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Preliminary Test Results of Rotordynamic Characteristics for 100 Watts Class Micro Power System

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Abstract

The rotordynamic characteristics of the micro power system supported by foil bearings were investigated. Stability analysis were performed by finite element method with the predicted dynamic coefficients of foil bearings. A preliminary test rig was developed to simulate the operating characteristics of the micro power system. It consisted of a rotor supported by two foil journal bearings and two foil thrust bearings, and an impulse drive turbine. The foil journal bearings had a diameter of 7 mm and a length of 7 mm (L/D=1). A test rig was operated very stably under various situations and speeds up to 300,000 rpm. The main portion of the rotor response was synchronous and the amplitude of synchronous vibration was about 5 ~ 20 μm. Further, the theoretical and experimental results for the unbalance response were compared. From this study, we showed the possibility of stable performance for the micro power system supported by foil bearings.

Keywords : Micro Power System, Foil Bearing, Rotordynamics, Stability

1 INTRODUCTION

The micro power system is a new portable power source, based on the Brayton cycle, which consists of a compressor, a turbine, a generator, and a combustion chamber. Air is compressed when it passes through the compressor and is transferred to the combustion chamber. Here, the combustion process occurs and the chemical energy of the fuel is transferred into the air. The high temperature gas which is produced by combustion has a high specific enthalpy. The turbine extracts a portion of the energy of the high-temperature gas and generates electrical energy with the generator. The micro power system requires an extremely high rotating speed to generate a sufficient compression ratio due to its small size. A small rotor of millimeter-scale diameter was also designed to operate at the surface speed of 500 m/s[1] and the temperature of turbine inlet component is very high (about 1,000 ℃). Existing rolling element bearings and conventional lubricated bearings have performance limits at this extreme environment of high operating speeds and temperatures. Gas bearings such as the externally pressurized gas bearing[2] or the hydrodynamic herringbone grooves and spiral grooves bearings also operate at higher surface speeds and higher operating temperatures[3].

However, they require additional air supply systems or extreme precision manufacturing procedures by MEMS fabrication process.

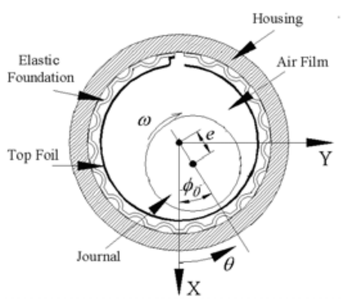


Figure 1. Foil journal bearing

To overcome the above mentioned problems, foil bearings were selected. Foil bearings (see Fig. 1) are self-acting compliant-surface hydrodynamic bearings that use ambient air or any process gas as a lubricating fluid. The hydrodynamic film pressure builds up in the small gap between the rotating shaft and the smooth top foil. The top foil provides a smooth bearing surface and is often supported by a series of bump foils that act as springs to make the foil bearings compliant. Because of the compliant bearing surface, there are certain advantages over the traditional rigid bearings including a higher load capacity for a given minimum film thickness, a lower power loss, and an increased stability. Foil bearings are less susceptible to damage due to large foreign particles in the lubricant flow, as foils can deform instead of seizing up. The compliant foil bearings are also more tolerant of misalignment and centrifugal/thermal growth, since compliant foils can accommodate these changes in shaft diameter and bearing clearance. With the development of a new foil bearing design and coating materials, foil bearings are now good candidates for high-temperature environments[4].

2 ANALYSIS OF FOIL BEARING

The micro power system is operated in a high-temperature environment with high eccentricity, so the rarefaction effect on the foil bearing is considered. When a high load is applied to the foil bearing under a high temperature, the local Knudsen number of the minimum film thickness may be greater than 0.01. In such a case, the slip flow effect becomes especially large as the molecular mean free path increases with temperature.

