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Gas Foil Bearings for Space Propulsion Nuclear Electric Power Generation

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This report is a formal draft or working paper, intended to solicit comments and ideas from a technical peer group.

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Gas Foil Bearings for Space Propulsion Nuclear Electric Power Generation

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Abstract

The choice of power conversion technology is critical in directing the design of a space vehicle for the future NASA mission to Mars. One candidate design consists of a foil bearing supported turbo alternator driven by a helium-xenon gas mixture heated by a nuclear reactor. The system is a closed-loop, meaning there is a constant volume of process fluid that is sealed from the environment. Therefore, foil bearings are proposed due to their ability to use the process gas as a lubricant. As such, the rotor dynamics of a foil bearing supported rotor is an important factor in the eventual design. The current work describes a rotor dynamic analysis to assess the viability of such a system. A brief technology background, assumptions, analyses, and conclusions are discussed in this report. The results indicate that a foil bearing supported turbo alternator is possible, although more work will be needed to gain knowledge about foil bearing behavior in helium-xenon gas.

Introduction

NASA's Vision for Space Exploration includes manned flights to the moon and beyond. The Prometheus Project includes efforts to develop advanced nuclear-based power and propulsion technologies to support Exploration. One approach for such propulsion is to use nuclear energy to power electric ion thruster engines. A heat-to-electricity conversion system is needed to provide large amounts of steady, reliable electrical power. A leading candidate for power conversion is the closed cycle Brayton turbo alternator. The Closed Cycle Brayton turbine, hereafter referred to as simply the Brayton turbine, is essentially a turbine engine integrated with a motor generator running with recirculating gas which derives its heat energy from a nuclear reactor and rejects heat through a radiator (refs. 1 and 2).

The proposed Brayton turbine is a closed loop system operating on a mixture of helium and xenon (HeXe) gas. By virtue of the fact that the cycle is closed, the turbo alternator requires a bearing support system capable of operating on the HeXe process fluid of the cycle. Gas foil bearings offer the ability to perform in this type of environment as well as exhibit long life and high reliability under extreme temperatures. These characteristics make gas foil bearings an excellent choice for the bearing system of the turbo alternator (ref. 3). Traditional oil lubricated bearings cannot be used in closed loop systems because oil can enter the process flow causing downstream coking problems.

Foil bearings, like the one shown in figure 1, are a special type of hydrodynamic sleeve bearing with a compliant surface on the inner diameter of the sleeve. In general, the compliant surface consists of two or more layers of superalloy sheetmetal, called foils. One layer provides stiffness (in this case the bumps act like springs), and the other is a smooth top layer providing the bearing surface. As the shaft rotates, a pressurized fluid film is generated that forces the foils to expand outward and separates the shaft from the top foil surface. The pressure in the fluid film increases with the speed of the shaft, and eventually supports the full weight of the rotor. The compliance of the foil structure allows it to grow radially in response to centrifugal and thermal growth that would otherwise seize a rigid geometry bearing. As motion of the shaft occurs relative to the sleeve, the bump and top foils deflect and rub against each other. The stiffness of the bumps controls the amount of deflection, and the rubbing adds Coulomb damping to

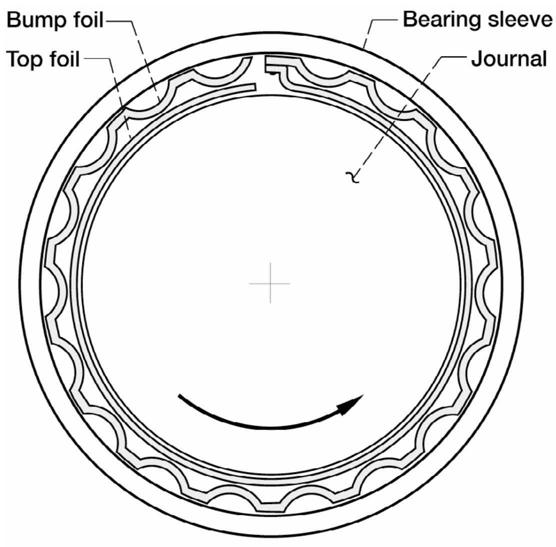

Figure 1. Foil bearing schematic.

