

Research of High Speed Shaft System Using Air Foil Bearing

Norio SHIMIZU*

Mitsuo MATSUSHIMA*

Ichiro ISHII*

Hitoshi IWADATE*

Hidehiko NAKATA*

ABSTRACT

The demand for a reliable, low maintenance, oil-free micro-gasturbine imposes significant demands on the bearing system. The air foil bearing provides low power loss, adequate damping for supercritical operation and long life with oil-free lubrication. In this application, foil bearings support a 9.2 kg, multi-component rotor operating at 77 000 rpm with a bending critical speed of approximately 42 500 rpm. Stable and reliable operation over full speed range on a shaft system rig and a micro-gasturbine has been demonstrated.

1. Introduction

While the small sized distributed generation is increasingly drawing attention, the micro-gasturbine has some superiority for its economical operation as the generator engine. Because its drive element is restricted to the main shaft, and it has few structural parts. To further increase this functional superiority requires an increased product life span and low level of maintenance. The air foil bearing has been targeted in meeting this demand.

The air foil bearing uses air flow to suspend the shaft thereby enabling rotation without additional mechanical contact support. Consequentially, eliminating the need for a lubrication system and achieving low maintenance due to a longer life are possible. Configuring an air foil bearing to a micro-gasturbine has become a key technology. Although this technology is currently applied to APUs on aircraft, the details of these applications are not made public. The aim of this research is to establish an air foil bearing structure that is

capable of achieving a high damping performance, and to configure this to an actual engine to verify its application on a micro-gasturbine.

2. Overview and Targeted Settings

2.1. Micro-gasturbine

Table 1 lists the specifications of a micro-gasturbine for which an air foil bearing is configured. Figure 1 shows the layout of a micro-gasturbine ^{(1), (2)}.

The engine shown in the figure adopts a symmetrical shaft layout in which high temperature components are positioned toward the center of the engine. This serves to suppress heat dissipation and prevent thermal stress. This layout has resulted in a shaft configuration that supports a high pressure compressor and turbine with extreme overhang thereby creating the severest bearing load conditions.

* Wako Research Center

Table 1 Specifications of micro-gasturbine

Items	Specifications
Rated engine speed	77 000 rpm (ISO conditions)
Allowable Max. speed	84 000 rpm
Life	60 000 hrs
Max. generator output	47.8 kW (ISO conditions)
Emissions (CO, HC, NOx)	< 3 ppm
Dimensions (D × L)	φ460 mm × 840 mm
Weight	145 kg

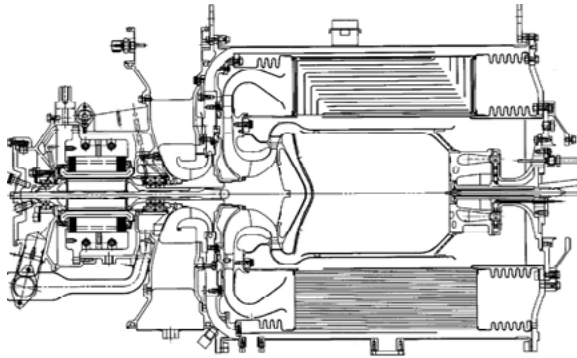


Fig. 1 Layout of micro-gasturbine

2.2. Targeted Settings

The greatest difficulty in applying the air foil bearing is the existence of the 1st bending critical speed on the shaft within the operating range which is caused by the low level of air foil bearing stiffness. Generally, when a low level of shaft stiffness is employed, critical speeds are avoided in low speed ranges. However, this requires a complex shaft structure and does not facilitate smooth substitution with rolling bearing. Consequentially, significant engine layout modifications are required.

In this research, the 1st bending critical speed has been allowed to set within the operating range, and passing through this critical speed at a low vibration level would make it possible to realize an air foil bearing capable of engine operation without making significant changes to the engine layout. Table 2 lists the targeted shaft system values for this engine. The 2nd bending critical speed was set above the maximum permissible engine speed of 84 000 rpm while the permissible amplitude for the bearing when the 1st bending critical speed is exceeded was set to 30 μm according to the cyclic fatigue strength of bump foil (refer to section 2.3.2). The bearing inside diameter was set to 48.8 mm in line with the magnetic rotor outside diameter on the applied engine.

Table 2 Shaft system performance target

Items	Performance target
Rated engine speed	77 000 rpm (ISO conditions)
1 st bending critical speed	Above lift up speed & As low as possible
2 nd bending critical speed	Above 84 000 rpm
Life	60 000 hrs
Bearing inner diameter	φ 48.8 mm

2.3. Air Foil Bearing

2.3.1. Operating principle

Figure 2 shows the operating principle of a typical air bearing with a cylindrical air bearing mechanism. As shown in the figure, the shaft's eccentric edging on the bearing produces tapered gaps. As the shaft rotates, dynamic pressure is generated by the air flow which is pulled in as working fluid to the tapered gaps and causes the shaft to be suspended. However, as dynamic pressure increases due to higher engine speed, the taper position in the bearing moves in the rotating direction. This creates a misalignment between the direction of the shaft weight and the direction of the load generated by dynamic pressure. As the misalignment increases, it becomes impossible to maintain the shaft's suspension. The resultant self-induced vibration, referred to as Whirl Instability, limits the operating range. In order to maintain the tapered gaps, the inner bearing ring must have compliance.

