NASA/TM—2001-211131

Rotordynamic Influence on Rolling Element Bearing Selection and Operation

Gilbert K. Queitzsch, Jr.
Boeing Propulsion Technology, Seattle, Washington

David P. Fleming
Glenn Research Center, Cleveland, Ohio

Prepared for the
International Symposium on Stability Control of Rotating Machinery (ISCORMA 2001)
sponsored by the Bently Rotor Dynamics Research Corporation
South Lake Tahoe, California, August 20–24, 2001

National Aeronautics and
Space Administration

Glenn Research Center

August 2001

Available from

NASA Center for Aerospace Information
7121 Standard Drive
Hanover, MD 21076

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22100

Available electronically at http://gltrs.grc.nasa.gov/GLTRS

Rotordynamic Influence on Rolling Element Bearing Selection and Operation

Gilbert K. Queitzsch, Jr.
Boeing Propulsion Technology
Seattle, Washington 98124

David P. Fleming
National Aeronautics and Space Administration
Glenn Research Center
Cleveland, Ohio 44135

ABSTRACT

Three case studies are presented that illustrate the importance of dynamic considerations in the design of machinery supported by rolling element bearings. The first case concerns a milling spindle that experienced internal rubs and high bearing loads, and required retrofit of an additional damped bearing. The second case deals with a small high-speed generator that suffered high vibration due to flexible mounting. The third case is a propulsion fan simulator rig whose bearings failed catastrophically due to improper bearing installation (which resulted in inadequate dynamic bearing stiffness) and lack of health monitoring instrumentation.

INTRODUCTION

Bearings frequently experience dynamic forces that are significantly larger than steady unidirectional loads. These dynamic forces are termed rotordynamic loads; they may arise from a number of different sources, and are often associated with vibration in rotating machinery. During the late 1800s, rotating machinery builders identified *critical speeds* as those operating speeds where very high rotor vibration amplitudes would develop. Subsequent work during the early 1900s investigated the relationship of rotor response to imbalance, and also phenomena leading to self-excited whirl. A summary of this early history was compiled by Gunter in 1966 [1]. In this paper we consider only the effects of imbalance and its effects on rotordynamic behavior. We will present two examples of machines that experienced imbalance-related vibration and were successfully redesigned to accommodate normal operating levels of imbalance. A third example illustrates how inattention to bearing installation and operation can lead to detrimental changes in machine dynamics and catastrophic bearing failure.

Imbalance and Critical Speeds

Mass imbalance is the measure of how far the mass center of a rotor is offset from the rotational center (geometric centerline). Unbalance force is produced when the offset mass center is forced to rotate about the spin axis of the rotor. In the design of rotating hardware, major rotor components are manufactured to standard balance limits based on machine application [2, 3]. Balance of smaller parts is often controlled only by manufacturing tolerance limits. The alignment and centering of each rotor component is controlled by the accuracy to which the rotor joints (curvics, pilots, etc.) are manufactured. The final as-built rotor imbalance is the sum of component imbalances plus the stack error resulting from joint tolerances. Since no hardware can be built perfectly, each machine will have some random distribution of imbalance. After the rotor is assembled, a trim balance may be performed using a limited number of balance planes to bring the rotor balance into specification [4, 5]

All mechanical systems have natural frequencies of vibration. When the speed of a rotating machine coincides with a natural frequency, it is termed a *critical speed* or *balance resonance*. Under this condition, any imbalance in the rotating components excites the natural frequency. Vibration amplitude is limited only by system damping which is typically quite low unless high-damping devices such as squeeze film dampers are employed. In particular, damping in rolling element bearings is quite low, generally much less than that from fluid film bearings. Normal practice is to operate subcritical (below the critical speed), or to rapidly transit critical speeds on the way to a supercritical (above the critical) operating condition. Dwelling on or near critical speeds without appropriate damping can be very destructive because the resulting rotor response amplitude, high bearing reaction forces, rotor rubs, and component dynamic misalignment will adversely impact machine life and performance.

CASE STUDIES

Three case studies are provided to illustrate some rotor dynamic issues that may influence dynamic bearing loads. The first is a milling machine spindle in which the motor critical speed was excited by the cutter tooth passing frequency. The second example illustrates how case mounting flexibility can lead to amplification of bearing forces in a small high-speed generator. The third example illustrates the interrelationships among bearing internal clearance, operating load, bearing dynamic stiffness, and rotor critical speed. Each of these machines had been balanced to accepted tolerances. However, unintended excitation of dynamic resonances initially prevented successful operation.

