a negative whirl. In fact, the steady-state whirl velocity has a negative mean value over the time of simulation. For the purpose of practical design the steady-state wear rates with the tighter clearances, 0.050 and 0.10 mm shown respectively in Figs. 3(a) or 3(b), may be used to estimate the time when the clearance may increase to 0.50 mm, at which time the cage will develop a gross instability as shown in Fig. 3(d). Thus, based on these results, some guidance for time to dynamic failure may be obtained.

Intuitively it might be expected that if the pocket clearance is further increased, perhaps beyond the ball excursion limits, the pocket interactions will eventually reduce and the pocket clearance will then have little or no effect on cage interactions. Although such a larger value of clearance may be used in oil lubricated bearings, the use of tighter clearances is essential in solid lubricated bearings with sacrificial cages where controlled ball/cage interaction is required for the release of solid lubricant from the cage. The above stability correlations are therefore more relevant to solid-lubricated bearings.

For the selected operating conditions and the range of cage clearances, the above results show a definite correlation

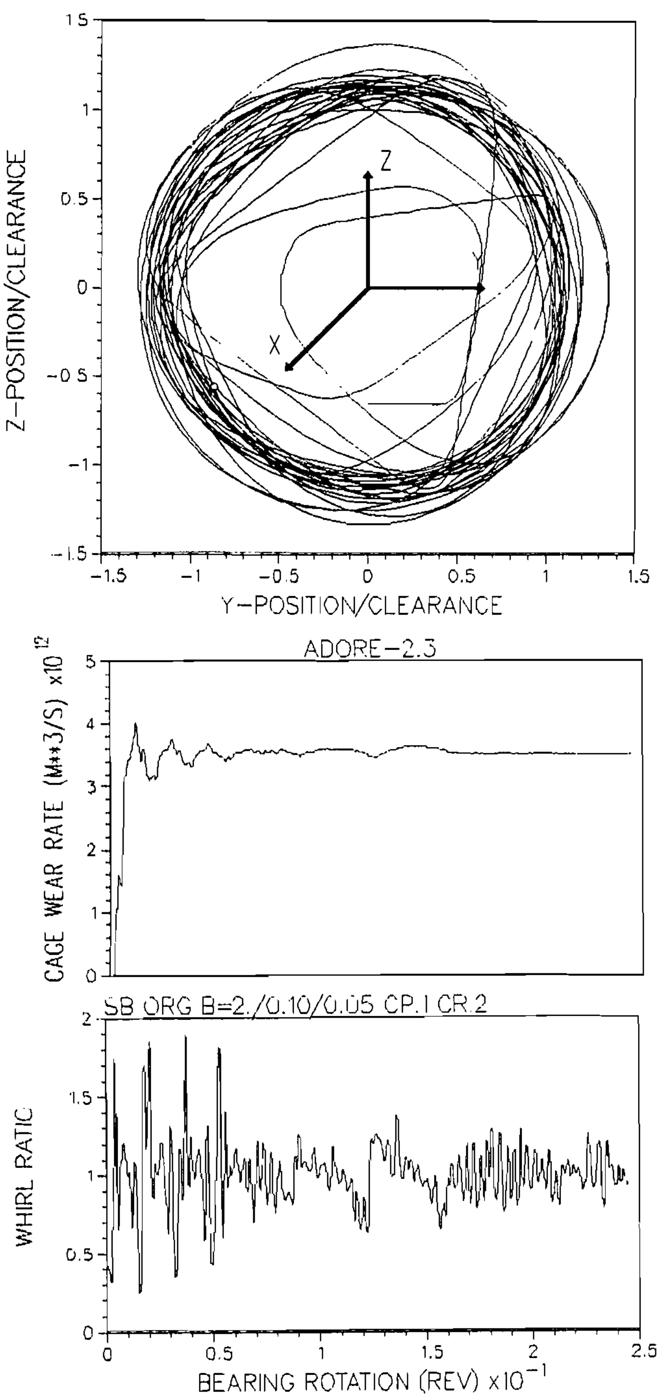

Fig. 3(b)—Bearing performance parameters with an outer race guided cage with a guide clearance of 0.20 mm and pocket clearance of 0.10 mm.

