Abstract
A split-type herringbone-grooved journal gas bearing has been developed to support the high-frequency rotations of a practical rotor for a micro machine such as a palmtop gas turbine generator currently in development. For this particular application, the critical mass of the rotor and the operable clearance of the journal and thrust bearings are designed to rotate at over 14 kHz (840,000 rpm). The test journal bearing and rotor have a radial clearance of 0.008 mm and a mass of 0.0042 kg. The roundnesses of the test journal bearing under the processed and reassembled conditions were measured using a 3-D coordinate measuring machine and compared to verify the repeatability of the bearing assembly; the difference in the measured roundness between each condition was about 0.003 mm, which includes axial assembly error. A rotational test of the rotor supported by the bearings was performed on a test rig. In this bearing, the maximum rotational frequency of the rotor reached 13.4 kHz (804,000 rpm). Therefore, the operational stability of the system was verified.

Key words: Hydrodynamic Gas Bearing, Roundness, High-Speed Rotation, Rotor Vibration, Power Source

1. Introduction
Small micro gas turbines with rotor diameters between 4 and 6 mm have been developed for small portable power generators. In order to achieve high power and be used in a self-contained robot, electric cart, etc., the turbine rotor and compressor impellers need to rotate at several hundred thousand rpm. Rolling contact bearings are not suitable for such machines since the rotating speed exceeds their maximum allowable rotating speed. Therefore, the use of gas bearings is necessary for these applications. So far, a hydroinertia journal gas bearing has been developed by Tanaka et al. to support small diameter rotors. They achieved 770,000 rpm using a 4 mm diameter rotor supported by hydroinertia journal gas bearings, resulting in a DN value of 3,080,000 (the DN value is expressed as the product of the rotor diameter (in mm) and the rotational speed (in rpm)). However, this type of bearing is not practical since it requires a gas supply from an external source. For these reasons, hydrodynamic gas bearings have been adopted since they do not require an external gas supply source.

Various types of hydrodynamic gas bearings have been studied for their application in small rotary machines. Mohawk Innovative Technology, Inc. studied hydrodynamic foil gas bearings to support the journal parts of a rotor with a 6 mm diameter and achieved 700,000 rpm (a DN value of 4,200,000). Isomura et al. studied herringbone-grooved hydrodynamic
Fig. 1 Construction of a practical rotor

To solve the abovementioned problems and still achieve a higher rotational speed (a target DN value of 4,800,000), we developed a split-type herringbone-grooved journal gas bearing in this study. The repeatability of the bearing shape (i.e., the roundness and the high-speed rotation capability) is important in these types of bearings that are to be used in practical rotor-bearing systems. This paper presents a bearing design (including dimensions and construction), the measured results of the bearing roundness (in both the manufactured and reassembled conditions), and experimental verification using a test rig.

Nomenclature

- \( c_{rg} \): Groove depth of journal bearing
- \( C_r \): Radial clearance of journal bearing
- \( f \): Rotational frequency
- \( h \): Clearance of thrust bearing
- \( h_g \): Groove depth of thrust bearing
- \( L \): Bearing length
- \( m_r \): Supportable mass
- \( M_s \): Dimensionless stability parameter
- \( N \): Rotational speed
- \( P_a \): Ambient pressure
- \( R \): Radius of journal bearing
- \( R_0 \): Outer radius of thrust bearing
- \( R_i \): Inner radius of thrust bearing
- \( W_t \): Load capacity of thrust bearing
- \( W \): Nondimensional load capacity of thrust bearing
- \( \alpha \): Groove width ratio
- \( \beta \): Groove angle
- \( \pi \): Circumference ratio
- \( \omega \): Angular velocity
- \( \Lambda_r \): Journal bearing number
- \( \Lambda_t \): Thrust bearing number
- \( \mu \): Gas viscosity coefficient

2. Construction of bearing test rig

In this paper, hydrodynamic journal and thrust gas bearings were designed for the ultra-high-speed rotation of vertical rotors according to the abovementioned study. A schematic illustration of the manufactured test rig is shown in Fig. 2, and the main dimensions are listed in Table 1. The vertical rotor is supported by the hydrodynamic journal and thrust gas bearings. The rotor has a mass of 0.0042 kg and is made of Ti-6Al-4V. The journal gas bearing has...
a 6 mm diameter and two journal parts. The target rotational speed is set at 800,000 rpm. The thrust gas bearings are positioned at the top of the rotor and have an outer diameter of 12.5 mm. The journal and thrust gas bearings are herringbone-groove and spiral-groove types, respectively. The rotor is driven by an impulse turbine positioned at the lower end. In this study, the compressor impeller was not installed in order to verify the bearing’s maximum rotational speed.

