Static and Dynamic Characteristics of a Micro Gas Turbine Rotor Supported by Radial Foil Bearings

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Abstract

In this paper, the design and manufacturing process of radial foil bearings for the rotor diverted from a commercially available turbocharger is introduced and then measuring techniques on investigating static and dynamic characteristics of a rotor supported by radial foil bearings for Micro Gas Turbine (MGT) use is discussed. The diameter of the rotor tested is 20mm and the surface of the rotor is coated by ceramic. The radial bearing used is composed of two elements; i.e. a housing and a multi wounded foil with projections of hemisphere on one side in the adequate interval. The bearing radial clearance between the rotor surface and the foil is 20μm. In the experiment, we measured rotational speed, load capacity and vibrations of the rotor in radial direction. Subsequently, natural frequencies corresponding to lateral, conical and the first bending mode were evaluated. Finally, the method for predicting the onset whirl rotational speed based on the data obtained by measurements is explained.

Key word: Foil bearing, Micro Gas turbine, Bearing Characteristics, Stability, Load Capacity

1. INTRODUCTION

In recent years, micro gas turbines (MGT) of 100kW or less are drawing close attention as power generators equipped with cogeneration features. Aerodynamic foil bearings are deemed to be the best candidate for the MGT due to its ease of maintenance. However, the details of design guidelines have not been clarified though this element is used in auxiliary power units (APU) for aircraft units. Therefore, we start working in designing and manufacturing radial foil bearings for the rotor diverted from a commercially available turbocharger, the diameter of which is 20mm. In this paper, first, we will show the details of the radial foil bearing tested in our experiment and introduce the method to measure load capacity and bearing stiffness coefficients to determine stability. We will also propose the method for predicting relevant vibrations such as the first bending natural frequencies taking account of bearing characteristics and whirl limit speed determined by measured lateral and conical mode vibration data.

2. EXPERIMENT

2.1 Radial foil bearing

The layout of the manufactured radial foil bearing is shown in Fig.1. This bearing is composed of two elements; i.e. a housing and a foil with projections of hemisphere on one side in the adequate interval. The foil is made from a phosphor bronze plate of 0.1mm in thickness, 20mm in width, 202mm in length with many hemispherical projections whose height is 0.2mm.

This plate is manufactured by wet etching process and projections are made by specially designed jig. When installing the foil to the housing, the plate was bent at the position of 3mm measured from the edge of the plate. Then, the foil was wound triply and fixed with the housing. Due to the rotation of the shaft, a wedge like space with high pressure is yielded between the shaft and the inner bearing surface, which levitates the rotating shaft. The bearing clearance is represented as C and is described by the equation as follows;

$$C = (D - d)/2 - 2h - 3t \qquad (1)$$

where C, D, d, h and t denotes radial clearance, inner diameter of housing, diameter of rotor, height of projections and thickness of the foil, respectively. The bearing radial clearance between axial surface and bearing surface is 20μm, because the foil is triply wounded.
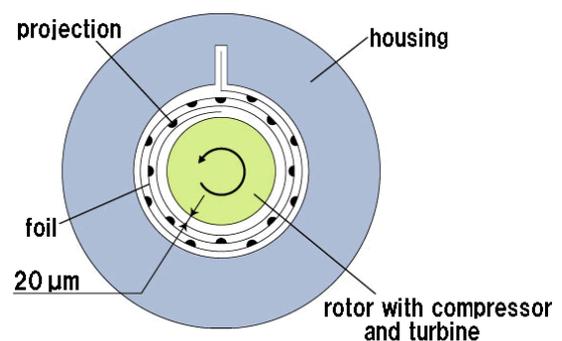