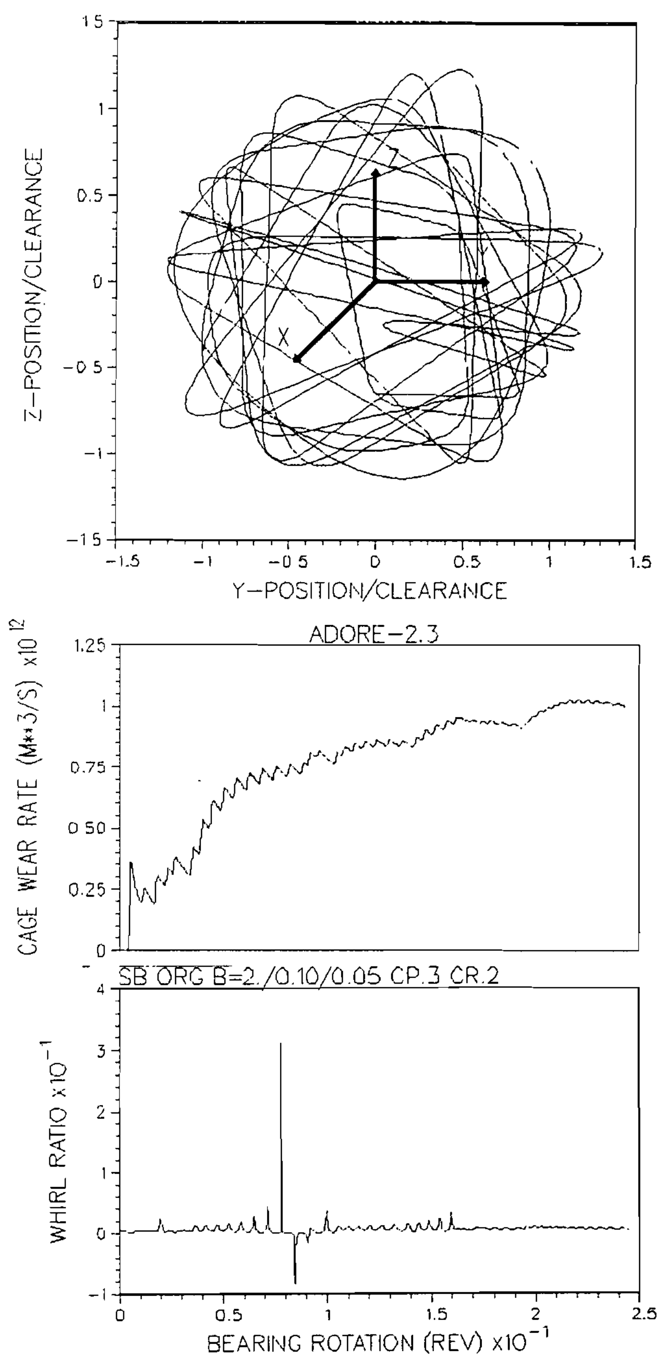

Fig. 3(c)—Bearing performance parameters with an outer race guided cage with a guide clearance of 0.20 mm and pocket clearance of 0.30 mm.

between stability and pocket clearance. Although such a finding differs from the results of Merriman and Kannel (*4*), who reported no effect of pocket clearance on stability, it is quite possible that they may be working in a range of pocket clearance which is beyond the ball excursion limits under the traction and operating conditions for their bearing. Thus, in view of several geometrical and operational parameters, it may be quite difficult to derive any universal conclusions which may be applicable to all conditions.

The results obtained with inner race guidance with identical cage clearances are quite similar to those discussed above for the outer race guidance. Figures 5(a) and 5(b) show the performance parameters for the pocket clearances of 0.30 and 0.50 mm respectively. Clearly, the cage is relatively unstable with the pocket clearance of 0.50 mm, although the instability is not as severe as that seen in Fig. 3(d) with outer race guidance.

