ULTRA-HIGH-SPEED TAPE-TYPE RADIAL FOIL BEARING FOR MICRO TURBOMACHINERY

K. Hikichi[1], S. Togo[2], K. Isomura[3], N. Saji[3], M. Esashi[1] and S. Tanaka[1]
[1]Tohoku University, Sendai, Japan
[2]Tohoku-Gakuin University, Tagajyo, Japan
[3]IHI Corporation, Tokyo, Japan

Abstract: This paper describes the design and test results of an ultra-high speed tape type radial foil bearing for turbo machinery. The outer and middle foil segments have some folds, where the section of the foil is made thinner by etching grooves. This automatically forms leaf springs just by wrapping the foil around a rotor three times. A foil bearing test rig with a shaft diameter of 8 mm was designed based on a design method developed in this study. The achieved maximum rotation speed was 642,000 rpm, corresponding to a DN product of 5,236,000. This is the highest *DN* product achieved by foil bearings to the best of our knowledge. The bearing performance showed no significant deterioration after 300 start-and-stop cycles.

Keywords: Foil bearing, Hydrodynamic bearing, Gas turbine generator, High speed rotation

INTRODUCTION

A portable power source with high energy density is indispensable for the practical use of mobile power-consuming machines such as self-powered robots and personal electric vehicles. As such a power source, a miniaturized micro gas turbine generator potentially has the advantages that exhaust gas is cleaner than that of reciprocating engine generators and power density is higher than that of fuel cells. In addition, continuous operation is enabled by refueling. Therefore, a lot of efforts have been made in the world for drastically downsizing gas turbine engines to coin size [1]-[3] or palm-top size [4]-[5]. Our group also has been studying a palm-top-size gas turbine engine [6]-[7], and demonstrated the establishment of Brayton cycle in this scale for the first time [7]. This world's smallest class gas turbine engine used externally-pressurized air bearings called "hydroinertia air bearings" [8]-[9] to support an inconel rotor with a shaft diameter of 8 mm up to 500,000 rpm or higher. For practical uses, however, the bearing must be hydrodynamic type.

Among several kinds of hydrodynamic gas bearing, a foil bearing is the most promising for miniaturized gas turbine engines, because it is tolerant of inevitable thermal expansion due to the compliance of foils, and also the friction between the foils produces moderate dumping to suppress rotor vibration at high rotation speed. There are several kinds of foil bearing including tape type [10], leaf type [11]-[12] and bump foil type [13]-[15]. The tape type foil bearing generally uses one piece of foil, which is wrapped two or more times around a shaft, forming a bearing surface and leaf springs. It is simple in structure, and thus basically suitable for small rotors, but there is few open literatures reporting its successful application to small high speed rotors.

This paper describes the design and test results of an ultra-high speed tape type radial foil bearing for a mobile gas turbine generator based on Ref. [7]. The shaft diameter is 8 mm, and the target maximum rotation speed is 600,000 rpm, corresponding to a *DN* product of 4,800,000, where *D* is rotor diameter in millimeter, and *N* is rotation speed in rpm.

DESIGN

Structure of tape type foil bearing

The structure of the tape type foil bearing developed in this study is shown in Fig. 1. A tape-shaped foil made of stainless steel is wrapped three times around a rotor, and then fixed in a housing. The outer and middle foils have some folds, where the section of the foil is made thinner by etching grooves. This automatically forms leaf springs just by rolling the foil in the housing. The leaf springs elastically support the most inner foil, which is called "bearing foil" in this paper.

Deflection and stiffness of foil springs

Assuming that the shape of the foil in the housing is shown in Fig. 2, the stiffness of the formed leaf springs is theoretically estimated. Deflection angle φ at the folded lines is given by

$$\varphi = \pi / (N + 0.5), \qquad (1)$$

where *N* is the number of partitions per one wrap of the spring segment of the foil. The addition of 0.5 to *N* in Eq. (1) represents a half pitch off-phase of the folded

