NASA/CR—1999-208689

Rotordynamics and Design Methods of an Oil–Free Turbocharger

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January 1999

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Prepared under Cooperative Agreement NCC3–409

National Aeronautics and
Space Administration

Lewis Research Center

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Acknowledgments

The author would like to thank Dr. Hooshang Heshmat of Mohawk Innovative Technology Inc. and Dr. Christopher DellaCorte of NASA LeRC for their assistance throughout this project. Thanks also go to the Timken Foundation whose financial assistance through the Timken Fellowship helped make the author's involvement in this project possible. Finally, the author's gratitude goes to the National Aeronautics and Space Administration and the Department of Energy for their continued financial support.

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ROTORDYNAMICS AND DESIGN METHODS OF AN OIL-FREE TURBOCHARGER

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SUMMARY

The feasibility of supporting a turbocharger rotor on air foil bearings is investigated based upon predicted rotordynamic stability, load accommodations, and stress considerations. It is demonstrated that foil bearings offer a plausible replacement for oil-lubricated bearings in diesel truck turbochargers. Also, two different rotor configurations are analyzed and the design is chosen which best optimizes the desired performance characteristics. The method of designing machinery for foil bearing use and the assumptions made are discussed.

NOMENCLATURE

C	bearing clearance
D	journal diameter
e	eccentricity
h	film thickness
h'	dimensionless film thickness = h/C
L	bearing length
L/D	length to diameter ratio
P_a	ambient pressure
R	journal radius
W	dimensionless load $W = w/(P_a R^2)$
w	load
α	compliance of the foils
ε	dimensionless eccentricity (e/C)
μ	fluid dynamic viscosity
ω	journal rotational speed
Λ	bearing number ($6m\omega/P_a\, R^2/C^2$), also called dimensionless speed

INTRODUCTION

A turbocharger is a turbine driven compressor that uses the waste energy from exhaust gas to increase the charge mass of air in the combustion chamber of an engine. This process allows more fuel to be burned and thus increases the power output of the engine. Nearly all commercially available heavy duty diesel truck engines use turbochargers for increased power output and to compensate for low presures at high altitudes where air is less dense (ref. 1). Also, by utilizing turbocharging, one engine displacement can be set up to provide a wide range of power levels.

However, turbochargers on diesel truck engines are vulnerable to failure. This is due, in part, to their oil lubrication systems. Turbocharger rotors rely on oil lubricated floating sleeve bearings and ring seals. Because turbochargers operate at high temperatures the oil is subject to degradation and coking. This can lead to catastrophic failure,

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increased particulate emissions from oil leaking into the air stream, and, in severe cases, engine fires. Because of these inherent problems and the recent emphasis on more stringent emissions standards, it has become desirable to eliminate the oil lubricated bearings in a turbocharger.

Foil bearings provide one potential means to accomplish this goal. Foil bearings are self-acting compliant hydrodynamic bearings lubricated by air and are well suited to high speed, light load applications. The turbocharger presents just such a set of bearing conditions and appears to be an attractive application.

Foil bearing technology has been used extensively in the last three decades in the air cycle machines (ACM) of many newer civilian airliners (ref. 2). An ACM is a turbocompressor used to provide cabin pressurization and air conditioning. These ACMs, which are powered by compressor bleed-off, operate under steady speed and load conditions and are not subjected to the demands of high temperature (ref. 3). To extend the technology of foil bearings to high speed and high temperature applications, the modification of a turbocharger has been proposed. It is hoped the knowledge gained from this application can lead to more complex uses for foil air bearings, such as gas turbine engines. Finally, turbochargers stand to benefit from the advantages offered by foil bearings over oil lubricated bearings, as there are virtually no maintenance requirements for the foil bearings, and their lower power loss (ref. 4) may lead to better fuel economy.

In this study, the goal is to show that air-lubricated foil bearings can be successfully utilized in a high temperature environment with variable loads. The work presented is to be used as a preliminary design study in an ongoing cooperative effort to design, build, and test a working oil-free turbocharger.

FOIL BEARINGS

The foil bearing, whose configuration is shown in figure 1, is comprised of a cylindrical shell lined with corrugated bump foils topped with a thin, flat foil. The purpose of the foil is to comply to the hydrodynamic pressure distribution inside the bearing resulting in a larger gap than would be present for an identically loaded rigid gas bearing. There are several benefits gained by this larger gap. The load capacity is increased because the foils deform, having the effect of spreading the load over a larger area. The larger gap leads to reduced shearing rates resulting in lower power loss than rigid gas bearings. Foil bearings are less susceptible to damage due to dirt particles not only because the gap is larger, but also because the foils can deflect to accommodate a large dirt particle instead of seizing. Foil bearings generally offer more damping than do identical rigid gas bearings. This increased damping is due to the Coulomb friction dissipation that occurs during shaft deflections causing the bump and top foils to rub together. Their increased damping gives them more stability at critical speeds. The compliance of foil bearings also makes them more tolerant of misalignment and centrifugal and thermal growth. This, in turn, lessens the importance of surface finish and precision tolerances which is especially crucial in the case of rigid gas bearings (ref. 5).