All of the above results are obtained with a cage/race clearance of 0.20 mm. For correlating the role of this guide clearance, the computer runs are repeated with several different cage/race clearances. The results are quite interesting when the cage/race clearance is just about equal to the pocket clearance. Figure 6(a) shows the performance parameters when both the pocket and guide clearance are 0.50 mm with outer race guided cage. Clearly the orbit is unstable, but the gross mechanical interaction of the cage, as indicated by the time-averaged wear rate, is well bounded and stable.

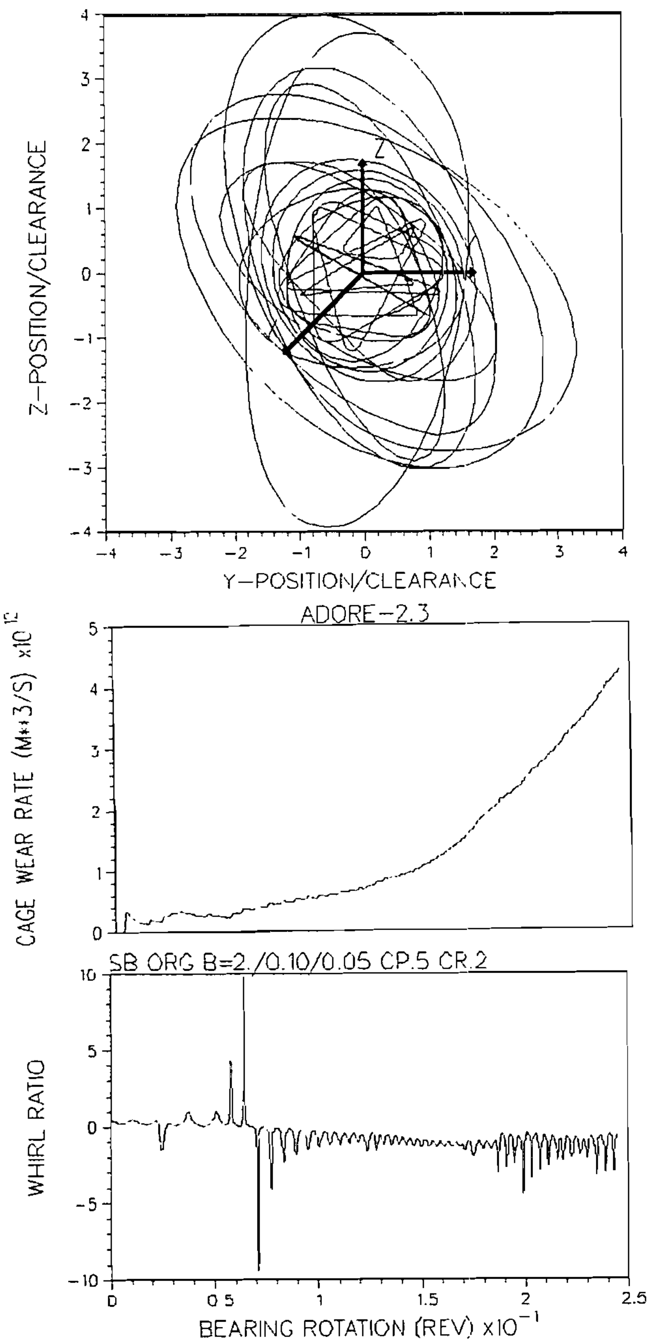

Fig. 3(d)—Bearing performance parameters with an outer race guided cage with a guide clearance of 0.20 mm and pocket clearance of 0.50 mm.

If the cage guidance is switched to the rotating inner race, the whirl orbit becomes quite well defined, as seen in Fig. 6(b). The wear rate now seems to show an instability, but since the orbit is well defined it is quite possible that the steady increase in wear rate may be a part of the transient to reach a higher value of steady-state wear rate which corresponds to steady cage/race contact. It is of course necessary to continue the simulation beyond the 25 shaft revolutions to firmly establish the steady-state behavior in this case. In any event, when the case of equal pocket and guide land clearance is compared to Fig. 3(d), where the pocket clearance is two and a half times larger than the guide clearance, it may be concluded that there may be a certain desirable ratio of the two clearances which provides favorable bearing performance.

