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CAGE STABILITY ANALYSIS FOR SSME HPOTP BEARINGS

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SUMMARY

A numerical model of cage motion (CAGEDYN) has been used to analyze the stability of bearing cages in the Space Shuttle main engine (SSME) high pressure oxygen turbopump (HPOTP). The stability of existing bearing geometries, as well as perturbations of these geometries, has been analyzed for various operating conditions. Results of the analyses show that some combinations of operating parameters, exacerbated by the sparse lubrication that exist in the HPOTP bearings, can cause unstable cage oscillations. Frequencies of cage oscillations have been predicted by the CAGEDYN numerical model by Fourier analysis of predicted cage motions. Under conditions that cause unstable cage motion, high frequency oscillations have been predicted that could cause premature cage failures.

INTRODUCTION

The bearings of the high-pressure oxygen turbopump (HPOTP) of the Space Shuttle main engine (SSME) are critical to the success of the engine. Improvement of these bearings is mandatory if the target design life of 27,000 seconds is to be met for future shuttle missions. While significant improvements have been made in the bearings, bearing life is still limited by marginal lubrication. One problem associated with marginal lubrication is that the bearing cage has a propensity for being dynamically unstable. The goal of this paper is to present examples of analytical predictions of HPOTP cage stability that have been made with the Battelle CAGEDYN bearing computer model.

BACKGROUND ON CAGE STABILITY ANALYSIS

Predictions of stability of the HPOTP bearing cages were made using the computer code CAGEDYN (CAGE DYNamics), developed at Battelle in the early

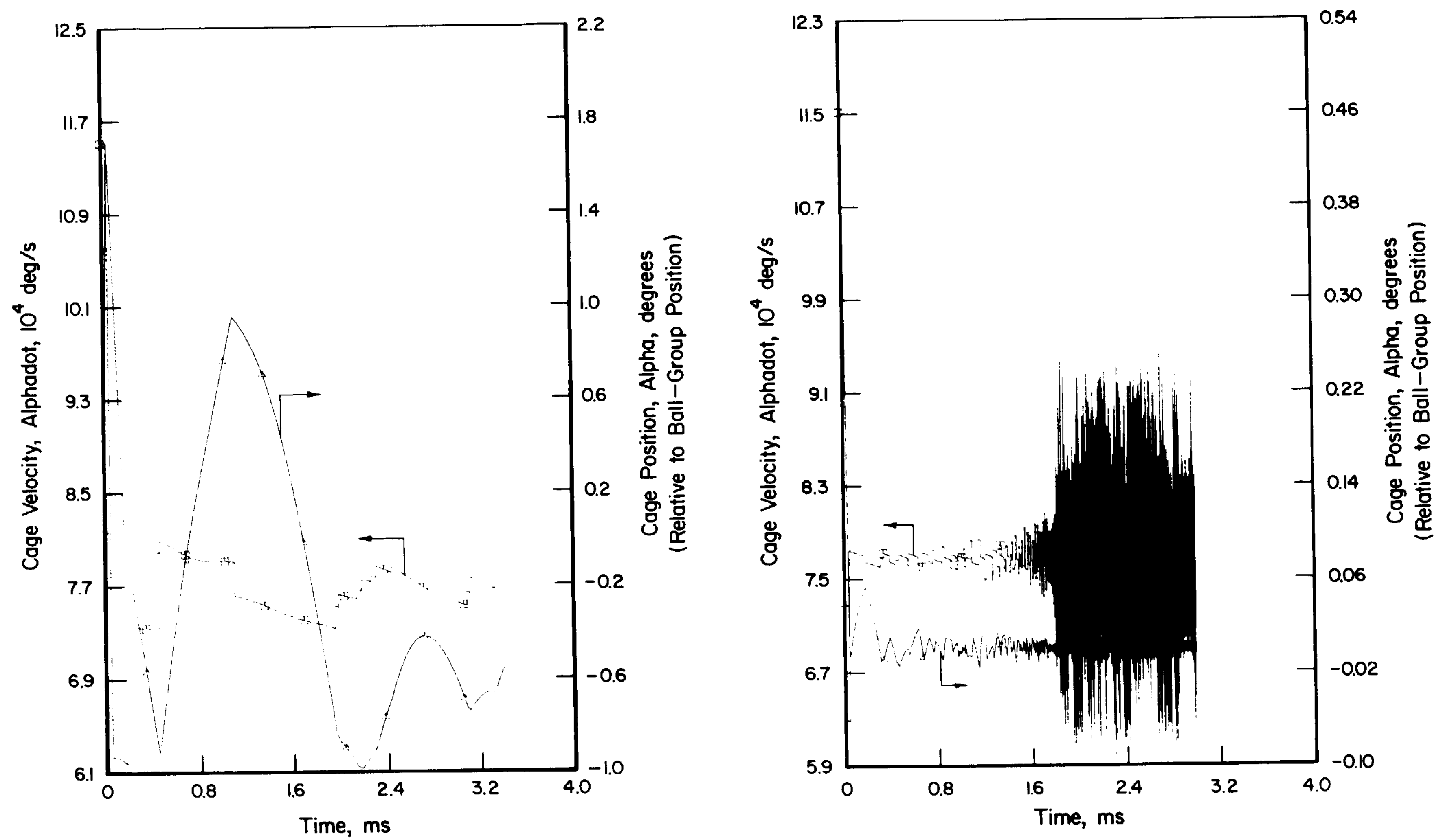
1970s. Since its inception, the computer code has been used to evaluate bearings for many industrial and military aerospace applications. Details of the analytical development of CAGEDYN are documented in Reference 1.