Fig.1 Schematic view and composition of radial foil bearing

These types of multi wounded foil bearing have advantages of easy fabrication and assemble compared with usual air bearing. In Fig.2, dimensions and compositions of the foil employed in the experiment are illustrated which has another advantage of changing bearing

characteristics by just accommodating the density and distribution pattern of the projections.
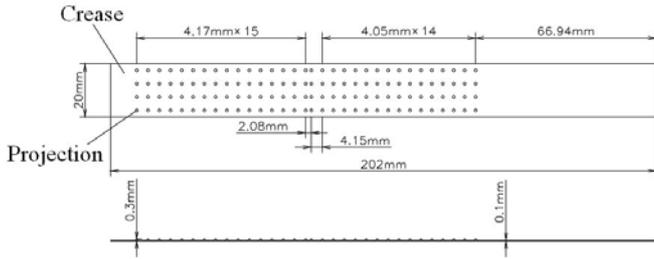

Fig.2 Radial foil with projections

2.2 Experimental apparatus

The composition of the rotor and rotor housing used in this experiment is shown in Fig.3. The rotating shaft, the weight of which is 0.5kg consists of the following four parts, the rotor with a turbine, a sleeve, a thrust disk and a nut for fixing the thrust disk to the rotor and the shaft is supported by two radial foil bearings and a spiral thrust bearing. The martensitic stainless steel was selected as a material of the sleeve, and the surface of the shaft is coated by ceramic. At the right end of the rotating shaft, a reverse thread was cut to fix the thrust disk and a through hole was drilled for measuring the rotational speed. The sleeve was inserted under pressure and fixed with the shaft.
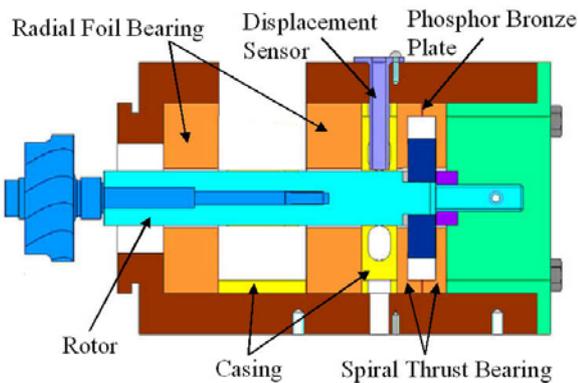

Fig.3 Experimental apparatus

2.3 Results on the vibration measurement

In Fig.4, measured vibration data in the radial direction are
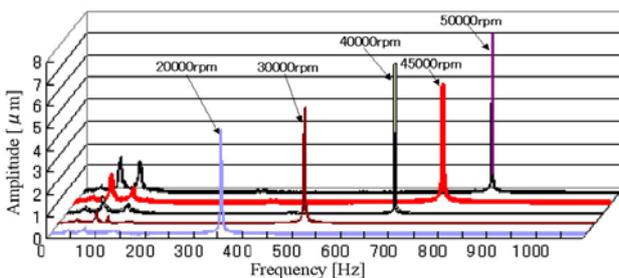

Fig.4 Measured frequency characteristics

shown where large amplitude peaks correspond to synchronous vibration components and two peaks around 100Hz correspond to conical and translational rigid modes, respectively.

3. INVERSE ANALYSIS ON BEARING COEFFICIENTS

In order to design the dynamic characteristics of the rotor supported by foil bearings, the information on bearing stiffness and damping is indispensable. We propose the method for calculating bearing stiffness and damping coefficients from the measured data with conical and translational rigid mode vibration and successfully obtained bearing coefficients as $k_b = 420000\,[\text{N/m}]$ and $c_b = 538\,[\text{N} \times \text{s/m}]$.

