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BASIC CHARACTERISTICS OF HYBRID GAS BEARING WITH WATER EVAPORATION FROM ULTRA-FINE POROUS MEDIUM

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Abstract. *A hybrid gas bearing with water evaporation from ultra-fine porous medium is newly proposed. The proposed bearing is considered to be applied to microturbines and have mainly three advantages suitable for the applications. The first is the stability improved by water evaporation from ultra-fine porous medium. The second is the effective lubrication by liquid water at start and stop of the journal rotation. The third is the cooling effect on the high-temperature journal due to water evaporation. Basic characteristics of the bearing are clarified on the basis of a numerical simulation for water vapor, and compared with the conventional bearings the improved performance is demonstrated. A preliminary experiment is conducted, and the stable start and stop of the journal rotation are confirmed.*

Keywords*. Gas bearing, Hybrid bearing, Water evaporation, Porous medium, Microturbine.*

1. Introduction

A bearing is one of the most crucial components of microturbines recently developed for distributed generation systems. This is because the operation condition is very tough due to very high rotation speed more than 100,000 rpm and a maintenance-free system is required as well. Therefore, a gas bearing (Czolczynski, 1999; Su and Lie, 2003), which has simple structure and very low friction, is considered to be ideal for the microturbines. Capstone, the pioneer of the microturbine, adopts a gas bearing for the most famous microturbine, Model 330. Gas bearings, however, have some demerits of poorer stiffness and stability than conventional oil bearings.

In this study we propose a novel gas bearing as shown in Fig.(1). In this bearing, water or water vapor discharged through ultra-fine porous medium are used as working fluid. The bearing surface is nearly hydraulically smooth, because the mean pore size of the porous medium considered in this study is about 1 μm. The bearing is enclosed by the water storage, and the water pressure is controlled to be higher than the atmospheric pressure. Therefore, at rest, the clearance is filled with liquid water. However, with increasing rotation speed of the journal, both viscous dissipation of water and heat transfer from the high-temperature journal to the bearing surface cause phase change from water to vapor. Since vapor evaporation from the bearing surface elevates local pressure in the clearance, stabilization effect due to "hydrostatic" pressure is superimposed on the conventional stabilization due to "hydrodynamic" pressure. Consequently, the proposed bearing is categorized into a "hybrid" bearing. Although a porous bearing is very popular as one of the hydrostatic bearings, the present bearing is largely different from the conventional ones in this point.

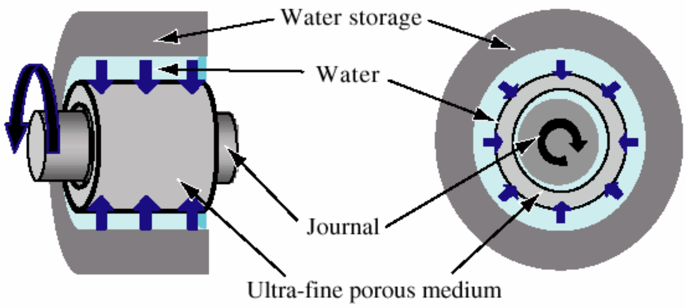

Figure 1. Hybrid bearing using ultra-fine porous medium.

One of the weakness of the hydrodynamic gas bearing is that the direct contact between the bearing and the journal occurs at rest or at very low rotational speed. The proposed bearing, however, is free from it because liquid water sustains the journal during those periods. Hence, the proposed bearing is also “hybrid” in terms of gas and liquid lubrications. Furthermore, in the application to microturbines in which the temperature of the journal exceeds 500 ^{o}C, the journal is effectively cooled by latent heat of evaporation.

In this paper, a numerical analysis and a preliminary experiment are carried out to clarify the basic characteristics of the proposed bearing system. As a base of the comparison, those of a plain gas bearing are employed.