2.3.2. Layout and function

To achieve the required level of compliance, this research has adopted a layout design based on bump foil as shown in Fig. 3. This layout adopts waveform bump foil made of a thin plate of high-strength Ni-Cr alloy (AMS5542) and a cylindrically shaped top foil that is laminated to the bearing surface. The bearing stiffness was set in accordance to the

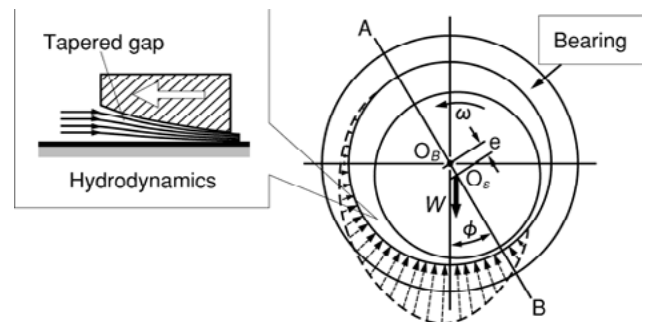
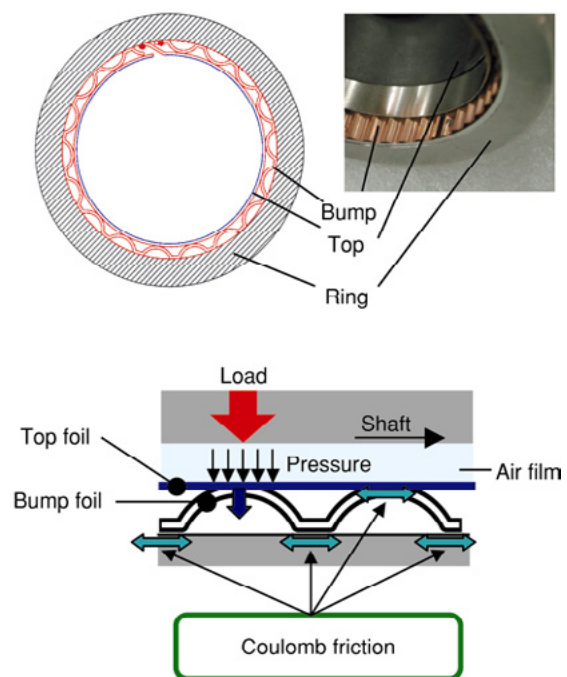
Fig. 2 Mechanism of air bearing⁽³⁾

Fig. 3 Compliant foil bearing

bump foil and air film stiffness. The damping performance was set in accordance to the sliding friction (coulomb friction)⁽⁴⁾ between the foils. The compliance standard was determined by the foil deformation according to the operating conditions.

2.3.3. Technical issues

As previously stated, the air foil bearing structure has a remarkably low level of shaft support rigidity compared to conventional rolling bearing configurations. Thus, the 1st bending critical speed exists within the operating range. To pass beyond this critical speed, the bearing must be able to accommodate an extremely high level of damping performance, which is also needed to avoid self-induced vibrations that occur at high speed ranges (refer to section 2.3.1). Additionally, in shaft speed ranges from zero to low speed levels (start-up or shutdown), measures must be employed to reduce friction and wear where contact with the shaft cannot be avoided since the formation of air film does not occur.

To suppress the shaft excitation force as the critical speed is being passed, the 1st bending critical speed was set to as low speed as possible within the operating range. This required that the 2nd bending critical speed should be set to levels that exceed the operating range.

3. Air Foil Bearing Structure

3.1. Damping Performance

Damping for the air foil bearing occurs because of the relative sliding friction between foils as shown in Fig. 3. A vibration test was conducted to verify specific areas where damping occurs, to evaluate the effect of the coating on the damping performance and the influence of the overall bearing structure. An excitation apparatus was used to excite a semicircle bearing sample and the damping performance was estimated from the resultant response. Figure 4 shows an overview of the vibration test rig.

3.1.1. Coating effect and identification of damping occurrence area

A coating was applied to the foils to increase the sliding friction and to improve the damping performance between the foils. A Cu coating was selected for its high friction coefficient characteristic, low cost and durability. Figure 5 shows the effects of the coating and the results of identifying the areas where damping occurred.

