

## SPIN BEARING RETAINER DESIGN OPTIMIZATION

Edward A. Boesiger\* and Mark H. Warner\*\*

## ABSTRACT

The dynamic behavior of spin bearings for momentum wheels (control-moment gyroscope, reaction wheel assembly) is critical to satellite stability and life. Repeated bearing retainer instabilities hasten lubricant deterioration and can lead to premature bearing failure and/or unacceptable vibration. These instabilities are typically distinguished by increases in torque, temperature, audible noise, and vibration induced into the bearing cartridge. Ball retainer design can be optimized to minimize these occurrences. A retainer was designed using a previously successful smaller retainer as an example. Analytical methods were then employed to predict its behavior and optimize its configuration.

## INTRODUCTION

Ball retainer instability, usually characterized by a squealing noise, is not uncommon with bearings operating at certain speeds and conditions. It has been observed in gyroscope bearings ("chatter") [1], momentum wheel bearings [2], and main shaft turbine engine thrust bearings [3], but is by no means limited to these types. Momentary periods of instability are normally non-destructive, such as noise heard during run-up. However, prolonged operation with instability can eventually lead to cage failure.

At the onset of retainer instability, ball-retainer collisions in a bearing occur with progressively increasing force, and the motion of the retainer departs from simple rotation (for more information about retainer instability see [4, 5]). These retainer motions can include random motions as well as repeating whirling-type behavior. The energy to drive the retainer unstable is believed to come from friction or viscous drag of the lubricant. Retainer instability is therefore aggravated by lubricant degradation or migration. When the instability begins, increased forces and friction in the bearing cause the power dissipation to increase, the ball slip along the races to increase, and thus the temperatures in the bearing increase. These increased

\* Lockheed Missiles and Space Company, Inc., Sunnyvale, California

\*\* Honeywell Corporation, Satellite Systems Operations, Glendale, Arizona

temperatures further the degradation of the lubricant, which propagates this instability cycle.

A life test of a control-moment gyroscope (CMG) spin bearing system was conducted under a relatively strenuous duty cycle. The bearings developed excessive drag torque prior to the end of their design life. Subsequent inspection of the test article revealed that a retainer had broken and that the lubricant had degraded. This failure prompted an investigation for an improved system.

Practice indicates that adequate lubrication reduces the risk of instability. A retainer can be designed to make it less susceptible to friction-induced instabilities. Therefore, a design effort was undertaken to optimize the retainer design for an existing bearing. To increase retainer strength, the land diameters of this bearing were set to their limits based on stress and spill-over in order to allow the use of the thickest practical retainer. Then, an "optimal" retainer was designed based on stability using a commercial dynamic bearing code in conjunction with other empirical tests. Predicted results generally correlated with bearing test data. The final configuration of the retainer is presently being manufactured and will be thoroughly tested.

## ANALYSIS

### Bearing Analysis Code

The modeling of the spin bearing was primarily accomplished using the commercially available ADORE (Advanced Dynamics of Rolling Elements) computer program [6]. This program simulates the dynamics (time and position dependent) of the bearing components, in contrast to other bearing computer programs that only calculate quasi-static (position dependent) conditions. A dynamic model is required to simulate retainer instability. Several dynamic bearing codes are available [5, 7]; however, ADORE was thought to be the most comprehensive and best suited for this application, although it is the most computer resource intensive.

To use ADORE for this design effort, pre- and post-processing code development was required. The retainer designs analyzed required accurate mass properties information. A pre-processor program was written to calculate the mass properties and transform them into ADORE format. A post-processor program was also written to assimilate the large amount of ADORE output data into a form where comparisons

among designs could be made. The many lessons learned about using the ADORE program were incorporated in a comprehensive user manual. Considerable training was found necessary to use ADORE effectively.

In addition, the ADORE program required modification to incorporate a retainer with a full inertia tensor and general mass center location. The retainer is a patented Honeywell design that incorporates a 'force' and 'moment' bias. The bias is achieved by offsetting the retainer mass center from the rotation center. A full inertia matrix (moments and products of inertia) is also required to characterize the retainer dynamics. Figure 1 shows an exaggerated schematic of a retainer with force and moment bias. The coupling of the products of inertia and the angular velocity of the retainer produce a 'moment bias' torque that tends to rotate the retainer out of the plane of the ball centers. The radial offset of the retainer mass center produces a force bias due to the centrifugal acceleration of the mass center. The force bias is thus proportional to the square of the instantaneous retainer rotational speed.

It was found that when modeling this retainer, ADORE was sensitive to small changes in input parameters, such as lubricant traction values, retainer stiffness, friction, and damping. Numerous tests and analyses were performed to identify accurate values for the system being modeled. These included ball-race lubricant traction parameter tests and retainer rub-block friction testing with the actual materials, surface finishes, and retainer dimensions. In addition, finite element modeling and tests for stiffness and modulus of the retainer material were conducted. Tests to determine damping coefficients, weight, and density were also performed. A thermal analysis was conducted for accurate temperature inputs into ADORE.

### **Design Evaluation**

The primary question asked of the analyst at every parameter change is, "Is the bearing now stable or unstable?" To obtain an optimal design, a refinement to this question is whether or not a particular change to the design is an improvement toward being more stable. A performance ranking method was needed to quantify small differences in the many indicators of stability (forces, wear, motion, etc.) between two cases that have similar, but not exactly the same, results. Two such methods were developed: using the critical friction threshold and combining stability indicators into a performance number. Both give

the same results, but the performance number is more efficient for optimization of the retainer.

