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Dynamics of Rolling Element—Bearings Experimental Validation of the DREB and RAPIDREB Computer Programs

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The general motion of the cage predicted by the computer models in an angular contact ball bearing operating up to two million DN is compared against experimental data. Both the computer predictions and experimental data indicate a certain critical shaft speed at which the cage mass center begins to whirl. The predicted and measured whirl velocities and orbit shapes are in good agreement. The axial and radial velocities of the cage mass center also agree within the tolerance band of the expected experimental error. Due to experimental difficulties the cage angular velocity could not be reliabily measured at high speeds. At low speeds, however, there is a fair agreement between the experimental data and the analytical predictions.

Introduction

Computer modeling techniques of the dynamic performance of rolling bearings have significantly advanced over the past decade. The conventional quasi-static equilibrium models of Jones [1, 2] have been replaced by real time dynamic simulations which integrate the classical differential equations of motion of each bearing element. Following the work by Walters [3] on ball bearings, Gupta [4] developed a fairly generalized dynamic model for both ball and cylindrical roller bearings. The model resulted in the computer program, DREB (Dynamics of Rolling Element Bearings). Similar dynamic analyses and computer programs have been subsequently pursued by a number of investigators, such as Brown et al. [5] and Conry [6].

One of the problems common to all transient dynamic models is the amount of computational effort required to integrate the differential equations of motion over practical time domains. Gupta [7], in an attempt to solve this problem, introduced certain equilibrium constraints to eliminate the very high frequency ball/race vibration, thereby significantly increasing the permissible step size. This resulted in a "rapid" version of the original computer program DREB and it has been well known, as RAPIDREB. Aside from the equilibrium constraints all the interaction models in DREB and RAPIDREB are identical. However, the increased efficiency of RAPIDREB in simulating the low frequency effects has been very effective in providing bearing performance simulations over several shaft revolutions for a wide range of practical applications.

Although the computer models, such as DREB and RAPIDREB, have proven to be very powerful in simulating some of the most sophisticated operating environments, a rigorous experimental validation of the predictive capabilities of such models is essential before they can be widely accepted for design purposes. It is necessary that the experimental validation be carried out at the most fundamental level and in view of the large number of inputs to these models, validations in terms of overall bearing performance as indicated by life or overall torque levels may not be sufficient. However, the measurement of the fundamental components of motion such as cage mass center velocities, cage angular velocities, ball or roller mass center and angular velocities etc, is an extremely difficult task and there is generally a large uncertainty band associated with any such measurement. Also, since all of the inputs to the model are not precisely known, the analytical results are also uncertain to a certain degree. A realistic experimental validation, therefore, becomes quite difficult.

The present investigation is an attempt to measure the various components of general motion of the cage in an angular contact ball bearing. The experimental results are compared to the analytical predictions obtained by the DREB and RAPIDREB computer programs. The operating conditions are kept as simple as possible in order to minimize the variance in both the experimental data and the analytical results.

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Ball Bearing Test Rig

Figure 1 presents a schematic description of the test rig used in the present investigation. The rig is driven by a 50 HP water cooled, variable speed drive electric motor through a 9.25-1 speed ratio gear box allowing shaft speeds to 30,000 rpm. The test bearing (A) is mounted in the housing and on the support spindle (H) which is supported by bearings (L) and (G) and a thrust retainer (I) which allows radial motion of the bearing (G) so that radial load may be applied through the pressure diaphragm (F). Axial load is applied directly to the test bearing through the pressure diaphragm (E). The bearing is lubricated through jets (C). The quantity of oil flow is kept to a minimum (7.5 liter/min) to eliminate any excessive churning effects. The oil is pumped from scavenge ports (D) into an accumulator where it is heated before recirculation to maintain a specified outer race temperature.

Instrumentation is available in the rig to measure shaft speed, applied loads, torque, and temperatues at various points within the system. A thin section of the drive shaft between the support bearing and the test bearing is instrumented with temperature compensated strain gages to measure bending and axial loads and torsion on the shaft. The temperature of the inner race is measured by a thermocouple. These signals are carried out through a slip ring assembly (K) located on a shaft extension (J). The shaft speed is measured by a magnetic pickup (M) connected to a digital pulse counter.