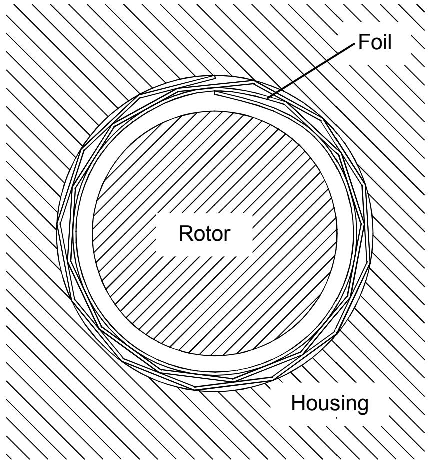


Fig. 1: Structure of a tape type foil bearing

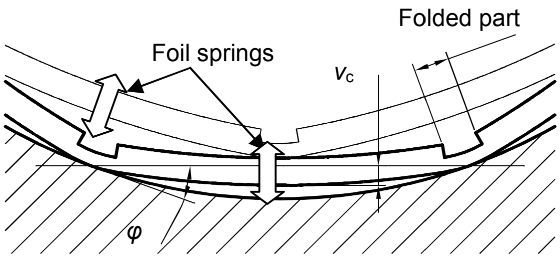


Fig. 2: Schematic of the foil in the housing

lines between the middle foil and the outer foil. From the relation between load and deflection at the center of the spring segment, the stiffness of the foil spring k_f is obtained. Figure 3 is the optical micrograph of the foil in the housing, indicating that the actual foil shape well agrees with the theoretical model shown in Fig. 2.

Load capacity of foil bearing

To estimate the load capacity, the foil bearing is modeled as shown in Fig. 4. In this model, it is assumed that the bearing foil is radially supported with springs, whose stiffness k_{sp} is given by

$$k_{sp} = 2k_f / (2 + \cos^2\varphi). \quad (2)$$

The radial stiffness of the bearing foil is also taken into account, but it is assumed that the bearing foil is rigid in the axial direction. Pressure distribution in the bearing clearance is simulated by divergence formulation (DF) method [16]. Simultaneously, the deformation of the bearing foil is calculated by transfer matrix method. These calculations are alternately repeated until they are converged.

An example of the calculation results of the pressure distribution is shown in Fig. 5. From this pressure distribution, the transformation of the foil is calculated. Figure 6 shows the tangential distribution of the bearing clearance and the pressure averaged along the axial direction. The load capacity is obtained by

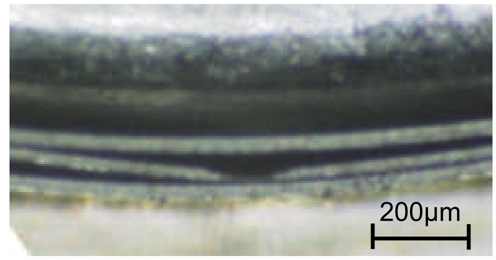


Fig. 3: Micrograph of the foil in the housing

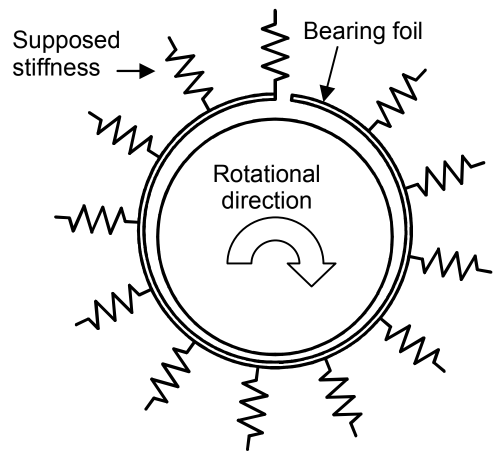
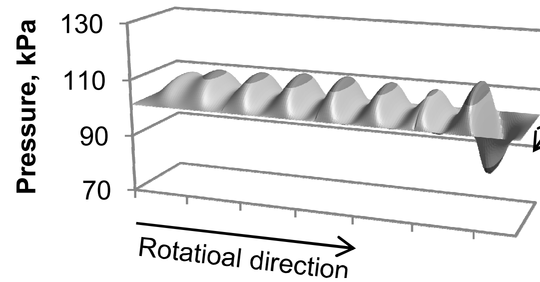
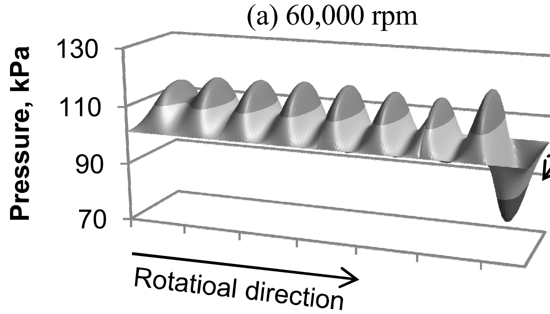