### **Critical Friction Threshold**

The hypothesis for the onset of instability (insufficient or degraded lubrication) led to an investigation of the friction level that causes instability. It was found that a critical level of friction exists at which a dramatic deterioration in motion stability and force magnitude is observed. This friction level is defined here as the critical friction coefficient. It was found that different retainer designs had different critical friction coefficients, and thus the retainer with a higher coefficient has more resistance to the effects of lubricant degradation.

In order to evaluate the critical friction coefficient of a particular design, as many as ten to fifteen computer runs were required. While this method of using the critical friction coefficient to compare designs appeared to work well, there were occasional inconsistencies in the results. The transition between the stable and unstable regions was usually a distinct friction coefficient value. However, there were instances where this transition occurred over such a large range that no meaningful comparisons to other designs could be made. In this range, increases in friction caused an unstable motion to become stable again. For example, a retainer design was stable with a friction coefficient less than 0.15, unstable with a friction level of 0.16, stable again for a friction coefficient of 0.17, and then unstable for coefficients greater than 0.19. The large number of simulations required and the inconsistencies observed led to the development of an alternate method that could be used for the optimization.

### **Performance Number**

In order to rank the stability of bearing designs quickly and quantitatively, a method using a 'Performance Number' (PN) was developed. The Performance Number permits direct comparison of stability for those cases where a clear distinction is not obtained by observation of the results.

The PN defined here answers the question "Is it now more or less stable than the nominal or previous case?" After running many parameter variation cases, it was realized that a definitive stable or unstable label was difficult to assign to each case. There are many requirements for a bearing; if the motion is not perfectly stable but the forces, torques, and wear are low, the bearing may be acceptable.

Although instabilities with irregular retainer motions may produce noise and torque variation, they may not result in mechanical failure of the retainer. Excessive forces, however, may produce structural failure. Generally, the more stable in both motion and forces, the better the bearing is for meeting requirements for full bearing life.

The Performance Number is intended as a comparison of the relative stability of designs, on the basis of both the motion and forces in the bearing. Two values used for the PN are calculated from the computer model output: the Stability Performance Number (SPN), based on retainer motion, and the Wear/Power Performance Number (WPPN), based on the forces in the bearing. The SPN and WPPN are combined to form the PN. Increased values of all three numbers relate to an increasingly stable bearing.

The PN provides a comparison relative to the nominal case. Thus, the PN does not measure absolute stability, only relative stability. If the nominal case is assigned a PN of 1, a case with a PN of 0.5 may or may not be "unstable," but is less stable than the nominal case. However, if the second case's PN is greater than 1 and if neither the WPPN nor SPN decreases significantly, the second case is considered more stable than the nominal case. This method is also applicable in optimization work where a modified case is compared with a previous case. The PN simply compares both to the nominal case, and the one with the higher PN is regarded as the most stable.

The SPN comprises the following indicators of retainer motion stability: retainer whirl ratio, retainer omega ratio, retainer mass center orbit shape, and the integration run time. Whirl and omega ratios give an indication of how the retainer is moving relative to the race rotation, and are constant when the retainer motion is stable [1]. A small, steady ball-retainer contact force produces a circular mass center orbit, while random or whirling retainer mass center motion results when the contact forces are large and have no defined pattern, as is the case during instability. Figure 2 illustrates the ADORE whirl ratio and orbit plots for a stable and unstable condition. Integration run time, though not an obvious stability indicator, is included because stable retainers have fewer and less severe contacts, and the integration proceeds more easily than if the retainer is unstable.

The WPPN is a measure of wear and power stability based on the forces, torques, and power losses in the bearing. Experience has shown

that stable bearings have lower drive torque, less power loss, and lower retainer contact forces than unstable ones. The WPPN is calculated by combining the values of drive torque, power loss, retainer wear rate, ball-retainer forces, and retainer-race forces. Torque, power, and forces in the bearing are somewhat obvious indicators of bearing stability. The slope and value of the retainer wear rate is also an indicator of retainer stability. If the forces and sliding velocities are increasing, the wear rate increases, and this generally indicates instability.

### **Optimization Methods**

After developing a performance criteria for evaluating retainer designs, an efficient method of iterating to an improved overall design was required. Common methods, such as Monte Carlo, and a full matrix of possibilities were rejected due to the large number of simulations needed. The ADORE simulations were run on a Cray computer and required approximately 1500 CPU seconds to run. The man-hours and computer time needed to make the hundreds or thousands of simulations for these methods was time and cost prohibitive. Another common method is to vary each parameter separately to the end of its tolerance range and, after observing its effect on the stability, combine all the parameters in a 'best-on-best' fashion. This method would have had limited success with this bearing dynamics problem, as it was found that the parameters have a significant amount of coupling among them. It was not possible to predict the effect of changing the value of one parameter when other parameters were also changed.

The optimization technique of steepest descent avoids the problems of these other methods. In this method, several variables are changed a small amount for each computer run rather than varying one parameter at a time. A systematic approach was desired, especially when the interactions of the parameters were unknown. Steepest descent is based on the premise that limiting the changes in variables to small fractions of their initial value tends to keep the effect on the output monotonic. This means that an optimal design can be found by iterating, using small increments of the variables in the direction of increased stability for each. Based on having the PN to determine relative stability among cases and the smaller number of runs required, the steepest descent method was chosen to do the retainer design optimization.

### **Optimization Results**

The strategy for this study was to find an improved design that

took into account the interactions of the various parameters. The optimization was based on changing only the retainer design. Changing the ball size or further changes to the races would have too great an impact on the system design (shaft, cartridge, housing, etc.). Thus, the following five primary characteristics of the retainer were chosen as variables:

- 1) retainer pocket-ball diametral clearance
- 2) retainer-race diametral clearance
- 3) number of balls
- 4) force bias
- 5) moment bias

The optimization study was carried out while operating under the nominal operating conditions, which included gravity, a small radial load, a shaft speed of 6000 RPM, and retainer-race and ball-retainer friction coefficients of 0.12.