Output from CAGEDYN is in the form of graphs of cage velocity, (Alphadot) relative to the cage center of mass, and the velocity of the cage center of mass (Betadot), relative to the cage geometric center. Integrating the cage velocities (Alphadot and Betadot) produces cage positions (Alpha and Beta) that are plotted relative to the ball-group position. From examination of graphs of the cage position and velocity, a prediction is made concerning cage stability.

Examples of plots of predicted cage motion are presented in Figure 1. In Figure 1a, the cage position (Alpha) initially oscillates between plus or minus 1.0 degree, leading or lagging the mean ball-group position, as a result of an initial energy input (for the example considered, the cage can typically lead or lag the ball group 1 degree before a ball-cage impact occurs). After approximately 1.0 ms the amplitude of oscillations is attenuated; this cage is predicted to be stable. Conversely, in Figure 1b after 1.8 ms both the amplitude and frequency of cage oscillations increases dramatically as the kinetic energy of the cage builds; this cage is predicted to be unstable.

Another phenomenon of concern, cage whirl, is derived from the CAGEDYN prediction of Beta, which is the position of the cage center of mass plotted relative to the bearing geometric center. Cage whirl is a steady rotational motion of the cage center of mass, generally at a greater frequency than the ball-group speed (Ref. 2). The results presented in Figure 2 show the characteristic behavior that indicates the absence or presence of a tendency towards cage whirl.