Milling Machine Spindle Resonance

A 55 hp, 2,000 to 5,000 rpm milling machine spindle was experiencing occasional motor rubs and random bearing failures. A review of spindle operation found no correlation of failures to cutting severity or spindle maintenance, so an analysis of spindle operating dynamics was initiated. The analysis predicted the first bending mode of the spindle to be at 7078 rpm. This coincides with the tooth passing frequency of a two-flute cutter operating at 3539 rpm. The spindle cross section is shown in figure 1 with the mode shape overlaid. The mode shape reveals that the cutter tip is at a location of relatively low response while the motor response and bearing loads (fig. 2) can be quite high, even for small stable cutter deflections.

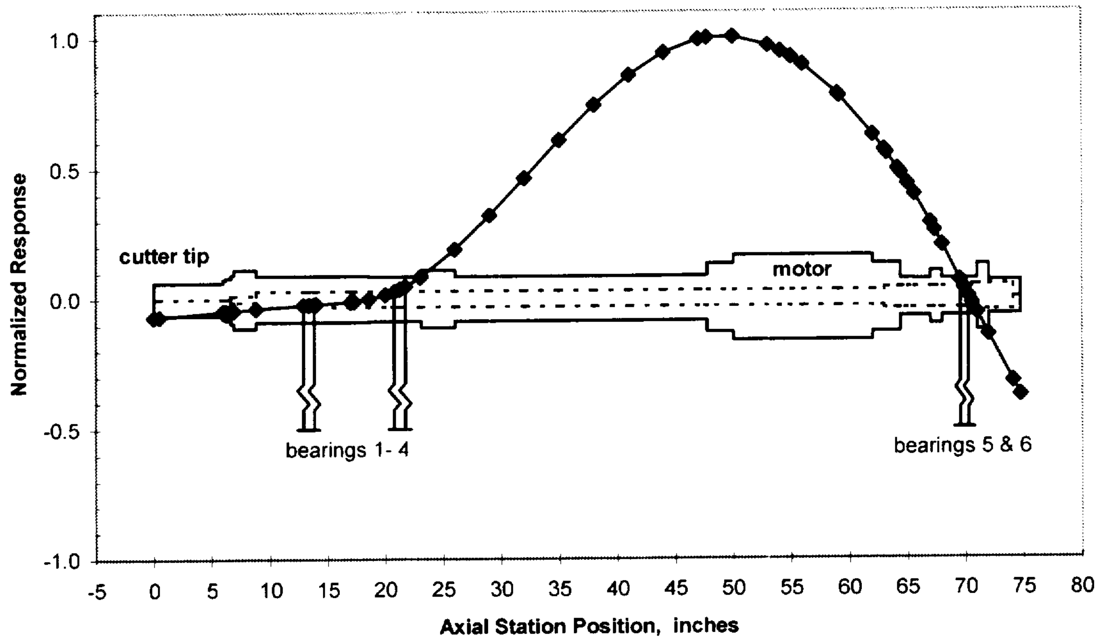

Figure 1. Milling Machine Spindle First Bending Mode

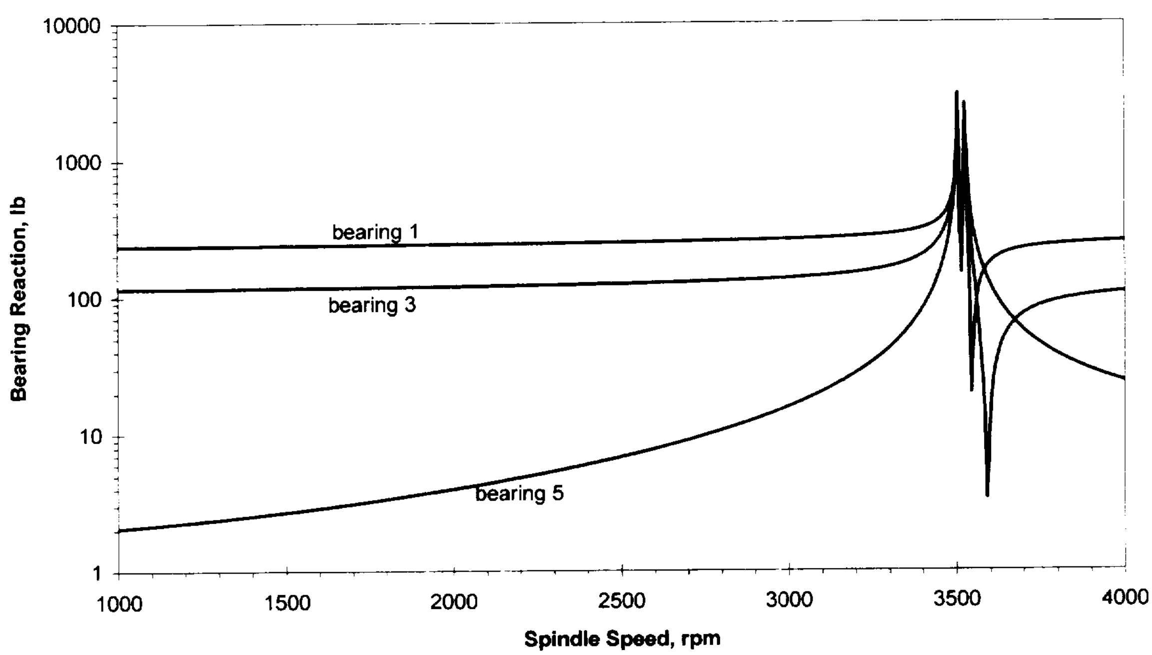

Figure 2. Bearing Reactions to 150 Pound Cutting Force

Cutting requirements for this spindle precluded avoidance of the 3539 rpm operating speed, so an alternative solution was sought to avoid excitation of the bending mode. The method selected was an additional bearing mounted in a squeeze film damper between bearing 4 and the motor. The additional bearing successfully eliminated the resonance problem by shifting the motor resonance well above the excitation frequency, while preserving the cutter tip compliance required for cutting stability. Dynamic amplification of cutting forces was virtually eliminated throughout the operating speed range.