In this study, the slip flow effect was considered in estimating the dynamic coefficients of an elastically-supported foil bearing (see Figure 1) when a high or allowable load was applied. The pressure distribution within the clearance of the foil bearing for the coordinate system in Figure 1 is determined using the following modified Reynolds equation:

$$\nabla \cdot \left(-\frac{1}{12\mu} \varphi^{p} ph^{3} \nabla p + \frac{\bar{U}}{2} ph \right) + \frac{\partial}{\partial t}\left(ph\right) = 0. \qquad (1)$$

φ^{p} denotes the molecular rarefaction coefficient.

In the model of a foil bearing, the smooth top foil is supported by an elastic foundation, the local deflection of which depended only on the pressure at the point of application. When the bending and membrane stresses on the foil were neglected, the elastic foundation including the equivalent viscous damping and the film thickness were represented by the following equations, respectively:

$$k_{e} \cdot w + c_{e} \cdot \frac{dw}{dt} = p - p_{a} \qquad (2)$$

$$h = c - e\cos(\theta - \phi) + w \qquad (3)$$

The variables of k_e and c_e are, respectively, foil structural stiffness and damping per unit area.

Applying the perturbation method, the dynamic characteristics of foil bearings were calculated. With the solutions for steady-state and perturbed pressure, the stiffness and damping coefficients were readily calculated as:

$$\begin{bmatrix} \bar{K}_{xx} & \bar{K}_{xy} \\ \bar{K}_{yx} & \bar{K}_{yy} \end{bmatrix} = \frac{c}{p_a R^2} \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} = \int_{-L/D}^{L/D} \int_{0}^{2\pi} \begin{bmatrix} \bar{p}_x \sin\theta & \bar{p}_y \sin\theta \\ -\bar{p}_x \cos\theta & -\bar{p}_y \cos\theta \end{bmatrix} d\theta d\bar{z} \qquad (4)$$

$$\begin{bmatrix} \bar{C}_{xx} & \bar{C}_{xy} \\ \bar{C}_{yx} & \bar{C}_{yy} \end{bmatrix} = \frac{c\omega}{p_a R^2} \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} = \int_{-L/D}^{L/D} \int_{0}^{2\pi} \begin{bmatrix} \bar{p}_{\dot{x}} \sin\theta & \bar{p}_{\dot{y}} \sin\theta \\ -\bar{p}_{\dot{x}} \cos\theta & -\bar{p}_{\dot{y}} \cos\theta \end{bmatrix} d\theta d\bar{z} \qquad (5)$$

3 ROTORDYNAMIC ANALYSIS

The rotordynamic model of the micro power system was established using a finite element program. The static equilibrium position, where the foil bearing load capacity is equal to the rotor weight, was found using the Newton-Raphson method. The stiffness and damping coefficients of the foil bearings at the static equilibrium position were calculated by solving four first-order equations. Using the

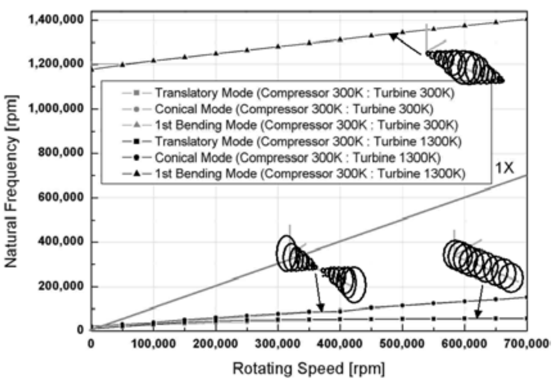


Figure 2. Campbell diagram of the micro power system

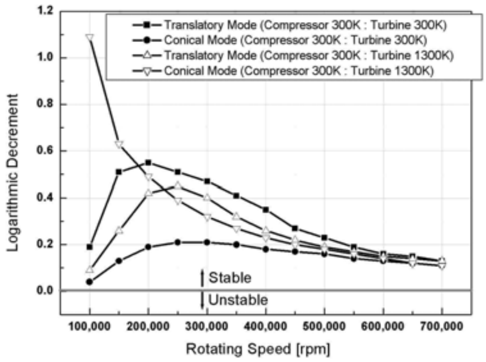


Figure 3. Logarithmic decrement vs. rotating speed

dynamic coefficients of foil bearings and the rotordynamic model of the micro power system, the vibration orbit was predicted through time integration.