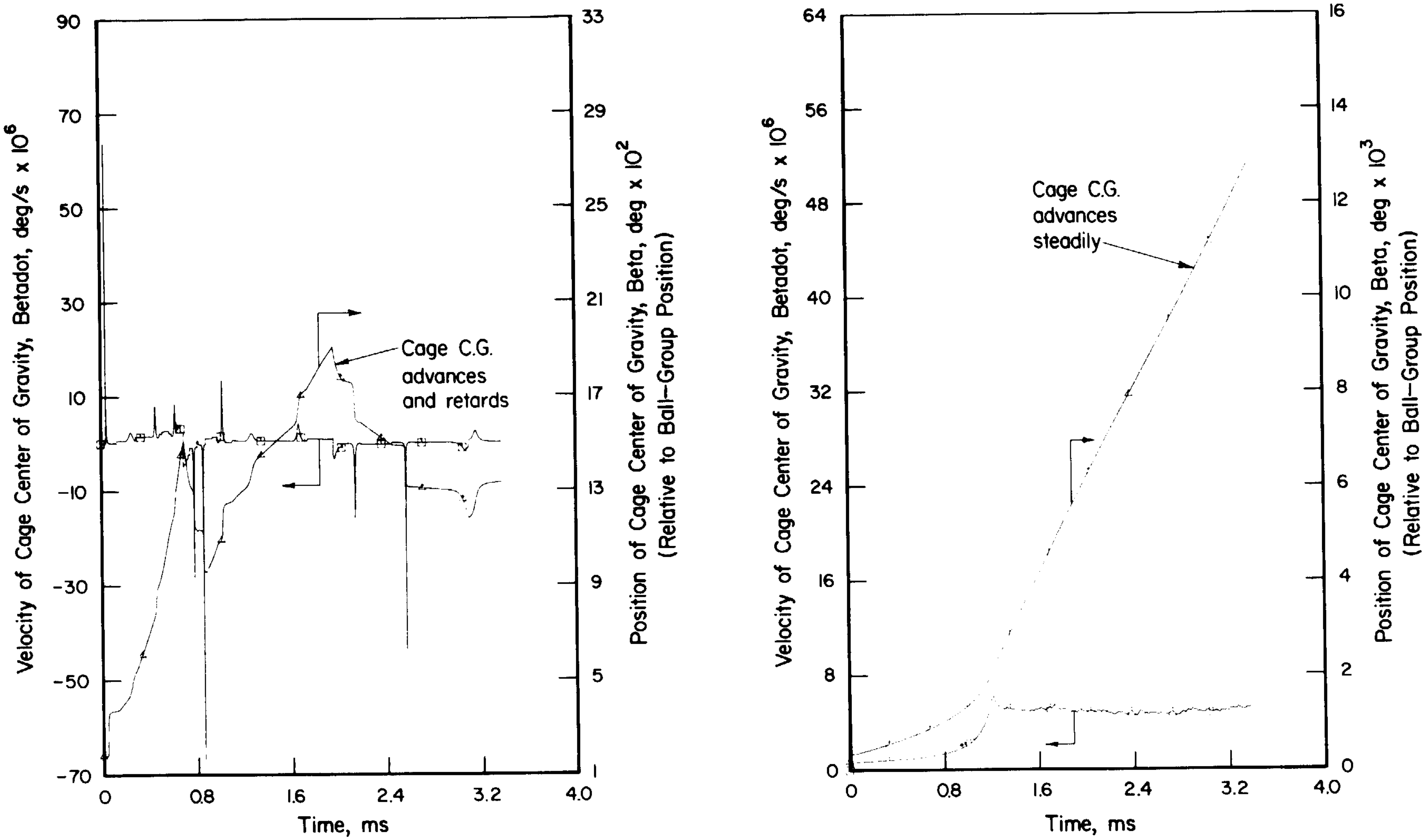
In the example of Figure 2a the motion of the cage center of mass advances and retards with time. This would produce nondescript frequency data in which particular frequencies are not dominant. Figure 2b shows a CAGEDYN prediction for a bearing where the position of the cage center of mass is steadily advancing over the ball-group position, which is interpreted as whirl. Since the motion is steady, a complete frequency analysis is not necessary. The slope of the plot of Beta can be taken directly from the plot and used to calculate a whirl frequency (19,100 Hz).



A. STABLE CAGE MOTION

B. UNSTABLE CAGE MOTION

FIGURE 1. EXAMPLES OF BEARING CAGE MOTIONS PREDICTED BY CAGEDYN COMPUTER MODEL



A. NO CAGE WHIRL PREDICTED

B. CAGE WHIRL AT 19,000 Hz

FIGURE 2. PRESENCE OF CAGE WHIRL AS INDICATED BY CHARACTERISTIC SHAPE OF PLOTS OF MOTION OF CAGE CENTER OF GRAVITY, BETA, PREDICTED BY CAGEDYN COMPUTER MODEL.

CAGE STABILITY ANALYSES FOR BASELINE HPOTP BEARING CONFIGURATIONS

Two bearing sets support the main shaft of the HPOTP. A set of 45 mm bore bearings, designated as bearing No. 7958, support the pump end of the HPOTP shaft. A set of 57 mm bore bearings, designated as bearing No. 7955, support the turbine end of the shaft. Figures 3 and 4 show results predicted by CAGEDYN for the two basic configurations. From examination of the figures, both the No. 7958 and No. 7955 bearing cages are predicted to be stable. In addition, steady whirling was not predicted for either cage for the operating conditions listed on the figures.

Although both of the basic bearing configurations are predicted to be stable under nominal operating conditions, further analyses of the bearing have indicated some combinations of operating conditions that theoretically promote cage instability or promote cage whirl. A discussion of operating parameters that can promote cage instability is presented below.

PARAMETRIC STUDIES OF CAGE STABILITY

In the parametric study, a nominal configuration was selected, one parameter was varied at a time, and the sensitivity of the cage stability to the parameter was observed. In some cases, the parameters were interrelated and the computer model must take these interrelationships into account. For example, a change in axial load causes changes in contact stresses, contact angles, and contact area geometry.

