DETERMINATION OF BALL BEARING DYNAMIC STIFFNESS*

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ABSTRACT

The dynamic radial stiffness characteristics of rolling element bearings are currently determined by analytical methods that have not been experimentally verified. These bearing data are vital to rotating machinery design integrity because accurate critical speeds and rotor stability predictions are highly dependent on the bearing stiffness. A tester was designed capable of controlling the bearing axial preload, speed, and rotor unbalance. The rotor and support structures were constructed to permit critical speeds that are predominantly determined by a 57 mm test bearing. A curve of calculated critical speed versus stiffness was used to determine the actual bearing stiffness from the empirical data. The results of extensive testing are used to verify analytical predictions, increase confidence in existing bearing computer programs, and to serve as a data base for efforts to correct these programs.

INTRODUCTION

The radial stiffness of a ball bearing is one of the key elements in the rotordynamic analysis of rotating machinery and is strongly dependent on axial preload, the constraints of the outer race and shaft speed. Attempts to determine bearing stiffness in the past by measuring applied loads and deflections have been complicated by the large required loads and very small deflections. Also, shaft rotation is difficult to include in such conventional tests. The method used in this experiment avoids these problems by utilizing a tester designed such that the lowest rotor critical speed is primarily a function of the test bearing stiffness. The overall purpose of this test program was to advance the technology of measuring ball bearing dynamic stiffness in support of theoretical rotordynamic analysis and rotating machinery design.

DESCRIPTION OF TESTER

The tester (Fig. 1) consists of a simple rigid rotor with control over the variables that affect the bearing stiffness. The significant features of the test rotor are as follows:

1. Axial preload can be accurately controlled by a hydraulic piston. This preload can be easily varied without disassembly of the tester.
2. Setscrew holes are provided in the large rotor disk for *in situ* balance corrections and access is provided in the supporting structure for easy corrections.
3. The rotor mass can be changed by substituting a heavy or lightweight spacer disk to shift the critical speed so that the stiffness can be found at various speeds. The rotor weight differential is 9.3 kg (20.5 pounds) between configurations.
4. The mass distribution of the tester rotor is proportioned so that the critical speed is effectively independent of the slave bearing stiffness.

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5. Locations are provided for two pairs of orthogonal displacement transducers to monitor rotor motion.

6. The test rotor is driven by a variable speed motor.

7. Different test bearing sizes can be accommodated by substituting bearing sleeves. The selection of a 57 mm test bearing was influenced by the radial stiffness effect of this bearing on the rotordynamic characteristics of the Space Shuttle Main Engine (SSME) High-Pressure Oxidizer Turbopump (HPOTP). Details of this turbine end bearing on the dynamic performance of this machine are documented in the literature by Rowan (ref. 1).

Figure 2 shows a schematic arrangement of the test setup, and Fig. 3 is a photograph of the actual installation. The rotor and supporting structures were designed so that their dynamic properties could be accurately predicted by analysis. Pertinent design data of the tester is presented in Table I.

ANALYTICAL APPROACH

A finite element model of the rotating assembly was generated, and a parametric computer analysis was performed to calculate the rotor critical speeds as a function of the test bearing stiffness to ground. The resulting critical speed plot is shown in Fig. 4, and the close spacing of curves A, B, and C indicate that the critical speed is nearly independent of the slave bearing stiffness. The critical speed can be easily detected by test, and by referring to Fig. 4, the test bearing stiffness can be determined. The slope of the curve in Fig. 4 indicates that the critical speed is highly dependent on the test bearing stiffness and should ensure accurate results. The lowest rotor critical speed mode shape is shown in Fig. 5.

By changing the mass of the rotor, the critical speed can be shifted so that a curve of test bearing stiffness versus speed can be plotted. Additional tests can be run with different axial preloads and/or unbalance to investigate these parameters.

TESTING

The general testing procedure consisted of determining the rotor residual unbalance, setting the hydraulic cylinder to maintain a given axial preload, and running a slow speed ramp through the critical speed while monitoring the rotor motion. The radial motion was monitored by two pairs of orthogonal displacement transducers that produced orbit plots on oscilloscopes. Photographic documentation of the orbits were taken every 1000 rpm starting at 15,000 rpm while speed was controlled manually to produce a series of step dwells to define the critical speed.