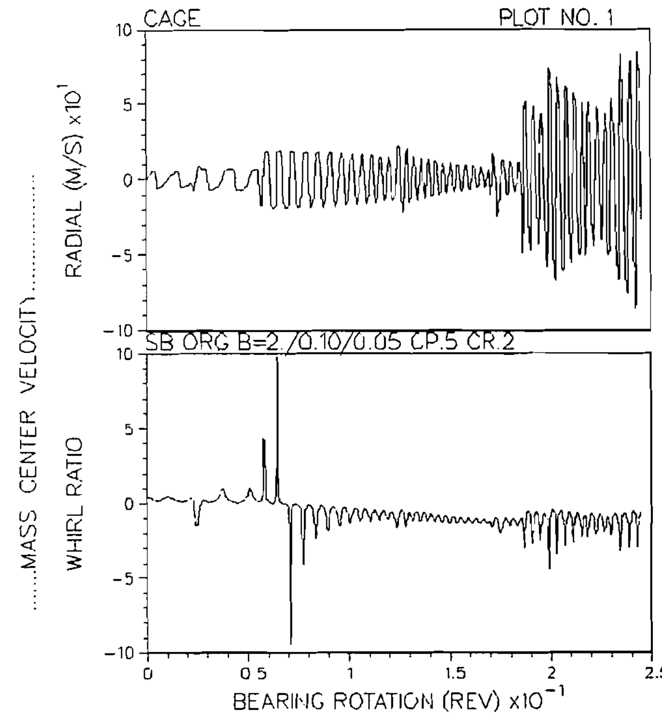

Fig. 4—Cage mass center whirl and radial velocities under unstable conditions.

Such an observation is in agreement with the findings of Meeks and Tran (*5*), although it is not possible to compare actual numbers due to differences in bearing geometry, operating conditions and traction models.

It is important to remember that the correlation between cage clearances and bearing performance discussed above, may very well be unique to the bearing geometry and operating conditions considered in this investigation. The development of any conclusions with more universal validity shall require considerably more performance simulations, and more so, a rigorous experimental validation of the computer predictions.

SUMMARY

Dynamic performance simulations for a high-speed ball solid-lubricated ball bearing, as provided by the computer program ADORE, are obtained over a range of cage clearances. Under the prescribed operating conditions cage instability has a definite correlation to cage pocket and guide land clearances. Under a given guide clearance, an increasing pocket clearance may trigger an instability when it reaches a certain critical value. Thus the wear rate solutions under stable conditions may be used to estimate the time required for the pocket clearance to increase to an instability threshold. For the purpose of practical design, parametric performance runs similar to the ones considered in the present investigation may lead to a certain preferable ratio of the two cage clearances for an overall acceptable bearing performance.