The nominal condition was run to establish a baseline. This nominal case has a PN of 1.0. Each of the five parameters was then varied either positively or negatively in separate runs. For each parameter, it was then determined whether the change had improved or degraded the retainer dynamic stability (i.e., by the PN). It should be noted that the steepest descent method was initially tested using a simpler bearing dynamic analysis program with good results [8]. However, when using ADORE, the steepest descent method with the PN required too fine a change in the parameters for the PN to accurately predict the direction to change the variables for increasing stability.

A parameter study and experimental data indicated the primary parameters that influenced the retainer stability. Of the five parameters listed previously for the steepest descent study, all except the moment bias had a significant effect on stability. Figure 3 illustrates the effect of two of these parameters (land and pocket clearance) on the retainer stability. The land clearance, pocket clearance, number of balls, and force bias parameters were then used in a 'partial matrix' optimization. The realistic range that these parameters could be varied was divided into three possibilities and all the combinations of these parameters were simulated (54 cases). The simulations were run with a high friction value to help distinguish between stable and unstable.

The high friction level eliminated a large number of the cases, and

there were only a few noticeably superior designs. This was confirmed with the PN of the cases. These few designs were then rerun with various operating conditions seen by the spin bearing system, including zero gravity, high radial loads, degraded lubricant, different speeds, and various press fits between the bearing and the shaft and housing. The results of these runs were more important than initially expected. It became obvious that one particular design was the best performer throughout the range of operating conditions. This design included smaller retainer pockets, larger retainer-land clearance, one less ball, and larger force bias.

## TESTING

The bearing dynamics problem (and simulation) is complicated, and there is considerable uncertainty associated with these types of calculations and model assumptions. As a result, it was believed that testing for confirmation and confidence in the model was essential. The optimized retainer design also had to be verified. Two bearing test fixtures were used to observe stability. An existing fixture, Universal Bearing Test System (UBTS), was considered a retainer screening tool. The Bearing Stability Tester (BeST) tests an end-item duplex bearing pair and was developed in parallel to complement the testing on the UBTS. Since the UBTS tests were done with one of the bearings from the duplex pair, the BeST more closely simulates the actual bearing system configuration.

### UBTS Fixture

The UBTS is a fixture that was developed to perform ball bearing dynamics research. The fixture was designed primarily as a single bearing screening device, but it can also be configured to accept a preloaded pair. Figure 4 shows the UBTS, along with its support equipment. The spindle is suspended on an air bearing, and the axial preload on the bearing can be varied by adjusting the gas pressure. Radial and axial induced vibration forces are measured using two piezoelectric load cells. In addition, two linear variable differential transformer deflectometers measure bearing drag torque. The amplitude variation of the induced vibration radial component at the retainer frequency, as well as audible noise and an increase in drag torque, give indications of retainer instability.

The UBTS was used in conjunction with a "rub block" fixture that measured the coefficient of friction in both the ball pocket and land



regions of the retainer. By accurately measuring these friction values and then evaluating the retainer's performance on the UBTS, a retainer's stability characteristics as a function of friction could be established. The retainer's friction coefficient could be increased by either cleaning with a solvent (Freon) or by vacuum baking the retainer. In this manner, a critical friction coefficient for a given retainer design was found. For friction levels above this value, the retainer exhibited unstable characteristics, and when below the critical value, the retainer appeared stable.

Figure 5 shows a plot of the Performance Number versus friction coefficient for both the old retainer and the new design optimized by the computer modeling. This friction coefficient is input in the model at both the ball/pocket and retainer/land interfaces. Superimposed on the graph are the actual UBTS-measured values for critical friction coefficients for the two retainers. While the actual values for measured and analytical critical coefficients are not exactly equal, the model did correctly predict that the new retainer would be less susceptible to increasing friction (i.e., the new retainer would have a higher critical friction coefficient). This empirical correlation is very important because as the lubricant degrades, friction forces on the retainer increase. The model-optimized design allows the retainer to remain stable for approximately 50% larger values of friction than the original design.

UBTS was also used to verify the model's prediction that the retainer stability was more sensitive to increasing friction in the ball pocket region than in the land region. Two retainers were completely cleaned using Freon TF and then run on UBTS. Both exhibited unstable behavior (audible noise, large variation in radial induced vibration, etc.). The friction coefficients at the ball-pocket and land-retainer interfaces were measured and found to be 0.36 and 0.44, respectively, on both retainers. The inner land of one of the retainers was then swabbed with a very light coat of lubricant, while the pockets of the other were lubricated in a similar manner. Both retainers were then rerun on UBTS. The retainer with its pockets lubricated did not exhibit any unstable behavior, while the retainer with the inner land lubricated behaved exactly as it did when it was unlubricated. The coefficients of friction of both retainers were then measured again. The retainer with its pockets lubricated had a friction coefficient of 0.23 in its pockets, and 0.44 in its land region. The retainer with its land region lubricated had a friction coefficient of 0.36 in its pockets, and 0.28 in its land

region. This verification of the model's prediction that the ball/pocket interface is the driver of retainer instability gave confidence in the output of the model.

### BeST Fixture

The BeST fixture was developed to simulate the end-item configuration while being able to measure the forces, motor current, accelerations, bearing temperatures, speed, and retainer motions. The bearings were rotated by a DC servo motor through a flexible coupling. The fixture was also designed to be compatible with mounting onto a shaker table. Figure 6 shows the BeST fixture and the associated wiring for the instrumentation.