The lessons to be learned from this example are that (1) excitation of resonances can occur far from critical speeds. In this case, the resonant vibration occurred at half of the critical speed due to the twice per revolution excitation of the two-flute milling cutter. (2) Observation of machine dynamics at only one location (e.g., the cutter tip) cannot provide accurate insight into the operational dynamics of the full machine. However, if an analytical or experimental understanding of the full machine is available, then test data from one or more locations can be used to scale the prior data and infer what may be happening at locations inaccessible during operation.

High Speed Generator Support Resonance

A high speed (30,000 rpm) generator was developed for an aerospace application that required compact size, low weight, and high reliability. To preclude critical speeds in the operating speed range, the rotor was designed to be compact and stiffly supported. Development rig testing demonstrated low rotor vibration. Unfortunately, when the generator was installed on the application gearbox, very high vibration was experienced. A cross section of the generator and gearbox model is provided in figure 3.

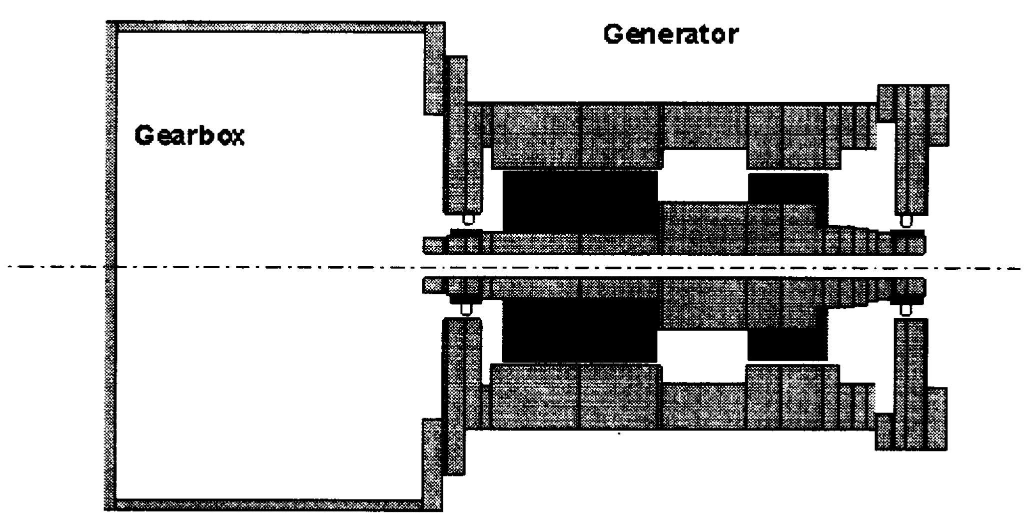

Figure 3. High Speed Generator Model

The generator was cantilevered from the drive gearbox, in keeping with typical aerospace practice. The generator manufacturer assumed that the gearbox would provide an effectively rigid mount for the generator. Based on this assumption, the first mode of the generator was calculated to be a bending mode at 47,791 rpm, 59% above the operating speed. With this critical speed margin, normal balancing procedures were expected to produce a vibration-free installation.