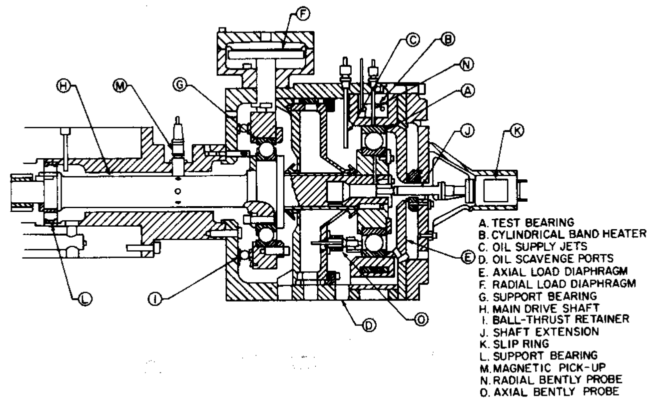

Fig. 1 Drawing of test rig cross-section

Thermocouples are located on the outer race of the test bearing, in the oil supply jets and scavenge ports, and in the oil sump to measure temperatures at those points.

Proximity probes, oriented as shown in Fig. 2, are used to measure the various components of cage motion. The measured axial and radial displacements are differentiated to obtain the cage mass center velocities which are transformed into axial, radial and whirl components. Also the measured radial displacements in the two planes (see Fig. 2) are used to determine the rotation of the cage about the transverse axes. The width of the cage is increased beyond the bearing lands so that the target surface on the outer diameter of the cage is fully exposed to the field of the sensors. This was also found to be necessary to eliminate the large signal picked up by the probes from the ball pockets.

To account for the sensitivity of the proximity sensors to temperature and to the presence of conducting surfaces other than the target, a calibration procedure is performed to obtain the relationship between sensor output and cage position. First the test housing, with the sensors installed, is mounted on a calibration fixture which is designed so that the cage could be positioned at a known location with respect to the probes. The outputs of the probes at various positions produced a calibration curve at room temperature. The second step in the calibration procedure is to assemble the test rig with the cage held in a known position, heat the bearing to the test temperature by pumping heated oil through the bearing cavity and reading the sensor output. The procedure is repeated for a few positions of the cage and the necessary thermal correction, which is applied to the room temperature calibration, is obtained.

For the purpose of measuring the angular velocity of the cage about the bearing axis and its variation in time, a black and white grid pattern consisting of 200 divisions is obtained by carefully etching the silver coating on the rim, or the end face of the cage. Two photonic sensors are placed 180 deg apart, as seen in Fig. 2, and directed at the etched grid pattern. The signals from the photonic sensors are averaged and passed through a frequency to dc convertor to obtain the cage angular velocity. The reason for the 200 divisions in the grid pattern is to obtain adequate resolution to detect an angular velocity variation of approximately ten times the nominal velocity of the cage. However, as will be discussed later, this technique did not result in a reliable measurement because with the presence of oil in the bearing the contrast between the black and white marks of the grid pattern was rather poor.