A servo motor controlled the rotating speed of the bearing pair. The drive motor current is an indirect measure of the torque required to turn the bearings. A three-axis accelerometer was mounted on the side of the housing to measure radial and axial accelerations. Thermocouples measure the outer race temperature at two places on each bearing. An infrared pyrometer measured the temperature of the inner race. Two three-axis piezoelectric load cells were intended to measure the forces and torques in the bearing. However, the poor quasi-static performance and high thermal sensitivity of the cells on the BeST fixture made it difficult to obtain useful steady state information.

The BeST fixture was used to test the bearings with the old retainer design by running up the bearings from 0 to 6000 RPM. The bearings did exhibit unstable behavior at some speeds during the run-up, which is consistent with the results observed during the bearing life tests. The instabilities were confirmed by audible noise, an increase in bearing temperature, an increase in motor current, and a sharp rise in acceleration. Figure 7 is a spectral waterfall-style plot for the axial direction during a run-up. The acceleration is plotted versus frequency for every 100 RPM. It can be seen that during certain speed ranges, a rise in acceleration at about 1500 Hz occurs. The same frequency peaks at instability are also seen in the other two accelerometer directions. This instability frequency has not been positively associated with a physical bearing characteristic, although it does correspond to the natural frequency of the ball-retainer collisions. At one of these speeds, 3400 RPM, the ADORE program was run at a high friction level to simulate the instability. The post-processing code was then used to produce retainer acceleration spectral plots for the radial, orbital and axial directions. The analysis found the predominant frequency during

instability to be 1500 Hz in the orbital and radial directions and 1700 Hz in the axial direction. These frequency values show general correlation between test and analytical results.

An important output of the computer simulations is the motion of the retainer mass center. It was desired to observe this retainer motion for both a stable and unstable condition on the BeST fixture. Cage motions were measured using non-contact proximity probes. A 0.008-inch thick aluminum ring was pressed onto the outer diameter of the inner-race guided retainer. Two proximity probes were mounted in the housing 90° apart, and were placed so that they could read the radial motion of the retainer. The bearings were rotated, and when no audible noise was heard, the proximity probe information was recorded. The proximity probe information was converted to mass center position and plotted as an orbit plot. The comparison of the ADORE prediction and the BeST test for this stable condition is shown in Figure 8.

The addition of the aluminum ring did change the retainer's dynamic characteristics. The ring was attached on the one side of the retainer not preloaded against the other bearing in the pair, and since the inner and outer retainer diameters are not concentric, the weight, force bias, and moment bias of the retainer were increased. These changes had the effect of stabilizing the retainer. The bearings had to be ultrasonically cleaned to remove any lubricant in order for the bearings to become unstable. Figure 9 shows the comparison between the ADORE prediction and the test result. Both mass center orbit plots show the whirling loops characteristic of the unstable retainer motion and there is good correlation between the two.

## CONCLUSION

An existing CMG bearing retainer design was optimized for stability using a bearing dynamics code in conjunction with experimental data. Bearing instabilities were observed during tests with proximity probes, accelerometers, and audible noise. Preliminary testing of retainer motion, instability frequency, and friction threshold have correlated with the computer modeling predictions. This design is being refined and fabricated and will be thoroughly tested in an end-item configuration. From this optimization effort, a number of important "lessons learned" became apparent:

- There is a critical friction level associated with each retainer design beyond which the retainer is unstable.

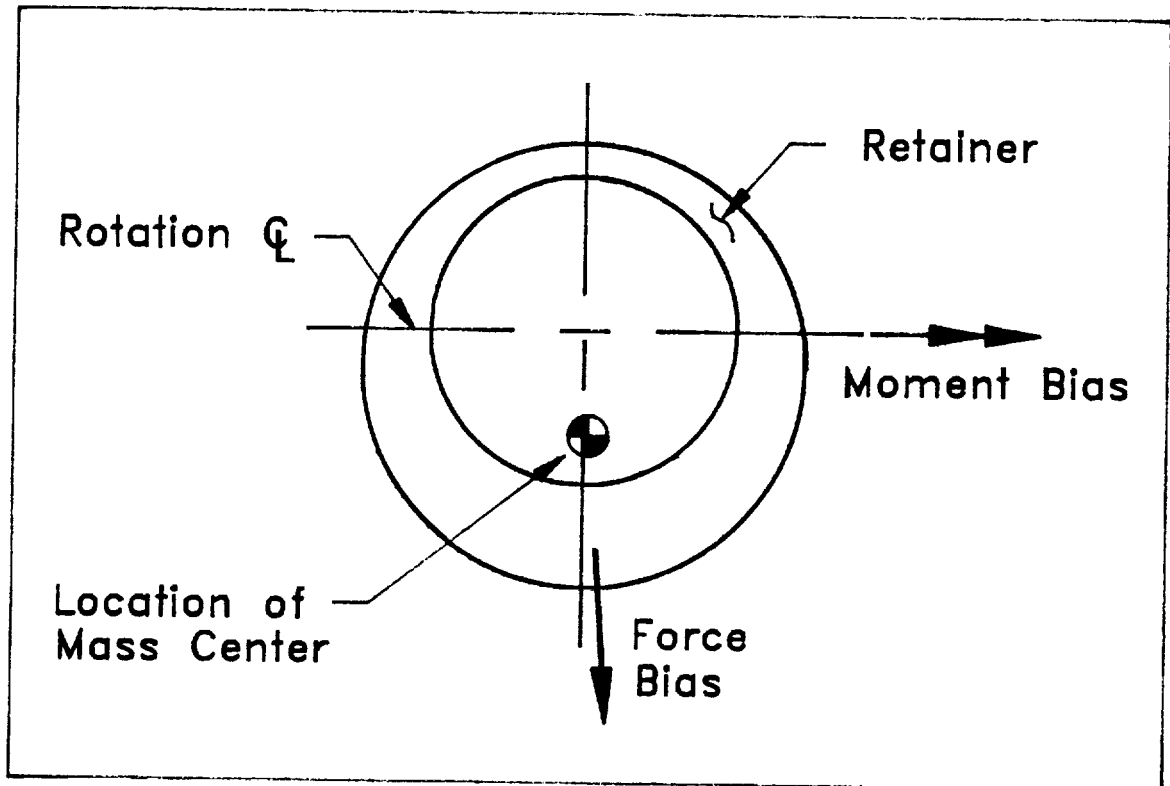
- Ball pocket friction is more critical to stability than retainer land friction for the bearing investigated.
- The nature of the bearing dynamics problem makes retainer optimization a very difficult task for the following reasons:
  - the parameters are coupled
  - the high accuracy of input data required
  - the specific operating conditions are important
- It is difficult to discern the effects of small changes on the indicators of stability from the bearing dynamics program.
- The steepest descent optimization method appears to be the most promising, but is hard to implement accurately for bearing dynamics.