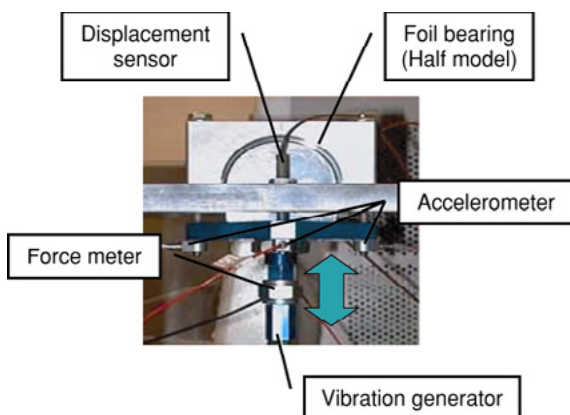


Fig. 4 Vibration test rig

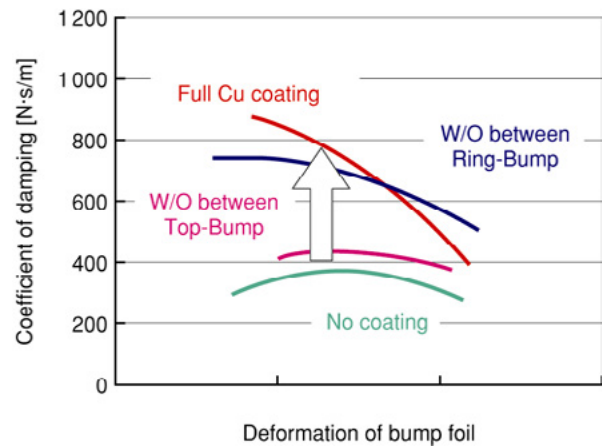


Fig. 5 Effect of Cu coating on damping coefficient

It is clear from this figure that the application of the Cu coating increased the sliding friction, thereby improving damping performance. It was found that when the Cu coating was removed between the bump foil and top foil as compared to the application of Cu coating to all contact surfaces of the foil, damping performance deteriorated to a level with no coating at all. On the contrary, when the coating was removed between the ring and bump foil, no difference in damping performance was observed. This indicates that damping is caused largely from the friction between the top foil and bump foil as the top foil moves. These results were used in support of applying Cu coating to the top and bump foils.

3.1.2. Effects of bearing structure to damping performance

3.1.2.1. Bump foil shape, thickness and separation

Figures 6, 7 and 8 show respectively the effects of the bump foil shape, thickness and separation on damping performance.

Figure 6 shows a comparison of two different bump foil shapes in which almost no significant difference in the damping coefficient was observed. Here, Shape A bump foil was selected because its high damping performance and surface contacts would restrain the top foil from falling between the bump foils when pressurized. This is thought to be effective in forming the tapered shape.

From Fig. 7, a bump foil that is the thickness of 4 mils was selected. Although there was almost no variation in the maximum damping coefficient, the bump foil thickness of 4 mils tended to alleviate the reduction in damping performance in areas where significant bump foil deformations appeared. On the other hand, a bump foil that is the thickness of 3 mils with its reduced level of foil strength had a low load limit.

Figure 8 shows the effect of bump foil separation on damping. A bump foil separation of 13 mm over 2 layers worked best in terms of the maximum damping coefficient. Separating the bump foil reduced the load level to facilitate functional performance. However, there was a remarkable drop in performance in areas with large foil deformation, which caused vibration as critical speed level was being passed. Narrowing the foil width reduced the allowable load limit and caused plastic deformation to the foil, which was of concern. For these reasons it was decided not to employ foils without separation.