Fig. 4: Calculation model of the foil bearing



(a) 60,000 rpm

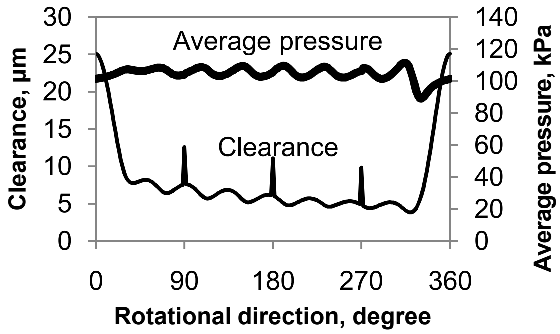


(b) 300,000 rpm

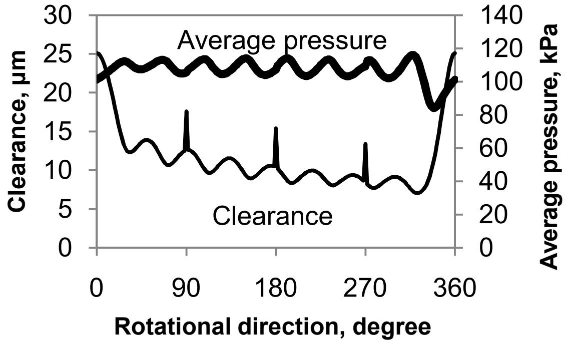
Fig. 5: Simulated pressure distribution in the bearing clearance

integrating the calculated pressure over the journal bearing surface. The calculated load capacity at an eccentricity of 1.0 μm is 0.12 N and 0.17 N at 60,000 rpm and 300,000 rpm, respectively.

Although it is not fully validated, yet, the simulation is generally known to underestimate the



(a) 60,000 rpm



(b) 300,000 rpm

Fig. 6: Calculated bearing clearance and pressure averaged along the axial direction

load capacity. Hence the real load capacity is expected to be not less than 0.12 N and 0.17N at 60,000 rpm and 300,000 rpm, respectively, and we judged this journal bearings to be capable of supporting the rotor with a weight of 17g (8.5g on each bearing).

EXPERIMENTS

Based on the abovementioned calculation method, we designed a tape type radial foil bearing test rig, which supported a Ti-6Al-4V rotor with an impulse turbine shown in Fig. 7. The shaft diameter is 8 mm, and the weight is 17 g. The rotor is coated with a ca. 1.6 μm thick sputter-deposited MoS_2/Ti composit coating. The thrust bearing is hydrostatic type, because this study focused on the radial bearing. Each side of the thrust bearing has 8 inherent orifice restrictors with a diameter of 0.4 mm. Each bearing dimension is compatible with the engine reported in Ref. [7].

Figure 8 shows a test result. When driving air generated enough torque to start the rotor, the rotor lifted off from the bearing foil, and the rotation speed rose quickly. The maximum achieved speed was 642,000 rpm (*DN* product is 5,236,000), which was limited by turbine performance. Figure 9 shows the relation between rotation speed and the vibration amplitude (difference of the maximum and the minimum value in one second) of the rotor, which was measured by an eddy current sensor. Rotor vibration increased with rotation speed, but its sudden increase was not observed all the way up to the maximum rotation speed. Figure 10 (a) and (b) show the waveform of rotor vibration at 300,000 rpm and 600,000 rpm, respectively. Whirl, which is the precession of the rotor at bearing resonant frequency (ca. 60,000 rpm), is found at 600,000 rpm, but the amplitude remains as small as 7 μm.

We repeated start-and-stop cycles 300 times to investigate the durability of the foil bearing. Figure 11 shows the radial amplitude of rotor vibration in the first and three hundredth cycles. No significant increase in vibration amplitude is found even after 300 start-and-stop cycles. Visual inspection of the tested foil bearing confirmed that a part of the coating on the rotor was lost, but transferred to the foil.