### ACKNOWLEDGMENTS

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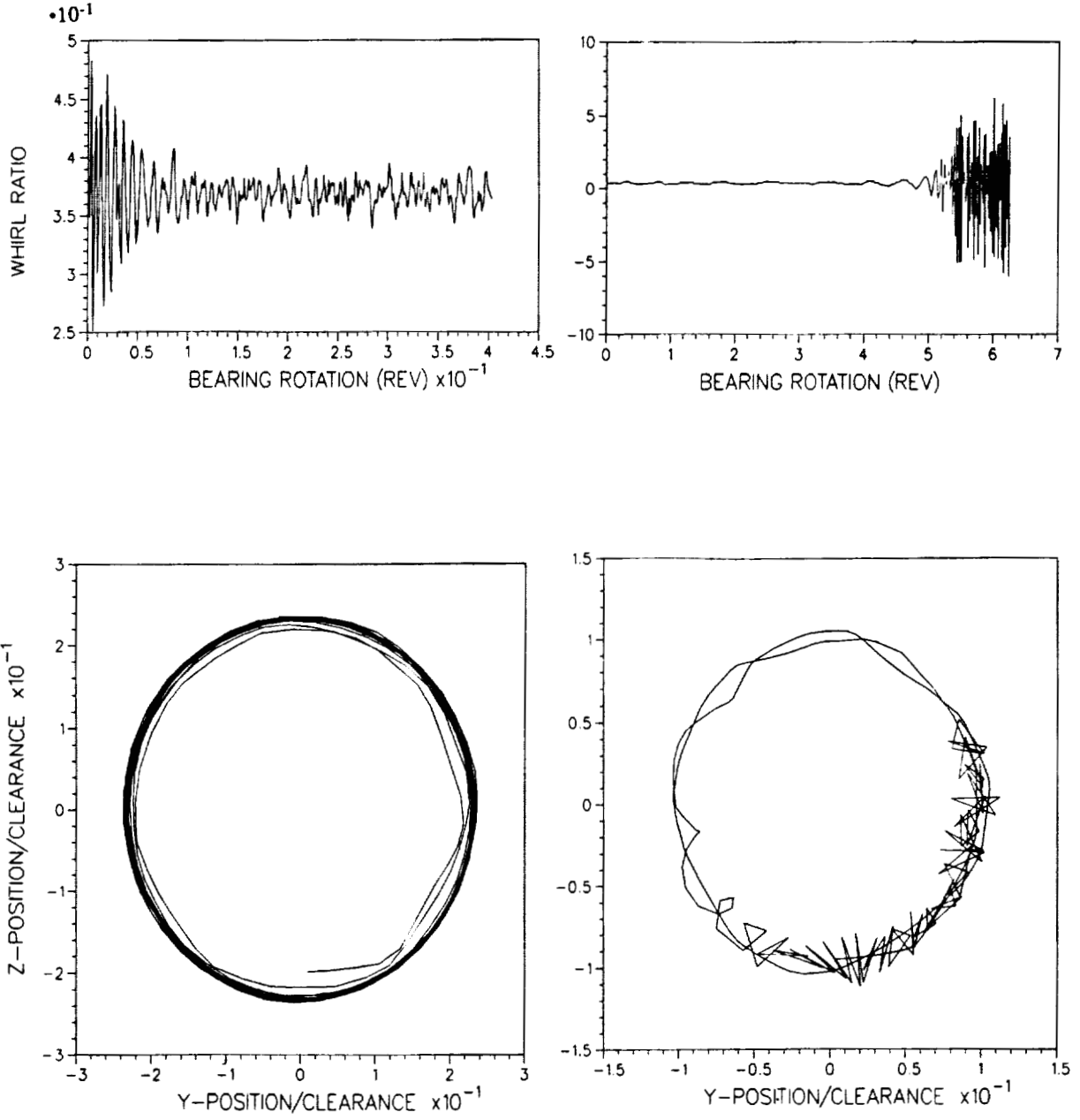
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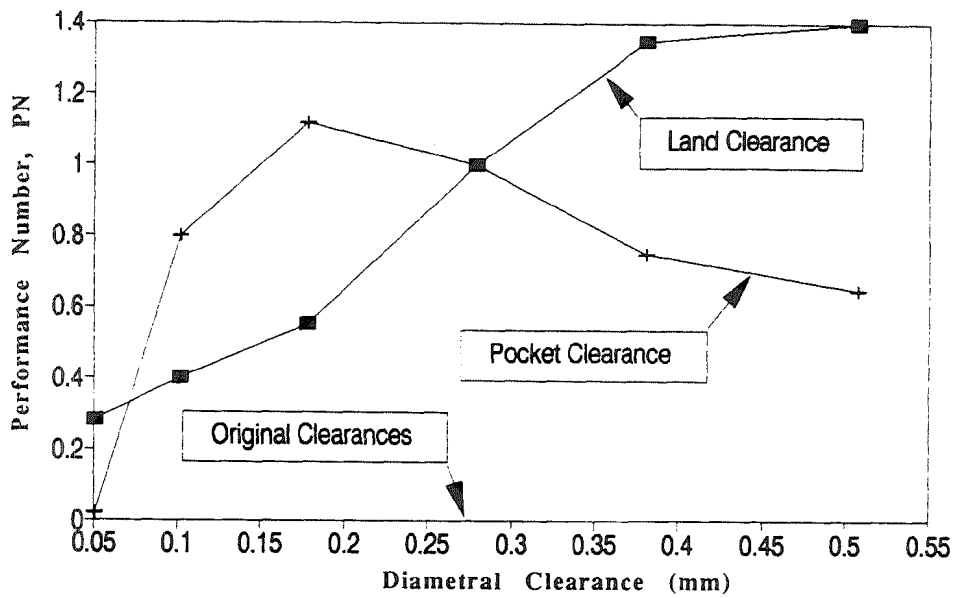
**Figure 1: Cage Force and Moment Bias**