Historically, gas foil bearings have had limitations that have kept them out of wide spread use in high speed rotating machinery. A serious drawback is the lack of high temperature start/stop cycle endurance (ref. 6). These bearings are very durable in high temperature environments once they are running, but starting and stopping requires endurance of high temperature metal to metal sliding. In addition, they lack the high load carrying capacity of oil-lubricated bearings or rolling element bearings. Typical air foil bearings have internal fluid pressures less than 0.7 MPa (100 psi) while oil film bearings regularly have pressures an order of magnitude higher. These obstacles have been largely overcome through the recent development of new bearing designs with enhanced damping and load capacity (ref. 7) and better solid lubricant coatings (ref. 8). Thus, the concept of an oil-free turbocharger warrants consideration.

APPROACH

An existing truck turbocharger will be used as the base design and will be modified as needed to accept the foil bearings. Since the turbocharger will be based upon an actual production unit, the design will allow for a direct comparison between the oil-lubricated and air-lubricated prototype bearings to determine the performance changes.

The first step in analyzing the oil-free turbocharger concept is to model the rotor system in a rotordynamics code. The code to be used, Analysis of Rotor Dynamic Systems (ARDS), was developed by Arizona State University (ref. 9). The code uses finite element analysis to determine rotor-bearing system response. A series of computer simulations is carried out with the direct stiffness coefficients (Kxx,Kyy) chosen over a wide range, and the cross-coupled stiffness coefficients (Kxy,Kyx) set equal to zero to map the undamped critical speeds of the rotor as a function of stiffness. This method will facilitate choosing desired stiffness values based upon critical speed requirements. Since these bearings do not tend to offer a great deal of damping compared to oil film bearings, it is desirable to design the rotor such that the third critical speed is above the maximum operating speed, and the second critical speed is below the lowest operating speed, typically 20 000 rpm, to improve the likelihood of stable operation.

The rotor/bearing system must also be designed to operate below the threshold speed of instability. This is the speed at which self-excited vibrations begin to occur. Unlike critical speeds, a rotating machine cannot be operated at or above the speed where self-excited vibrations begin. However, it is possible to shift the speed at which this occurs up or down by increasing or decreasing the amount of damping in the system. With a rotor system chosen based upon the previously explained stiffness requirements, damping is added to the model in varying degrees in order to plot the log decrement as a function of bearing damping. The log decrement is a measure of vibration decay with time, and thus gives an indication of system stability. If the log decrement of the system is positive, the vibration amplitude is decreasing with time. However, if the log decrement is negative the amplitude is increasing with time and this corresponds to self-excited vibration. Experience described in the literature has shown that in order for the rotor to be stable, the log decrement must be positive and greater than 0.20 (ref. 10).

The stability analysis is done at a spin speed of 117 000 rpm, the maximum speed of the turbocharger. It is assumed that since the bearings will be the least eccentric at the maximum speed, if the rotor is stable there, it will be stable at lower speeds. This assumption is only valid because the operating speed range does not include a critical speed where more damping would be needed.

ROTOR GEOMETRY

Two rotor geometries were analyzed in this study to determine an appropriate size for the bearings. The first design analyzed (called the short rotor) has 2.54 cm (1.0 in.) long bearings, and a shaft 9.680 cm (3.8 in.) long to maintain the existing overall length of the current oil-lubricated rotor. The diameter of the bearings is 2.54 cm as well. This diameter was chosen to give an L/D ratio of 1. Some of the reasons for this are that long bearings with L/D >> 1 offer little tolerance to misalignment. They also limit cross flow forced convection cooling. Short bearings, L/D << 1 suffer from edge losses which reduce load capacity. Short bearings may also be hard to manufacture because the foils tend to warp and distort. Therefore, an L/D ratio of 1 appears to be a rational selection.

The inside diameter of the shaft is determined by the requirement that the bending critical speed be above 117 000 rpm. Based upon this consideration, the required inside diameter is 1.9 cm (0.75 in.). This value was determined by choosing an inner diameter and running the computer code to find the third critical speed. If the bending critical speed was lower than 117 000 rpm, the inner diameter was decreased. This rotor geometry is shown in figure 2.