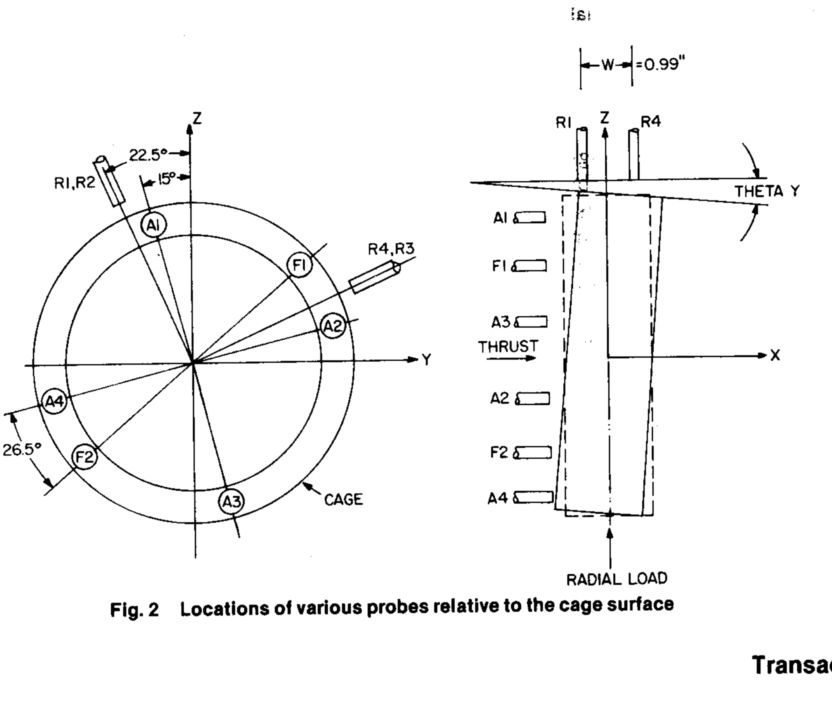

Fig. 2 Locations of various probes relative to the cage surface

Table 1 Bearing geometry

PARAMETER	UNITS	VALUE
Bore	mm	100.
Outside Diameter	mm	180.
Ball Diameter	mm	19.05
Number of Balls		18
Pitch Diameter	mm	140.
Contact Angle	degrees	25
Curvature, Outer Race		0.52
Curvature, Inner Race		0.54
Cage O. D.	mm	148.8
Cage I. D.	mm	129
Cage Width	mm	27.3
Diametral Cage/Race Clearance	mm	2.0
Diametral Ball Pocket Clearance	mm	0.826
Cage Weight	gm	636
Ball and Race Elastic Modulus	N/m^2	2.0×10^{11}
Ball and Race Density	Kg/m^3	7750.
Cage Elastic Modulus	N/m^2	2.0×10^{11}
Cage Density	Kg/m^3	7750.

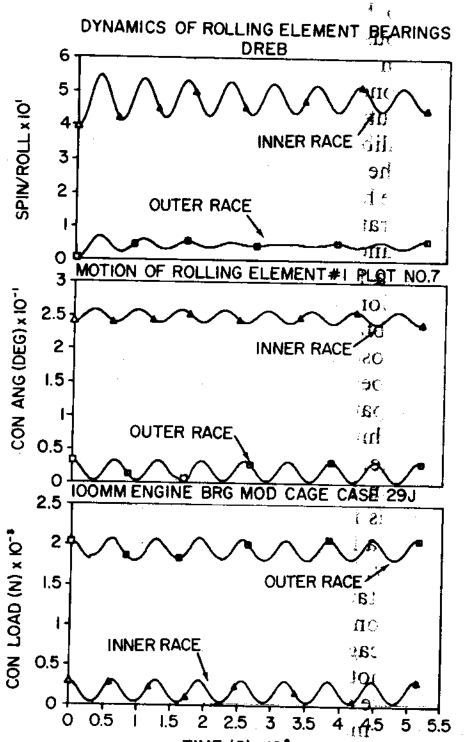

Fig. 3 Ball/race interaction with a combined axial and radial load of 1112N at 20,000 rpm

Test Bearing and Operating Conditions

The 100 mm bore, split inner race, angular contact ball bearing representative of a main shaft turbine engine thrust bearing operating at three million DN is used in the present investigation. The basic geometry of the bearing is outlined in Table 1. The races and balls of the bearing are made of M-50 tool steel heat treated to a hardness of Rockwell 62C. The cage is constructed of AISI 4340 steel with hardness of Rockwell 28-32C, and it is guided on the inner race. A silver coating of approximately 25-50 micron thickness is applied to the cage in areas of rubbing between the inner race land and cage inner diameter and in the ball pockets. The cage surface exposed to the probes and the outer race was left uncoated for the reasons of probe sensitivity. The lubricant used is MIL-L-7808, a synthetic ester turbine engine lubricant having a kinematic viscosity of 3×10^{-6} m^2/s (3 centistoke) at 99°C. Based on the available traction data for the MIL-L-7808 type oil, the friction coefficient at the silver coated ball/cage and cage/race interface is assumed to be 0.005 in all the analytical solutions.