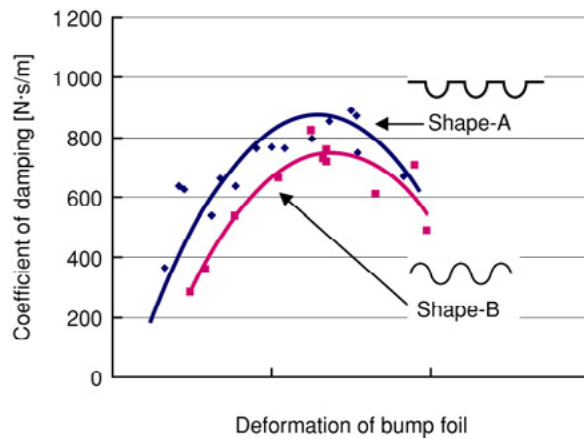


Fig. 6 Effect of bump foil shape on damping coefficient

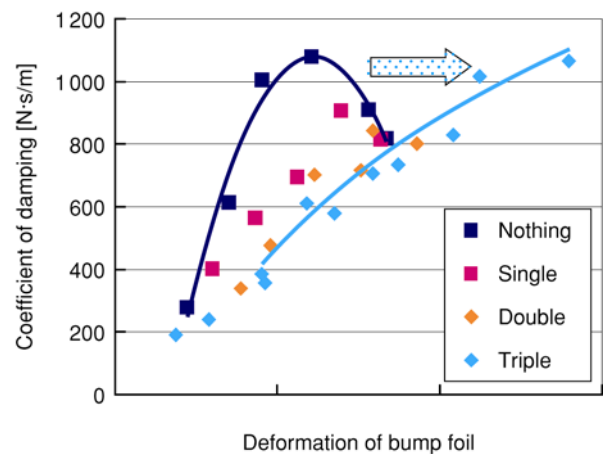


Fig. 9 Effect of multiplex middle foil on damping coefficient

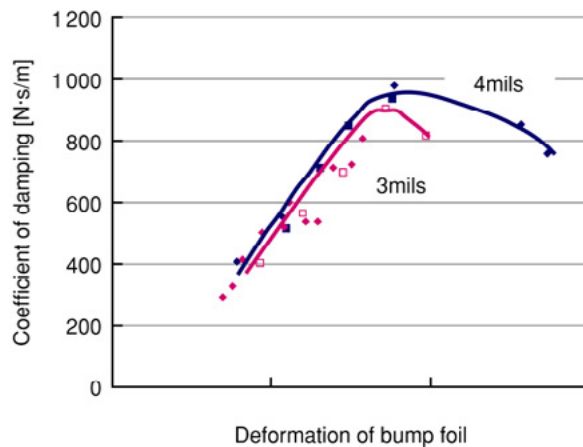


Fig. 7 Effect of bump foil thickness on damping coefficient

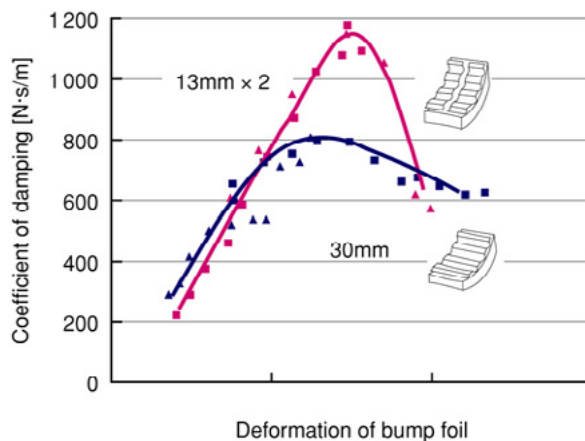


Fig. 8 Effect of bump foil separation on damping coefficient

3.1.2.2. Adding a middle foil

Damping between the top foil and bump foil is a result of movement of the top foil. By adding a middle foil between the foils, an improvement in the damping performance was achieved. Figure 9 shows the changes in the damping performance according to the number of middle foils applied. Increasing the number of foils created a shift of the peak damping coefficient points to the areas where the large foil deformation appeared. Where three middle foils were

applied, the damping performance tended to moderately improve along with the foil deformation. In other cases, the damping performance peak was exceeded which led to a drastic reduction in performance. This has shown that the application of three middle foils promotes high damping performance where the greatest vibration occurs as the critical speed level is being passed. Therefore, the three middle foils specification was adopted.

3.2. Selecting a Top Foil Rubbing Coating

In selecting a top foil coating to resist rubbing, low friction and wear resistance were focused on.

Figure 10 shows the results of the rubbing test conducted on coatings. In the test, a rotating shaft was rubbed up against the foil on which a load was exerted and changes to the wear coefficient were measured. The instances where fluctuation was recorded were then used in determining the wear limit. A period of 20 000 seconds (10 seconds $\times$ 2 000 cycles) was set as the standard period of wear resistance for the allowable period of continuous rubbing action. A hard coating of DLC (Diamond like carbon) was applied to the shaft and PTFE (fluorocarbon resin) was applied to the foil since this combination satisfied the standard and demonstrated the lowest wear coefficient.

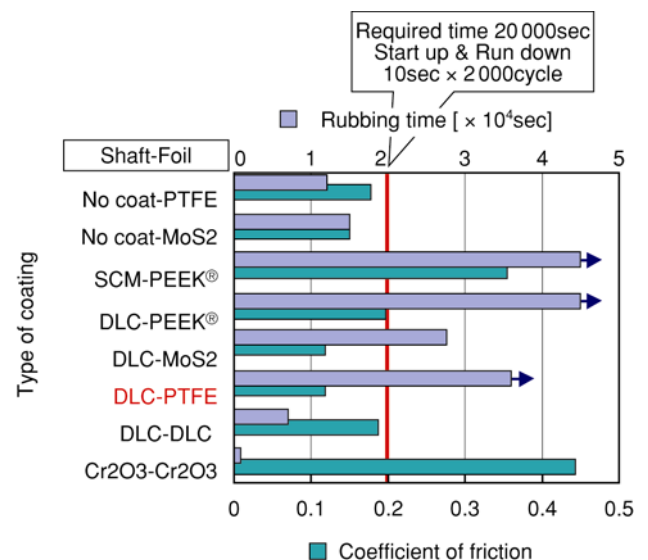


Fig. 10 Results of coating rubbing test

3.3. Air Foil Bearing Specifications

Table 3 and Figs. 11 and 12 provide the air foil bearing specifications and layout design determined from the above test results. The positioning of the shaft support for the air foil bearing provides an extremely efficient level of damping and achieves maximum sliding between the foils. More specifically, the air foil bearings were positioned at the edge of the shaft and the section corresponding to the loop of the 1st shaft bending mode where maximum amplitude is observed (See Fig. 13 for details).