Figure 2 is a Campbell diagram of the micro power system. While the operating environment was under room temperature, 300 K, the first rigid mode (the translatory mode) and the second rigid mode (the conical mode) were predicted at 10,100 rpm and 12,300 rpm, respectively. On the other hand, the first rigid mode and the second rigid mode considering the turbine inlet temperature of 1,300 K were predicted at 13,750 rpm and 25,7000 rpm, respectively. In any case, the first bending mode is always predicted over 1,000,000 rpm regardless of temperature conditions. The micro power system supported by foil bearings operates under conditions within the range of 100,000~700,000 rpm. This indicates that the operating speed of the micro power system is beyond the first and second critical speed (i.e., the translatory mode and the conical mode) and much lower than that of the third mode, the first bending mode.

Generally, the damping characteristics of foil bearings are lower than those of oil-lubricated bearings. Thus, the assurance of stability is an essential consideration. Figure 3 shows logarithmic decrement versus rotating speeds. It shows that the stability of the rotor-bearing system is expected to be positive over the entire operating speed range, which means that the micro power system can be operated at a regular rotating speed, 700,000 rpm, with dynamic stability. Actually, foil bearings are tolerant of external impacts and transient dynamic conditions.

4 PRELIMINARY TEST RIG

A preliminary test rig of the micro power system (see Figure 4) was developed to simulate the operating characteristics of the micro power system. It consisted of a rotor supported

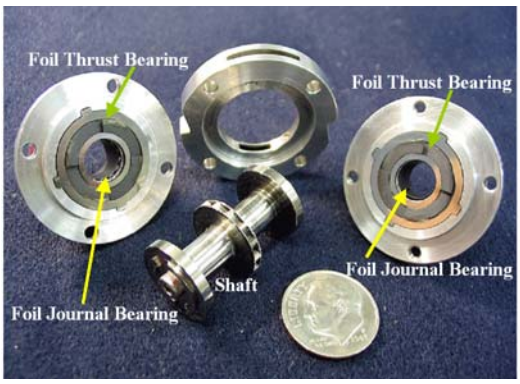


Figure 4. Preliminary test rig of the micro power system

by two foil journal bearings and two foil thrust bearings, and an impulse drive turbine. The rotor was made of Inconel 718 for high-temperature operating, had a length of 26 mm, and a weight of 14.74 grams. The residual mass unbalance was 0.0035 g·mm. Dummy disks were located at each end of the rotor. These were used to simulate the turbine and compressor. This preliminary test rig could be operated at 300,000 rpm given the limit of the rotating force based on the impulse turbine configuration and structure.

The foil journal bearings had a diameter of 7 mm and a length of 7 mm (L/D=1). The foil thrust bearings had an inner diameter of 10 mm and an outer diameter of 15 mm. A verified high-reliability coating was applied to the smooth top foil.

To measure the vibrations of the rotor, fiber optic displacement sensors were positioned in the horizontal and vertical directions at each end of the rotor. Figure 5 shows detected points while test operating.

5 TEST RESULTS AND DISCUSSION

The main purpose of the experiments was to verify the stability of the micro power system supported by foil bearings, and to identify the prediction of the vibration orbit by analytical results compared to those of the experiments.

The micro power system was tested at horizontal and vertical attitude. These tests showed that the performance, while dynamic conditions were imposed on the micro power system, adapted to dynamic applications such as those of Miniature Aerial Vehicles (MAVs), mobile robots for dangerous work, and others. Figure 6 and Figure 7 show the frequency spectrums waterfall plots and vibration orbits measured from fiber optic displacement sensors for each of the operating test to validate the running characteristics of the micro power system. The detected points(X1, X2, Y1) indicated in Figures 6 to 7 were defined in Figure 5. Considering these test results, the main portion of the rotor response was synchronous and the amplitude of synchronous vibration in all directions was about 5~20 μm. No subsynchronous or other responses appeared. These clear frequency spectrums indicated a well-damped rotor-bearing system. Especially, the rotor was mainly supported by foil thrust bearings while operating in a vertical attitude. As shown in Figure 7, not only the foil journal bearings but also the foil thrust bearings had sufficient load capacity, stiffness and damping characteristics to maintain the system with stability.