Influence of Radial Load

For the smaller bore No. 7958 (pump end) bearing the axial load was fixed at 2224 N (500 lb) and the radial load was varied between 0 N (0 lb) and 6673 N (1,500 lb). The results showed that an increase in radial load tends to stabilize the cage. At the highest radial load the cage lagged significantly behind the ball group speed and a steady cage whirl was predicted.

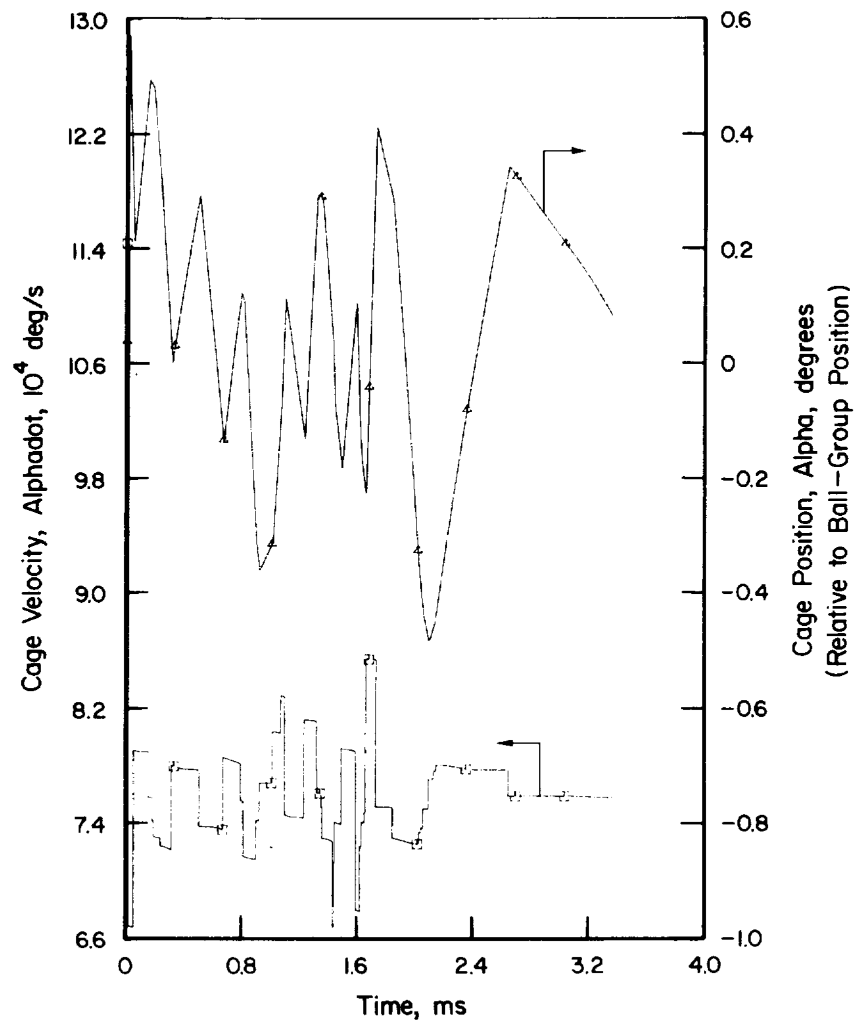


FIGURE 3. CAGEDYN PREDICTION OF CAGE MOTION FOR NO. 7958 (PUMP END) BEARING.

Number of Balls: 13
Pitch Diameter: 65.0 mm (2.56 in.)
Outer Race Curvature: 0.52
Inner Race Curvature: 0.55
Axial Load: 2224 N (500 lb)
Radial Load: 3114 N (700 lb)
Cage-Outer Race Diametral Clearance: 0.203 mm (0.008 in.)
Ball-Pocket Diametral Clearance: 0.635 mm (0.025 in.)
Shaft Speed: 30,000 rpm