Using the critical speed map (fig. 3), a range of possible bearing stiffness is found to be roughly 175 N/cm to 21 000 N/cm (1 000 to 12 000 lb/in.) in order to keep the operating speed range (20 000 to 117 000 rpm) clear of critical speeds. Now, it is necessary to determine the bearing properties. For the short rotor geometry, the static load on each journal bearing is about 6.7 N (1.5 lb). From the bearing data in the literature, it can be seen that a load of 6.7 N (which corresponds to dimensionless load W = 0.408) can be accommodated in several ways. Unlike rigid bearings, which only require the load, speed, and geometry to be specified in order to determine the eccentricity, foil bearings also require specification of an additional parameter, the compliance (α). The compliance is a dimensionless measure of the rigidity of the foils; zero being very rigid, ten being very soft. However, the compliance and eccentricity are related to each other. This relationship has been numerically determined and is described in reference 11. Therefore, for a fixed speed and load, only matched pairs of compliance (α) and eccentricity (ε) are possible. Consequently, the primary design considerations can be categorized into two groups: the rotor environment (load and speed), and the bearing characteristics (compliance and eccentricity).

Since the design is being carried out at the maximum operating speed, it is desirable to choose a small eccentricity so that when the speed is decreased, the bearing will still have a relatively small eccentricity. This is important because the bearing needs to be able to tolerate shock loads that could be much larger than the dead weight of the rotor. Therefore, it is best to design the bearings to be lightly loaded (low eccentricity) at maximum operating speed so that when the bearings are operated at low speed they still have some tolerance to dynamic loads. It is anticipated, based upon the critical speed requirements (already discussed) and the data taken from reference 11, that the stiffness required of the journal bearings will dictate a compliance near 1. Under the load and speed range encountered, a compliance of 1 leads to an eccentricity of 0.45 at maximum speed. At minimum speed, the eccentricity increases to 0.90.

The clearance can be determined from the dimensionless speed. The analysis is done for dimensionless speed (Λ) of 1 and the design is being carried out at 117 000 rpm, therefore $\Lambda = 1$ must correspond to 117 000 rpm. From this, the clearance can be determined to be 0.0048 cm (0.0019 in). Now, all bearing parameters are known: D = L = 2.54 cm (1 in.), clearance = 0.0048 cm (0.0019 in.), $\alpha = 1$, $\varepsilon = 0.45$.

SHORT ROTOR RESULTS AND DISCUSSION

The undamped critical speeds of the rotor with the bearing geometry as given above are acceptable. The first and second critical speeds are both below the lowest operating speed of 20 000 rpm. The third, or bending, critical speed is roughly 150 000 rpm which is safely above the highest operating speed.

The rotor is modeled in ARDS to determine the amount of damping needed for stability. The log decrement, as calculated by the computer code, is shown in figure 4 as a function of damping values. From the plot, it can be seen that if the damping coefficients of the bearings are above 315 Ns/m (1.8 lb sec/in.), the system will be stable at 117 000 rpm. It is assumed that the rotor will be stable at all other operating speeds as stated earlier.

It is important to ensure that the maximum stress in the shaft is less than the material's yield strength. It is also necessary to determine how much radial growth the shaft will have at the maximum speed. The maximum centrifugal radial growth of the shaft (not including the center of the thrust disk) at 117 000 rpm is 0.00091 cm (0.00036 in.). The associated stress in the shaft is 185 MPa (26 800 psi) (ref. 12) which is well below the 758 MPa (110 000 psi) yield strength of one candidate rotor material, Inconel 713LC, at 538 °C(1000 °F) (ref. 13).

The expected temperature in the journal at the compressor end is 461 °C (861 °F), and is 394 °C (742 °F) at the turbine end. The temperature in the bearing housing at the compressor end is 409 °C (768 °F), and is 369 °C (696 °F) at the turbine end. The higher bearing temperatures at the compressor end are due to the different paths taken by the cooling air flowing through them. The compressor end journal bearings receive cooling air that has already flowed through the loaded side of the thrust bearing, thus it is hotter than the cooling air for the turbine end journal bearing which flows through the unloaded side of the thrust bearing. Under these conditions, the shaft and the housing will grow different amounts. These expansions will cause the clearance to change. The compressor end clearance will decrease by 0.0008 cm (0.0003 in.) and the turbine end clearance will decrease by 0.00025 cm (0.0001 in.). When combined with the centrifugal growth, this decrease amounts to less than half of the original clearance, and will not pose a problem as the foils can elastically deform by several times the original clearance (ref. 14). This characteristic of foil bearings that gives them their tolerance of misalignment and dirt particles also provides room for clearance reduction of that magnitude.

Thus, it seems that this configuration will work provided the bearings are capable of delivering at least 315 Ns/m (1.8 lb sec/in.) of damping. Damping values have been estimated using a test rig at the National Aeronautics and Space Administration Lewis Research Center to be on the order of 350 to 525 Ns/m (2 to 3 lb sec/in.) (ref. 15). Therefore, a stable design appears achievable.