However, the gearbox was more flexible than anticipated, so instead of the first mode being a rotor bending mode, the installed first mode was a generator rigid body rocking mode near 30,000 rpm (fig. 4). When the generator and gearbox were tested, acceleration at the cantilevered end of the generator was measured as high as 35 g. The resulting loading of the bearings was excessive, and limited bearing life. To raise the first mode frequency and alleviate the high bearing loads, a support strut was installed between the cantilevered end of the generator and the gearbox. This additional support raised the rocking mode frequency out of the operating speed range (fig. 4). Figure 5 compares the acceleration and bearing forces at the generator cantilevered end for the original and strut-supported conditions. Both acceleration and bearing loads are reduced to practical levels by the additional support.

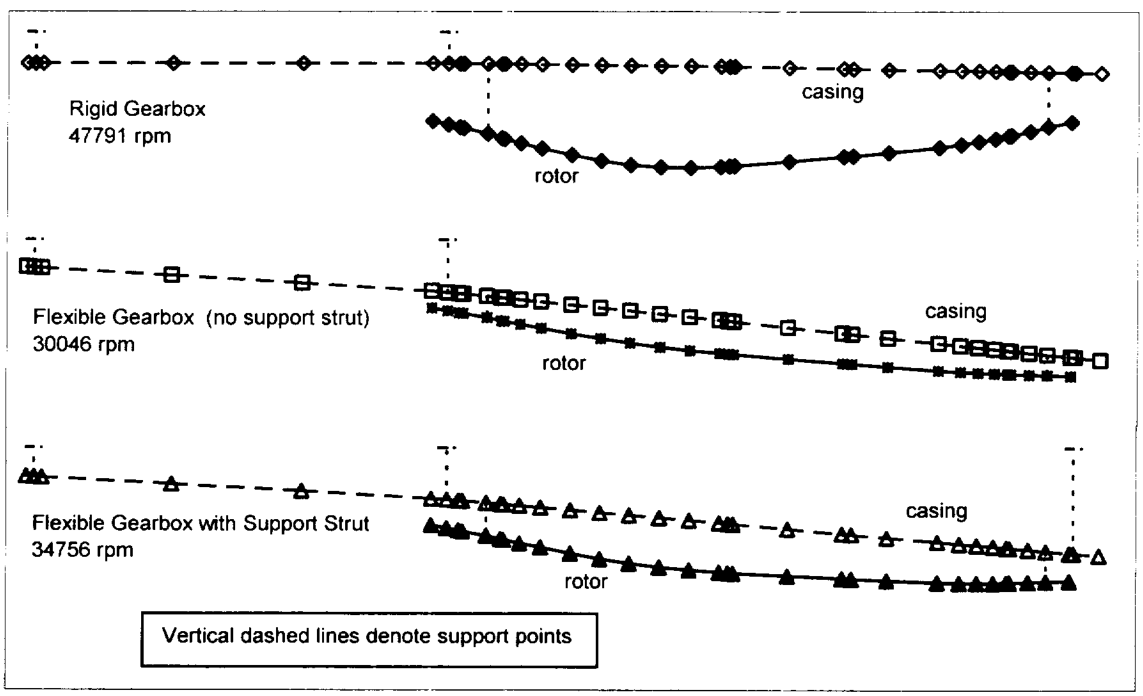

Figure 4. Generator/Gearbox First Critical Speed Modes – Effect of Attachment Flexibility

The generator in this case study is effectively mounted on a flexible support. Thus it is instructive to understand the characteristics of machines mounted on isolators [6]. An isolator is a soft support used to reduce force transmission from a machine to its foundation. The soft support allows the machine some flexibility. In many cases, this results in reduced bearing forces because the bearings are not forcing the rotor to operate about its geometric center. However, if the machine is operated at the soft support resonance, as in this example, the resulting casing motion can amplify bearing forces.

Therefore when encountering unanticipated high vibration, the entire system must be studied, including mounting arrangements. Even though each component is carefully designed, integrating the larger system may result in unanticipated dynamic loads that significantly shorten bearing life.