2. Theoretical analysis of steady state

2.1. Governing equations

For the actual operations the working fluid is liquid at rest or at very low speed. In this chapter, however, the working fluid is assumed to be water vapor throughout the operation for simplicity. Therefore, for the compressible fluid in the clearance, the classical Reynolds equation based on the cylindrical coordinate (r, q, z) is expressed as follows:

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left(\rho h^3\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\rho h^3\frac{\partial p}{\partial z}\right)=12\eta\frac{\partial\rho h}{\partial t}+\frac{1}{r}6\eta U\frac{\partial\rho h}{\partial\theta}-12\eta\rho v \tag{1}$$

where h denotes local clearance, p pressure, U velocity of journal surface. The last term in the right-hand side of Eq. (1) denotes the hydrostatic stabilization due to the evaporation. Here, the velocity of the water vapor from the bearing surface v is given by the following equation:

$$v=\frac{\tau_0 U+q}{\rho L+\rho c_p(T_{sat}-T_{water})} \tag{2}$$

where the first term of the numerator denotes the heat due to the friction and the second the heat transfer from the high-temperature journal to the bearing surface.

On the other hand, the motion of the journal is horizontal coordinate x and vertical coordinate y:

$$mc\ddot{\varepsilon}_x = f_x \quad (3),\qquad mc\ddot{\varepsilon}_y = mg+f_y \quad (4)$$

where c is clearance, ε_x and ε_y eccentricities of the journal, m mass of journal, f_x and f_y forces exerted by working fluid.

The forces f_x and f_y are given by the following equation based on local pressure p obtained from Eq. (1):

$$f_x=-\int_0^L\int_0^{2\pi}p\cos\theta rd\theta dz\ , \quad (5)\qquad f_y=-\int_0^L\int_0^{2\pi}p\sin\theta rd\theta dz \quad (6)$$

Although in the actual case large temperature difference exists between the journal and bearing surfaces, we assume that the vapor temperature is equal to the local temperature of the bearing surface, and thus the temperature of the working fluid is not a function of r.

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2.2. Boundary and initial conditions

On the bearing surface, we assume that liquid water and vapor are in equilibrium state. Hence, temperature is specified by the saturation value corresponding to pressure.

At the edge of the bearing surface the boundary condition for pressure was given by

$$p(z=0)=p(z=l_b)=p_0(\text{atmo. press.}) \tag{7}$$

where l_b denotes bearing length.
As the initial condition, the eccentricity of the journal was fixed at

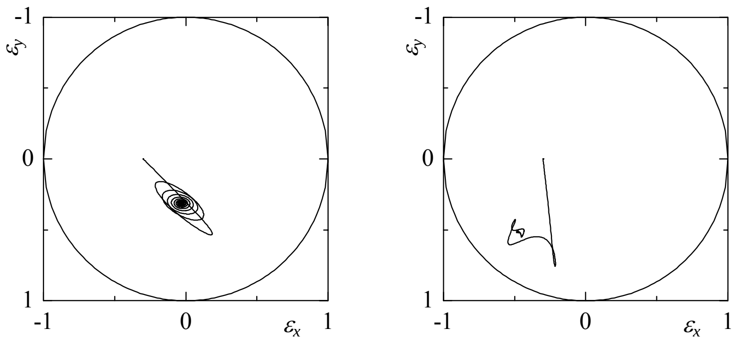
$$\varepsilon_x=-0.3,\qquad \varepsilon_y=0 \tag{8}$$

Hereafter, the eccentricity is expressed in dimensionless value based on clearance c.

The rotation speed was given by a step function with constant. Although treatment is not realistic, this corresponds to the severest case for stability. The length of the bearing and the clearance is 15 mm and 10 μm, respectively. The diameter and mass of the journal is 15 mm and 1 kg, respectively. The temperature of the journal surface is fixed at 100 ^{o}C corresponding to the atmospheric pressure.

2.3. Results and discussion

To obtain the fundamental characteristics of the bearing, we show the results for *Ro* = 10,000 rpm. Figure (2) compares the trajectory of the journal center. Here, "plain bearing" means the conventional gas bearing, the viscosity of which is the same with the vapor. As explained above, the dimensionless start point locates at ε_x = -0.03 and ε_y = 0. The final eccentricity corresponding to a steady-state operation for the hybrid bearing is smaller than that for the plain bearing. Therefore, we can confirm the higher load carrying capacity of the hybrid bearing.



(a) Hybrid bearing (b) Plain bearing

Figure 2. Trajectory of journal center (10,000rpm).