Since there is potentially more damping available than is needed for stability, it may be wise to test for other possible bearing configurations that will need more damping, but will offer other benefits to make them a better choice than this design. For instance, since these bearings are somewhat heavily loaded at maximum speed, they are even more heavily loaded at 20 000 rpm (eccentricity 0.90). This could present a problem with shock loads at low speed. The dynamic loads could be several times the static loads in an application such as truck engine turbochargers, so it would be beneficial to design the bearings with extra load capacity at slow speeds.

LONG ROTOR GEOMETRY

With this in mind, it is desirable to consider larger bearings that would be less heavily loaded at low speed in order to gain shock load capacity. This approach has a disadvantage. The larger bearings would also be lightly loaded at high speed thus requiring more damping to be stable. For the larger diameter bearings, the rotor length must be increased to retain the desired L/D ratio of one. This rotor design is designated as the "long rotor" and is shown in figure 5.

The length of the rotor is limited by the amount of space in the engine compartment. The maximum distance between the compressor wheel and turbine wheel that can be accommodated in the truck engine compartment is 12.8 cm (5.04 in.). Therefore, the use of 3.8 cm (1.5 in.) long bearings is possible. Keeping with the L/D = 1 constraint, the shaft diameter must also be 3.8 cm.

Just as before, the bending critical speed dictates the inner diameter of the shaft. In order to keep the third critical speed above the maximum operating speed, the inner diameter must be 3.3 cm (1.3 in.). This geometry is shown in figure 5.

Using the critical speed map (fig. 6) it can be seen that bearings with stiffness coefficients of 175 N/cm to 31 500 N/cm (1 000 to 18 000 lb/in.) will keep the critical speeds out of the operating speed range. The load on each bearing in this configuration is roughly 8.9 N (2.0 lb), corresponding to a dimensionless load of 0.242. Immediately one can see that these bearings are less heavily loaded, as the dimensionless load for the shorter bearings was 0.408.

The next step is to determine the rest of the bearing parameters. From $\Lambda = 1$, the clearance is determined to be 0.0074 cm (0.0029 in.). As with the short rotor bearing design, the eccentricity will be chosen to be small in order to maintain a relatively light loading at low speed. Choosing the eccentricity = 0.3 is a good choice because α will again be 1 dictated by the stiffness requirements stated earlier. All of the bearing parameters for this design are now known: D = L = 3.8 cm, clearance = 0.0074 cm, $\alpha = 1$, $\varepsilon = 0.30$.

LONG ROTOR RESULTS AND DISCUSSION

The long rotor critical speed map looks very similar to the short rotor critical speed map. The first two modes are both well below 20 000 rpm, and the third mode is about 159 000 rpm.

Using the ARDS code, the log decrement is calculated for the first two rigid body modes and plotted versus damping coefficient. The plot (fig. 7) shows that in order for the turbocharger to be stable in the long rotor configuration, the bearings need to supply at least 482 Ns/m (2.75 lb sec/in.) of damping, which is within the 350 to 525 Ns/m range for which they are capable of being designed. Therefore, this rotor should be stable.

The radial growth of the shaft is about 0.0025 cm (0.001 in.), and the reduction of the clearance due to differential thermal growth is 0.0010 cm (0.0004 in.) and 0.0008 cm (0.0003 in.) for the compressor and turbine ends respectively. When combined, they are again roughly equal to half the original radial clearance. The maximum centrifugal stress in the shaft (again not including the thrust disk) is about 418 MPa (60 600 psi) which is only half of the 758 MPa (110 000 psi) yield strength of Inconel 713LC as stated previously.

The eccentricity at the low end of the speed range in this case is approximately 0.70, indicating that the bearings have a somewhat higher tolerance to shock loading than in the previous case in which the eccentricity was 0.90. From numerical studies of these types of bearings, maximum load capacity typically corresponds to an eccentricity of 4.0 (ref. 14).

An analysis with 5.08 cm (2.0 in.) bearings was considered,but based upon some preliminary calculations, using a shaft that large would cause the centrifugal shaft growth to be roughly the same as the initial size of the clearance. Also, the maximum stress in the shaft increases (752 MPa (109 000 psi) for D = 5.1 cm) because the outside diameter is larger and because the wall is not as thick. Therefore, it is not feasible to further increase the size of the bearings.

SUMMARY REMARKS

The results of this project suggest that a successful design of air-foil bearings for use in a turbocharger is feasible. The work also describes some of the steps in such a design process using published bearing data and public domain computer software. The results obtained in this manner are preliminary, but represent a starting point for a more detailed design. This analysis is valuable in reducing the detailed design effort, and eliminating some of the guess work.

