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Hydrostatic Asymmetric Journal Gas Barings for Seismic ACROSS Transmitters

-Improvement of Safety Operation by Supply Gas Pressure Control under the Rotational Frequency Modulation-*

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Abstract

The ACROSS (Accurately Controlled Routinely Operated Signal System) is used to detect subtle changes in the physical state of Earth's structure. The signal is sinusoidal elastic waves generated by the rotation of a rotor with eccentric mass set in ACROSS transmitters. In currently working transmitters, since the rotor is supported by rolling contact bearings, bearing losses are so large that the cooling, lubrication management and bearing life become serious problems. To solve these problems, hydrostatic asymmetric journal gas bearings have been proposed to support the eccentric mass rotors. This paper describes the experimental investigation of rotational characteristics of proposed hydrostatic asymmetric journal gas bearings. It is verified that the rotor amplitude in the journal bearings can be zero at a certain rotational frequency by using trial test rig. In order to decrease the gas consumption and increase more safely operation, test of rotational frequency modulation using electro-pneumatic pressure regulator was carried out. In fixed supply pressure condition, rotor amplitude largely changes in accordance with change of rotational frequency. In controlled supply pressure condition, the rotor amplitude decrease under the operation and the gas consumption also largely decreased.

Key words: Hydrostatic Gas Bearing, Asymmetric Bearing, Rotor Amplitude, Gas Consumption, Large Eccentricity, Frequency Modulation, Seismic Source

1. Introduction

The ACROSS (Accurately Controlled Routinely Operated Signal System) is a new technical method to detect subtle changes in the physical state of Earth's structure using precise and continuous signal transmitter-receiver systems[(1)-(5)]. The signal is a type of sinusoidal elastic waves. The waves are generated by a centrifugal force caused by the rotation of an unbalanced rotor with large eccentric mass set in ACROSS transmitters. In

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currently working transmitters, the unbalanced rotors are supported by rolling contact bearings. A currently working transmitter generates centrifugal force of 2×10^5 N at rotational frequency of 25 Hz. An unbalanced rotor has magnitude of unbalance of 8.0 kg·m and is driven with frequency modulation within a range between 10.2537 Hz and 19.2537 Hz. With such type of bearings, frictional losses are so large that the cooling, lubrication management and bearing life become serious problems. To mitigate the problems, use of hydrostatic journal gas bearings to support largely unbalanced rotor is proposed.

In preliminary test with a trial ACROSS transmitter using hydrostatic gas bearings, it is shown that hydrostatic journal gas bearings support safely a largely unbalanced rotor. However, the load capacity of hydrostatic gas bearings is much smaller than rolling contact bearings, the bearing dimensions become large and as the result, large gas flow rate is required. To improve this demerit, a new type of hydrostatic gas bearing with asymmetric bearing surface is investigated[6]. With such type of bearings, rotor amplitude becomes zero at a certain rotational frequency, which changes depending on the bearing supply gas pressure. If supply gas pressure can be changed in accordance with the rotational frequency modulation, it is expect that the keeping of small rotor amplitude under the operation becomes possible. Moreover, it is expected that the gas consumption under the operation can be decreased.

This paper presents the experimental verification of the hydrostatic asymmetric journal gas bearings using 2000 N class trial ACROSS transmitter and that of applying electro-pneumatic pressure regulator.

2. Hydrostatic asymmetric journal gas bearings

2.1 Construction and working mechanism of the bearing

In previous study, conventional symmetric type journal gas bearings with double admission slot restrictor have been adopted[7] to support large centrifugal force because it has high load capacity in comparison with other restrictors such as inherent orifice type, however, gas consumption was extremely large. Then a new type of journal bearings with asymmetric bearing area (hereafter called as an asymmetric bearing) has been proposed to support large load. A schematic construction of the asymmetric bearing and its approximately expressed gauge pressure distribution are shown in Fig.1 (a) and that of double admission slot type conventional symmetric bearing is shown in Fig.1 (b) for comparison. Unbalance of a rotor, i.e., notched parts of the rotor generates centrifugal force by the rotation. That is, the centrifugal force works to lower side in this state. In conventional symmetric type bearing, the notched parts are located at the outside of the bearings as shown in Fig.1 (b). Then the bearings support full cylindrical part of rotor. Now,