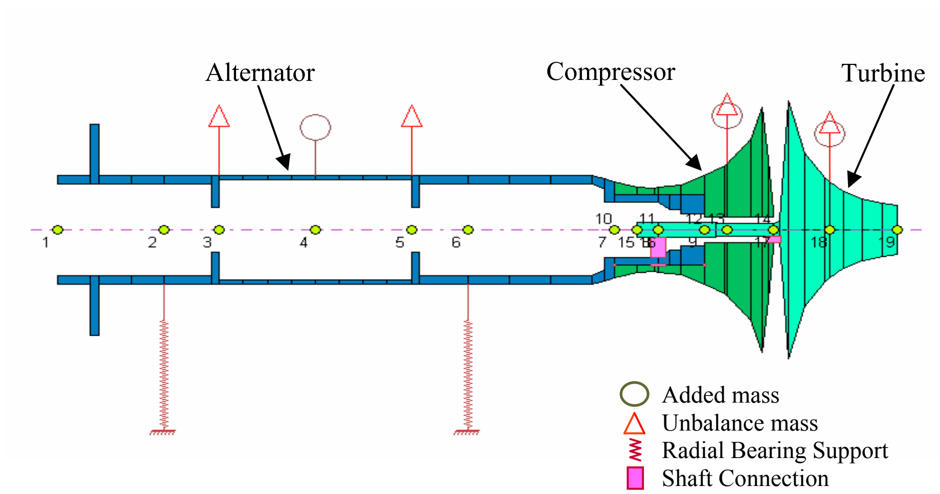

Figure 2. Rotordynamic model of proposed Prometheus I rotor layout.

the system. Thus, the foil structure gives the bearing designer the ability to modify the stiffness and damping properties of the bearing, within certain limits, to meet the demands of the system rotordynamics.

In addition to developing and demonstrating the Brayton turbine power conversion system currently under consideration, this project serves as a technology development test bed for planetary surface power systems.

In an effort to assist in and augment the contractor led design work for the turbo alternator power conversion unit of the Prometheus I Program, NASA Glenn Research Center has conducted an independent rotor dynamics analysis of the proposed shaft system (fig. 2). The goal of this analysis is to assess the feasibility of a gas foil bearing rotor support system, and to outline initial design constraints and potential concerns. The proposed unit is a centrifugal machine with a back-to-back turbine and compressor outboard of two foil journal bearings. A permanent magnet alternator is mounted integral to the shaft between the two bearings. Axial loads are constrained by a thrust bearing located on the opposite end of the shaft to the aero wheels, outboard of the journal bearings. The assumptions, analyses, and results of the NASA study are contained in the following discussion.

Assumptions

The analyses are done computationally using a commercially available finite element-based rotordynamic computer code (ref. 4). The code uses the geometry of the rotor, material properties, bearing properties, and forces such as unbalance, etc. to build a model of the system. The code calculates critical speeds and corresponding mode shapes, unbalance vibration response, stability, and time transient vibration response.

The assumptions made in modeling the rotor system have a profound impact on the results, and thus are important to discuss. It has been shown for a range of loads (eccentricities) that bearing stiffness changes with load. Foil gas bearings become more stiff as the eccentricity (eccentricity is the displacement of the journal center relative to the center of the bearing) increases (ref. 5). This effect is neglected in the current analysis. A consequence of this assumption is that vibration amplitudes calculated using linear bearing properties may be overestimated. The stiffening effect will tend to decrease the amplitude of vibration as the motion becomes large. Also, linear bearings will behave in a linear manner with respect to unbalance loads. If unbalance loads double for example, unbalance response will double. In reality, non-linearities in the bearings will cause a doubling of unbalance loads to result in less than double response amplitudes. Both of these consequences are conservative, meaning that the linear bearing assumption tends to result in overestimated vibration amplitudes.

Bearing properties, in particular stiffness and damping, are unknown in HeXe fluid. Preliminary testing at NASA suggests that load capacity may not change as drastically as might be expected for gases with viscosities different from air. Therefore, stiffness and damping estimates for foil journal bearings in air are used in the analysis as an approximation. The result of this assumption is that the properties could be incorrect for a system operating in HeXe having implications on the critical speeds, response amplitudes, and stability of the system. It is anticipated that the eventual design of the bearings will take this factor into consideration. Planned experimental testing of foil bearings in HeXe will eliminate many of the unknowns, and is therefore very important to successful design. In any case, the rotordynamic models developed here will be available for refinement and reassessment once HeXe data is obtained.