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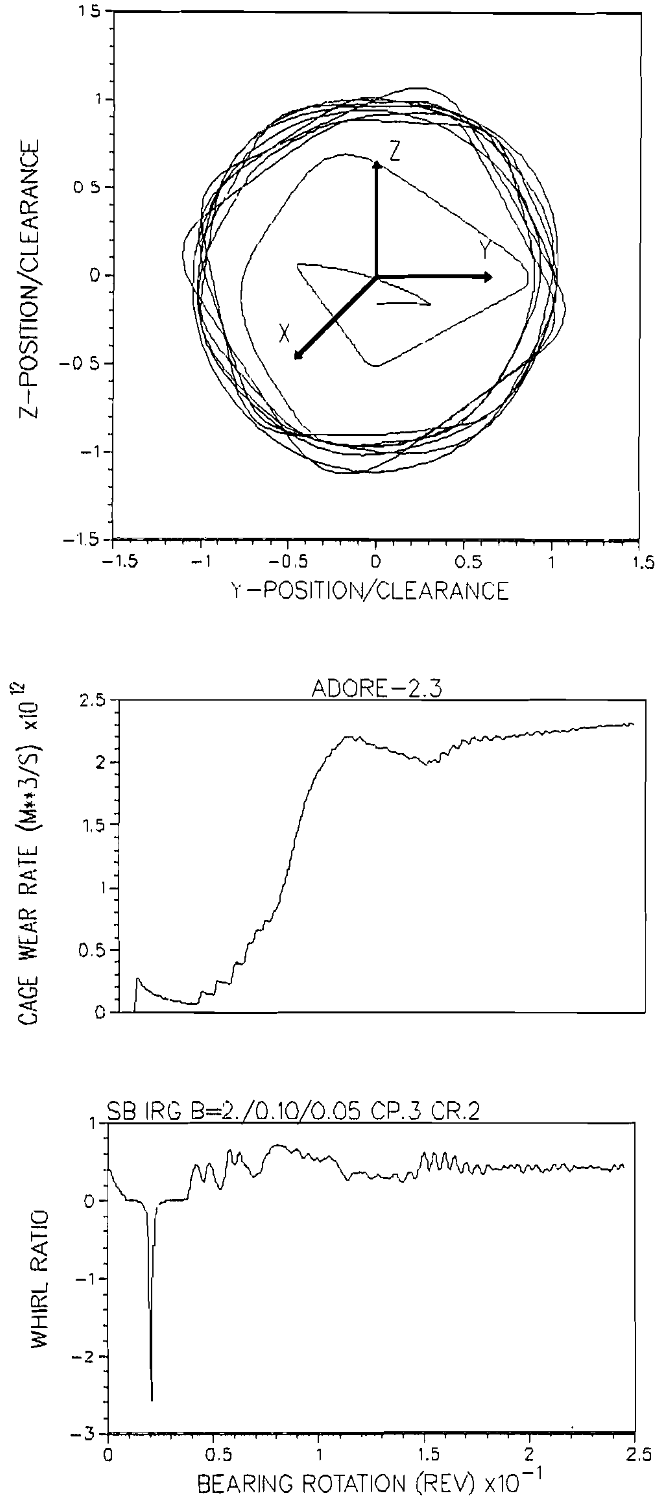

Fig. 5(a)—Bearing performance parameters with an inner race guided cage with a guide clearance of 0.20 mm and pocket clearance of 0.30 mm.