3. Hydrodynamic gas bearings for ultra-high-speed rotation

The main dimensions of the hydrodynamic journal and thrust gas bearings are designed for operating at a target ultra-high rotational speed of 800,000 rpm. The procedures in this study are described as follows.

3.1. Herringbone-grooved hydrodynamic journal gas bearing

The stability of the bearing’s rotational speed range depends mainly on the rotor mass and radial clearance. The design was conceived using the procedure given by D. P. Fleming et al.\(^5\). The target bearing design conditions were set as follows (considering manufacturing accuracy): \(D = 6\) mm, \(L = 6\) mm, \(N = 800,000\) rpm, and \(C_r = 0.008\) mm. \(A_r\) is calculated using the following formula:

\[
A_r = \frac{6\mu\omega}{P_0} \left( \frac{R}{C_r} \right)^2
\]

For example, if a rotor operates at 800,000 rpm, \(A_r\) becomes 13.1 using the above dimensions. The value of \(M_s\) for each bearing is calculated using the following formula:

\[
M_s = \frac{m_0P_0}{2\mu^2L} \left( \frac{C_r}{R} \right)^5
\]
The stability range of the rotor mass based on rotational speed is obtained using the above formulas and the numerically calculated results provided by D. P. Fleming et al. The result is shown in Fig. 3, where the other dimensions are as follows: $\alpha = 0.6$, $c_{gr} = 0.022$ mm, and $\beta = 26$ degrees. The curve in the figure indicates the operational stability boundary, where values above the curve indicate unstable operation, and values below the curve indicate stable operation. The red line represents the supportable rotor mass at a rotational speed of 800,000 rpm. We can see that the rotor can be operated at over 800,000 rpm if the rotor mass $m_r$ supported by one bearing is less than 0.0024 kg. Therefore, the manufactured bearing can support a rotor mass of 0.0048 kg at 800,000 rpm since the bearing has two journal parts, as mentioned above. The bearing is made using these calculated results and assumed dimensions.

### 3.2. Spiral-grooved hydrodynamic thrust gas bearing

A spiral-grooved hydrodynamic thrust gas bearing was designed using the procedure given by Togo(6). The rotor is supported by two thrust plates to obtain high stiffness. Therefore, the loading-side bearing plate (upper side) must support the counter-loading force of the
opposite-side bearing in addition to the rotor load. The bearing dimensions are set as follows: $h = 0.020$ mm, $R_o = 6.25$ mm, and $R_i = 3.5$ mm. $\lambda_i$ is given by the following formula:

$$\lambda_i = \frac{3\mu\omega}{P_o} \left( \frac{R_o^2 - R_i^2}{h^2} \right)$$

(3)

For example, if the rotor operates at 800,000 rpm, $\lambda_i$ becomes 3.12. The load capacity of the bearing $W_t$ can be calculated using Fig. 4 and the following formula:

$$W_t = \bar{W}\pi R_o \left( R_o^2 - R_i^2 \right)$$

(4)

The other bearing dimensions needed to obtain these curves are as follows: $\alpha = 0.6$, $h_g = 0.026$ mm, and $\beta = 18$ degrees. In this study, the bearing radius ratio $R_i/R_o$ becomes 0.56, and $\bar{W}$ can be designed by interpolating the curves of Fig. 4. Let us assume that the loading-side clearance is 0.019 mm, and the counter-loading-side clearance is 0.021 mm. The difference between the load capacities of each bearing becomes 0.13 N. Therefore, the rotor can be supported at an approximately neutral position for a total bearing clearance of $2h$ since the rotor mass is small relative to the load capacity.