The rotor is modeled as three separate components connected together. The compressor is connected to the main shaft, and the turbine is connected to the compressor. The stiffness of each connection is assumed to be very high (1.75^{10} N/m, 1^{8} lb/in.) such that the components are essentially modeled as rigidly connected at their common nodes. Also, the main shaft is modeled as being welded together. A shaft with non-welded coupled components will likely have a much lower bending stiffness, potentially affecting this model adversely.

The alternator magnets, compressor blades, and turbine blades are assumed to not add to the structural strength of the shaft. Therefore, they are modeled as extra mass placed at the location of their respective centers of gravity. This is believed to be a reasonable assumption.

Any radial forces acting on the rotor due to the permanent magnet alternator are neglected. The magnitude of these forces is unknown, and therefore may or may not be significant. A detailed design study of the rotor should at least consider these forces to determine if they should be included. These types of forces can be destabilizing resulting in the need for additional damping in the radial bearings.

Unbalance is assumed to be 7.2^{-6} Kg-m (0.01 oz-in.) for each of the major rotating components: alternator, compressor, and turbine. This level of unbalance should be attainable, for this size class rotor. However, if the actual unbalance is greater (or smaller), the vibration amplitudes estimated in this analysis can be approximated to change proportionately in keeping with the linear bearing assumption.

Analyses

The analysis philosophy follows that of a previously successful effort to develop an Oil-Free Turbocharger using the same type of bearing technology (ref. 6). There are three main aspects of rotor behavior that must be taken into account in order to assess the feasibility of such a system. First, it is important to know the critical speeds of the rotor system. A critical speed is a speed at which the rotational frequency of the rotor is equal to a natural frequency of the rotor system (ref. 7). A critical speed can be a dangerous speed at which, or through which, to operate as the synchronous forcing function due to unbalance can excite the natural frequency of the rotor system and cause catastrophic vibration levels. Therefore, it is general practice to design rotor systems such that there are no critical speeds within the operating speed range of the machine, if possible. In gas foil bearing machines, this is typically done by designing shaft systems to be relatively light weight and rigid. The use of hollow, high strength tubular elements is one method to achieve high rigidity and low weight. Coupling lightweight, rigid shafts with relatively soft foil bearings generally results in two rigid body critical speeds at low frequencies, which are below the minimum operating speed. Third and higher bending critical speeds occur above the maximum operating speed (refs. 6 and 8). In this manner, one can usually design a rotor system that avoids sustained running at critical speeds, and traversing the rigid body modes occurs fast enough to not damage the system.

Following the above design philosophy, the critical speeds of the turbo alternator are calculated using a range of expected bearing stiffness. The bearing stiffness and damping are estimated using a third party computer code that calculates bearing performance as a function of speed, load, bearing geometry, and structural stiffness (i.e., foil stiffness) (ref. 4). For this analysis, three different structural stiffness levels were used to calculate rotor dynamic response; low, medium, and high structural stiffness.

In terms of their effect on critical speeds, the difference between the three stiffness values is not significant. The desire to keep critical speeds out of the operating speed range is satisfied with all three structural stiffness levels. The importance of stiffness will become apparent later when considering gravitational effects on the rotor position and stability. Figure 3 shows a critical speed map that illustrates the effect of bearing stiffness on the speed of the first three mode shapes. One can see that there exists a large range of bearing stiffness for which safe operation is possible, represented on the chart as the

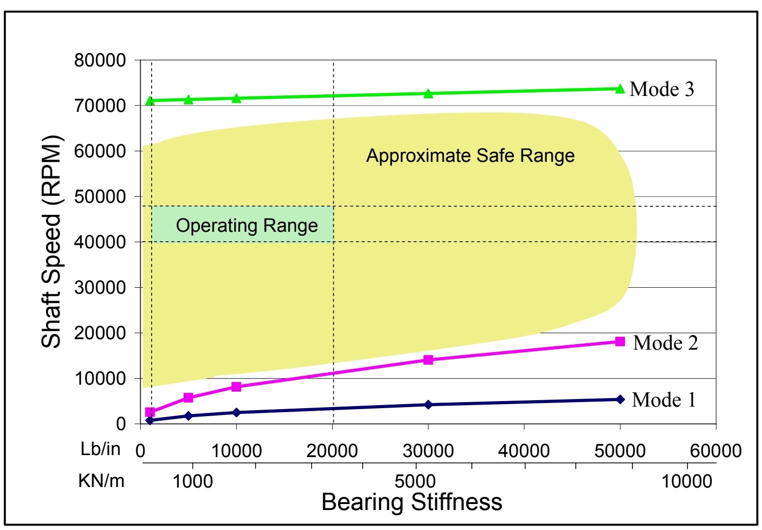

Figure 3. Critical speed map of the proposed Brayton Turboalternator.