Table 3 Specifications of air foil bearing

Items	Specifications
Bearing inner diameter	$\phi 48.8$
Bearing width	30 mm
Top foil	Thickness 4 mils PTFE & Cu coating
Middle foil	Thickness 3 mils 3 pieces
Bump foil	Thickness 4 mils Cu coating 7 pieces
Foil material	AMS 5542

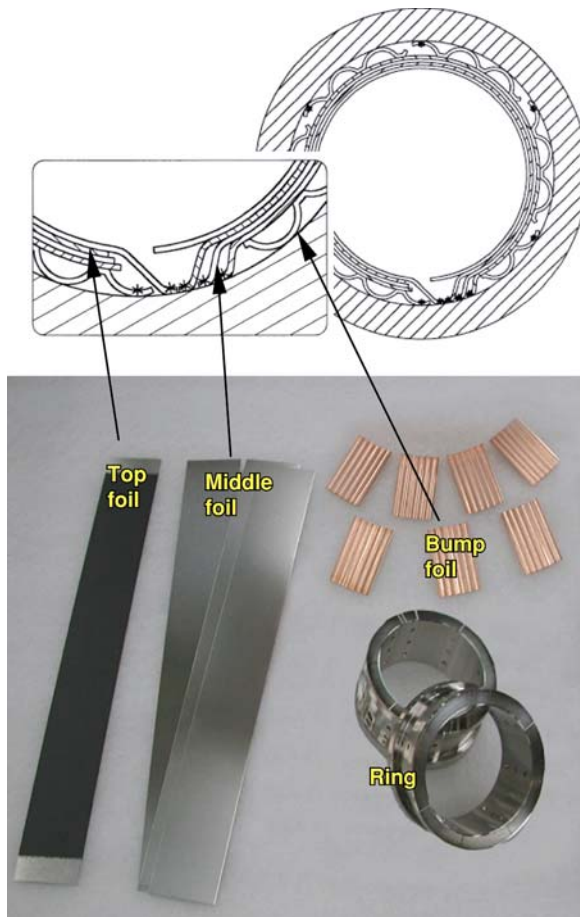


Fig. 11 Air foil bearing (Component)



Fig. 12 Air foil bearing (Cut model)

4. Shaft Structure

4.1. Shaft Structure Applied to Air Foil Bearing

Figure 13 shows the layout of the shaft to which the air foil bearing has been applied. In accordance with the conducted FEM analysis, the engine shaft with air foil bearings was configured with a uniform shaft diameter and an extended shaft length, in contrast to the engine shaft employing rolling bearings. The shaft weight was set to a weighty 9.2 kg. This configuration succeeded in reducing the critical speed to a minimum. Also, since the engine shaft was constructed so that multiple components, including permanent magnet, compressor rotor, and turbine rotor, were firmly tightened by tie shaft tension, a reduction in the stiffness of contact sections between shaft components and the effects of bending caused when the 1st bending critical speed is exceeded had to be taken into consideration. In addressing these issues, an experimental modal analysis was used to determine the optimal shaft layout. Figure 14 shows the critical speed map and the results of this analysis from which a design capable of accommodating the targeted values for each of the critical speeds was achieved.