Figure (3) shows the pressure distribution on the bearing surface. Here, the vertical direction is axial, the horizontal direction circumferential; if we assume clockwise rotation of the journal, θ = 0 or 360 deg corresponds to three o'clock, θ = 90 deg six o'clock, θ = 180 deg nine o'clock, θ = 270 deg twelve o'clock. Owing to the axial flow of vapor generated in the axially central area, pressure at inside area is higher than that at the edge. Furthermore, as expected from Fig. (2), the highest pressure is found at the center position along θ = 90 deg where the local clearance takes minimum value. Although the temperature distribution on the bearing surface is not shown, it is qualitatively similar to the pressure distribution because of the equilibrium assumption.

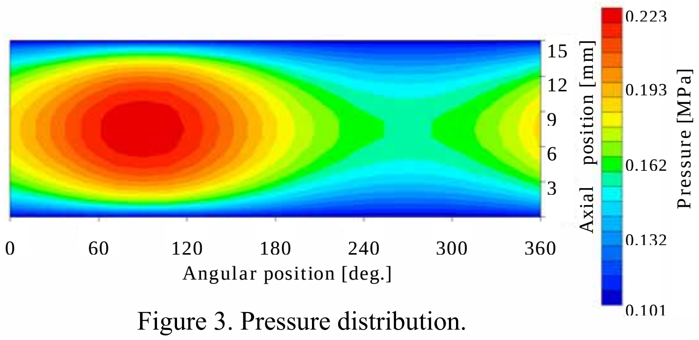


Figure 3. Pressure distribution.

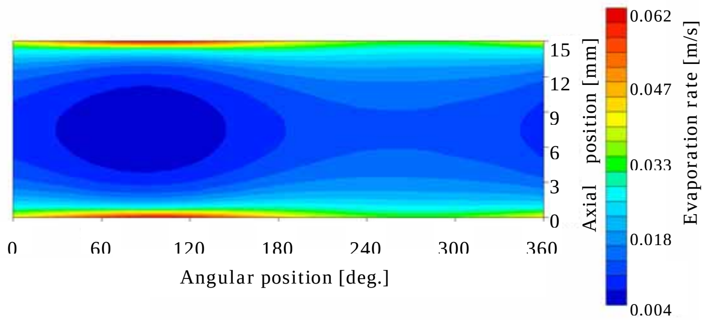


Figure 4. Evaporation rate distribution.

Figure (4) shows the distribution of the evaporation rate *v* given by Eq. (2). Since the temperature of the journal surface is assumed to be uniform, the temperature difference between the journal and bearing surfaces decreases with increasing temperature on the bearing surface. Furthermore, in Eq. (2) the heat flux from the journal to the bearing surface *q* dominates the viscous dissipation. Consequently, the distribution of the evaporation rate is in contrast to that of pressure distribution.

Figure (5) shows the trajectory of the journal center at *Ro* =30,000 rpm. For the hybrid bearing the final location of the journal center is ε_x = -0.0 and ε_y = -0.2, whereas for the plain bearing unstable behavior occurs and the eccentricity

$$\varepsilon = \sqrt{\varepsilon_x^2 + \varepsilon_y^2} \tag{9}$$

finally reaches unity, which means the journal collides with the bearing surface.

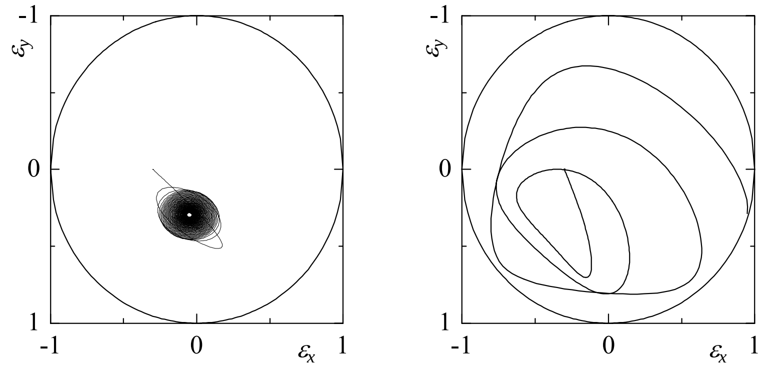


Figure 5. Trajectory of journal center (30,000rpm).