The rotor was installed with the heavyweight spacer, and the initial run indicated that no balance correction was needed because the residual was sufficient to produce a resonance, without excessively loading the bearings. All tests conducted are listed in Table II and the bearing stiffness was found by using the Fig. 4 curve, which relates the critical speed to test bearing stiffness. Table II also contains the analytically predicted stiffnesses in accordance with the procedures of ref. 2 for comparison.

Typical radial displacements from Test No. 3 are shown in Fig. 6 for 445 Newtons (100 pounds) axial preload and an estimated 6.4 gm-in. unbalance. This test indicated a critical speed at 20,000 rpm and a resulting test bearing stiffness of 9.1×10^7 N/m (0.52×10^6 lb/in.). Bearing outer race clearance in the fixture is evident by the abrupt post-critical drop in displacement above 22,000 rpm. This agrees with the theoretical behavior of a rotor with deadband bearing installations.

It was found during testing that the facility gearbox drain was not large enough to allow free oil flow above 23,000 rpm. Since increased preload or use of the lightweight spacer would tend to raise the current critical speed in the 19,000 to 20,000 rpm range, testing was limited to that shown in Table II and further testing suspended.

CONCLUSIONS

The tester and very practical testing technique functioned as intended with the exception of the gearbox lubrication, which is not related to the tester design. The data indicate that the dynamic stiffness of this 57 mm ball bearings is on the order of 8.75×10^7 N/m (0.5×10^6 lb/in.) for 445 to 890 Newtons (100 to 200 pounds) axial preload in the vicinity of 20,000 rpm. The value may differ in the usual spring preloaded installation since the preload piston in this test may resist bearing tilting more than a conventional preload spring and this tilt resistance may be responsible for some of the difference.

The analytically calculated value with the race free to tilt is approximately 20 percent lower than the measured value for the larger rotating radial load situation. A closer lower stiffness correlation is obtained for the lower dynamic loading cases. Stiffness increases for larger loading are typical of angular contact ball bearings and a nonlinear system. The data do indicate some sensitivity to the nonlinearity of the clearance between the outer race and the fixture.

Computer program bearing stiffness such as calculated per ref. 2 are adequate approximations for rotordynamic analyses if adequate design margin is incorporated. It is recommended as a guideline that rotor systems be designed so that the operating speed is not within ±20 percent of any calculated critical speed.

FURTHER TESTING

To increase scope, the tester is designed so that tests with various clearances between the support and outer race can be conducted. This clearance is generally referred to as the bearing deadband. The effect of outer race constraint can be investigated by machining the diameter of the test bearing support so that an increased clearance is produced and installing a preload spring between the hydraulic piston and outer race. The rotor would then be ramped through its critical speed with similar axial preload and unbalance conditions for comparison with the original clearance. This nonlinear effect of the deadband produces an apparent shift in the critical speed and influences the effective bearing stiffness. As of this time, these tests are still in the planning stages, and no schedule is available. Budget and the rotordynamic community interest will be the determining factors.

REFERENCES

1. Rowan, B. F.: "Rotordynamics Analysis of the Space Shuttle Main Engine High-Pressure Oxidizer Pump," presented at the 21st Structures, Structural Dynamics, and Materials Conference, Seattle, Washington, May 1980.

2. Jones, A. B.: "A General Theory for Elastically Constrained Ball and Radial Roller Bearing Under Arbitrary Load and Speed Conditions," Journal of Basic Engineering, June 1960.