CONCLUSION

This paper presented the design and test results of a tape type foil bearing at very small scale. The shaft diameter is 8 mm, and other dimensions of the foil

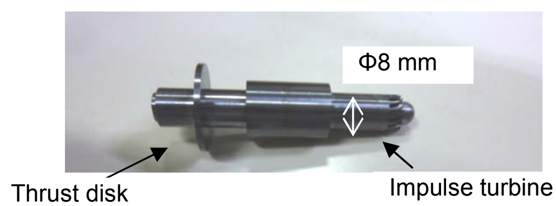


Fig. 7: Rotor

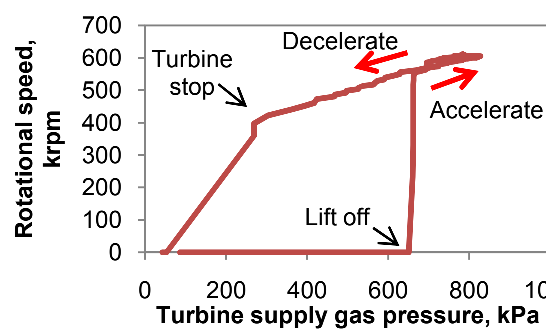


Fig. 8: Relationship between turbine supply pressure and rotation speed

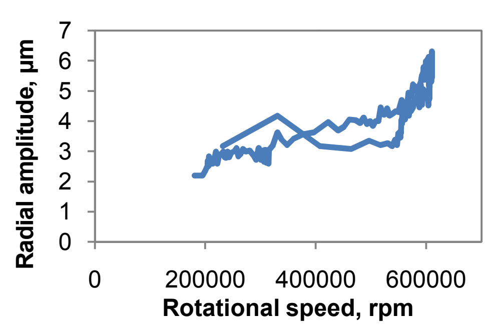
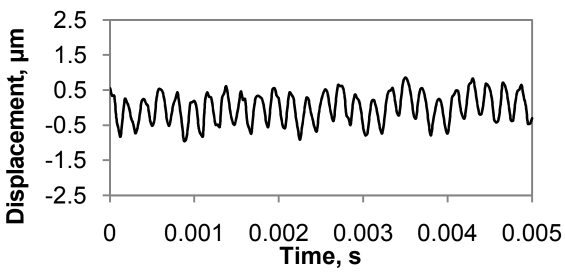
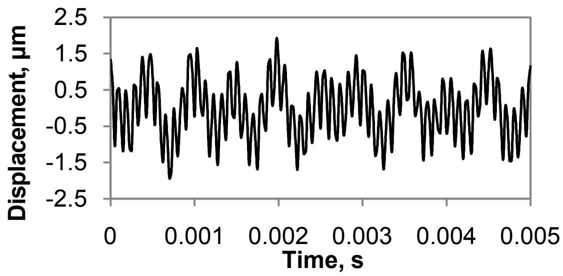


Fig. 9: Radial amplitude of rotor vibration as a function of rotation speed

bearing are compatible with the gas turbine engine by which we previously demonstrated the establishment of Brayton cycle. First, we developed the design method of the foil bearing using the calculation model that the bearing foil was radially supported by supposed springs. Using this design method, a foil bearing was designed, and its load capacity was estimated. The foil bearing was tested, and the maximum rotation speed of 642,000 rpm, which was limited by turbine performance, was recorded. The corresponding *DN* product is 5,236,000, which is the highest value achieved by foil bearings to the best of our knowledge. Rotor rotation was stable with moderate whirl even at such a high rotation speed. In addition, no significant deterioration of the bearing performance was found after 300 start-and-stop cycles.



(a) 300,000 rpm



(b) 600,000 rpm

Fig. 10: Waveform of radial rotor vibration

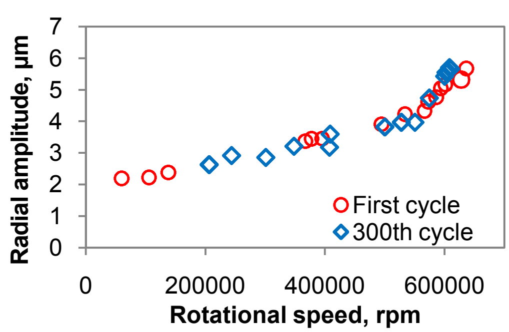


Fig. 11: Radial amplitude of rotor vibration in the first and three hundredth start-and-stop cycle

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