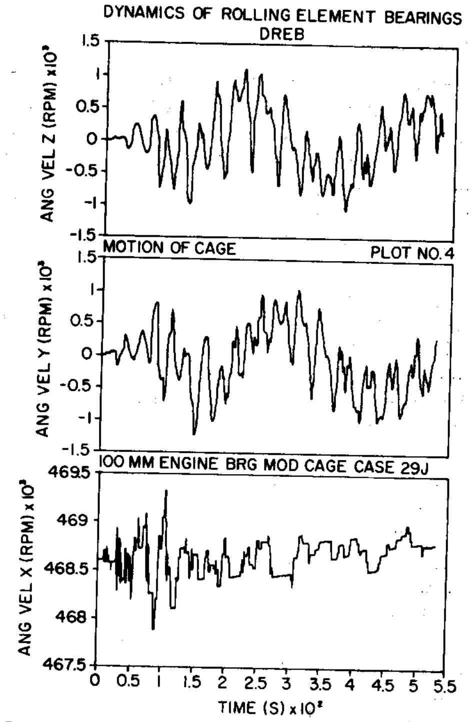

Fig. 4 Cage angular velocity variation with the combined axial and radial load of 1112N and a shaft speed of 20,000 rpm

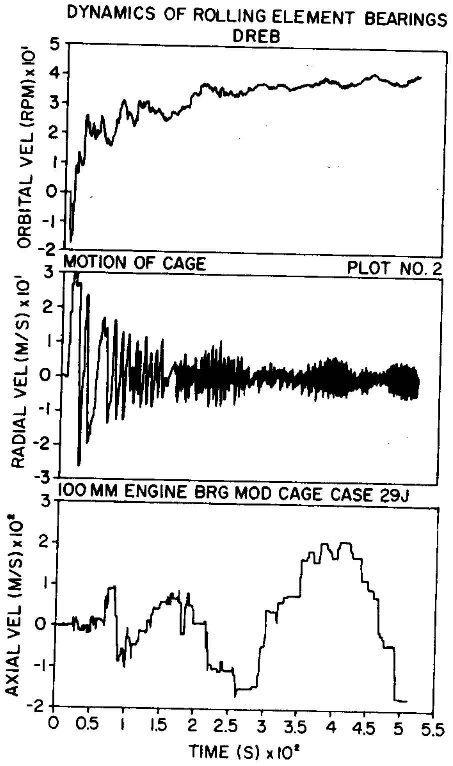

Fig. 5 Cage mass center velocity variation with the combined axial and radial load of 1112N and a shaft speed of 20,000 rpm

Although the experimental data was collected over a range of operating loads and speeds, two sets of loads (a pure thrust load of 1112 N and a combined thrust and radial load of 1112 N) and four operating speeds of 2500, 5000, 10,000, and 20,000 rpm are considered for the present investigation. These

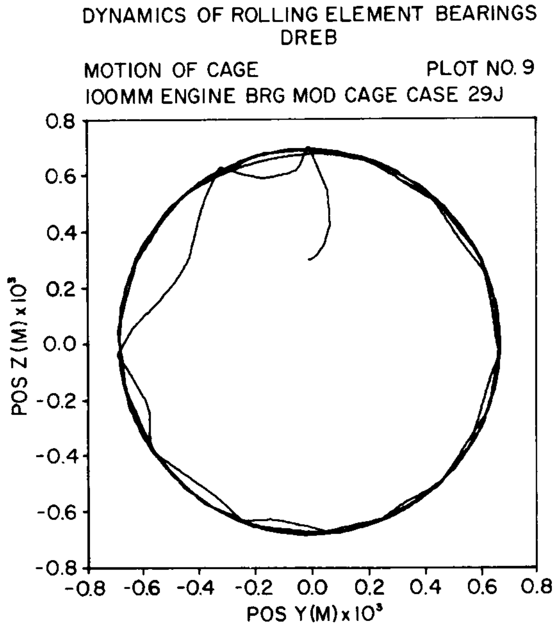

Fig. 6 Cage mass center orbit as simulated by RAPIDREB under the combined axial and radial load of 1112N and a shaft speed of 20,000 rpm