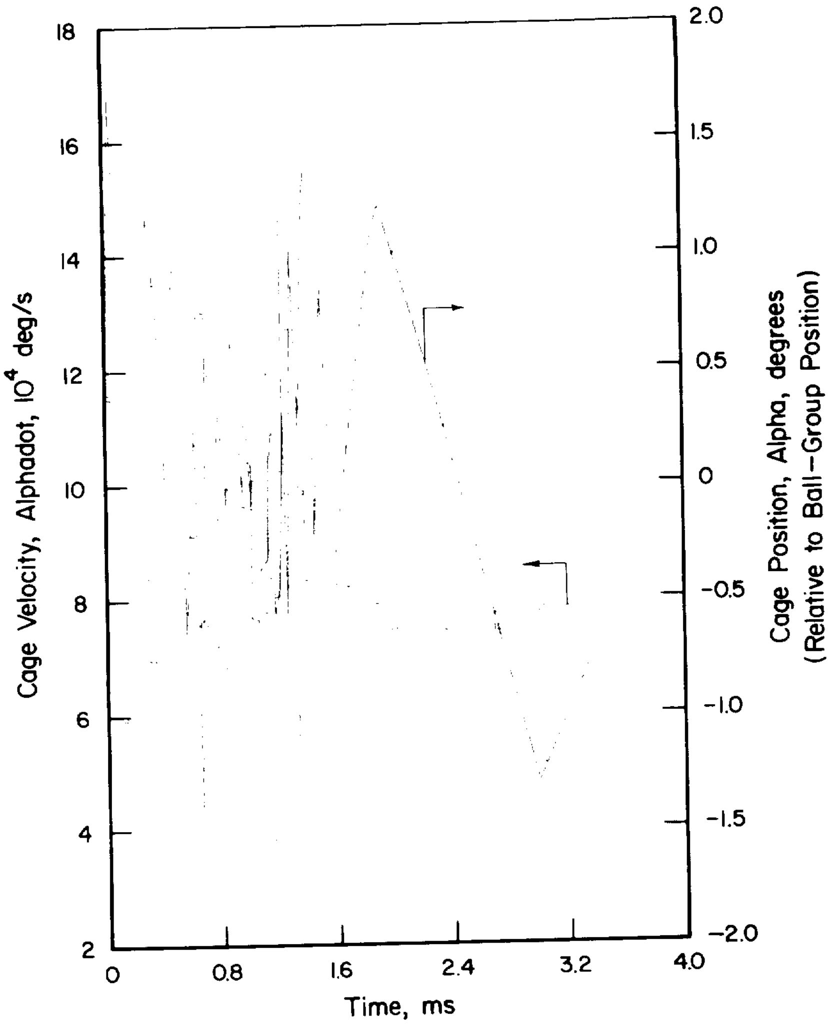


FIGURE 4. CAGEDYN PREDICTION OF CAGE MOTION FOR NO. 7955 (TURBINE END) BEARING.

Number of Balls: 13
Pitch Diameter: 80.5 mm (3.17 in.)
Outer Race Curvature: 0.53
Inner Race Curvature: 0.53
Axial Load: 2224 N (1100 lb)
Radial Load: 3114 N (1400 lb)
Cage-Outer Race Diametral Clearance: 0.737 mm (0.029 in.)
Ball-Pocket Diametral Clearance: 2.41 mm (0.095 in.)
Shaft Speed: 30,000 rpm

Influence of Cage to Outer Race Clearance

For the No. 7955 (turbine end) bearing the diametral piloting clearance was varied from 0.254 mm (0.010 in.) to 1.27 mm (0.050 in.). The results showed that the No. 7955 bearing cage had a propensity for instability at the larger piloting clearances. For the largest clearance used in the analysis, reduced cage stability and an unsteady cage whirl mode at approximately 3,040 Hz were predicted. Previous work with this bearing, as well as No. 7955 bearing, suggest a hypothesis that the cage whirl frequency is related to the cage-race clearance.

Influence of Friction

For the CAGEDYN analysis there are three frictional interfaces of interest on the cage: the ball-cage contact, the cage-outer race contact, and the ball-race interface. Analyses of the pump-end and turbine-end bearing has shown that of the three frictional interfaces, the ball-race contact has the most significant impact on cage stability.

The ball-race contact requires somewhat complex analyses to characterize (Ref. 3). The results of the analyses of Reference 3 were used in determining a ball-race traction coefficient, which was related to an average effective viscosity for use in the CAGEDYN analysis. The solid film used for the analyses was polytetrafluoroethylene (PTFE). The analyses predicted an effective viscosity of 1×10^6 cp, for PTFE, for a traction coefficient of 0.1 to 0.2. Effective viscosities were also calculated for alternate solid lubricants such as molybdenum disulfide; the effective viscosities varied by a factor of 2 or 3 from one solid lubricant material to another. The cage stability was seen to be sensitive to this type of variation in the effective viscosity. A trend was established that the cage stability would degrade in parallel with a degraded lubricant condition at the ball-race interface.