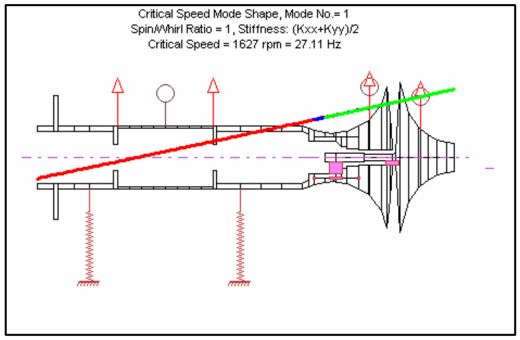

Figure 4. First critical speed mode shape.

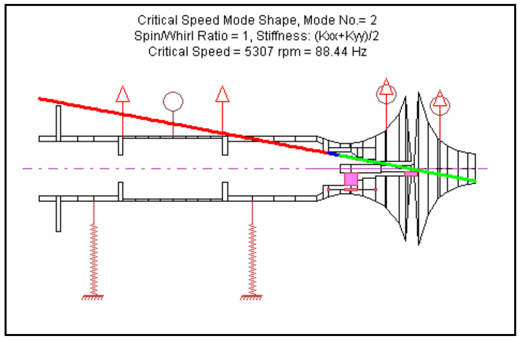

Figure 5. Second critical speed mode shape.

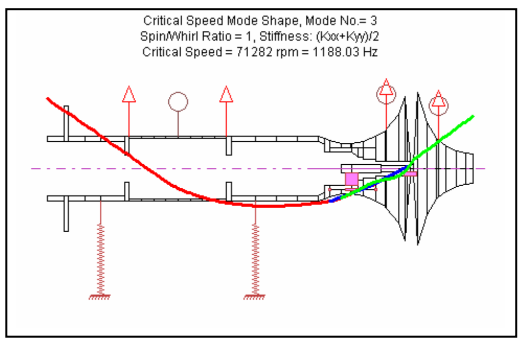

Figure 6. Third critical speed mode shape.

approximate safe range. The range of expected foil bearing stiffness values ($3.5^5 – 3.5^6$ N/m, 2,000 – 20,000 lb/in.) and operating speeds for the machine are shown as the operating range on figure 3. The first rigid body mode, whose mode shape is shown in figure 4, has a frequency of approximately 27 to 40 Hz (1600 to 2400 rpm) depending upon the stiffness used. The second rigid body critical speed (fig. 5) occurs at 88 to 110 Hz (5300 to 6600 rpm). The third critical speed, or the first bending critical speed (fig. 6), occurs at 1188 to 1193 Hz (71,300 to 71,600 rpm). With an operating speed range of 40,000 to 48,000 rpm, it is clear that no critical speeds exist within the speed range regardless of bearing stiffness.

Another consideration for rotor design is ensuring that the motion of the rotor does not exceed the clearances between various components and their housings. Usually, the most critical of these is the compressor. The clearance between compressor blade tips and the housing is often small to keep pressure losses low, thereby improving overall efficiency. There is always some motion of the rotor relative to its housing. It is the job of the rotor support system, the bearings, to control that motion to a tolerable level. For fluid film bearings, such as gas foil bearings, there are two major contributors to the displacement of the shaft center relative to the bearing center. First, is the static displacement of the shaft relative to the housing. Second, is the dynamic displacement. The static displacement of the rotor is dictated by the stiffness of the bearings, and the weight of the shaft. Stiffer bearings result in less static displacement for a given rotor weight. The turbo alternator in this report is designed for operation only in space, where there is no gravity. Therefore, the weight of the shaft will not produce any static displacement. However, gravity will be a concern for any ground-based testing of the rotor system. In order to design a robust system, two analyses are done. One analysis considers the effects of gravity and is applicable to ground-based operation of the turbo alternator. The other analysis does not include gravity, and is relevant to space-based operation of the machine.