TABLE I. - TESTER DESIGN DATA

ROTOR MATERIAL	INCO 718 STEEL
TEST BEARING	57 mm
SLAVE BEARING	45 mm
BEARING SPACING	34.3 cm (13.5 INCHES)
DUMMY MASS OUTER DIAMETER	20.32 cm (8.0 INCHES)
HOLLOW SHAFT OUTER DIAMETER	10.16 cm (4.0 INCHES)
ROTOR WEIGHT:	
HEAVY DISK	33.3 kg (73.5 POUNDS)
LIGHT DISK	24.0 kg (53.0 POUNDS)
MAXIMUM SPEED	35,000 RPM

TABLE II. - 57mm BEARING DYNAMIC STIFFNESS

TEST NO.	TEST DESCRIPTION	CRITICAL SPEED, RPM	MEASURED BEARING STIFFNESS* N/m X 10^{-8} (LB/IN. X 10^{-6})	ANALYTICAL BEARING STIFFNESS** WITH TILT AT 20,000 RPM N/m X 10^{-8} (LB/IN. X 10^{-6})	% ERROR
1	445 N (100 LB) PRELOAD ESTIMATED UNBALANCE = 2.8 GM-IN.	18,000	0.70 (0.40)	0.61 (0.35)	-12.5
2	890 N (200 LB) PRELOAD ESTIMATED UNBALANCE = 2.8 GM-IN.	19,000 TO 20,000	0.857 (0.49)	0.945 (0.54)	+10.2
3	445 N (100 LB) PRELOAD ESTIMATED UNBALANCE = 6.4 GM-IN.	20,000	0.91 (0.52)	0.70 (0.40)	-23.0

*SLAVE BEARING STIFFNESS CAN PRODUCE ±5% ERROR. AVERAGE CURVE B IS USED (FIG. 4)
**REF. 2, CALCULATED BY FRANCIS LEE, ROCKETDYNE.

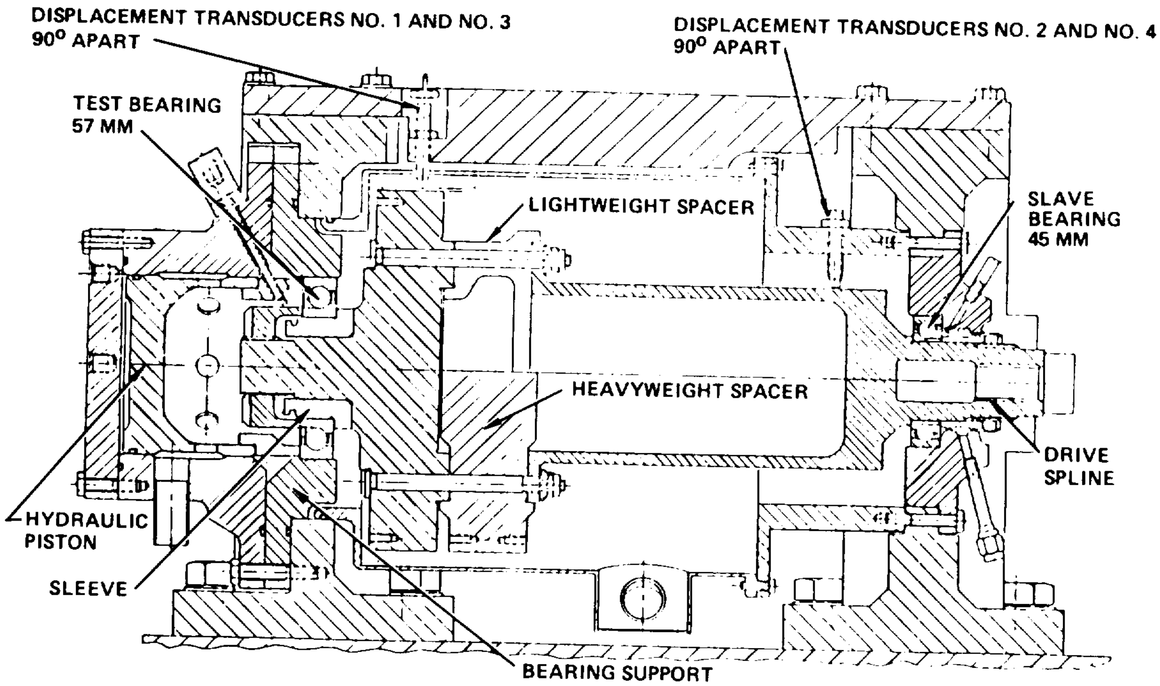

Figure 1. - Bearing tester.