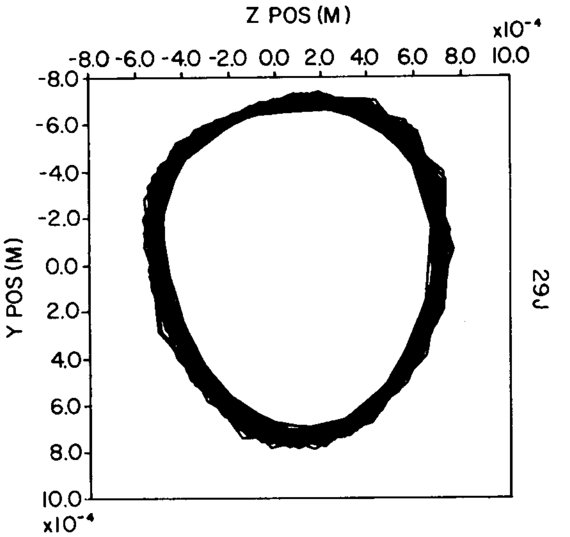

Fig. 7 Cage mass center orbit as derived from the experimental data obtained at 20,000 rpm with a combined axial and radial load of 1112N

operating conditions result in a maximum DN value of two million and radial to thrust load ratios of zero and one.

The experimental data are recorded on a FM tape recorder and later digitized for subsequent analysis to determine the various cage motion components. Typically the data are recorded at each test condition for about 10 seconds. The digitizing rate is 2000 samples per second. This results in an upper cut-off frequency of about 1 KHz, which is found to be more than sufficient; a spectral analysis of the analog data demonstrated that the data does not really contain any frequencies higher than about 500 Hz.

Results

Corresponding to each experimental condition, analytical simulations of the cage motion are obtained by the computer program RAPIDREB. By exercising the equilibrium constraint in RAPIDREB it is possible to generate solutions over several shaft revolutions with computing effort, equivalent to

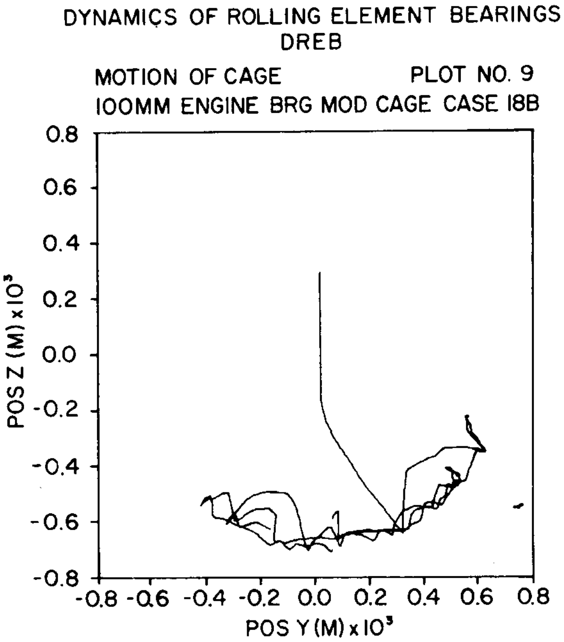

Fig. 8 Cage mass center orbit as simulated by RAPIDREB under a thrust load of 1112N and a shaft speed of 5000 rpm

about one to two orders of magnitude less than that required by the original DREB program. It should be remembered that except for the ball motion constraints RAPIDREB is identical to the original computer program DREB. Thus the experimental validations presented herein apply to both DREB and RAPIDREB programs.