On Earth, the rotor has weight in the downward direction. This results in a non-zero static load on the bearings. Since bearing stiffness increases with load, bearing stiffness will be higher on Earth than in space. In the absence of gravity, the bearings become unloaded and are inherently less stable than when they are loaded radially. This characteristic will be discussed more thoroughly later. When gravity is included in the analysis, the weight of the shaft dictates that bearings with a high structural stiffness are desired to maintain acceptable static deflection of the shaft. Higher structural stiffness results in higher overall bearing stiffness, and the critical speeds of the rotor system are at the upper end of the ranges above. As stated, the operating speed range is not adversely affected by the use of the higher stiffness bearings. The static deflection however, is affected, and is what drives the desire for stiffer springs for ground-based operation of the turbo-alternator-compressor (TAC).

For space-based operation of the TAC, softer bearings can be used with similar steady state unbalance response amplitude. One could design the machine to use stiff bearings for both operating regimes, but the design may have more stability margin using softer springs for space operation. Likewise, soft bearings could be used for operation in space and on Earth, as long as the static deflection with the softer bearings on Earth is acceptable for the blade tip clearances. Another alternative would be to conduct Earth-bound testing with the rotor in a vertical position, assuming sufficient thrust bearing capacity exists. In any case, system stability may ultimately dictate the choice of bearing design. Bearing designer experience and proper understanding of foil bearing supported systems in lightly loaded versus heavily loaded conditions will be necessary for successful implementation of the Brayton.

In both cases, the steady state unbalance response can be adequately controlled to satisfy the tip clearance constraints on the rotor. Unbalance response was calculated up to a maximum speed of 58,000 rpm, which corresponds to a 20 percent overspeed condition. Figure 7 shows a representative Bode plot of steady state response amplitude at the turbine end of the shaft. At 58,000 rpm, vibration amplitude is approximately 4.1^{-5} m (0.0016 in.) peak to peak (pk-pk). At the maximum design speed of 48,000 rpm, amplitudes are slightly more than half as large. With tip clearances in the compressor of approximately 2.0^{-4} m (0.008 in.), the vibration amplitude due to unbalance is not expected to be a problem even at 20 percent overspeed. As stated above, the unbalance is modeled as being located at the CG or ends of each of the three components (turbine, compressor, alternator). Figure 8 shows the same turbine end Bode plot for the analysis that includes gravity. One can see that vibration amplitude is similar in magnitude. In addition to the unbalance mass, the angular location of the three unbalance masses relative to each other also affects the vibration amplitude. Several configurations are analyzed to approximate the worst case scenario. It turns out the best configuration with respect to vibration amplitude is when all three weights are in phase, i.e., all located at the same angular position. In which case, the vibration amplitude at 48,000 rpm is about 1.2^{-5} m (0.00048 in.) pk-pk, a little less than half the amplitude of the worst case. The worst case shown above, with pk-pk amplitude of 2.5^{-4} m (0.001 in.) at 48,000 rpm, occurs when the masses are indexed from one another with the second weight 180° from the first, the third 0° from the second, and the fourth 180° from the third. Figure 9 shows the shaft at 48,000 rpm with vectors representing the unbalance loads. Considering the shape of the shaft at this speed, one can easily see why the above unbalance configuration results in the maximum shaft deflection. This indicates that even more unbalance can be tolerated than assumed, if its angular position can be controlled to a favorable orientation. Figures 10 and 11 show the orbit plots for the turbine end without gravity and with, respectively, that correspond to 48,000 rpm on the above Bode plots. Looking at the orbits, it becomes clear that without gravity, the rotor is centered around zero as the static deflection is zero due to no rotor weight. With gravity, the orbit is about the same size, but is centered around a point about 1.3^{-4} m (0.005 in.) below, and slightly to the right of the bearing clearance center. This illustrates the importance of understanding how bearing stiffness can affect static deflection as it pertains to blade tip clearances.

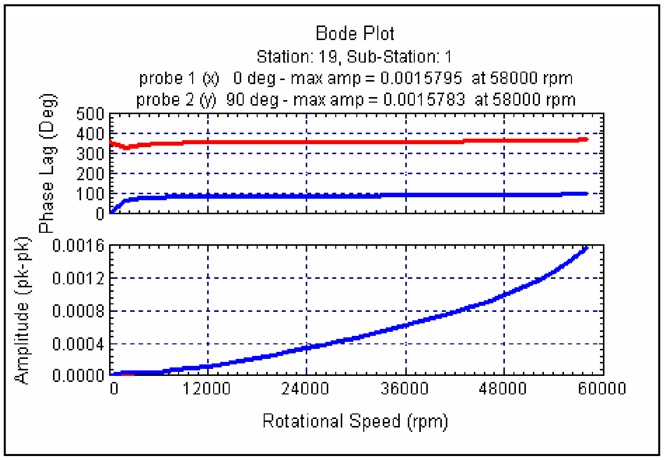

Figure 7. Bode plot: vibration at turbine end with no gravity.