4. CALCULATION OF THE NATURAL FREQUENCIES OF FIRST BENDING MODE

Employing FEM and making use of the bearing coefficients obtained by the inverse analysis, we calculated the natural frequencies of the first bending mode of the rotating shaft supported by radial foil bearings. The results are shown in Fig.5. As can be found on this figure, backward and forward natural frequencies are obtained as approximately 130000 rpm and 175000 rpm, respectively.
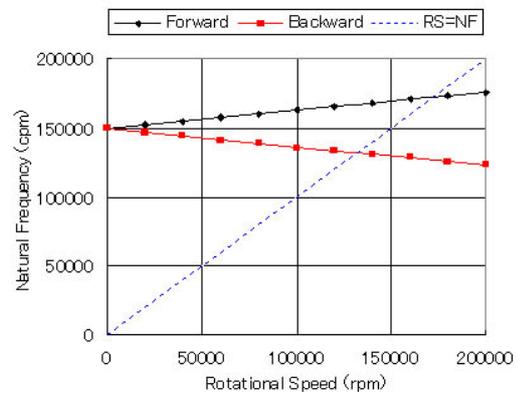

Fig.5 Calculated natural frequencies of the first bending mode

5. LOAD CAPACITY TEST OF THE RADIAL FOIL BEARING

In order to determine the size of the bearing, the information on the load capacity is necessary. Then, we measured the load capacity of the foil bearings by changing the additional cylindrical weight attached to the rotating shaft by a pojirock as shown in Fig.6. Measured results are shown in Fig.7 as a function of eccentricity ratio.
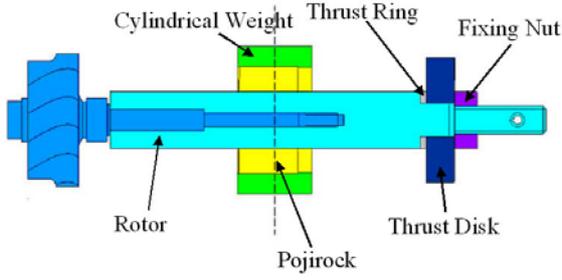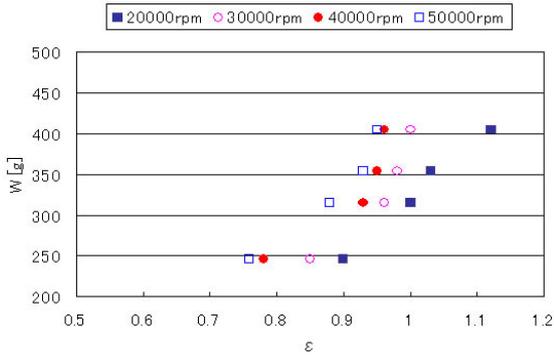
Fig.6 Rotor used for measuring load capacity

Fig.7 Load capacity vs eccentricity ratio

Fig.7 shows the fact that eccentricity of the bearing decreases with increasing rotational speed under constant load and that eccentricity exceeds 1.0 under constant rotational speed due to the deformation of the foil, which is the feature of the foil bearing.

6. MEASUREMENT OF STIFFNESS COEFFICIENT

In an actual test to determine the stability limit, we have to estimate the critical speed before the test to avoid the failure. We tried to predict the whirl limit speed based on the data obtained during safe operation. Then, we measured the bearing stiffness coefficients composed of direct term k_{xx} and k_{yy} and cross term k_{xy} and k_{yx} by changing the applied forces. The coefficients of k_{xx} and k_{xy} can be determined by using the apparatus shown in Fig.6. However, to obtain k_{yy} and k_{yx}, we had to design and manufacture special apparatus making use of the air jet shown in Fig.8 and measured the reactive force by load cell. Using this apparatus, we could successfully obtain necessary coefficients.

In Fig. 9, obtained stiffness coefficients are shown as a function of rotating speed where open square, closed square, closed circle, and closed triangle correspond to to k_{xx}, k_{xy}, k_{yx} and k_{yy} and solid lines show extrapolated curves. Now, we can discuss the stability criterion based on such information.
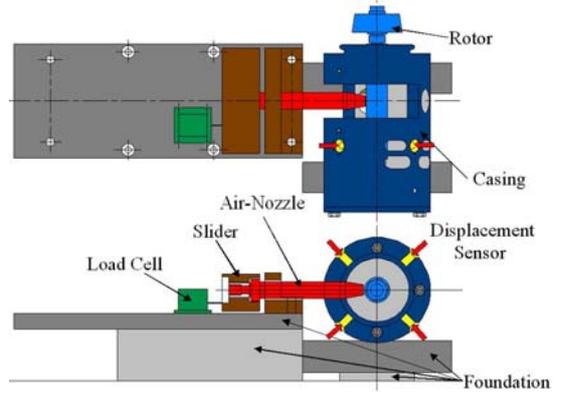