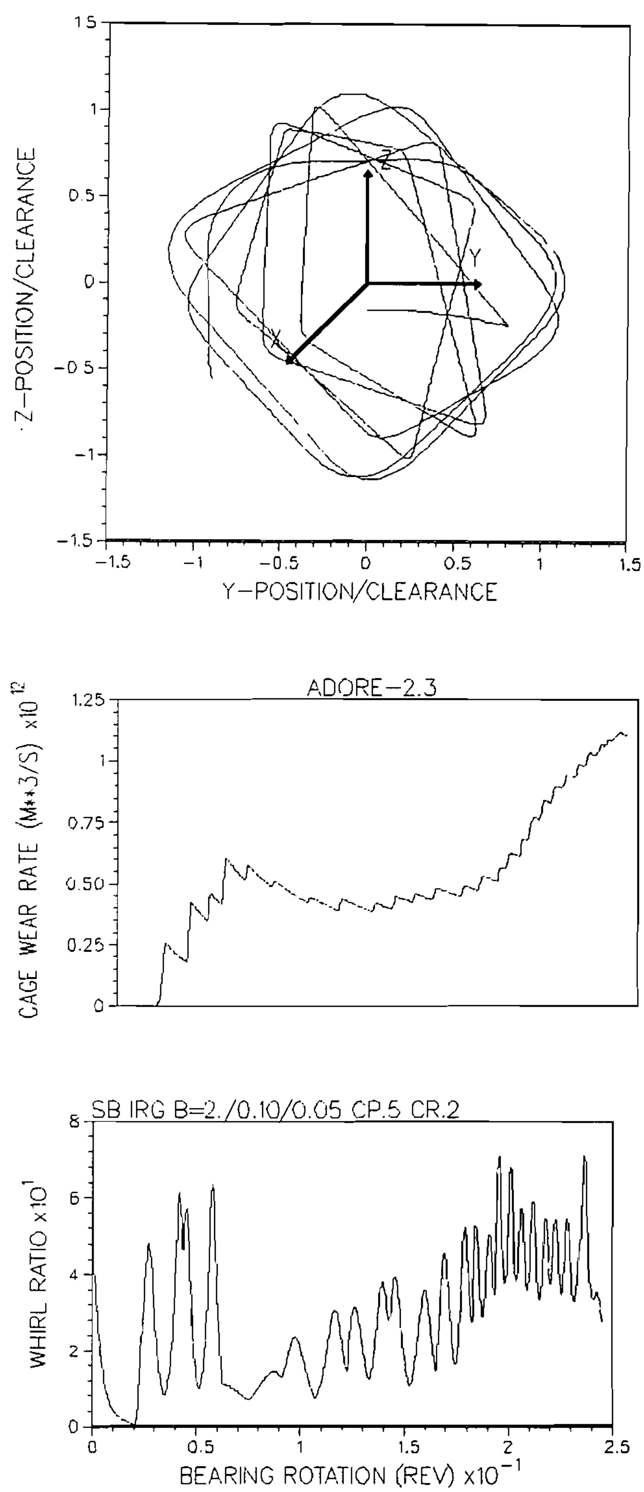

Fig. 5(b)—Bearing performance parameters with an inner race guided cage with a guide clearance of 0.20 mm and pocket clearance of 0.50 mm.

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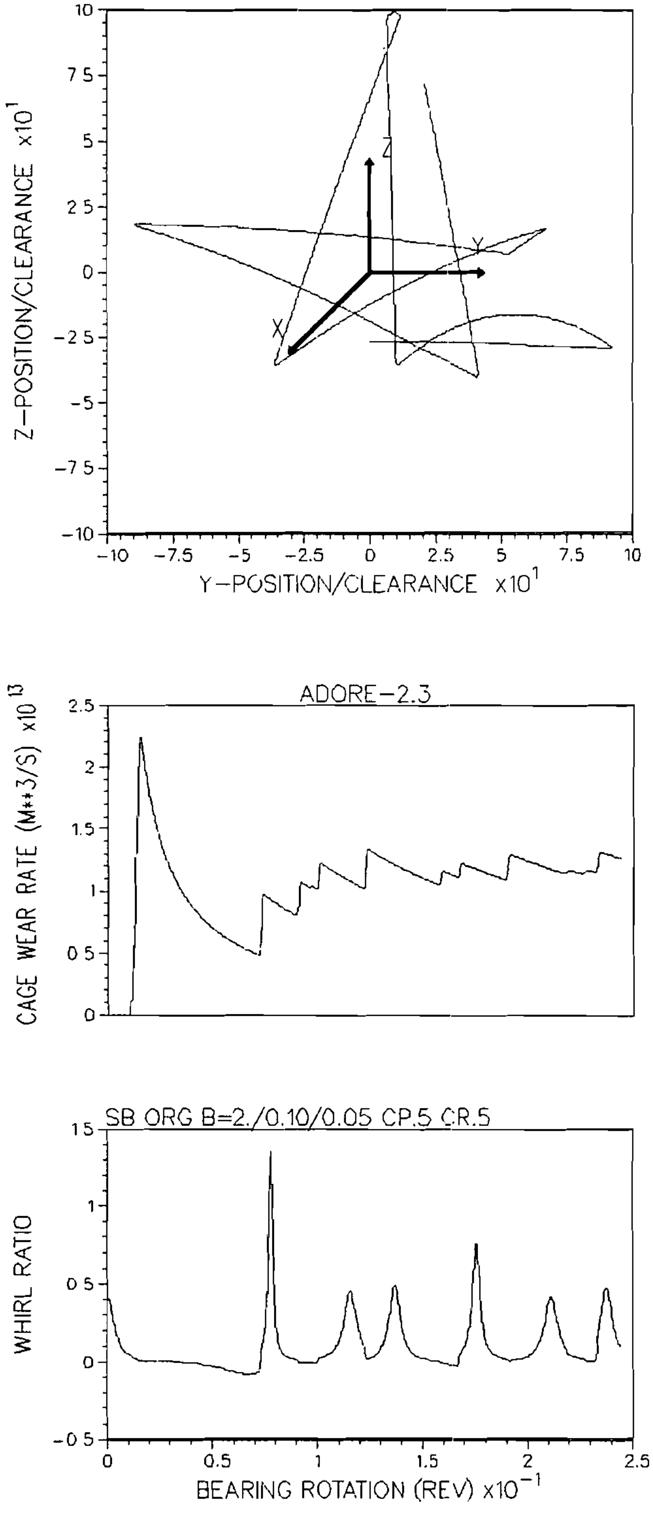