4. Split-Type Herringbone-Grooved Hydrodynamic Journal Gas Bearings

4.1. Construction and roundness of journal parts

A test journal bearing with split construction was made to solve the problems associated with practical small rotary machines. The outer view is shown in Fig. 5. The axially split bearing parts were assembled using taper pins and cap bolts to maintain the roundness of the journal parts, even during reassembly. The bearing has two of the journal parts shown in Fig. 2 (i.e., on the thrust bearing side and the drive turbine side). The bearing is made of self-lubricating copper alloy to prevent contact damage at high rotational speed.

The roundness was measured from the thrust bearing side (upper surface of Fig. 5) along the axial direction of the journal parts at 1 mm steps using a 3-D coordinate measuring.
The measured roundnesses of the two journal parts are shown in Fig. 6 and Fig. 7, respectively, and the values are listed in Table 2. In these figures, the x-axis represents the split surface. This is the result after being reassembled about ten times. The average roundness in the processed condition is 0.0038 mm, whereas it is 0.0061 mm in the reassembled condition. This variation seems to be large for the processed bearing’s radial clearance. However, in these figures, no significant misalignment occurred at the split surface. Therefore, it is thought that the bearing surface in the reassembled condition performs equally to that in the processed condition. As shown in Fig. 6 and Fig. 7, the roundness in the reassembled condition becomes oval in shape. It is thought that the variation caused by assembly error in the axial direction influenced the 3-D CMM measurements.

4.2. Test instrumentation

The test journal bearing was installed in the test rig as shown in Fig. 2. The test was carried out by driving the rotor with the impulse turbine placed at bottom of the rotor, as shown in Fig. 2. The turbine has eight nozzles (each with a diameter of 0.6 mm). The rotor vibration was measured using an eddy-current-type displacement meter (AEC Corporation Gap-Sensor PU-03A) installed in the journal bearing metal. The output was analyzed using an FFT analyzer. The output of the displacement meter was calibrated before the test using a rotor of the same diameter and materials mentioned in the former section.
4.3. Rotational characteristics of the rotor in the split bearings

Figure 8 shows a waterfall diagram of the measured rotor vibration frequency. The horizontal axis and the vertical axis represent the frequency and drive turbine supply pressure, respectively. The lines represent the measured results, in which the supply gas pressure of the drive turbine is 0.05 MPa to 0.775 MPa with a step size of 0.025 MPa. A sampling frequency of 51.2 kHz was used in this test to prevent aliasing. The rotation frequency reaches 13.4 kHz (804,000 rpm: a DN value of 4,824,000) at 0.775 MPa. Stable rotor operation is obtained in this test, though whirl vibration is seen in the low-frequency range. Figure 9 shows a part of the measured time history of the rotor vibration at 10.7 kHz, 12.4 kHz, and 13.4 kHz. The rotor amplitude only exceeds about 0.0002 mm, so a relatively small amplitude is obtained. These results should include the effects of rotor tolerance. The designed roundness tolerance is 0.001 mm at the measuring position. Considering this value, the amplitude seems to be too small. At this time, a confirmation of roundness could not be conducted since the rotor is too slender. It is thought that a conical mode of vibration occurred since the rotor has a thrust collar at one end and is driven at the other end. Since the displacement meter is located near
the center of the rotor, as shown in Fig. 2, it is thought that a small amplitude was detected at this position. Thus, the rotor might vibrate several μm at the edge. It was shown that the split bearing provides rotational stability at ultra-high-speed rotation. Therefore, this type of bearing would be very effective in the practical rotors of the micro gas turbines currently being developed.

5. Conclusion

In order to solve the problems associated with a practical rotor for micro rotary machines while achieving a higher rotational speed than the current gas bearings, a split-type herringbone-grooved journal gas bearing has been developed. The bearing was designed and manufactured, and the repeatability of the journal bearing roundness was measured. In addition, an ultra-high-speed rotation test was performed using a manufactured bearing. The obtained results are listed as follows:

(1) The manufactured split-type herringbone-grooved journal gas bearing has remarkable roundness repeatability under the reassembled condition.

(2) The rotor was operated at 804,000 rpm (a DN value of 4,824,000).

(3) Stable rotor operation was obtained in the test, though whirl vibration was seen in the low-frequency range.

Therefore, this type of bearing would be very effective in the practical rotors of the micro gas turbines currently being developed. We will aim at 1,000,000 rpm (a DN value of 6,000,000) as the next phase of research.

References