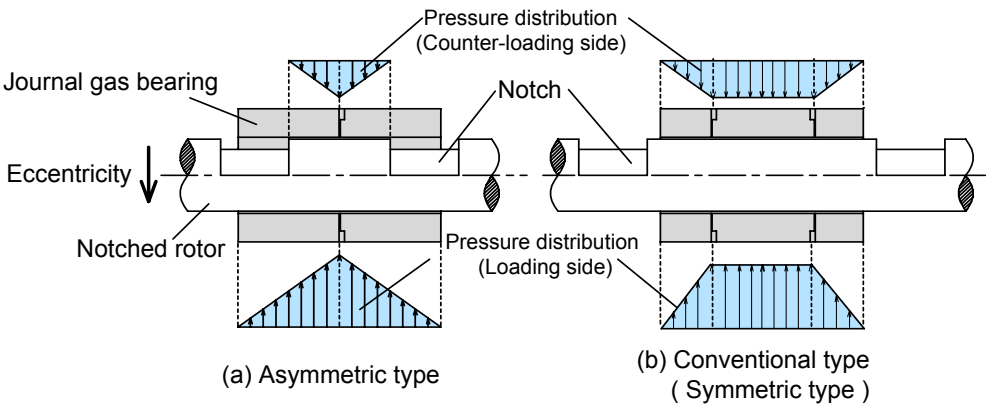

Fig.1 Construction and working mechanism of asymmetric gas bearing

let us assume that the rotor displaces slightly to lower side (loading side) from the bearing center by rotation of the rotor. Approximately, the net load capacity of journal gas bearings depends on the difference of the sum of lower side (loading side) gauge pressure and that of upper side (counter-loading side) in the bearing clearance. In conventional symmetric bearings, the bearings should support the generated centrifugal force add to the counter-loading force generated by the bearings themselves so that they are required high gross load capacity. On the other hand, in asymmetric bearing, the notched parts are located at the inside of the bearing as shown in Fig.1 (a). Since the counter-loading force can be decreased by decreasing the bearing surface of the counter-loading side, the net load capacity expected to be largely increased. In addition, since the centrifugal force acts on the side of the rotor without the notched parts, the bearing will stiffly support the rotor by increasing the effective bearing area in this side.

2.2 Characteristics of the asymmetric bearing

Characteristics of the asymmetric bearing were calculated and compared with the conventional symmetric bearing. Notations of the asymmetric bearing are defined in Fig.2 and the dimensions are listed in Table 1. Calculated load capacity W and flow rate Q of the asymmetric bearing are shown in Fig.3. In the figure, eccentricity ratio ε of horizontal axis represents the rotor displacement (loading side from bearing center is positive). Load capacity W = 1000 N at allowable eccentricity ratio ε = 0.5 is taken up. In the figure, red lines and black lines correspond to load capacities of asymmetric bearing and conventional symmetric bearing, respectively. In asymmetric bearing, divergence formulation method is applied for this calculation[8]. In conventional bearing, analytical approximate calculation is applied[9]. The calculations are conducted by using air as working gas. Figure 3 (a) shows the load capacity W against eccentricity ratio ε. When W = 0 N, this means the unbalance

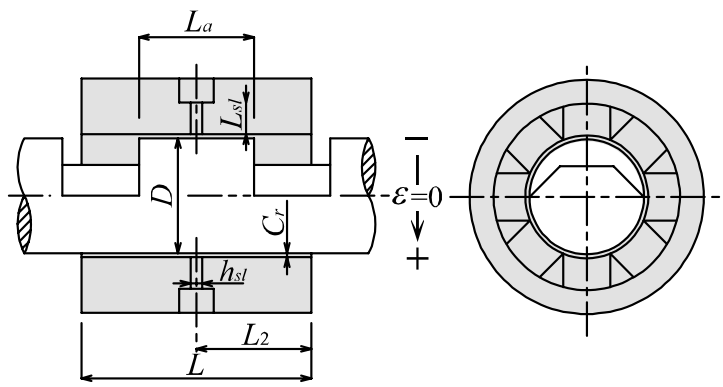

Fig.2 Hydrostatic asymmetric journal gas bearing

Table 1 Bearing dimensions for the calculation

	Asymmetric (Single admission)	Conventional (Symmetric) (Double admission)
Bearing diameter D	60 mm	
Bearing length L	120 mm	
Slot length L_{sl}	10 mm	
Radial clearance C_r	0.03 mm	
Length between slot and bearing edge L_2	60 mm	30 mm
Slot clearance h_{sl}	0.020 mm	0.015 mm
Counter-loading side length L_a	60 mm	
Bearing supply pressure p_s	0.5 MPa	