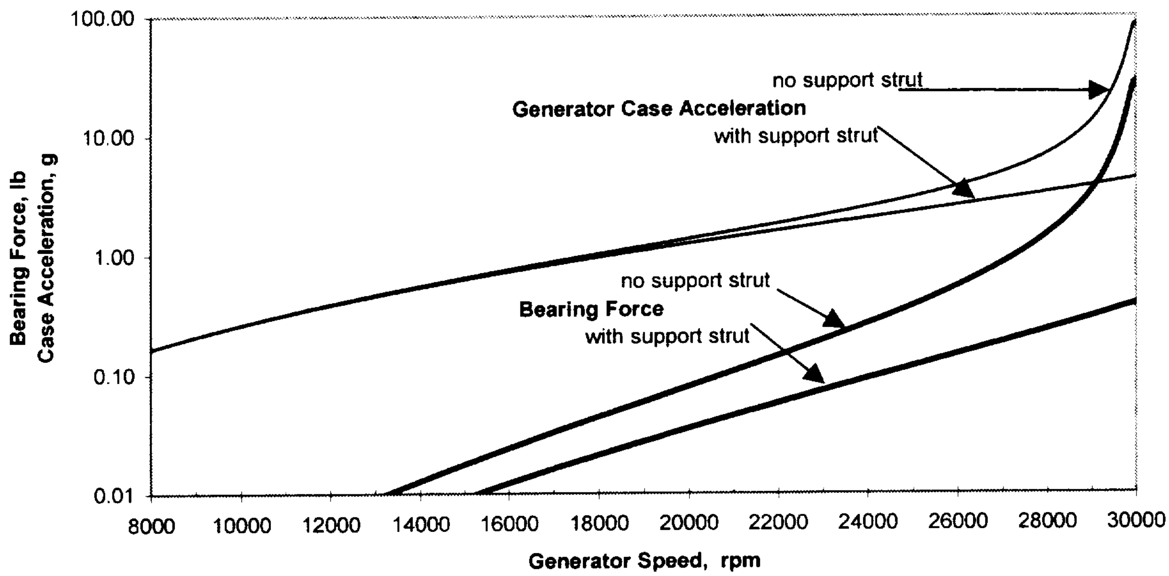

Figure 5. Generator Case Acceleration and Bearing Forces

Propulsion Fan Simulator

An air turbine driven propulsion fan simulator, used to test scale models of aircraft engine fans, experienced a catastrophic bearing failure during testing near maximum power and speed. Prior to the bearing failure the health monitoring accelerometers had malfunctioned, so a critical indicator of impending failure was not available to the operators. In this example, the events leading up to the failure are discussed and secondary warnings of the impending failure identified.

The rig design requirements called for a minimum operating life between overhauls of 100 hours, an operating speed range of 8,000 to 26,000 rpm with occasional short duration operation to 28,000 rpm, and a 20% margin on the first critical speed. The turbine and rig housing hardware and air-oil mist lubrication system were pre-existing, so the challenge was to select high speed bearings to fit the housing bore locations (axial locations fixed) and design a shaft to preclude critical speeds in the operating speed range. Within these constraints the rotor design evolved to the configuration modeled in figure 6.

A critical speed map was constructed to evaluate the influence of bearing stiffness on critical speeds (fig. 7). The desired 20% margin above 28,000 rpm means that the first critical speed must be above 33,600 rpm. This was not possible within the existing rig hardware constraints, so a less stringent goal of 20% margin at 26,000 rpm and 10% at 28,000 rpm was accepted. A minimum critical speed of 31,200 rpm was thus needed. The test plan would then require careful monitoring of the high speed conditions between 26,000 and 28,000 rpm. As shown in figure 7, the relaxed margin requirements could be met if the bearings provided at least 700,000 lb/in dynamic stiffness.