Cage design offsets the mass center from the rotation center. At speed, the design induces forces and moments on the retainer.



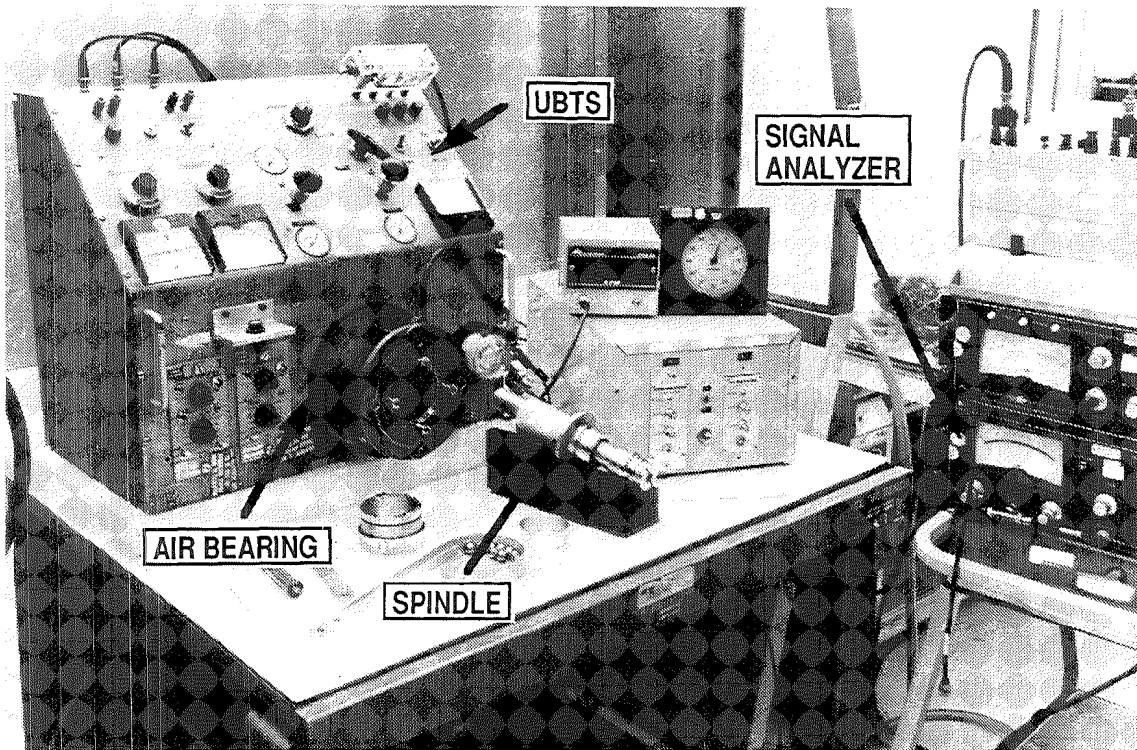
**Figure 2: Typical ADORE Output**

Shown are two indicators of retainer stability for runs where the retainer was stable (left column) and unstable (right column). The retainer whirl ratio is the ratio of the retainer mass center speed to the race speed. The orbit plots are the position of the retainer mass center relative to the bearing center.