A bearing system has its maximum rotational speed, below which it can sustain the journal stably. Therefore, it is important to know the threshold above which the bearing system goes unstable. In order to make the analysis more general, the following dimensionless parameters are introduced [1].

Bearing Number: $$\Lambda = \frac{6\eta\omega}{\left(\frac{c}{r}\right)^2 p_0} \tag{10}$$

Mass Parameter: $$\overline{M} = \frac{mp_0}{72l\eta^2}\left(\frac{c}{r}\right)^5 \tag{11}$$

Although the bearing number represents the hydrodynamic effect, for a fixed geometry, it is regarded as the dimensionless form of the rotational speed. On the other hand, the mass parameter is regarded as the dimensionless form of the mass of the journal.

Figure (6) depicts the stability boundary against the mass parameter for the various journal temperatures. The stability boundary increases almost linearly as the mass parameter increases. In an ordinary hydrodynamic bearing operation, however, it is known that the stability decreases with increasing mass parameter. Therefore, it is considered that the proposed bearing system is more strongly based on the hydrostatic effect rather than the hydrodynamic effect. Furthermore, as can be seen from Fig.(6), the proposed bearing system is more stable when the temperature of the journal is high. As indicated in Eq. (2), this is because the increase in heat flux *q* corresponding to high temperature enhances the water evaporation and makes hydrostatic effect greater. This characteristic is very favorable for an actual microturbine operation where the journal temperature exceeds 500 ^{o}C.

From the present numerical analysis assuming that the clearance is fully filled with water vapor, it is confirmed that the hydrostatic contribution due to the water evaporation is very effective.

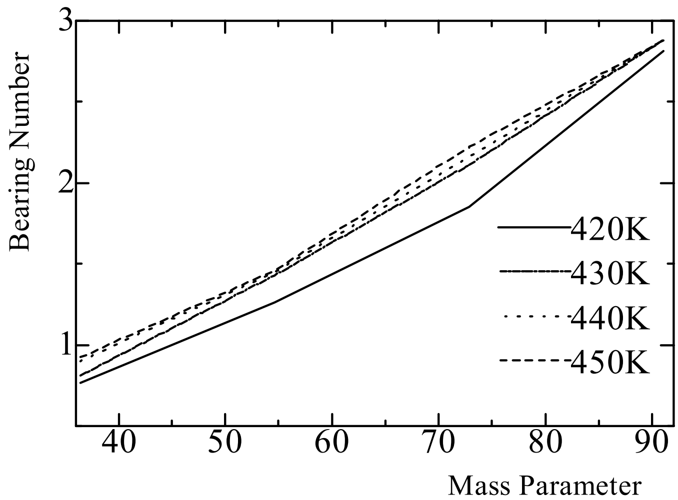


Figure 6. Stability boundary for $T_{journal}$ = 420, 430, 440 and 450 K.

3. Experiment

3.1. Experimental apparatus

A preliminary experiment was conducted to check the applicability of the ultra-fine porous medium to a bearing material, and also to confirm the stable start and stop of the bearing system. Figures 7 and 8 show a schematic and a photograph of the experimental apparatus. To eliminate the external vibration as much as possible, the journal was designed to be driven by an air turbine located at the center of the two bearings. Figure 9 shows the photograph of the air turbine attached on the journal. The journal was made of aluminum; its diameter, length and weight are 16 mm, 379 mm and 206 g, respectively. The distance between the two bearings is 165 mm. At the both ends of the journal two thrust bearing were installed. Two burners located near the thrust bearings were used to heat the journal to enhance the evaporation at the water surface inside the bearings.