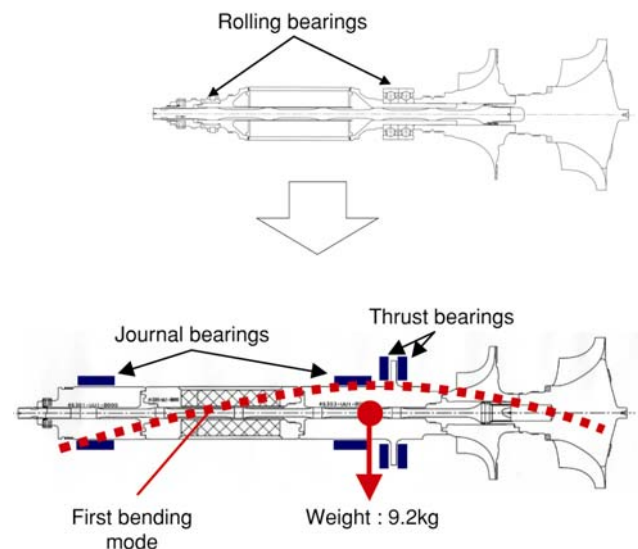


Fig. 13 Multi-component shaft for air foil bearing

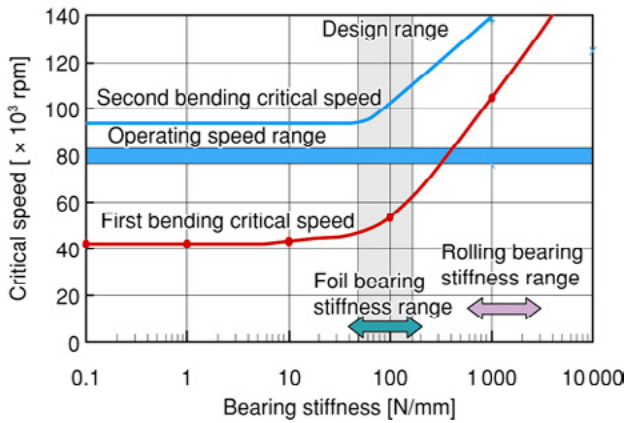


Fig. 14 Critical speed map

5. Unit Testing of Shaft System

5.1. Shaft System Unit Test Stand

A shaft system unit test was conducted to evaluate the air foil bearing performance and the configuration of the shaft itself. Figure 15 shows the shaft system test stand. The shaft is driven by the air turbine mounted on the right end of the shaft. The thrust bearing in the center of the shaft restricts the shaft movement to the shaft direction. During operation, displacement sensors mounted to the shaft components detect shaft displacement, and a thermocouple measures the air foil bearing outer ring temperature, foil temperature and thrust bearing temperature. The load cell mounted on the bearing stand was used to measure the friction torque of the air foil bearing.

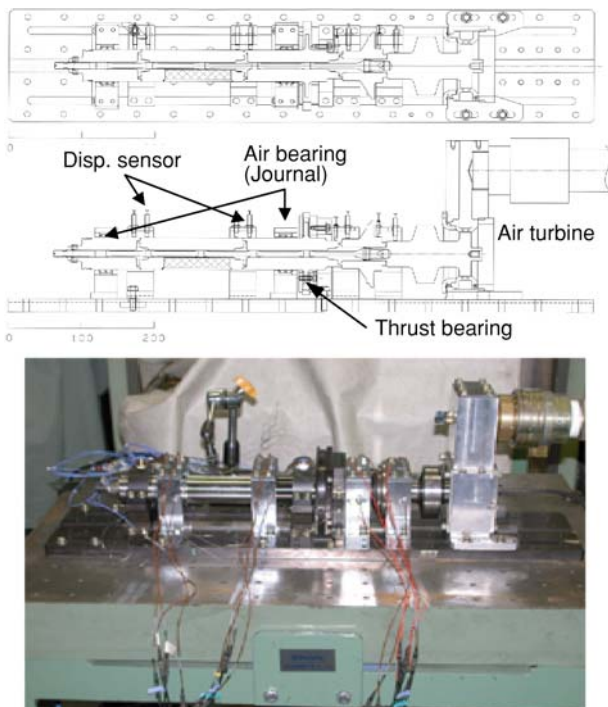


Fig. 15 Shaft system test stand

5.2. Unit Test Results

5.2.1. Speed test results

Figure 16 shows the results of the shaft system test. The graph in the upper part of Fig. 16 shows tracking data which indicates that by improving the damping performance on the air foil bearing, the 1st bending critical speed of the shaft was passed at around 42 500 rpm with a low amplitude level, and a rated speed of 77 000 rpm was attained. The contour diagram shown in the bottom of Fig. 16 indicates that the level of whirl instability induced by the frequency of the shaft's rigid body mode at rated speed was low, and a stable shaft rotation was achieved. The test results shown in Fig. 17 provide the damping ratio required for surpassing the 1st bending critical speed (the damping coefficient being represented by a dimensionless value). Verifying the dynamic characteristics of shaft (including shaft 1st bending critical speed and damping ratio) at speeds lower than the 1st bending critical speed level enabled estimation of the tested bearing performance and amplitude for surpassing the 1st bending critical speed.

Figure 18 shows the effect of the air bearing ring cylindricity and the fitting clearance with the shaft in respect to whirl instability. In order to reduce the level of whirl instability (indicated in the figure by W. I. Amplitude) to less than 4 μm , a cylindricity of less than 65 μm was set as the targeted bearing dimensions (indicated in the blue dots). The fitting clearance was set to $-25 \sim 0 \mu\text{m}$ after taking the startup torque into consideration.