The ball/race contact load solutions presented in Fig. 3 indicate typical length of simulation; each cycle represents one ball orbit around the bearing or approximately two shaft revolutions. Typically each RAPIDREB simulation required about 2000 time steps with an average step size of 250 microseconds; the increased computing efficiency of RAPIDREB is evident when this is compared to the maximum permissible step size of about 10 microseconds in the original DREB program. The cyclic variation results from the applied radial load, which in this case is equal to the thrust load. Typical cage motion solutions for this case are presented in Figs. 4 and 5, which plot the simulated angular and the mass center velocities of the cage. It is seen that the analytical solutions indicate both significant axial and out-of-plane coning motion of the cage under the combined axial and radial loads.

In terms of the overall motion of the cage both RAPIDREB simulations and the experimental data indicate that there is a certain critical shaft speed, around 5000 rpm, below which there is no whirl of the cage and above which the cage whirls in a fairly circular orbit. The radius of the orbit is almost equal to the cage/race clearance. Thus there is a continued interaction at the cage/race interface and the whirl orbits are fairly stable. However, a visual examination of the cage after testing did not indicate any significant metal to metal contact; this suggests that a thin lubricant film was always maintained at the race/cage interface. Typical mass center orbits as simulated by RAPIDREB and those obtained from the experimental data are shown in Figs. 6 and 7, respectively. Clearly, these two solutions are closely similar. The small differences in the radius and shape of the orbit can be attributed to the uncertainties in the calibration of the radial probes and the actual operating clearance at the cage/race interface. It should be noted in Fig. 6 that the cage moves, rather rapidly, into a circular orbit; typically it took only about 20 percent of the total simulation time, or about ten milliseconds, to reach this steady-state.

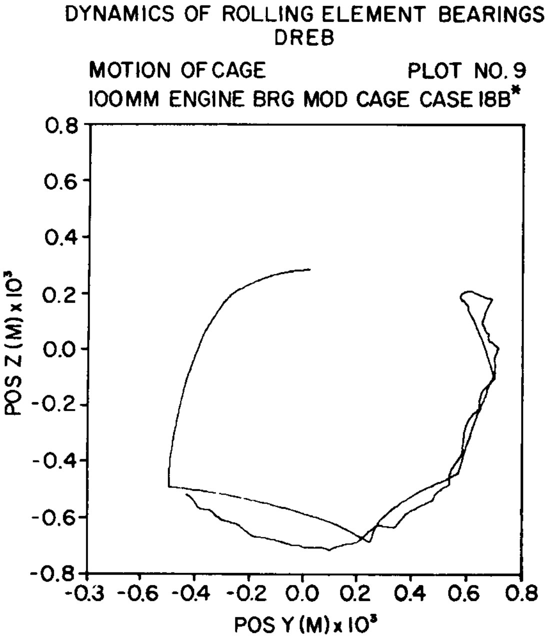

Fig. 9 Cage mass center orbit with the ball/cage and cage/race friction coefficient increased to 0.010 while all other conditions identical to those of Fig. 8

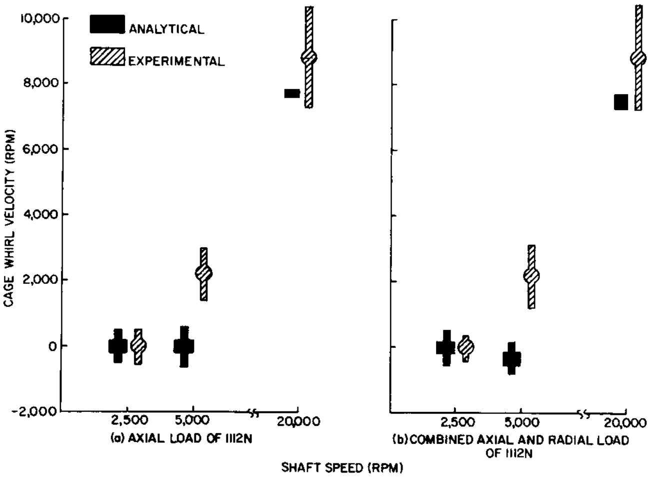

Fig. 10 Comparison of the cage mass center whirl velocities

At a speed of about 5000 rpm, the experimental data indicated that the cage mass center just moved into a stable orbit, similar to that shown in Fig. 7, while the analytical results indicated a somewhat oscillatory motion of the cage mass center as shown in Fig. 8. This difference is attributed to the coefficient of friction of 0.005 assumed at the ball/cage and cage/race contacts. This is confirmed by rerunning RAPIDREB simulations with this coefficient of friction increased to 0.010. The resulting cage orbit is shown in Fig. 9, which indicates that the increased friction coefficient results in the increased amplitude of the oscillatory motion of the cage mass center. Based on these results it may be concluded that the ball/cage and cage/race friction may have some impact on the critical speed at which the cage starts whirling.