Two possible design solutions were analyzed in the feasibility study to determine if an air-lubricated turbocharger is possible. The results are that both designs, the short rotor and long rotor designs, are capable of offering a solution. However, the long rotor design is a better compromise between the desired performance characteristics. It can tolerate more shock load while maintaining acceptable shaft stresses, though it does require more damping to be stable at the maximum operating speed of the system.

ACKNOWLEDGMENTS

The author would like to thank Dr. Hooshang Heshmat of Mohawk Innovative Technology Inc. and Dr. Christopher DellaCorte of NASA LeRC for their assistance throughout this project.

Thanks also go to the Timken Foundation whose financial assistance through the Timken Fellowship helped make the author's involvement in this project possible.

Finally, the author's gratitude goes to the National Aeronautics and Space Administration and the Department of Energy for their continued financial support.

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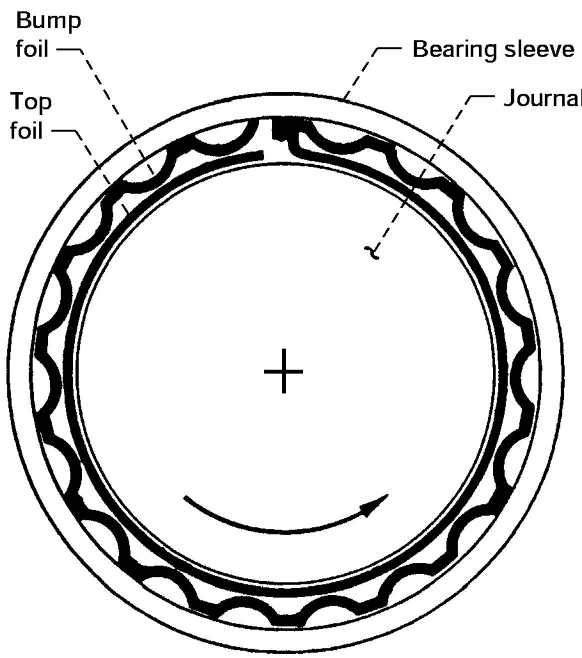

Figure 1.—Foil bearing schematic.

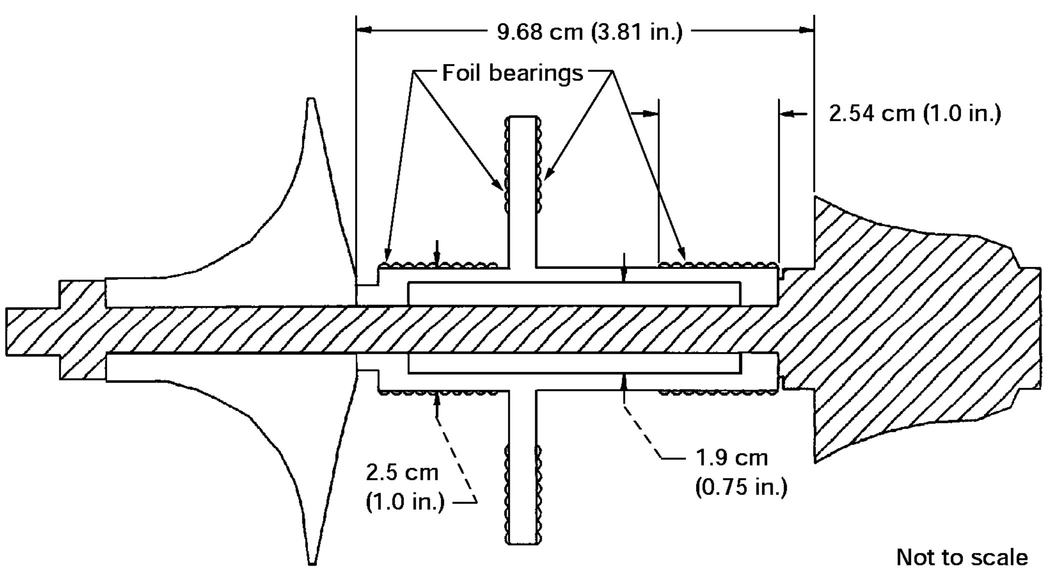

Figure 2.—Short rotor configuration.