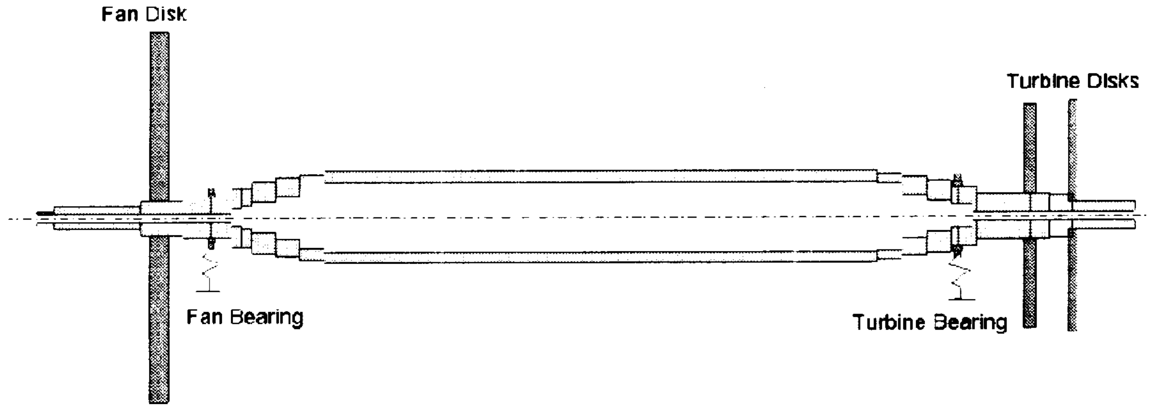

Figure 6. Propulsion Fan Simulator Model

Bearing radial stiffness varies with the bearing axial load, as shown in figure 8. The difference in thrust reactions between the turbine and fan results in net axial loads ranging from 200 pounds at low speed to 1,300 pounds at maximum speed. With these loads, the resulting radial stiffness for the thrust bearing was found to be adequate over the operating speed range, inasmuch as the requirement of 700,000 lb/in is not needed until speed rises to where the thrust load is greater than 300 pounds.

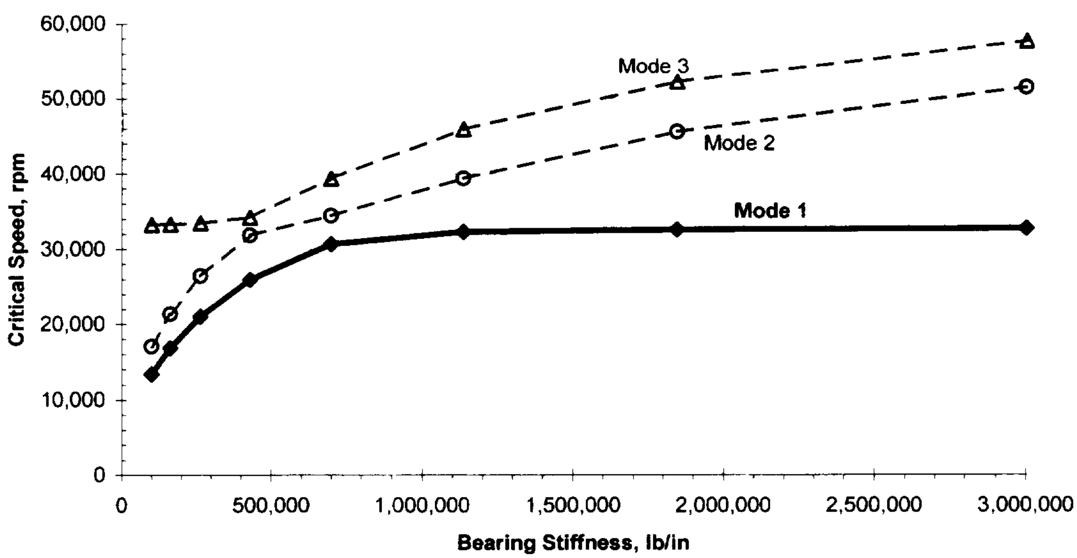

Figure 7. Fan Simulator Critical Speed Map

To avoid operating on the inner race split line and to ensure adequate radial stiffness, the second (non thrust-carrying) bearing required a preload spring. Figure 8 shows that the preload spring needed to provide at least 350 pounds to meet the 700,000 lb/in minimum stiffness requirement at high speed. The final critical speed analysis, performed with the anticipated bearing stiffness characteristics, determined that the first bending critical speed was 32,160 rpm. The mode shape is provided in Figure 9; it shows that this is a bending mode. The analysis indicated 15% critical speed margin at 28,000 rpm and 24% margin at 26,000 rpm. This comfortably met the revised design goal.