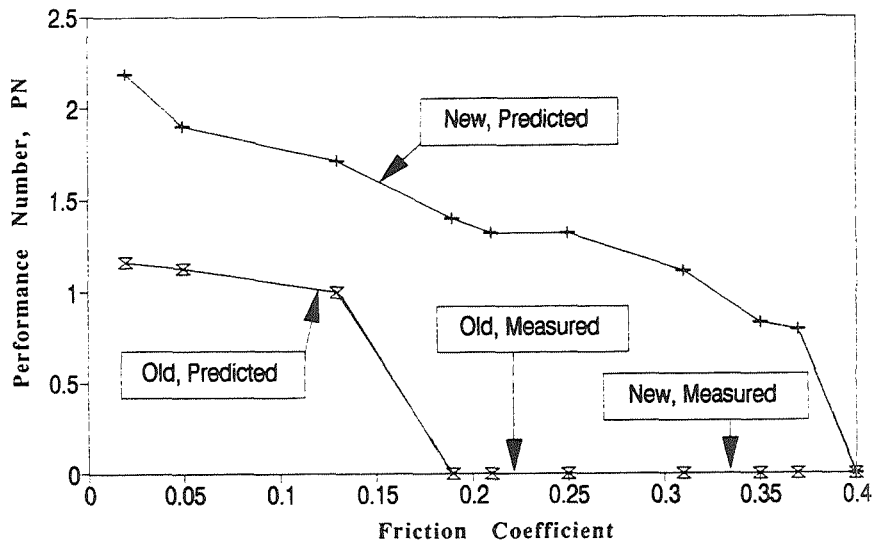
**Figure 3: Parameter Study Results**

Perturbations from the original design were studied to determine the sensitivity of the land and pocket clearance. Nominal land and pocket clearances of the original design were both 0.28 mm.



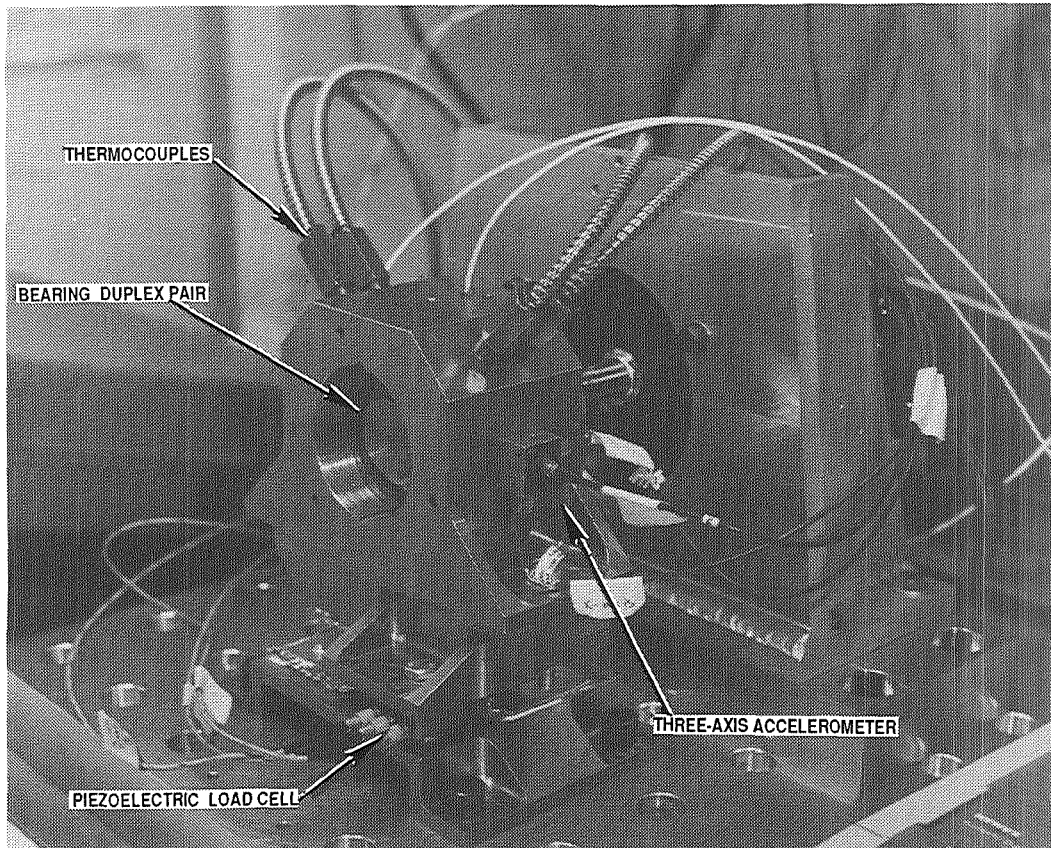
**Figure 4: UBTS Fixture**

Pictured is the UBTS fixture used for friction threshold bearing tests.



**Figure 5: Optimization Results**

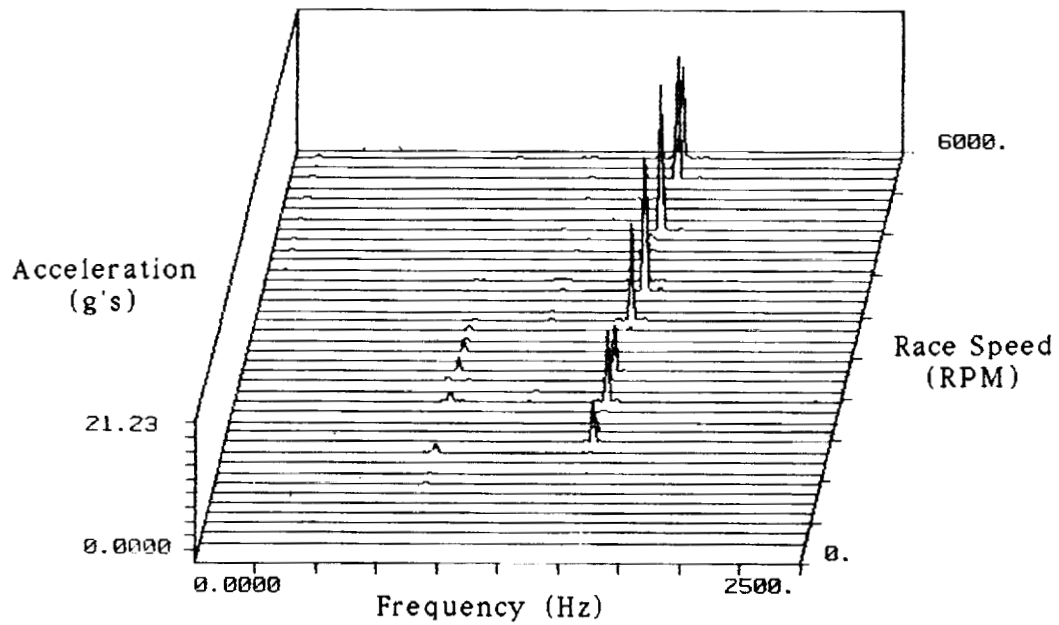
The optimal ("new") retainer is shown to be superior to the original ("old") design by the Performance Number, and by both the analytical and tested critical friction threshold.