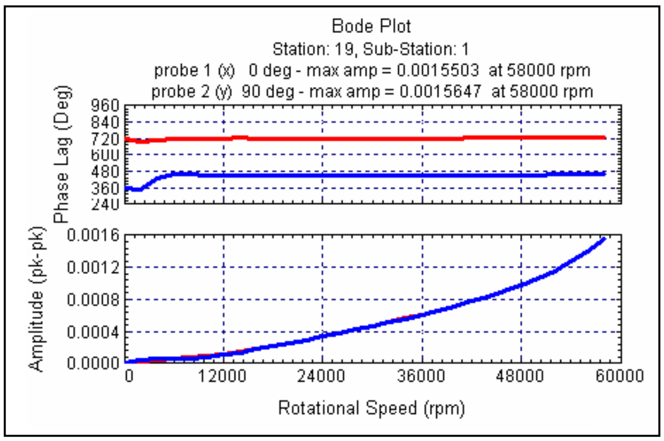

Figure 8. Bode plot: vibration at turbine end with gravity.

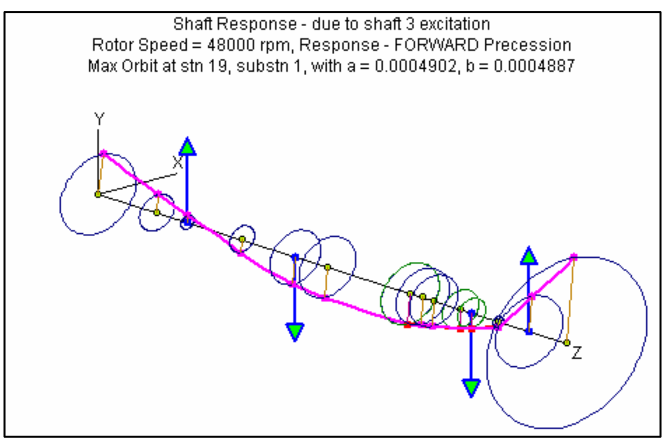

Figure 9. Schematic of the rotor at 48,000 rpm showing orientation of unbalance masses which results in maximum shaft deflection.

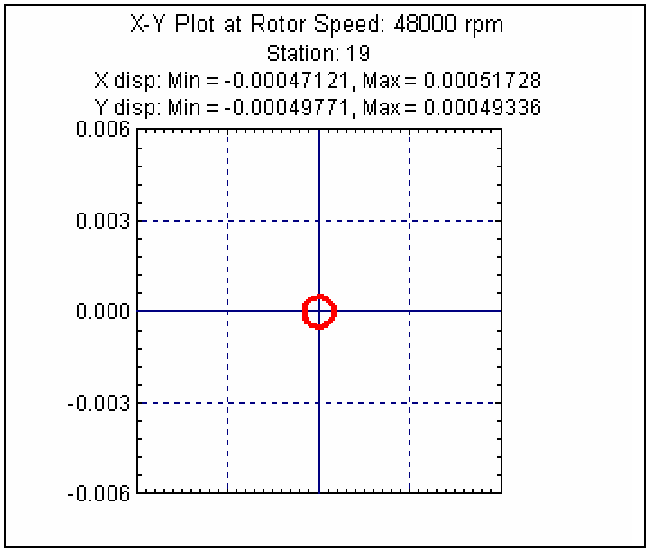

Figure 10. Orbit plot at turbine end with no gravity.

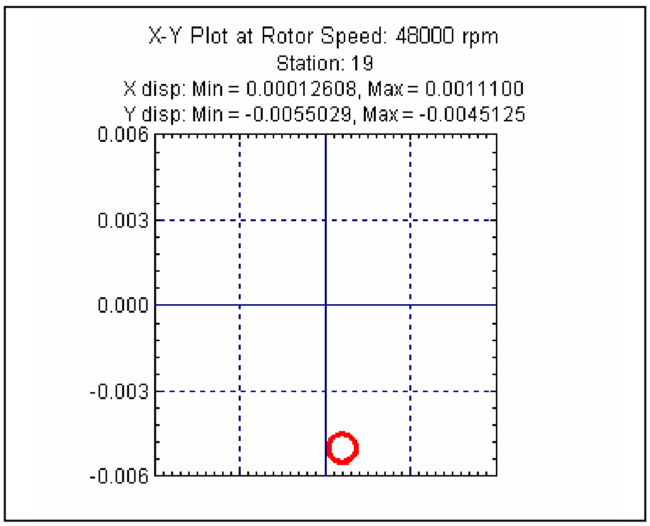

Figure 11. Orbit plot at turbine end with gravity.