At speeds less than 5000 rpm, both experimental and analytical results indicated no whirl motion at all. At higher speeds, when the orbits are well established the ball/cage and cage/race friction did not effect the circular shape of the cage mass center orbit, although some differences in the whirl speeds are noticed. This further establishes the significance of the friction at the ball/cage and cage/race interactions.

Figure 10 shows a comparison between the measured and predicted whirl velocities. Within the uncertainties in the experimental data and the friction behavior at the ball/cage and cage/race interfaces, RAPIDREB predictions of the cage

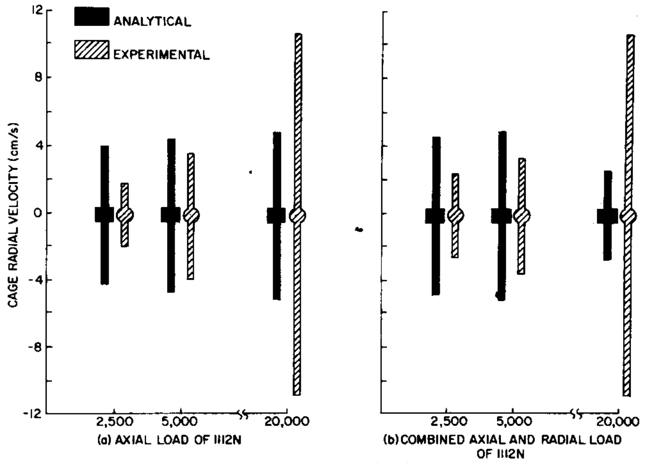

Fig. 11 Comparison of the measured and predicted radial velocities of the cage mass center