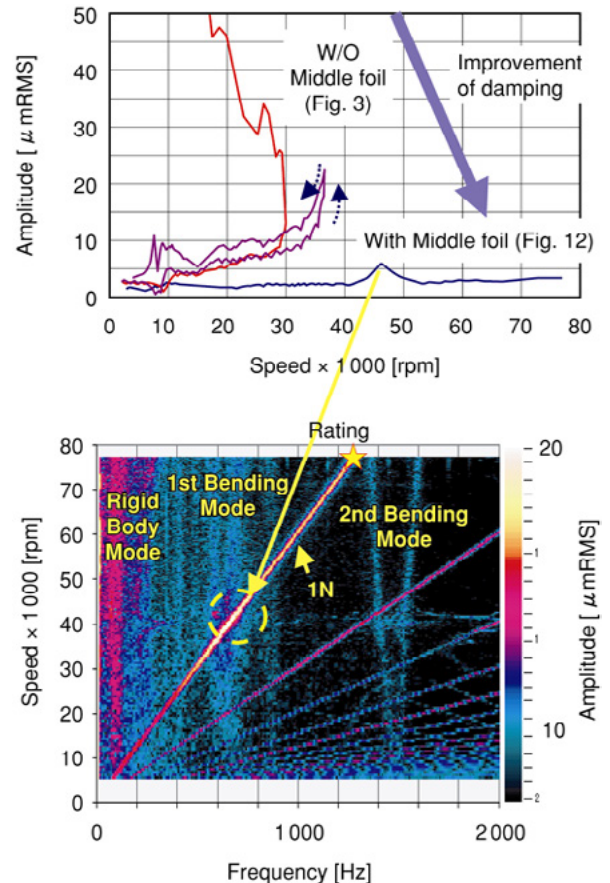


Fig. 16 Shaft system test results

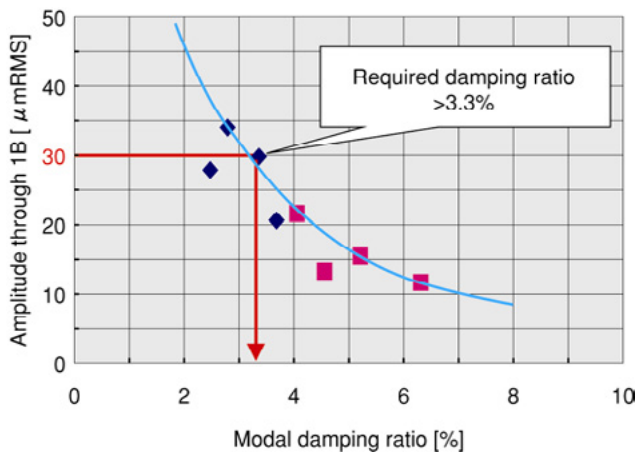


Fig. 17 Required damping ratio

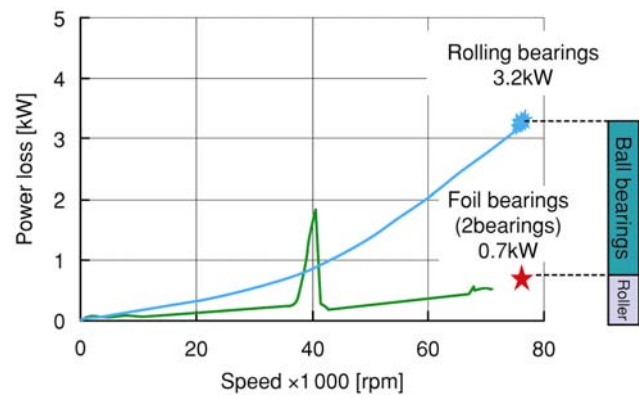


Fig. 19 Comparison of friction for foil and rolling bearings

6. Application to Micro-Gasturbines

6.1. Application to Micro-Gasturbines

In the shaft system test, it has been verified that the shaft's 1st bending critical speed was passed and a stable performance was achieved at high speeds. This has allowed the air foil bearing to be applied to a micro-gasturbine. Figure 20 shows an example of the micro-gasturbine layout.