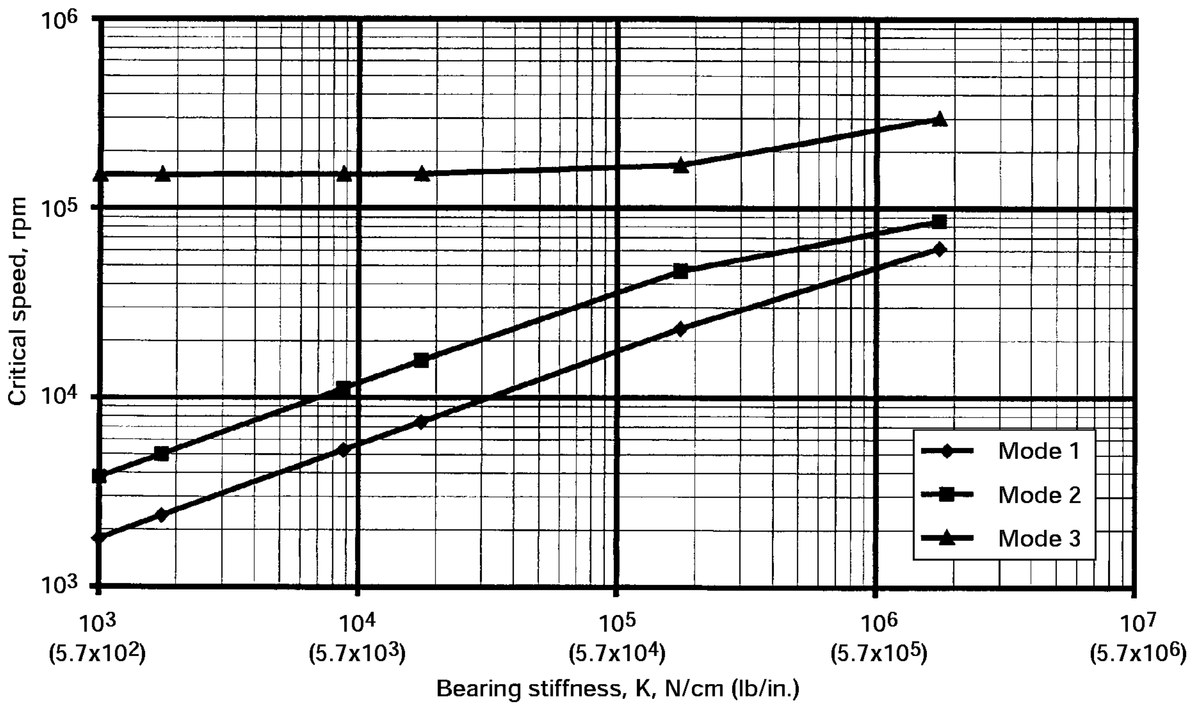

Figure 3.—Critical speed map for the short rotor.

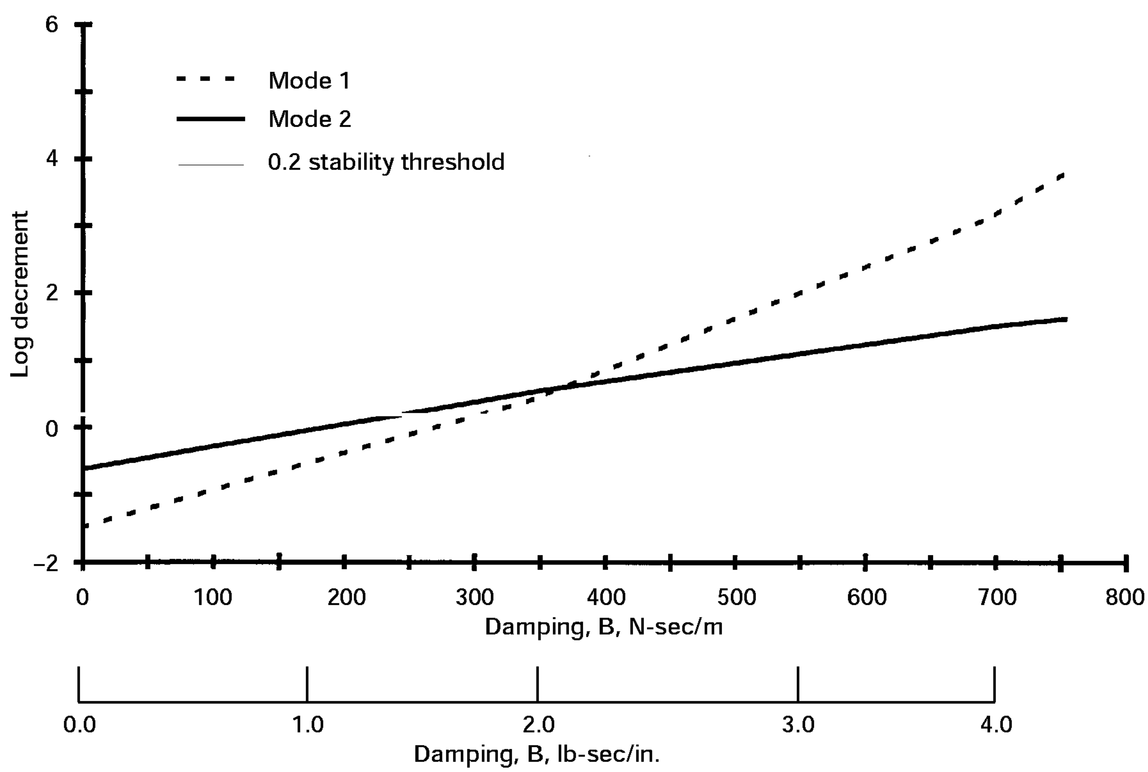

Figure 4.—Log decrement versus bearing damping for the short rotor.