**Figure 6: BeST Fixture**

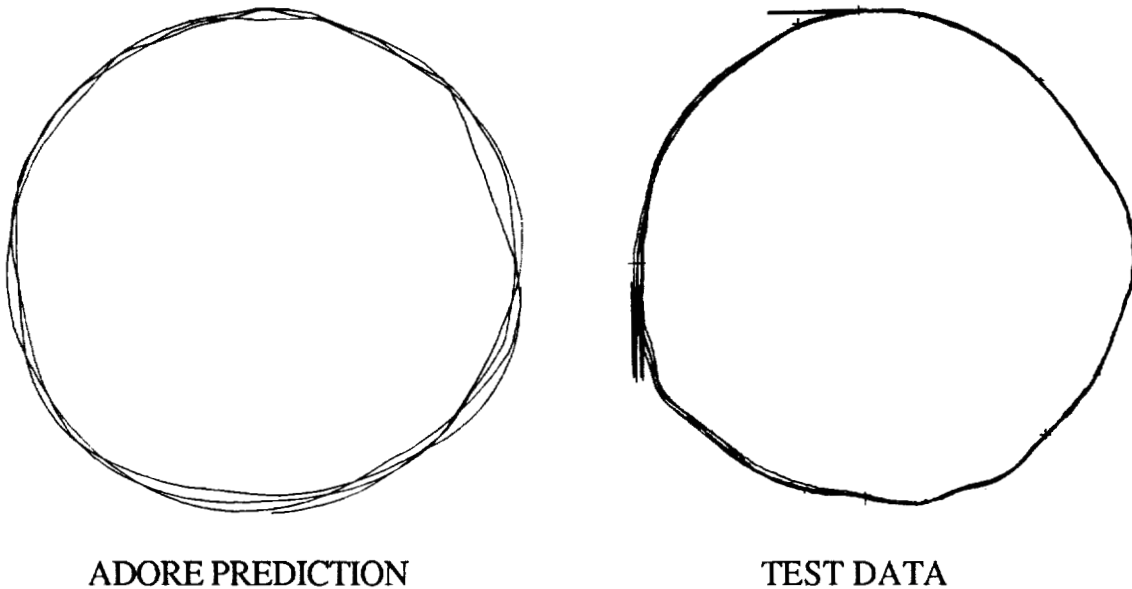
Pictured is the BeST fixture used for retainer acceleration and motion tests.





**Figure 7: Typical Accelerometer Spectral Plot**

A spectral plot of the axial direction is shown. Acceleration in g's is plotted versus frequency in Hertz for every 100 RPM from 0 to 6000 RPM. Note periods of instability distinguished by the high accelerations.

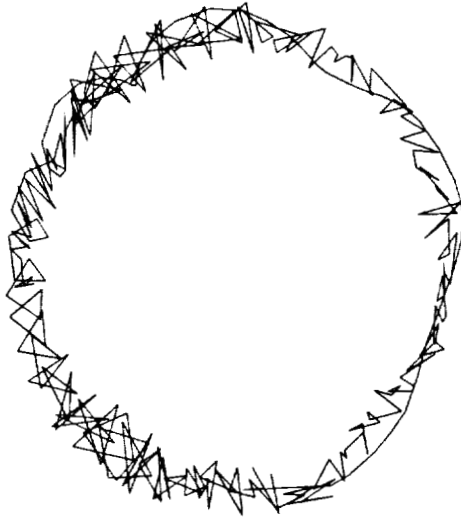


ADORE PREDICTION

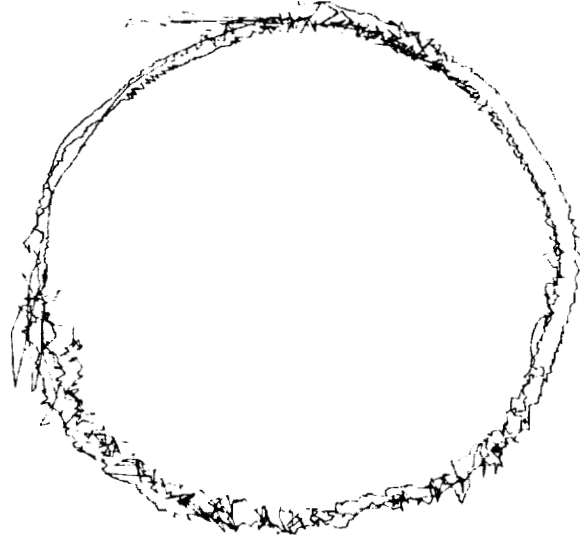
TEST DATA

**Figure 8: Stable Cage Mass Center Orbit Plots**

Mass center orbit plots from the analysis and test data for a stable retainer condition at 1750 RPM show a qualitative correlation.



ADORE PREDICTION



TEST DATA

**Figure 9: Unstable Cage Mass Center Orbit Plots**

Mass center orbit plots from the analysis and test data for an unstable retainer condition at 1780 RPM show a qualitative correlation.