The final constraint on the design of a feasible gas lubricated foil bearing supported TAC is stability. Any rotor system must be able to operate under a steady state condition for an unlimited amount of time in order to be considered stable. In addition, it must be able to tolerate transient external stimuli. In the case of the TAC, external forces can come in the form of straight line vehicle accelerations, gyroscopic loads from vehicle maneuvering, or intermittent magnetic loading from the alternator to name a few. The stability of a rotor system is basically dependant on the amount of damping in the bearing system. Sufficient damping must be provided to counteract any inherent instabilities in the bearings themselves, as well as the above mentioned transient perturbations from steady state. For the Prometheus TAC, the main stability concern will be in a no, or low, gravity environment. As stated earlier, if improperly designed, hydrodynamic bearings can be unstable when they are not loaded in the radial direction. The cause of this instability is negative cross-coupled bearing stiffness and low or non-existent damping under no radial load. As the radial load increases, the cross-coupled stiffness tends to becomes less important than the direct stiffness, either by its magnitude becoming small in comparison, or becoming positive. In any case, the design of a rotor system operating in space will have to take into account the possibility of this type of instability. There are methods to deal with this problem, such as constructing the bump foil

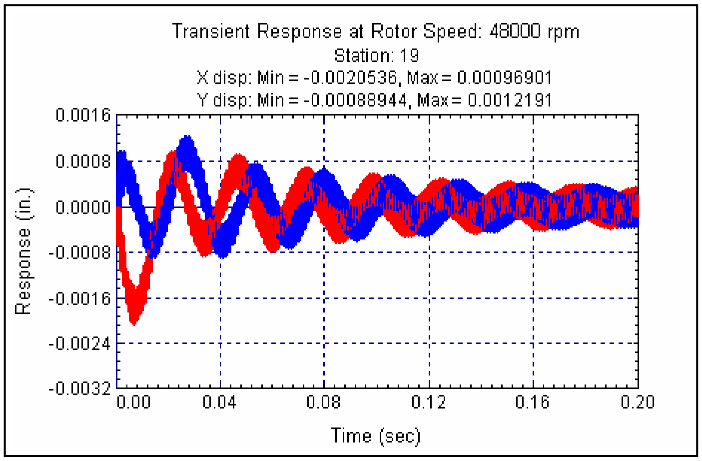

Figure 12. Transient response of rotor at turbine end showing decay of vibration.

geometry in a way that there is a bias on one side of the bearing that essentially applies a load. Although not considered a major issue, attention during design is warranted.

In order to determine the feasibility of the rotor system, the stiffness of the bearings is modeled as if there were a small radial load to produce bearing dynamic coefficients. Using soft bearings in this model, rotor stability is attainable with damping coefficients of 1700 N-sec/m (10 lb-sec/in.) or higher. Figure 12 shows the transient response at the turbine end of the rotor. The decay in amplitude of vibration indicates that the rotor is stable at this condition. If the rotor were unstable, the vibration amplitude would grow in time rather than decay.

For earth-bound operation, as in a test rig configuration, the negative cross-coupled stiffness tends to decrease due to sufficient radial loads. However, choosing stiffer bearings to constrain static deflection, as described earlier, negatively impacts stability. The amount of damping needed for stability is marginally higher at 1900 N-sec/m (11 lb-sec/in). Stiffer foil bearings typically exhibit less damping than softer foil bearings because some of the damping comes from Coulomb friction in the foil structure. In stiff foil bearings, there is less foil motion and less frictional rubbing resulting in less damping. This analysis leads one to realize that stiffer bearings may be better for ground-based operation while softer bearings may be more appropriate for space operation. Bearing and rotor designers will need to determine the suitable trade-off to ensure successful ground-based and space-based rotordynamic stability. One possibility is to optimize the system for space, and do all ground testing with the rotor in a vertical position. This will require proper attention to the trust bearing such that it can support the weight of the shaft as well as the thrust loads. In any case, while it is difficult to predict the exact amount of damping supplied by a given bearing design, the levels of damping needed for stability of this system, roughly 1700 to 3500 N-sec/m (10 to 20 lb-sec/in.), are attainable for foil bearings (ref. 9).