rotor is in rest, $\varepsilon = 0$ for conventional type, while for asymmetric bearing, $\varepsilon = -0.5$. This means that the notched rotor displaces to the counter-loading side. When $\varepsilon = 0$ for asymmetric type, W becomes 500 N. When $\varepsilon = 0.5$, W =1000 N for both types. This means that asymmetric type has almost the same load capacity with conventional type although asymmetric type is the single admission type. Figure 3 (b) shows the gas flow rate Q against eccentricity ratio ε. The flow rate of asymmetric type is almost half that of conventional type in spite of the same load capacity. This means that asymmetric type may support large load with low gas consumption. Therefore, asymmetric bearing has a merit to decrease practical risk to crash the bearing because the rotor approaches to the bearing center when the load acts to bearing. The use of asymmetric bearing to the ACROSS transmitter will increase safety margin in operation.

2.3 Experimental verification of rotational characteristics of the asymmetric bearings

Rotational test of the asymmetric bearings was carried out by using trial test rig as shown in Fig.4. The test rig is designed that an unbalance rotor generates centrifugal force of 2000 N at rotational frequency f of 50 Hz. The unbalance of the rotor is made by partially notching the rotor surface. The magnitude of unbalance is 0.02 kg·m. The notched rotor has the length of 520 mm and the diameter of journal parts is 60 mm, which is positioned vertically and is driven by a variable speed motor connect with a flexible coupling. Since two journal gas bearings support centrifugal force, one bearing must shares the centrifugal force of 1000 N. Designed dimensions of the asymmetric bearings are the same as Table 1.

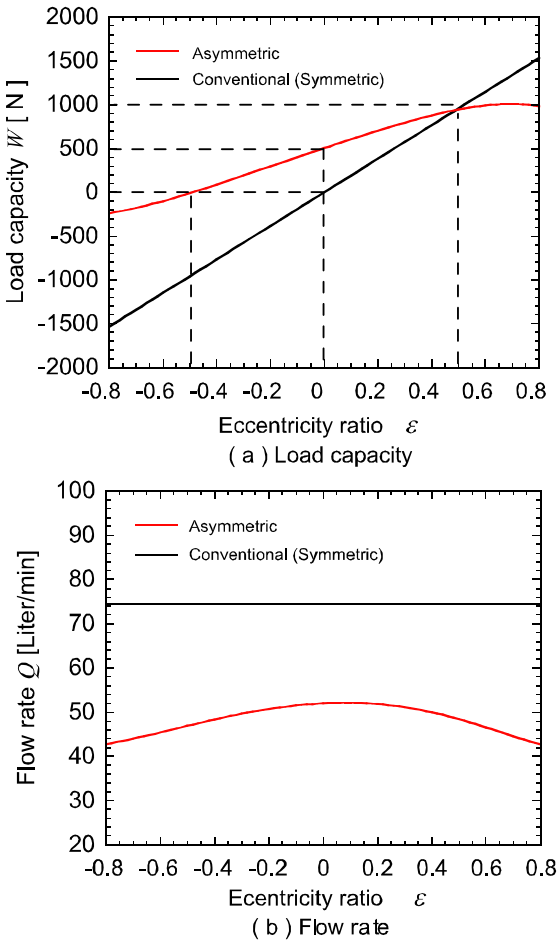

Fig.3 Calculated bearing characteristics

Actual average values of the radial clearance C_r and the slot clearance h_{sl} are 0.038 mm and 0.035 mm, which is caused by manufacturing and assembly error. Calculation of the rotor amplitude is conducted by using above values. An inherent orifice type circular thrust gas bearing is applied to support mass of a notched rotor vertically, which is positioned at bottom of the rotor. Generated centrifugal force is supported by a high stiffness holder.