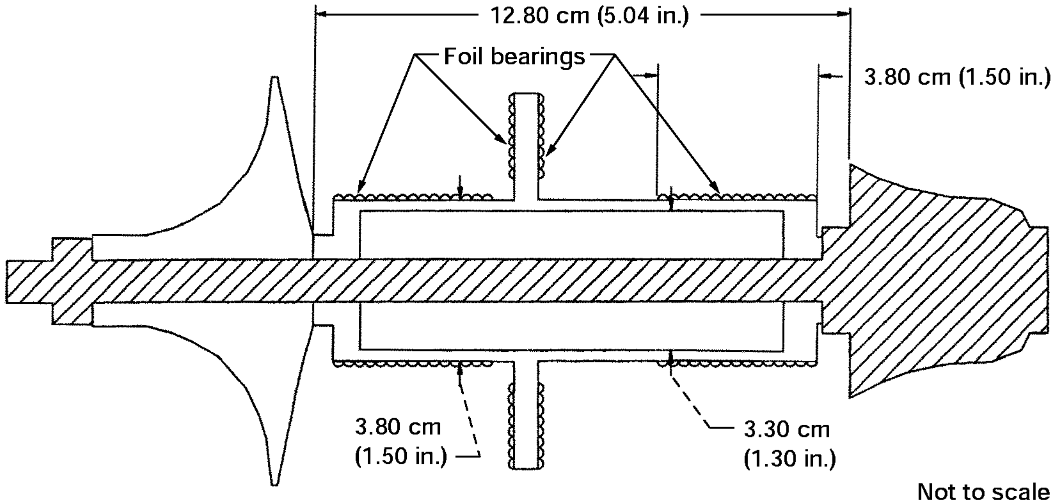

Figure 5.—Long rotor configuration.

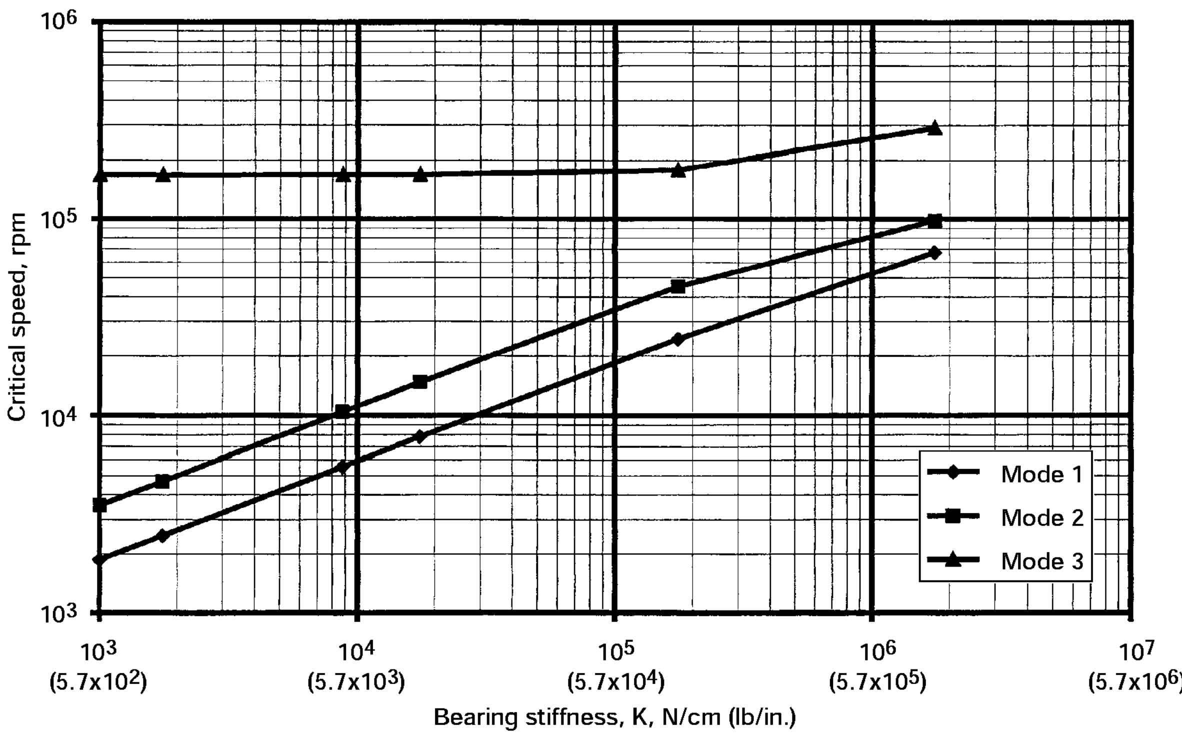

Figure 6.—Critical speed map for the long rotor.