Conclusions and Recommendations

This preliminary analysis shows that the proposed system for power conversion in a Brayton cycle turbo alternator is feasible from a rotordynamic standpoint. The design is within the scope of what has been done with similar machines. Good, detailed design work is needed from designers familiar with the nuances of foil bearing systems. Persons should be involved who have experience designing rotors for high speed electrical machines to account for the effects of the alternator. Foil bearing designers should also have experience dealing with the special requirements of very lightly loaded bearing designs, such as space applications or vertical rotor machines. Also important will be the effects of the process fluid, HeXe, on the bearing performance. Laboratory testing in HeXe gas will be critical in assessing designs and providing feedback to designers. Ground based simulation of potential rotor systems can provide invaluable insight into performance, and model validation. As such, component level and system level testing is viewed as indispensable to ensure a working design. In addition, the model developed here will be continually refined and used throughout the project.

The prospects of designing a robust gas foil bearing rotor system for a closed cycle Brayton turbo-alternator are good. Coupling foil bearings with a rigid rotor offers the ability to avoid critical speeds at all speeds in the operating range. Traversing the low frequency rigid body critical speeds should occur fast enough to prevent damage. With careful rotor and bearing design and high fidelity balancing, steady state unbalance response amplitude should be well within acceptable margins. System stability, while at the edge of current state-of-the-art in foil bearing design, should be attainable with careful attention to detailed design.

References

1. Mason, L.S., "A Power Conversion Concept for the Jupiter Icy Moons Orbiter," 1st AIAA/ASME/IEEE International Energy Conversion Engineering Conference, Portsmouth, Virginia, August 17–21, 2003, AIAA–2003–6007.
2. Hervol, D., Mason, L., and Birchenough, A., "Experimental Results from a 2kW Brayton Power Conversion Unit," NASA/TM—2003-211999.
3. Heshmat, H., "Advancements in the Performance of Aerodynamic Foil Journal Bearings: High Speed and Load Capability," ASME Journal of Tribology, Vol. 116, pp. 287–295, April, 1994.
4. Chen, W.J., Foil Bearing Modules for "DyRoBeS" ver. 3.3., Copyright 1991-2000 Eigen Technologies Inc., Bearing modules developed for Mohawk Innovative Technologies, Inc. (MiTi).
5. Howard, S.A., DellaCorte, C., Valco, M.J., Prahl, J.M., and Heshmat, H., "Dynamic Stiffness and Damping Characteristics of a High-Temperature Air Foil Journal Bearing," Tribology Transactions, Vol. 44, Num. 4, pp. 657–663, 2001.
6. Howard, S.A., "Rotordynamics and Design Methods of an Oil-Free Turbocharger," NASA/CR—1999-208689.
7. Vance, J.M., "Rotordynamics of Turbomachinery," John Wiley & Sons, Inc., New York, 1988.
8. Heshmat, H., "Gas-Lubricated Foil Bearings for Heat Pumps with Counter-Rotating Heat Exchangers," ASME Paper 82–DET–139.
9. Heshmat, H., "Operation of Foil Bearings Beyond the Bending Critical Mode," STLE/ASME Tribology Conference, Orlando, FL, Oct. 10–13, 1999, ASME Paper 99–TRIB–8.

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13. ABSTRACT *(Maximum 200 words)*

The choice of power conversion technology is critical in directing the design of a space vehicle for the future NASA mission to Mars. One candidate design consists of a foil bearing supported turbo alternator driven by a helium-xenon gas mixture heated by a nuclear reactor. The system is a closed-loop, meaning there is a constant volume of process fluid that is sealed from the environment. Therefore, foil bearings are proposed due to their ability to use the process gas as a lubricant. As such, the rotor dynamics of a foil bearing supported rotor is an important factor in the eventual design. The current work describes a rotor dynamic analysis to assess the viability of such a system. A brief technology background, assumptions, analyses, and conclusions are discussed in this report. The results indicate that a foil bearing supported turbo alternator is possible, although more work will be needed to gain knowledge about foil bearing behavior in helium-xenon gas.

14. SUBJECT TERMS

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