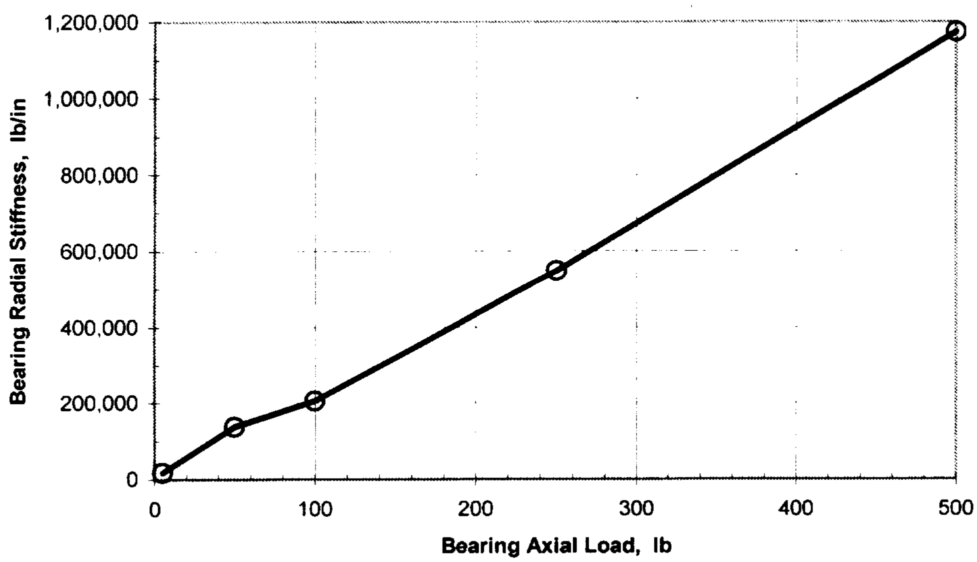

Figure 8. Bearing Stiffness as Function of Axial Load (high speed operation with rotor gravity load)

It remained to choose the thrust bearing location. The turbine bearing encounters the broadest range of environmental operating temperatures as a result of turbine and aft body seal leakage. To minimize risks associated with changes of bearing internal radial clearance as temperature changes, the simulator was to be configured with the fan bearing as the thrust bearing; the turbine bearing would be installed with an axial preload spring. However, when the simulator was assembled, a very tight test schedule combined with miscommunication led to the bearing retainers being installed such that the turbine bearing reacted thrust while the preload spring was not installed at all. As shown in figure 8, lack of axial preload on the fan bearing prevents development of adequate radial stiffness and potentially allows the bearing to operate on the split line.

The simulator incorporated bearing health monitoring accelerometers and outer race thermocouples. Unfortunately, before the rig was run at high speed the accelerometers failed. With the bearings improperly installed, the first critical speed fell within the operating speed range. As rig speed increased, the operators observed decreasing turbine efficiency and a reduction in turbine bearing temperature. They also noted that the proximity probe signal used for a tachometer was becoming saturated with a 1/rev signal. The efficiency loss was attributed to seal leakage while the tach anomaly and counter-intuitive drop in bearing temperature for the thrust-loaded turbine bearing were ignored. As the rig approached its critical speed, the turbine seals were rubbed out causing severe loss in turbine efficiency, which led to very low exhaust temperatures (below -30°F). Cold turbine air leaking past the seals cooled the turbine bearing and back pressured the air-oil mist lube system, effectively cutting off turbine bearing lubrication. Continued operation resulted in catastrophic bearing failure.

During post-failure evaluation, the reduced critical speed margin accepted for this design was not considered a significant factor in the failure. The improper bearing installation so deviated from design intent that even with 20% - 30% margin the rig would still have failed. Furthermore, the radial bearing loads were probably not a primary failure cause because even operating near the critical speed the bearings were near node points as shown in figure 9. The significant factor observed in the post-mortem hardware examination that identifies the probable location of primary failure was thrust bearing stress ellipse migration over the shoulder of the inner race due to excessive internal bearing clearance. This resulted from excessive cooling of the bearing inner race and led to extrusion of the race shoulder, severe ball distress, and cage failure.