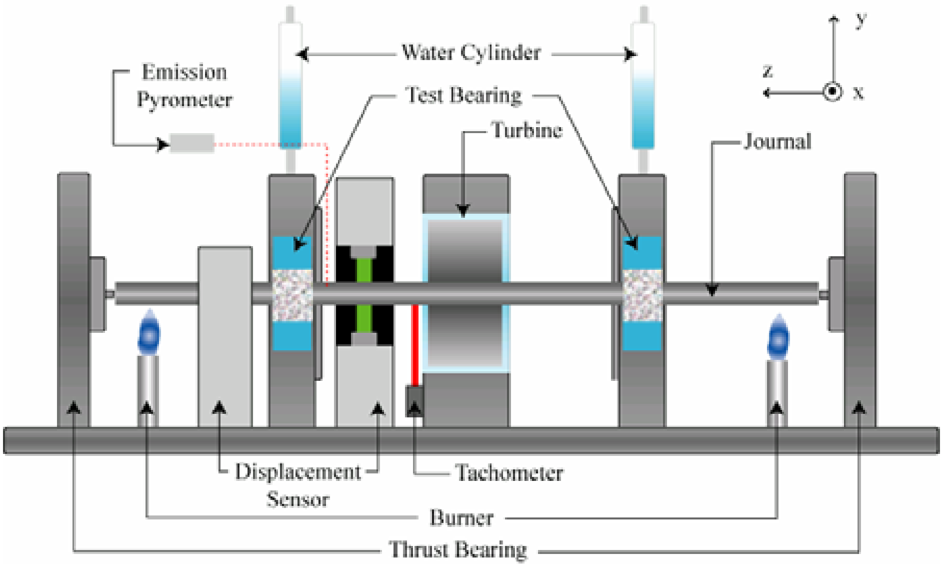


Figure 7. Schematic of experimental apparatus.

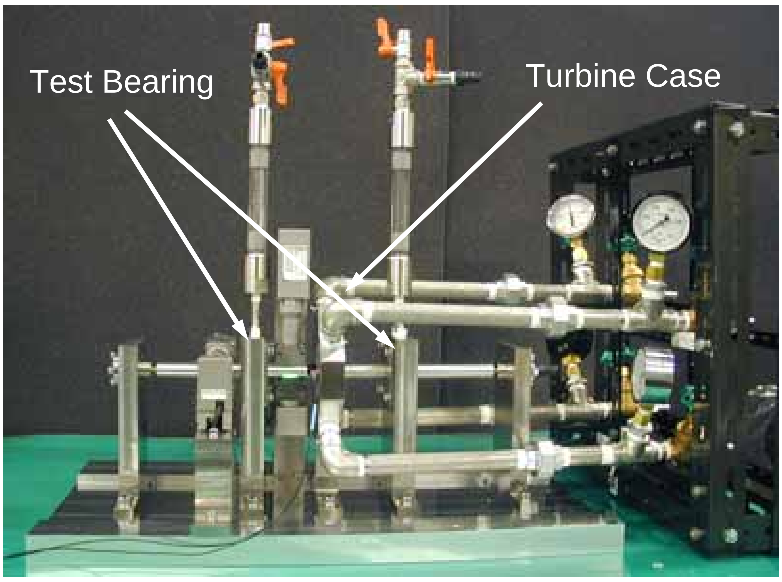


Figure 8. Photograph of experimental apparatus.

Figure 10 shows a photograph of the ultra-fine porous medium which were supplied by Nippon Seisen Co., Ltd. (http://www.n-seisen.co.jp/). The porous medium are manufactured using ultra-thin stainless-steel fiber of 2-5 μm in diameter (trademark name: NASLON); the mean pore size is about 1 μm and the porosity is 35 %. Although this ultra-fine porous medium looks like purely solid material, one can easily confirm the capillary effect if it is partly submerged in water. The inner diameter and the axial length of the bearing was 16 mm and 25 mm, respectively. The clearance varies depending on the journal temperature owing to thermal expansion of the journal material. When both the journal and the bearings are maintained at room temperature, the clearance was about 40 μm. For operations with evaporation, however, the journal temperature was controlled by the two burners, and the clearance was adjusted less than 20 μm. To fill the clearance with water, the water was externally pressurized in this experiment.

Figure 9. Air-turbine blade

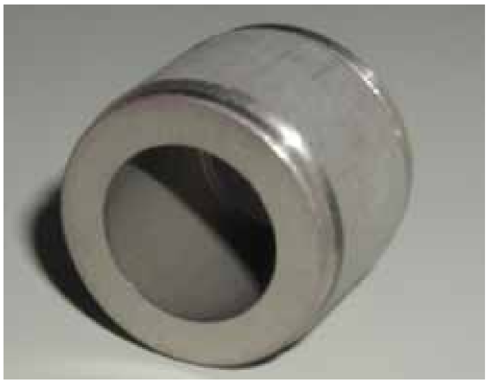


Figure 10. Photograph of ultra-fine porous medium.