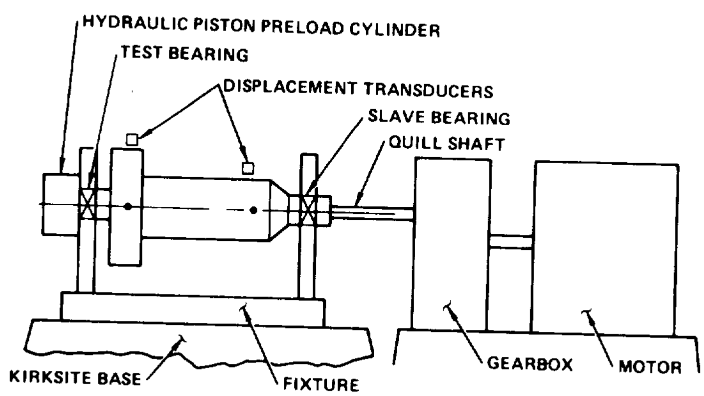

Figure 2. - Test setup schematic.

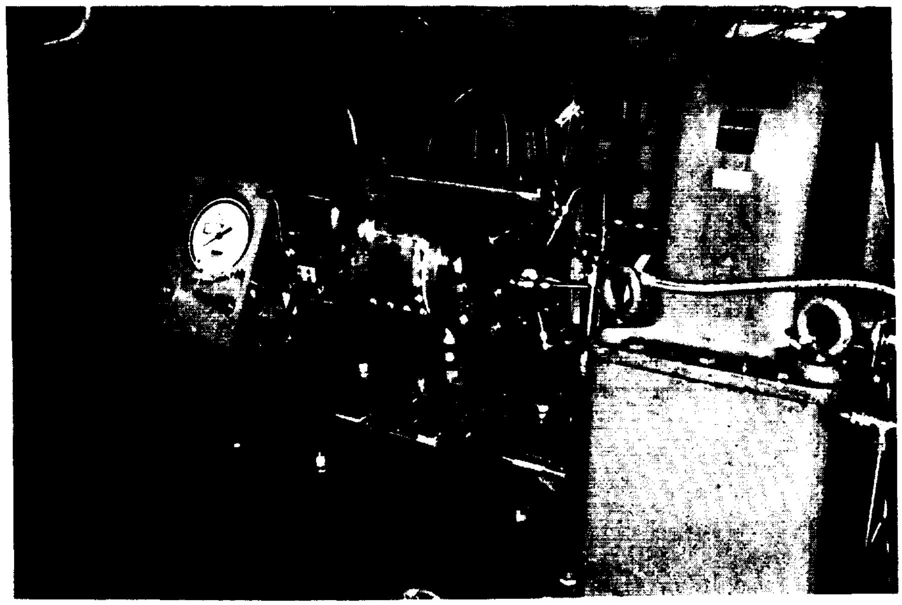

Figure 3. - Test setup.

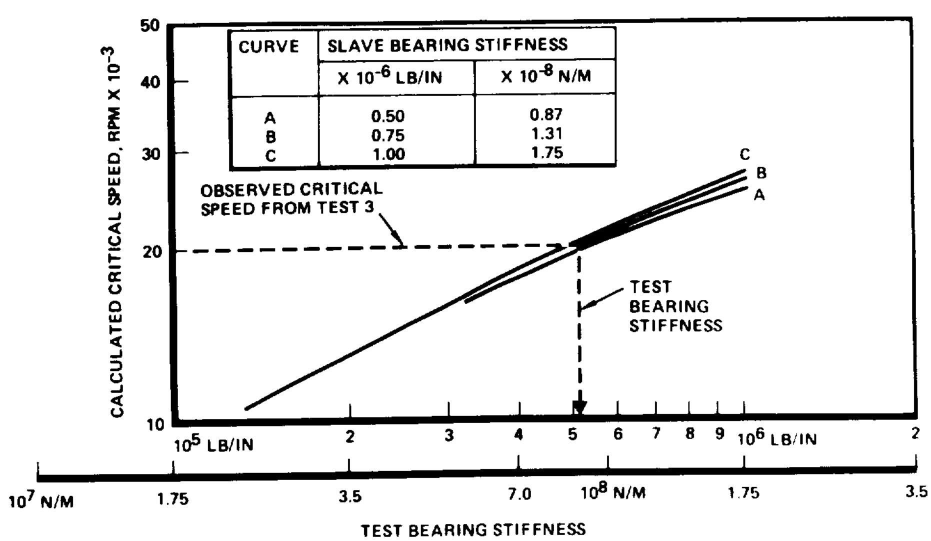

Figure 4. - Heavyweight spacer critical speed plot.