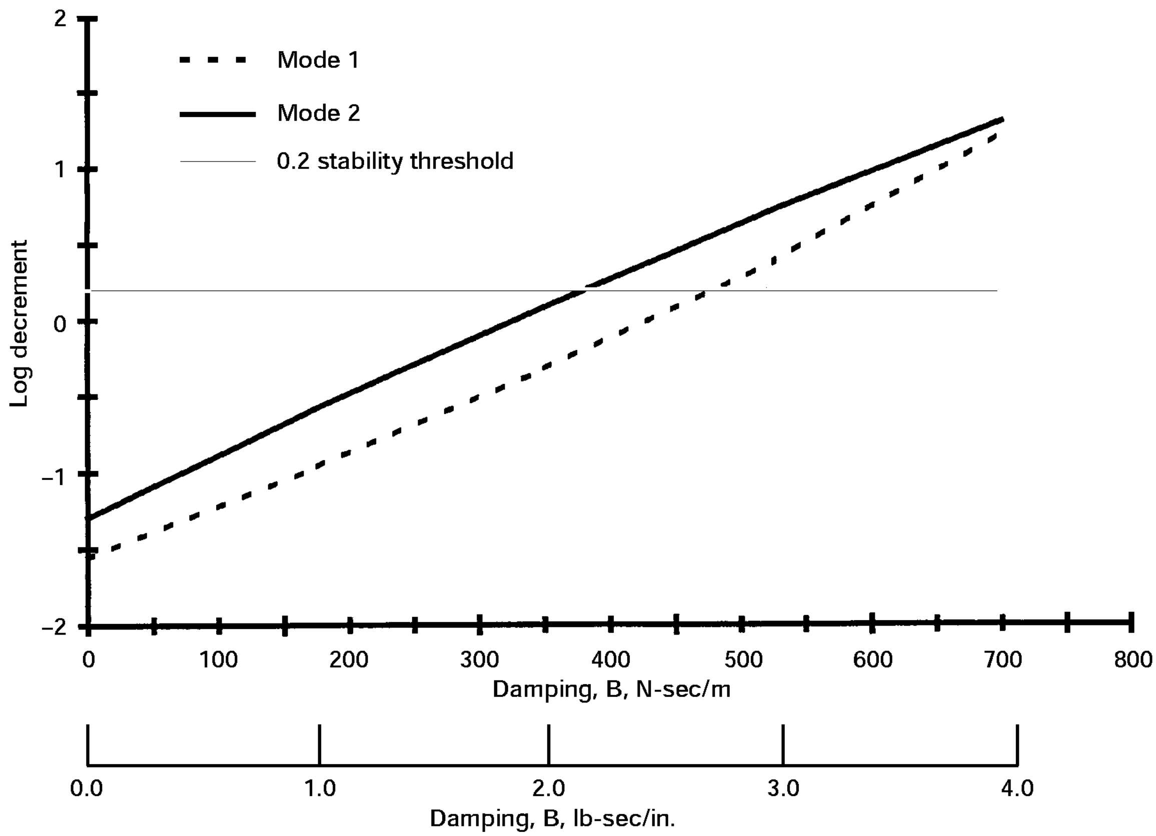

Figure 7.—Log decrement versus bearing damping for the long rotor.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	January 1999	Final Contractor Report

4. TITLE AND SUBTITLE

Rotordynamics and Design Methods of an Oil–Free Turbocharger

5. FUNDING NUMBERS

WU–523–22–13–00
NCC3–409

6. AUTHOR(S)

Samuel A. Howard

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

Case Western Reserve University
10900 Euclid Avenue
Cleveland, Ohio 44106

8. PERFORMING ORGANIZATION REPORT NUMBER

E–11476

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

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135–3191

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA CR—1999-208689

11. SUPPLEMENTARY NOTES

Project Manager, Christopher DellaCorte, Materials Division, NASA Lewis Research Center, organization code 5140, (216) 433–6056.

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Unclassified - Unlimited
Subject Category: 23 Distribution: Nonstandard

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

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14. SUBJECT TERMS

Bearings; Rotordynamics; Lubrication

15. NUMBER OF PAGES

16

16. PRICE CODE

A03

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