Figure 5 shows the experimental results of the asymmetric bearing using trial test rig[10].

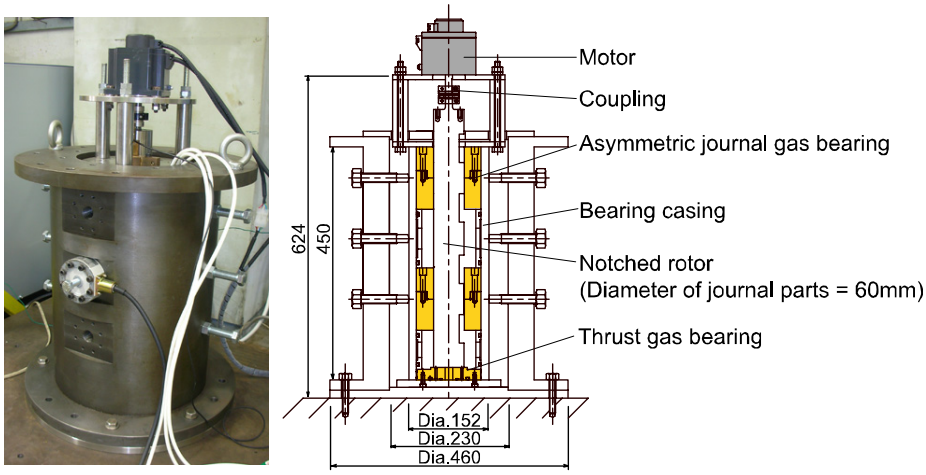

Fig.4 The trial ACROSS transmitter using asymmetric bearings

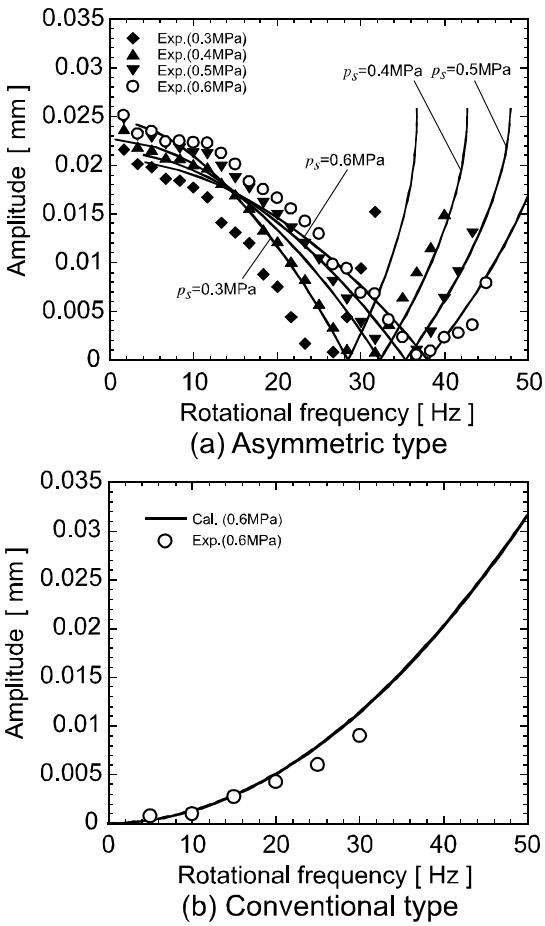

Fig.5 Rotational characteristics of the asymmetric bearings

The test is conducted by using air as working gas. The test was carried out using supply gas pressure p_s of 0.3 MPa to 0.6 MPa. Measured rotor amplitude in the asymmetric bearings is shown in Fig.5 (a). Solid lines represent calculated rotor amplitude at several bearing supply gas pressure. These curves were obtained by equating generated centrifugal force and load capacity as shown in Fig.3(a). Calculation of the bearing characteristics was conducted under the static condition since the rotational frequency of the rotor is comparatively small. In the figure, the negative inclination of the curves means that the eccentricity ratio ε is negative in Fig.3 (a), i.e., the rotor displaces to counter-loading side from bearing center in this state. Plots represent the measured values. Measured values agreed with calculated values. The differences between the calculated and measured values are thought to be caused by the errors in roundness and concentricity of the rotor. As other causes, it is thought that the shaft rotate with slightly incline from perpendicular caused by difference of characteristics of the bearings. In asymmetric bearings, since the amplitude becomes zero at a certain rotational frequency, the amplitude can be controlled to zero according to the design at the practical rotational frequency. For comparison, measured result of conventional symmetric bearings is shown in Fig.5 (b) as an example[11]. The rotor amplitude increases uniformly with rotational frequency. Therefore, the practical utility and the safely margin in driving of the asymmetric bearings are proved.

3. Supply pressure control of asymmetric bearings under the operation

In practically working ACROSS transmitters, an eccentric rotor is driven with rotational frequency modulation. In the case of hydrostatic asymmetric journal gas bearings, the rotational frequency that the theoretical amplitude becomes zero changes as shown in Fig.5 (a). If supply gas pressure changes automatically in accordance with the rotational frequency modulation, it is thought that the keeping of the small amplitude under the operation become possible. Simultaneously, it is expected that the gas consumption decrease in comparison with fixed pressure condition at 0.6 MPa. Then, experimental verification of pressure control under the operation was carried out by using an electro-pneumatic pressure regulator.

3.1 Measuring system of the test

Figure 6 shows the measuring system of the test applying electro-pneumatic pressure regulator. Black lines and blue lines represent electric signal and gas flow, respectively.