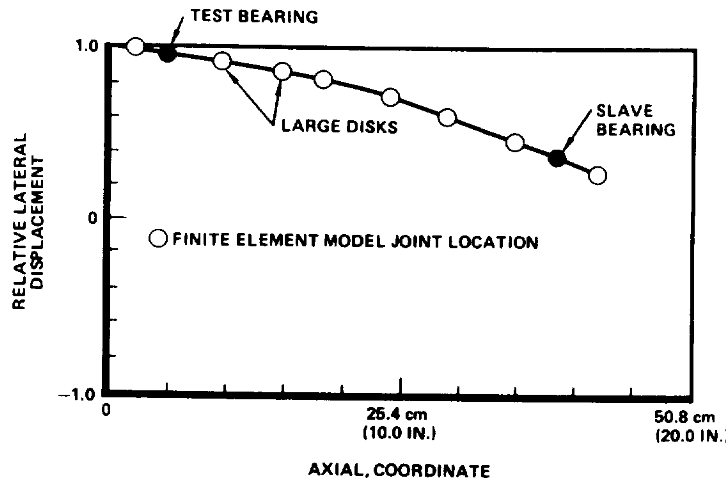

Figure 5. - Test rotor lowest mode shape relative displacement versus axial coordinate.

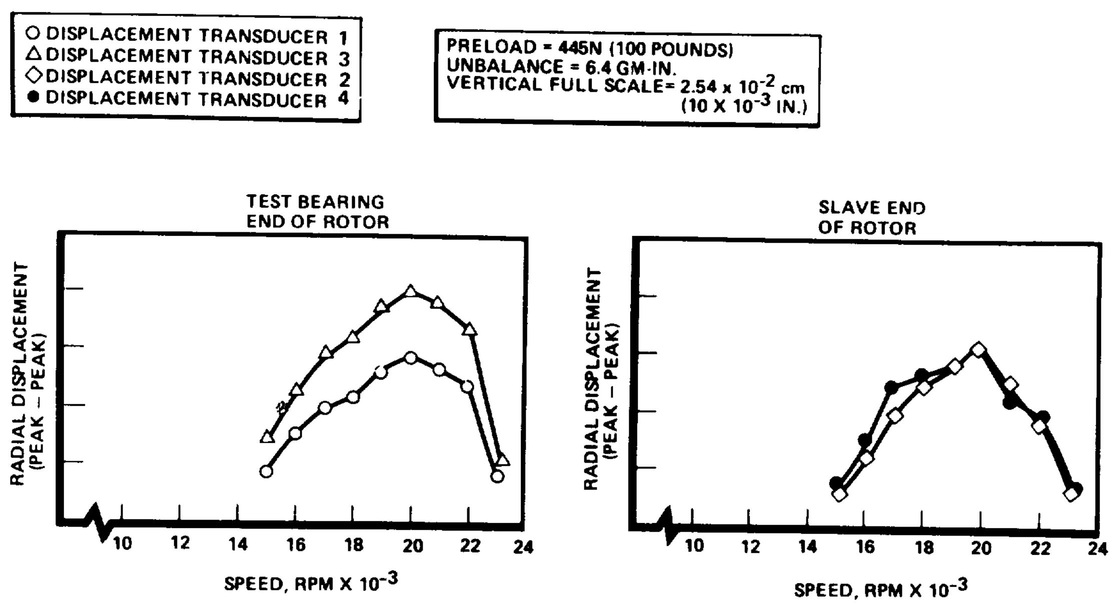

Figure 6. - Test no. 3 displacements vs. speed.