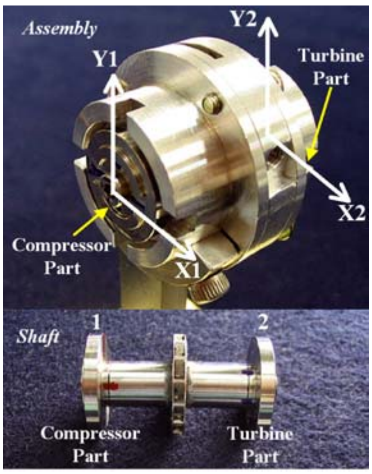
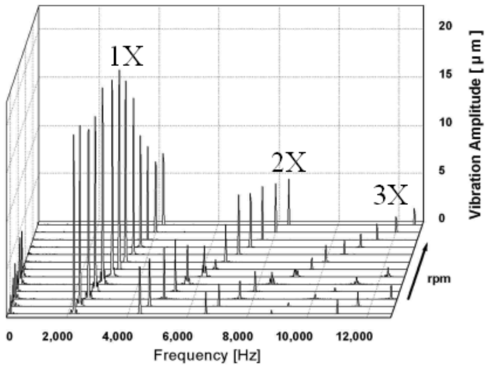
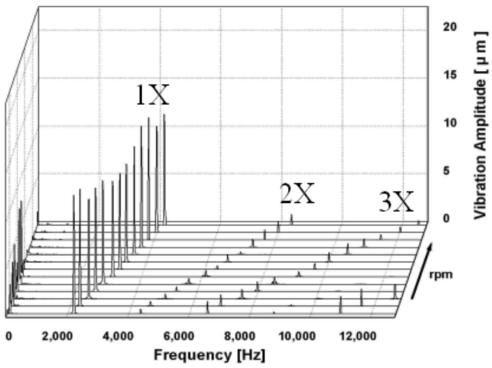


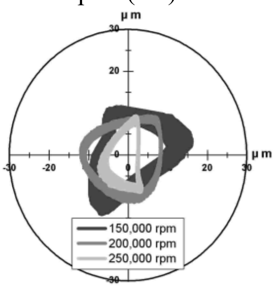
Figure 5. Detected points while test operating



(a) Compressor part (X1) waterfall plot



(b) Turbine part (X2) waterfall plot



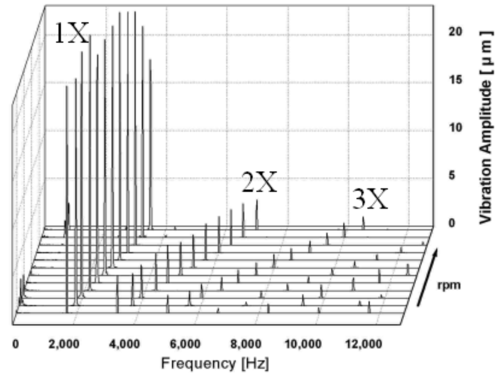
(c) Compressor part (X1-Y1) vibration orbit

Figure 6. Waterfall plot and vibration orbit for horizontal operation

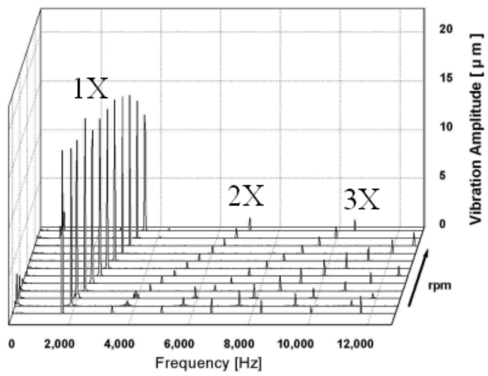
Figure 8 compares experimental and theoretical orbits at the end of the rotor. As shown in this Figure, the experimental vibration orbit had decreasing trends while the rotating speed was increasing, and the theoretical vibration orbit had increasing trend while the rotating speed was increasing. These results explained the experimental trend that an increased rotating speed could increase the stiffness of the elastic foundation (bump foil) in the foil bearing. It also explained that the vibration amplitude grew by an unbalance mass while the rotating speed was increasing. Considering these results, it was expected that the actual stiffness of foil bearings would be a little larger than the predicted stiffness, according to an analysis in the high-rotating-speed region.