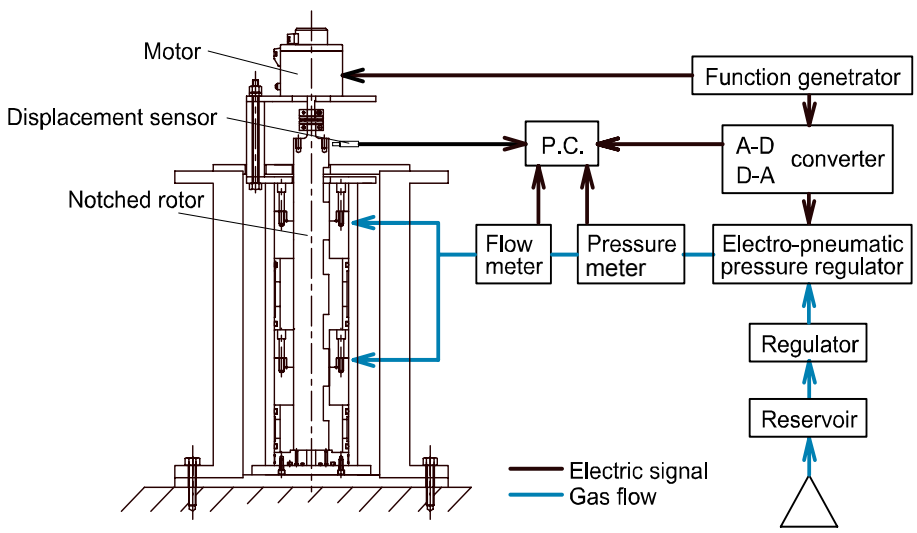

Fig.6 Measuring system of the amplitude and gas consumption

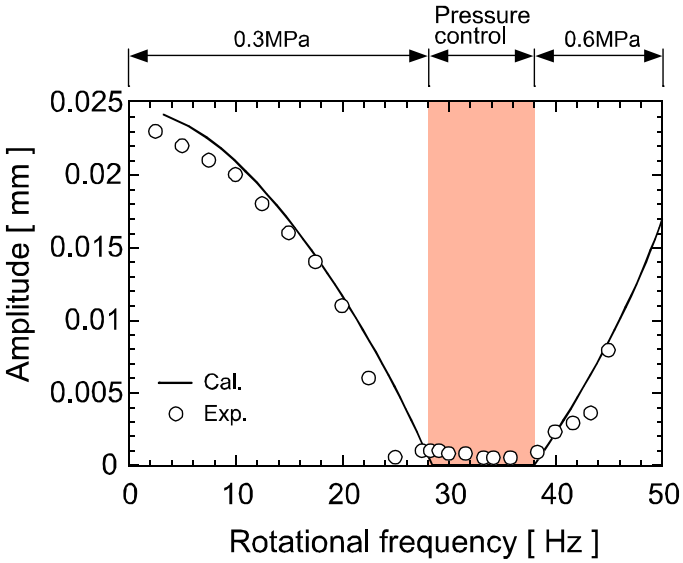

Fig.7 Measured rotor amplitude under the quasi-static condition

Rotor amplitude is measured by a non-contact type displacement sensor placed at upper part of the rotor. Voltage signal to drive a motor is generated by a function generator. The signal is also input to an electro-pneumatic pressure regulator through an A-D and a D-A converter to control the input voltage. Bearing supply gas pressure is controlled in accordance with rotational frequency modulation by this system. The test is conducted by using air as working gas. Pressurized gas is supplied to the bearings through a gas reservoir and regulators. Gas flow rate and supply gas pressure are measured by a flow meter and a pressure meter, respectively.

3.2 Test of the quasi-static condition

The test of quasi-static condition was carried out before rotational frequency modulation test to verify the operation of the electro-pneumatic pressure regulator. The rotational frequency range of pressure control was set from 28 Hz to 38 Hz in the test. In lower rotational frequency than 28 Hz, supply pressure is fixed to 0.3 MPa, and higher rotational frequency than 38 Hz, that is fixed to 0.6 MPa. The pressure control range corresponds to the range of supply pressure from 0.3 MPa to 0.6 MPa when the calculated amplitude becomes zero in Fig.5 (a). The bearing supply pressure is controlled that the rotor amplitude becomes zero in this range in accordance with the rotational frequency modulation.

The measured result is shown in Fig.7. Solid line and plots of open circle represent calculated amplitude and measured values, respectively. Colored frequency range represents the pressure control range. Measured values agreed with calculated values. It is verified that the amplitude became almost zero in pressure control range.

3.3 Test of the rotational frequency modulation

The test of the rotational frequency modulation is carried out using above frequency range, i.e., 28Hz to 38Hz. Figure 8 shows the measured time history of the rotor amplitude under the rotational frequency modulation. Figure 8 (a) represents time history of rotational frequency and (b) and (c) represent amplitude under the fixed supply pressure at 0.6 MPa and controlled pressure condition, respectively. The rotational frequency (a) and the amplitude of the figure (b) and (c) are synchronized. Considering allowable torque of a motor, sinusoidal modulation and a cycle of 10 seconds are applied. In fixed supply pressure condition (Fig.8 (b)), rotor amplitude largely changes synchronizing rotational frequency. Measured amplitude became 0.01 mm at 28 Hz, and almost zero at 38 Hz. These