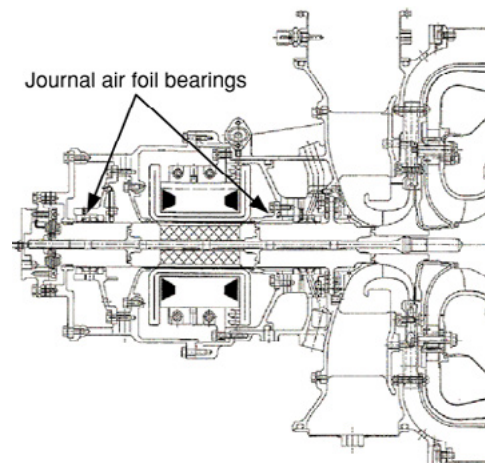


Fig. 20 Micro-gasturbine layout with air foil bearing

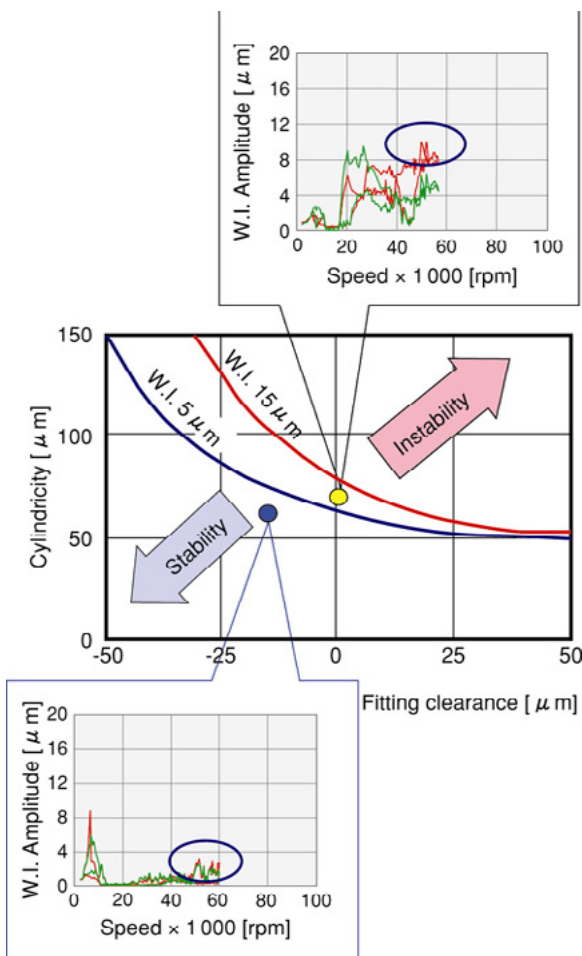


Fig. 18 Whirl Instability toughness

5.2.2. Friction measurement results

Figure 19 shows the friction measurement test results where the bearing loss on the air foil bearings is 0.7 kW at a rated speed of 77 000 rpm. Correlating this loss to a loss of one roller bearing of the rolling bearing specifications has shown that friction can be reduced by employing an air foil bearing. Friction increases when the 1st bending critical speed is passed because the shaft shakes and the air film diminishes. However, since the 1st bending critical speed is passed in an instant and is out of the normal speed range, it has been judged not to present any concern.

6.2. Operating Results of Micro-Gasturbine

Figure 21 shows the measurement results of shaft vibration on the micro-gasturbine. The amplitude where the 1st bending critical speed is surpassed is less than 15 μm and the amplitude at the rated speed is less than 5 μm, indicating that the targeted values have been achieved. In applying an air foil bearing to a micro-gasturbine, there were concerns with the effects to the air bearing performance because of the exciting force brought on by load fluctuation when the permanent magnetic generator is in operation, the exciting force by the aerodynamic element and the overriding effects of heat from high temperature engine parts. Although the test results have shown that the effects of the magnetic force during magnetic generator operation created differences in the surpassing amplitude at speed increases and decreases, this was shown not to be a significant concern.

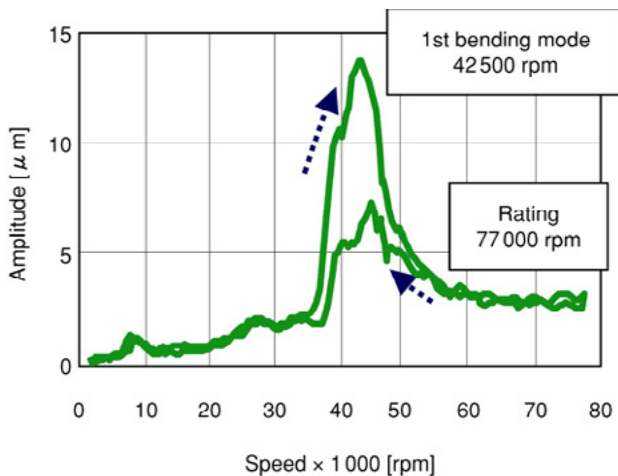


Fig. 21 Shaft vibration of micro-gasturbine

7. Conclusion

In improving the level of maintenance required for the micro-gasturbine, testing was conducted on the shaft system and the air foil bearing from which the following results were achieved.

- (1) In the testing of the shaft system in which an air foil bearing of unique configuration was used, it was verified that the 1st bending critical speed that occurs within the operating range of a 9.2 kg shaft with supported overhang was surpassed at a low amplitude with a moderate μm measurement, and stable driving performance was possible at high speeds.
- (2) An air foil bearing was fitted to a micro gasturbine engine and a similar performance level to the results of the test was reproduced. This configuration has been successfully verified for adoption in an actual engine.

Acknowledgements

In carrying out this research, the authors would like to extend their deepest appreciation to the individuals, both internal and external, for their invaluable support. Particular acknowledgement is extended to those in the technical testing section for their overwhelming assistance and support in fabricating air bearings.

References

- (1) Shinmura, K., Yamaguchi, K., Kimura, H., Tsuzuki, S., Kato, T.: Research of Micro-gasturbine Generator, Honda R&D Technical Review, Vol. 15, No. 2, p. 7-16 (2003)
- (2) Shinmura, K.: Micro-gasturbine Research at HONDA, Presented in 2003 ASME TURBO EXPO panel session (2003)
- (3) Togo, S.: Gas Bearings design and manufacture, Kyoritsu Shuppan Co., Ltd., P. 195 (1984)
- (4) Yamamoto, T., Ishida, Y.: Linear and Nonlinear Rotordynamics, Modern Treatment and Application, Koronasha (2001)

■ Authors ■



Norio SHIMIZU



Mitsuo MATSUSHIMA



Ichiro ISHII



Hitoshi IWADATE



Hidehiko NAKATA