3.1. Results and discussion

First, lubrication purely due to liquid water is discussed. Since the journal temperature is the same with that of the bearing, there is no possibility of evaporation on the bearing surface. Figure (11) shows transient characteristics at the starting period. In the left figure, the red line shows the number of the revolutions, while the blue line shows the eccentricity ratio. In the right figure, the trajectory of the journal center is depicted. Initially, the journal center locates at $\varepsilon_y = 0.75$, because the static pressure is applied by surface water. With increasing revolution number, moderate vibration of the journal center, the frequency of which is the same with the revolution number, occurs.

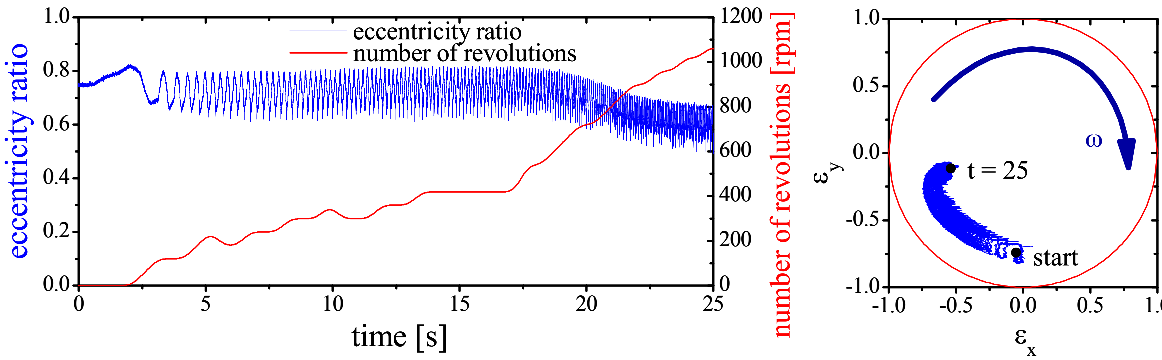


Figure 11. Transient characteristics at starting period.

From Fig. (12) for the constant revolution number of 4000 rpm, periodic small vibration is observed. Under this condition, hydrodynamic pressure associated with the journal rotation becomes more effective than hydrostatic pressure, and the eccentricity ratio decreases to about 0.2.

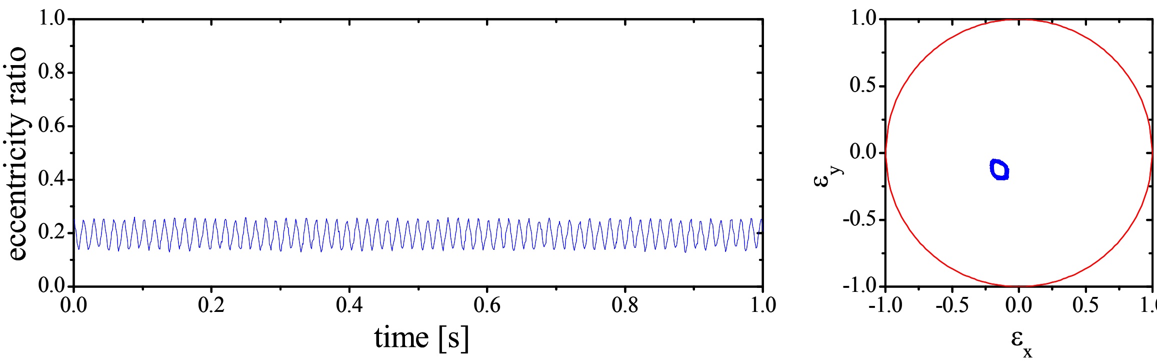


Figure 12. Characteristics at constant revolution number (4000 rpm).

Figure (13) shows the transient characteristics at the stopping period. The frequncy of the vibration of the journal center is always the same with that of the revolution number, and hence the observed response is considered to be nearly quasi steady.