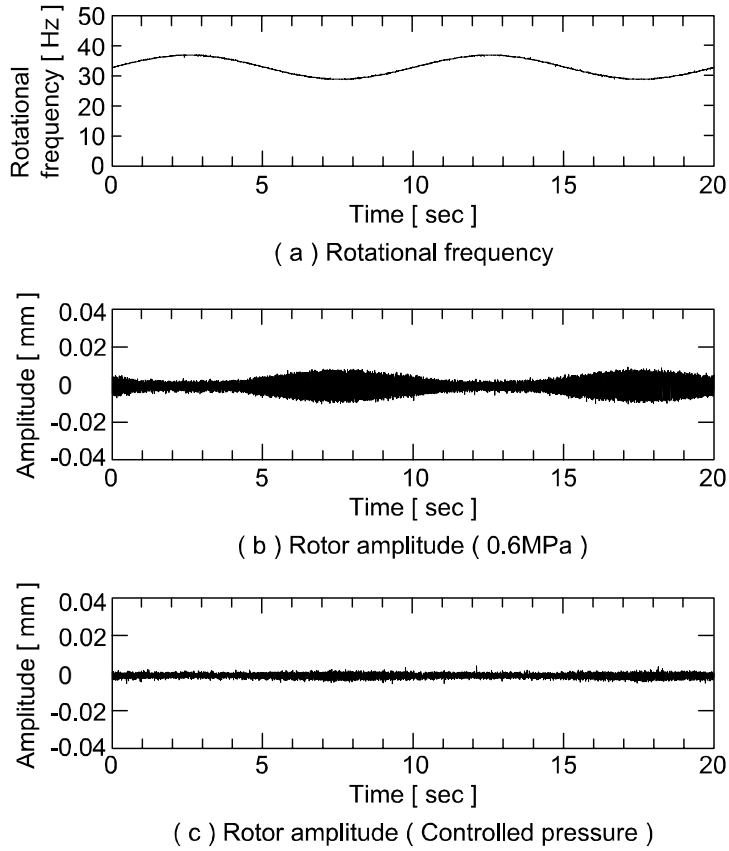

Fig.8 Time history of rotor amplitude under the rotational frequency modulation

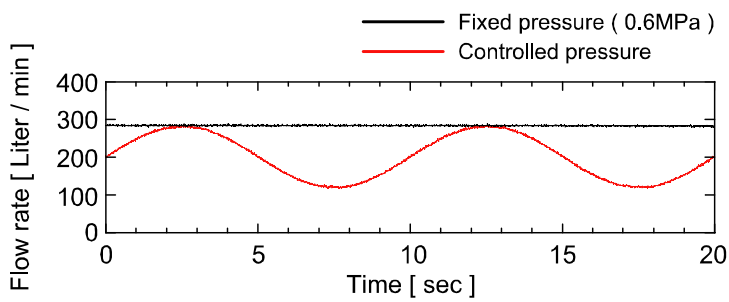

Fig.9 Measured gas flow rate of the asymmetric bearings

values agreed with above measured result as shown in Fig.5 (a). In controlled supply pressure condition (Fig.8 (c)), rotor amplitude decrease with the rotational frequency. Thus, it is proved that the effectiveness of the decrease of the rotor amplitude under the operation by using the electro-pneumatic pressure regulator.

Figure 9 shows the measured time history of the consumed gas flow rate of the asymmetric bearings. Vertical axis represents gas flow rate at one minute. Black line and red line represent gas flow rate under the fixed and controlled pressure condition, respectively. In case of controlled pressure condition, gas flow rate largely decreased in comparison with the fixed pressure condition. Gas flow rate at one cycle is calculated by integrate during the one cycle (10 seconds). In fixed pressure condition, the consumed gas flow rate becomes 47.4 liter and that of controlled pressure condition, it is 33.5 liter at a cycle. In case of the test bearings, gas flow rate decreased 30 percents in comparison with the fixed supply pressure at 0.6 MPa. Since this brings the decrease of the gas compressor power, it is thought that the supply gas pressure control of the asymmetric bearings is very effective.

4. Conclusion