An interesting comparison with the solid-lubricated bearing performance is the CAGEDYN prediction of cage motion for a liquid lubricated bearing. Conditions in the HPOTP bearing preclude the use of liquid lubrication other than a flow of cryogenic fluid diverted through the bearing. Experimental evaluations of the elastohydrodynamic film forming capabilities of cryogenic fluids has shown that only meager lubrication can be achieved with these fluids (Ref. 4), thus the cryogenic fluid has little (if any) beneficial effect on bearing stability. Even if a conventional high viscosity fluid could be used in the HPOTP, the high contact pressures in the bearing would cause the lubricant viscosity to approach the value approximated for the solid film. The CAGEDYN stability prediction for such a case would be therefore similar whether the solid-film PTFE lubricant or a conventional lubricant were used. If a low viscosity lubricant could be used, the predicted stability of the HPOTP bearing cage would be greatly improved. However, this viscosity would have to be sufficient to allow for the formation of an elastohydrodynamic film.

Influence of Ball-Pocket Clearance

For the No. 7955 (turbine end) bearing, the ball-pocket diametral clearance was varied over a range from 0.635 mm (0.025 in.) to 2.54 mm (0.100 in.). Changes in the ball-pocket clearance did not cause significant changes in the predicted cage stability for this bearing. The primary effect of an enlarged ball pocket was to allow larger ball excursions before ball-cage impacts occurred.

Influence of Shaft Speed

For the larger bore No. 7955 (turbine end) bearing, the influence of shaft speed on cage stability was estimated by varying the shaft speed from 26,000 rpm to 30,000 rpm. The results showed that the bearing may be only marginally stable; small percentage changes in speed cause a change in predicted stability. This behavior has been seen in previous analyses at different speeds and indicates that the bearing speed sensitivity is probably interrelated with other bearing parameters. At 29,000 rpm occasional oscillations occur, which indicate that the bearing is near a critical frequency. At a low speed, 26,000 rpm, the cage motion is stable but has a propensity for whirl.

FREQUENCIES OF OSCILLATION OF THE BEARING CAGE

By examining the plots of cage motion predicted by the CAGEDYN computer model, interpretations are made as to whether a cage is stable or unstable. In order to determine characteristic frequencies of cage oscillations, which can be compared with experimental data such as from accelerometers, frequency analysis computer software was added to the CAGEDYN model. An explanation of the function of the frequency analysis model is given below followed by examples of frequency analysis of the No. 7955 bearing at various operating speeds.

Details of the Analytical Development of the Frequency Analysis

To perform a frequency analysis with the CAGEDYN computer model digital records of cage motion in the time domain are analyzed. The number of points and the size of time step used in the CAGEDYN analysis determine the resolution and total bandwidth of the frequency analysis. A frequency of 50,000 Hz was selected as the point at which frequency analysis of the HPOTP cage motion data was cut off. This value is 10 times typical shaft speeds and the amplitude of coefficients from frequency analysis were found to have decayed substantially by the 50,000 Hz point. To convert the time domain cage motion data to frequency domain, the entire cage motion data record is first multiplied by a Hanning window. The Hanning window is an accepted digital signal analysis tool (Ref. 4 and 5) which is necessary because the CAGEDYN motion data given for a few revolutions of the cage is only a window of continuous time signal simulating a longer time history. The cage motion data may therefore contain discontinuities that would be troublesome to the Fourier analysis technique. The purpose of the Hanning window is to remove the effect of discontinuities at the edges of the continuous time signal windows prior to Fourier analysis.