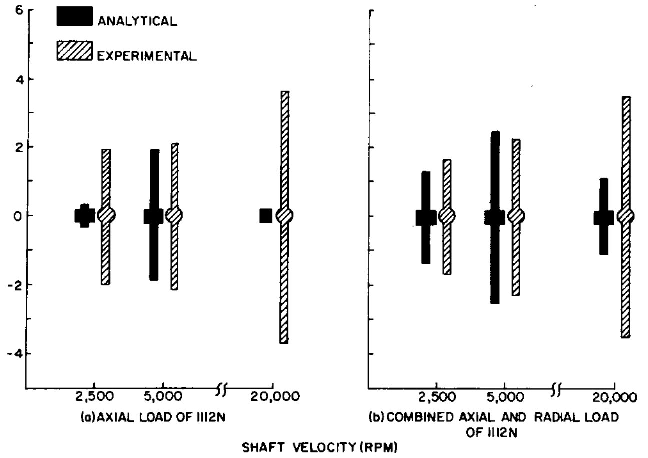

Fig. 12 Measured and predicted axial velocities of the cage mass center

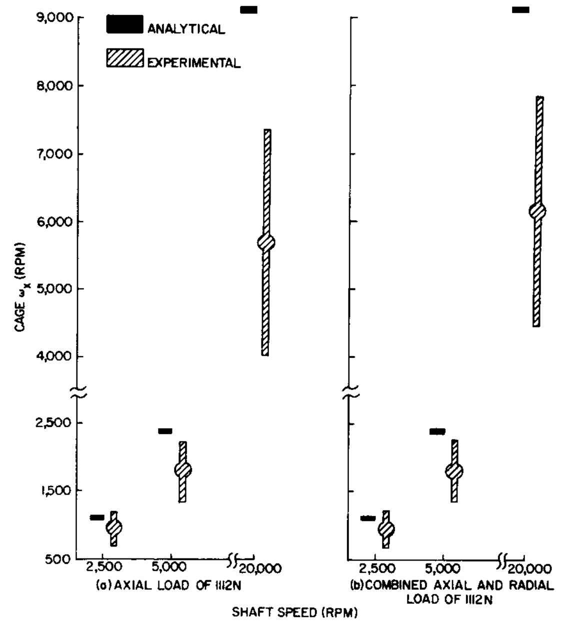

Fig. 13 Comparison of the cage angular velocities

mass center whirl may be considered to be in agreement with the experimental data. The radial and axial velocities of the cage mass center are shown in Figs. 11 and 12, respectively. Once again the agreement is fair. The differences in the radial velocities, particularly at high speeds, may be attributed to the vibrations of the housing which holds the radial probes. A rotor dynamic analysis of the test rig showed that there is a critical speed around 18,000 rpm. This could explain the differences in the radial motion at 20,000 rpm. A close examination of the axial probe data indicated a certain beat frequency which led to suspect axial vibrations of the plate holding the axial probes. This may explain the observed differences in the axial velocities. However, the test rig has yet to be instrumented to confirm the presence of any vibration problems at high speed.

The measurement of the angular velocity of the cage about the bearing axis was the most difficult task, particularly at high speeds. As the speed increased the contrast between the black and white grid pattern on the cage reduced and the response of the photonic sensors was increasingly poor. Perhaps the presence of oil in the bearing is responsible for this difficulty, although the quantity of the lubricant was held to a bare minimum. The significantly lower measured velocities of the cage, as shown in Fig. 13, are attributed to this poor response of the photonic sensors. Post-test inspection of the test bearing indicated no skidding damage on the ball and the races which confirms that the measured angular velocity of the cage is indeed in error.

The predicted coning motion of the cage about the transverse axis is similar to the experimental observations in the sense that the variations in the cage coning angles almost cover the entire range permissible by the cage/race clearance. However, the angular velocities derived from the data obtained from the probes in the two radial planes demonstrated large scatter compared to the variations predicted by RAPIDREB. The mean velocities are of course zero in all cases.

Another significant finding in the present investigation, is that the motion of the cage is relatively unaffected when a radial load of a value equal to the applied thrust load (1112 N) is added. This implies that the clearances in the ball pocket and the cage/race interfaces are large enough to accommodate the ball excursions due to the radial load. This warrants further investigation at reduced clearances.

In general, it must be pointed out that this investigation simply presents a first step in truly validating the sophisticated computer models of the dynamic performance of a ball bearing as represented by the DREB and RAPIDREB programs. The fair agreement between the analytical predictions and the experimental observations is encouraging and it substantiates the need for further improvements in the experimental techniques and the development of both experimental and analytical methods of determining the crucial input parameters to the DREB and RAPIDREB programs, such as the ball/cage and cage/race friction.

Conclusions

Both the experimental observations and the predictions of the RAPIDREB computer program indicate that there is a certain critical shaft velocity at which the cage mass center goes into a whirl orbit. At operating speeds below this critical speed, there is no whirl and at speeds higher than the critical speed the whirl orbit is almost circular with the orbit radius to the cage/race clearance. Both the predicted nature of the whirl orbit and the whirl velocities of the cage mass centers are in reasonable agreement with the experimental observations. Analytical results show that the friction behavior at the ball/cage and cage/race interactions is significant in determining the critical speed at which the cage mass center goes into the whirl mode. The measured axial and radial velocities of the cage mass center are in fair agreement with the analytical predictions. The measured angular velocities of the cage are significantly lower than those predicted by the computer model, particularly at the higher shaft speeds. This difference is attributed to the experimental difficulties and the poor response of the photonic sensors resulting from the reduced contrast of the grid pattern on the cage which forms the primary input source to the photonic sensors. For the test bearing considered, the general motion of the cage is insensitive to the applied radial load on the bearing. This is in complete agreement with the computer predictions. It is expected that the clearances in the ball pocket and at the cage/race interface are large enough to accommodate the ball excursions due to the radial load.

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