Fig. 6(a)—Bearing performance parameters with an outer race guided cage with a guide land and pocket clearances of 0.50 mm.

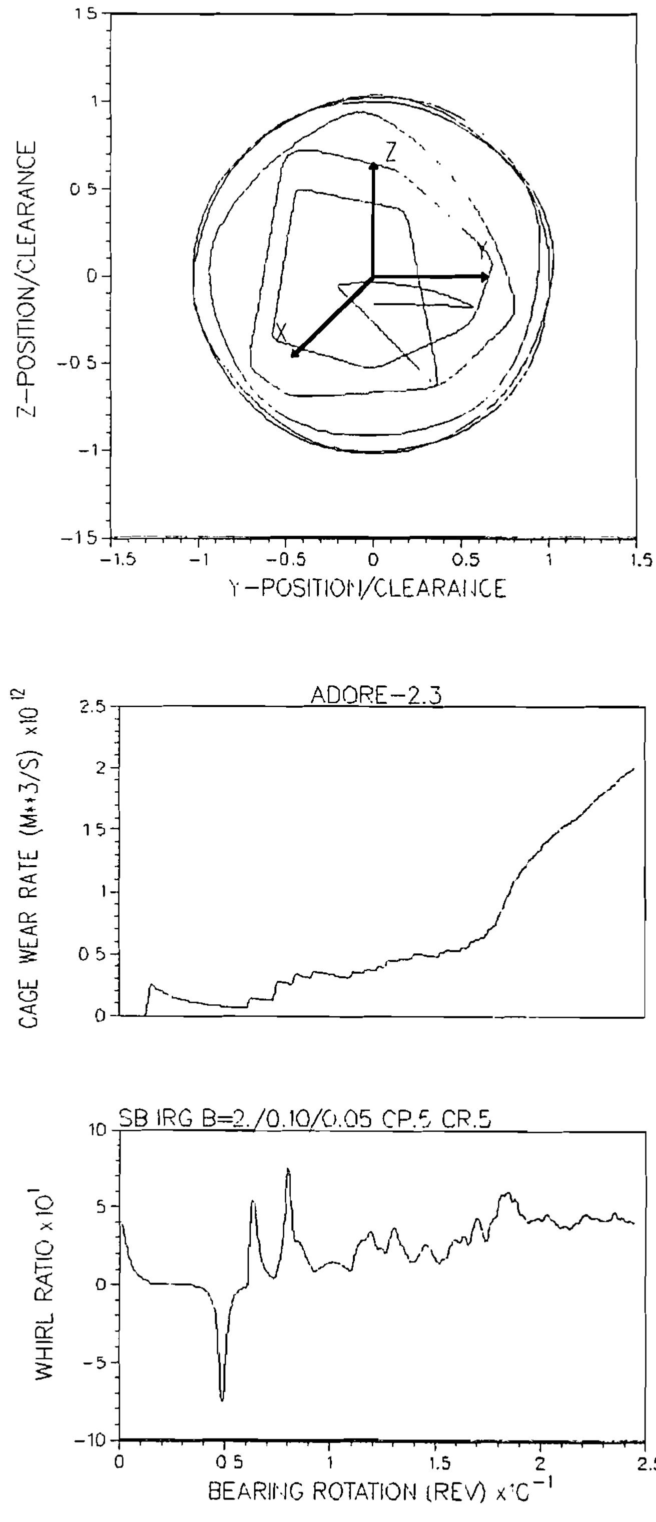

Fig. 6(b)—Bearing performance parameters with an inner race guided cage with a guide land and pocket clearances of 0.50 mm.

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