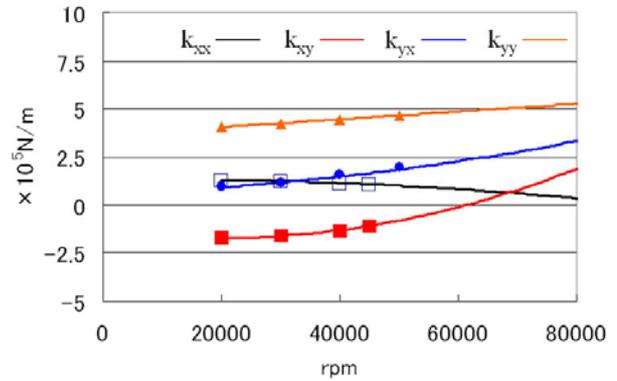Fig.8 Apparatus for measuring bearing stiffness parameters

Fig.9 Bearing stiffness coefficient

7. PREDICTION OF WHIRL LIMIT SPEED

In our measuring system, bearing damping coefficients are not measurable. In the following, we will limit our discussion the case neglecting damping effect. When we disregard the bearing damping coefficients, the equation of motion of the rotating shaft can be described as

$$\begin{bmatrix} m' & 0 \\ 0 & m' \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix}\begin{bmatrix} x \\ y \end{bmatrix} = 0 \qquad (2)$$

where m' denotes the mass. The characteristic equation of the above equation is obtained as,

$$\begin{vmatrix} m's^2 + k_{xx} & k_{xy} \\ k_{yx} & m's^2 + k_{yy} \end{vmatrix} = 0 \qquad (3)$$

Expanding the above determinant, we will obtain,

$$a_4 s^4 + a_2 s^2 + a_0 = 0 \qquad (4)$$

$$a_4 = m'^2 \qquad (5)$$

$$a_2 = m'(k_{xx} + k_{yy}) \tag{6}$$

$$a_0 = k_{xx}k_{yy} - k_{xy}k_{yx} \tag{7}$$

Now, we may determine the whirl limit speed by Hurwitz criteria and the critical condition can be obtained as the condition that the coefficient a_0 of Eq.(7) is equal to zero. If we plot a_0 in the ordinate axis and rotational speed in the abscissa axis, the curve representing Eq.(7) are represented by Fig.10. As is shown by the figure, in the current case, the system will get unstable at the speed of 72000 rpm. However, in reality, the whirl limit is higher than the predicted speed because of the existence of the bearing damping.
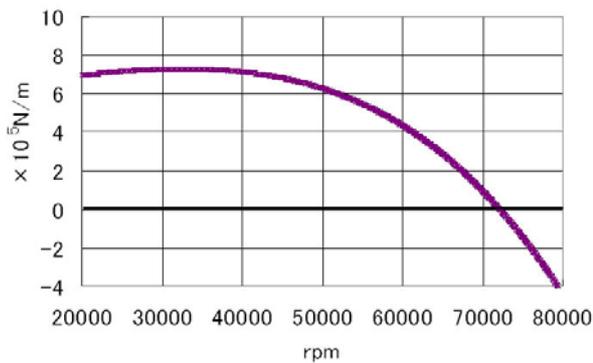

Fig.10 The coefficient a_0 determining stability

8. CONCLUSIONS

(1) We designed and manufactured radial foil bearings for high speed and small size rotating shaft system.

(2) We proposed inverse method for determining bearing coefficients of radial foil bearings.

(3) We also proposed simplified method to predict the whirl limit speed based on the data during safe operation.

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