6 CONCLUSION

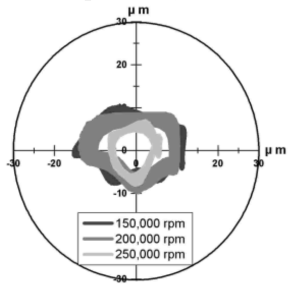
In this paper, a rotordynamic analysis and the stability of the micro power system supported by foil bearings were investigated. How numerical predictions can effectively explain



(a) Compressor part (X1) waterfall plot

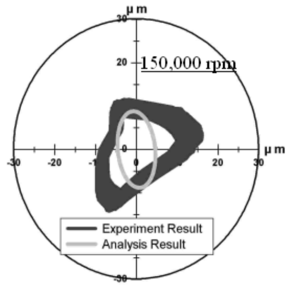


(b) Turbine part (X2) waterfall plot

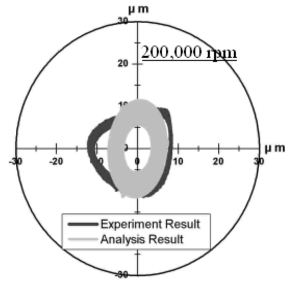


(c) Compressor part (X1-Y1) vibration orbit

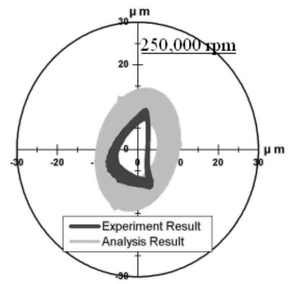
Figure 7. Waterfall plot and vibration orbit for vertical operation



(a) At 150,000 rpm



(b) At 200,000 rpm



(c) At 250,000 rpm

Figure 8. Comparison of vibration orbit between the analysis results and the experimental results

the experimental observations was also investigated by comparing the experimental and theoretical research findings on the unbalance response of the rotor-bearing system for the micro power system with foil bearings.

The rotor-bearing elements were established using a finite element analysis, and the behaviors of the system were predicted. The micro power system had the first and second critical speeds (i.e., the translatory mode and the conical mode) in the range of 10,000~30,000 rpm according to various environmental temperatures, and the third critical speed (the first being mode) over 1,000,000 rpm. These showed that there was no resonance in the regular operating speed region.

Also, the operating characteristics were simulated using a test rig of the micro power system with foil bearings. The test rig was operated very stably at various speeds and under various situations. The foil journal bearings and the foil thrust bearings had enough load capacity, stiffness and damping characteristics to maintain the stability of the system.

According to a comparison of experimental and theoretical vibration orbits, the actual stiffness of foil bearings was a little larger than that shown by analysis. This limitation will be addressed by an advanced foil bearing analysis and an extensive structural dynamic test of the elastic foundation.

REFERENCES

[1]Sangkyun Kang, "Fabrication of Ceramic Components for Micro Gas Turbine Engines", Ph.D. thesis, Stanford University, Stanford, CA, 2002.

[2]Alan H. Epstein, "Millimeter-scale, Micro-Electro-Mechanical Systems Gas Turbine engines", Journal of Engineering for Gas Turbines and Power, Vol. 126. pp. 205-226, 2004.

[3]K. Isomura *et al*. "Component development of micromachined gas turbine generators", Power MEMS 2002 Technical Digest, pp. 32-35, 2002.

[4]Mohawk Innovative Technology Inc. Development Newsletter, "Mesoscopic Turbojet Simulator Tested at Speeds Above 700,000 rpm on Air Foil Bearings", Vol. 17, pp. 1-4, 2003.