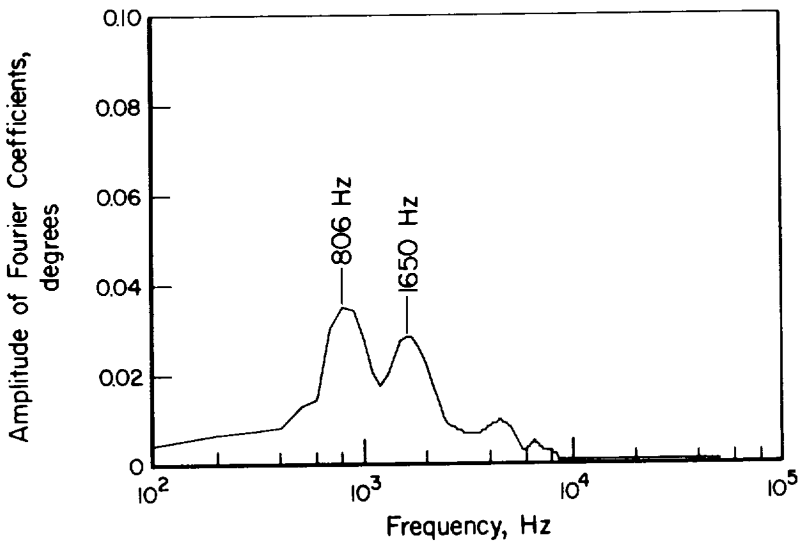
General Trends Observed in Frequency Analysis

As a result of frequency analysis of the HPOTP bearing under various operating conditions, several characteristics observed in the plots of

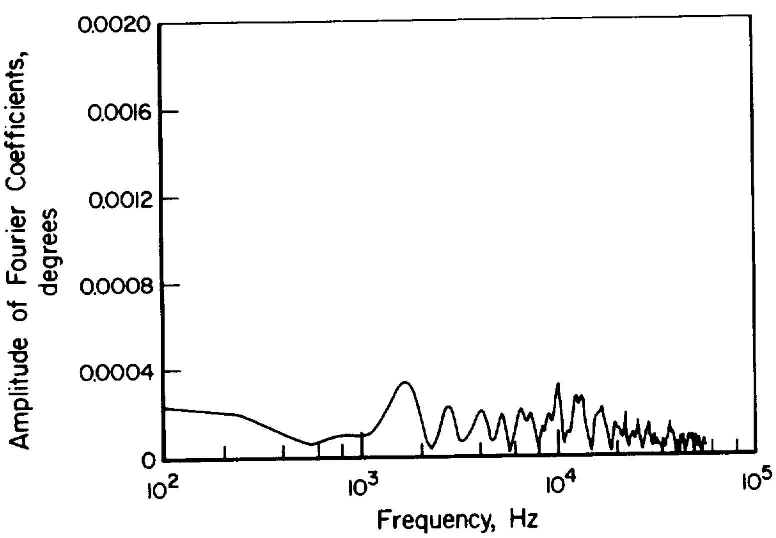
Fourier coefficients of the cage motion indicated what mode of operation the bearing was experiencing. Figure 5 presents a comparison of frequency analyses predicted for stable and unstable cage oscillations. A stable case, such as in Figure 5a, typically produces a frequency plot with one or two dominant peaks in the 500 Hz to 2,000 Hz range. Amplitudes of oscillations are substantially attenuated below 10,000 Hz. Conversely, an unstable case, such as in Figure 5b, does not show peaks of oscillations at a dominant frequency. Instead, amplitudes of Fourier coefficients are of lower amplitude, more randomly scattered, and extended to much higher frequencies.

CONCLUSIONS

The CAGEDYN computer model predicts the bearing cages of the SSME HPOTP to be marginally stable. Analyses have shown that perturbations of various operating parameters can increase the likelihood of an instability or a cage whirl mode. Lubrication at the ball-race interface has shown a strong influence in controlling stability and improvements in this area might make the cage stable over a wider range of conditions. Frequency analysis of the predicted cage motions has shown characteristic frequency spectra under stable operation with one or more dominant oscillation peaks in the 500 Hz to 2000 Hz range. Under unstable cage operation, peaks of oscillation are more randomly scattered and are predicted to extend to higher frequencies.



A. EXAMPLE OF CHARACTERISTIC SHAPE OF FREQUENCY SPECTRUM PREDICTED FOR STABLE CAGE MOTION



B. EXAMPLE OF CHARACTERISTIC SHAPE OF FREQUENCY SPECTRUM PREDICTED FOR UNSTABLE CAGE MOTION

FIGURE 5. COMPARISON OF EXAMPLES OF FREQUENCY ANALYSIS OF STABLE AND UNSTABLE CAGE MOTIONS PREDICTED BY CAGEDYN COMPUTER MODEL

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