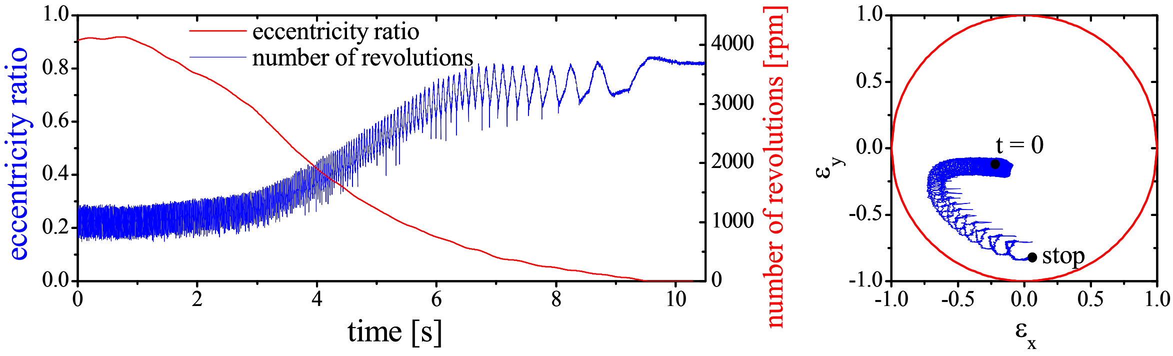


Figure 13. Transient characteristics at stopping-down period.

Although the final target of this study is the clarification of the performance at very high rotation number on the order of 10^5 rpm using a high-temperature journal, it is difficult to conduct measurements for such conditions at the present stage. This is mainly due to the difficulty in the precise control of the clearance in connection with the precise control of the journal temperature. Hence, in this study, the measurement for the vapor-lubricated case was restricted to the condition of relatively low temperature and low revolution number. Although the temperature of the journal surface was measured by an infrared thermometer, there exists large nonuniformity as well as measurement uncertainty. Consequently, the clearance was not accurately evaluated during the heating experiment. Hence, we assume that the time-averaged position of the journal center is the position of $\varepsilon_x = \varepsilon_y = 0$ μm. In Fig. (14), the eccentricity is depicted in dimensional form unlike Figs. 11-13. As shown in Fig. (14), the trajectory of the journal center irregularly deviates, and finally the journal contacts the bearing surface and stops.

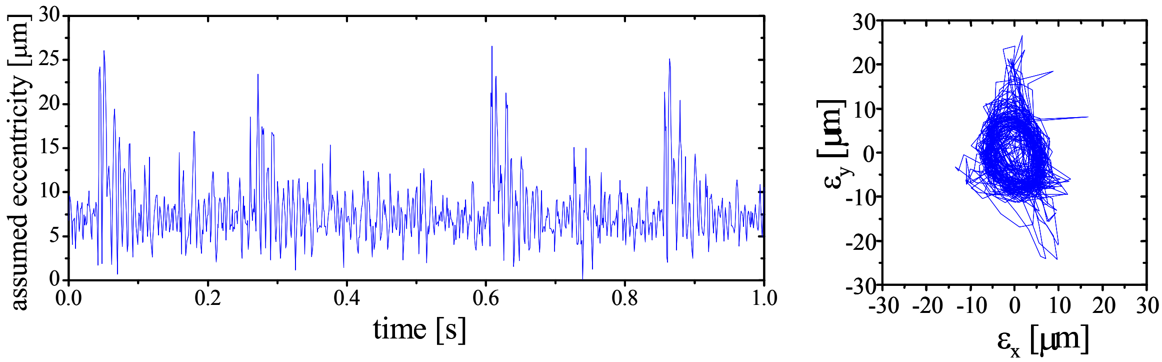


Figure 14. Characteristics under vapor-lubricated condition (5500 rpm).

5. Conclusions

A novel hybrid gas bearing was proposed and its basic characteristics were clarified theoretically and experimentally.

The theoretical analysis for the vapor-lubricated condition revealed that the proposed hybrid gas bearing is superior to conventional gas bearings in terms of load capacity, stiffness and stability. Concerning the transient state including phase changes from water to vapor or vice versa, a more sophisticated analysis is currently in progress.

Experimentally, the stable starting-up and stopping-down operation was confirmed under the water-lubricated condition. However, for the vapor-lubricated condition with a high-temperature journal and high revolution number, experiments have not yet been successful. The principal reason is considered to be the nonuniform temperature distribution in the journal and the resulting nonuniform clearance. Therefore, the temperature control for the journal as well as the bearing and water is the key for the improved experiment.

6. Acknowledgement

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