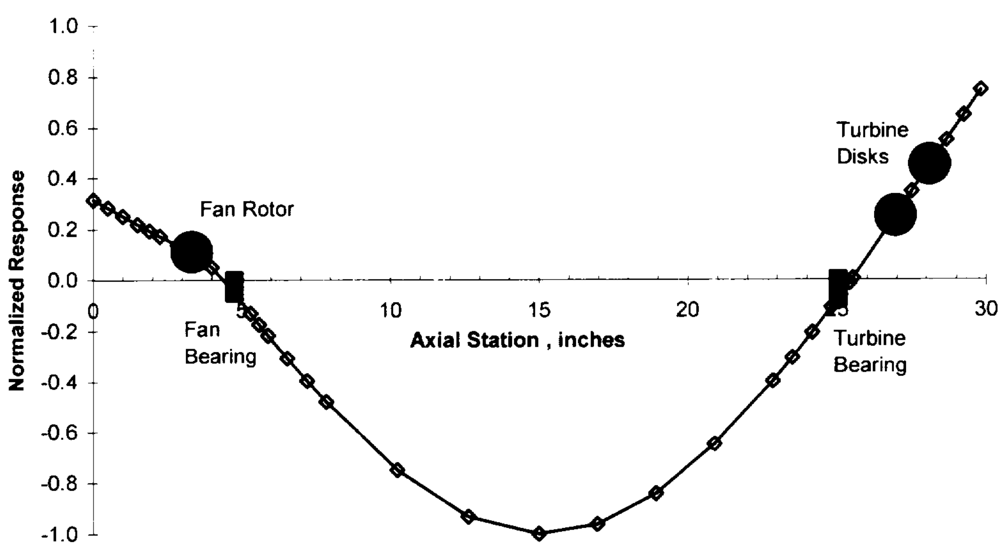

Figure 9. Fan Simulator Critical Speed Mode Shape

This case study reinforces the need to ensure that special bearing installation requirements are understood and achieved when a prototype machine is assembled. Then, if normal health monitoring data are not available when monitoring machine operation, be vigilant for secondary indicators of anomalous behavior.

CONCLUDING REMARKS

These examples show that rotordynamics must be heeded in the design and operation of even seemingly simple machines. Basic critical speed analysis and inclusion of adequate elements of the dynamic system are important to successful operation of rotating machinery. Careful monitoring during startup is essential to verify that the machine is operating as designed; continued operation after failure of heath-monitoring instrumentation can have catastrophic results when the machine characteristics have not been experimentally proven.

REFERENCES

1. Gunter, E.J., *Dynamic Stability of Rotor-Bearing Systems*, NASA SP-113, 1966.

2. ISO1940/1-1986(E), "Mechanical Vibration - Balance Quality Requirements of Rigid Rotors - Part 1: Determination of Permissible Residual Unbalance"

3. ISO1940/2-1997(E), "Mechanical Vibration - Balance Quality Requirements of Rigid Rotors - Part 2: Balance Errors"

4. API Publication 684, "Tutorial on the API Standard Paragraphs Covering Rotor Dynamics and Balancing: An Introduction to Lateral Critical and Train Torsional Analysis and Rotor Balancing", American Petroleum Institute, 1996.

5. Jackson, C., *The Practical Vibration Primer*, Gulf Publishing, 1979.

6. Harris, C.M., *Shock and Vibration Handbook*, 4th edition, McGraw Hill, 1996.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	August 2001	Technical Memorandum

4. TITLE AND SUBTITLE

Rotordynamic Influence on Rolling Element Bearing Selection and Operation

5. FUNDING NUMBERS

WU–712–30–13–00

6. AUTHOR(S)

Gilbert K. Queitzsch, Jr., and David P. Fleming

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
John H. Glenn Research Center at Lewis Field
Cleveland, Ohio 44135–3191

8. PERFORMING ORGANIZATION REPORT NUMBER

E–12980

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Washington, DC 20546–0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA TM—2001-211131

11. SUPPLEMENTARY NOTES

Prepared for the International Symposium on Stability Control of Rotating Machinery (ISCORMA 2001) sponsored by the Bently Rotor Dynamics Research Corporation, South Lake Tahoe, California, August 20–24, 2001. Gilbert K. Queitzsch, Jr., Boeing Propulsion Technology, Seattle, Washington 98124; and David P. Fleming, NASA Glenn Research Center. Responsible person, David P. Fleming, organization code 5950, 216–433–6013.

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified - Unlimited
Subject Category: 37 Distribution: Nonstandard

Available electronically at http://gltrs.grc.nasa.gov/GLTRS

This publication is available from the NASA Center for AeroSpace Information, 301–621–0390.

12b. DISTRIBUTION CODE

13. ABSTRACT ***(Maximum 200 words)***

Three case studies are presented that illustrate the importance of dynamic considerations in the design of machinery supported by rolling element bearings. The first case concerns a milling spindle that experienced internal rubs and high bearing loads, and required retrofit of an additional damped bearing. The second case deals with a small high-speed generator that suffered high vibration due to flexible mounting. The third case is a propulsion fan simulator rig whose bearings failed catastrophically due to improper bearing installation (which resulted in inadequate dynamic bearing stiffness) and lack of health monitoring instrumentation.

14. SUBJECT TERMS

Rotordynamics; Bearing selection; Shaft vibration; Bearing loads; Balancing

15. NUMBER OF PAGES

16

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102