To solve serious problems of the practically working ACROSS transmitters, a hydrostatic asymmetric journal gas bearing has been proposed. Rotational test of 2000 N class trial ACROSS transmitter using asymmetric bearings was carried out. As the result, amplitude of the notched unbalance rotor in the asymmetric bearings can be zero at a certain rotational frequency. In conventional symmetric bearings, since the amplitude increase with increase in rotational frequency, use of asymmetric bearings more effective to support large unbalance rotor.

In addition, test of the rotational frequency modulation with the bearing supply pressure control was carried out. As the result, rotor amplitude is largely decreased under the operation by control of the bearing supply pressure. Gas consumption is also largely decreased in comparison with the fixed supply pressure condition. Therefore, the system is thought to be useful for the decrease of the gas compressor power in the continuous operation.

References

(1) Kumazawa, M. et al., ACROSS: Theoretical and Technical Developments and Prospect to Future Applications: *JNC technical review*, Vol.9, (2000), pp.115-129, (in Japanese with English abstract).

(2) Kunitomo, T. and Kumazawa, M., Active Monitoring of the Earth's Structure by the Seismic ACROSS -Transmitting and Receiving Technologies of the Seismic ACROSS-, *Proceedings of the 1st International Workshop on Active Monitoring in the Solid Earth Geophysics*, No.S4-04 (2004-7), pp.181-184.

(3) Kunitomo, T. and Kumazawa, M., Active Monitoring of the Earth's Structure by the Seismic ACROSS -Development of the Seismic ACROSS and Continuous Transmission at Tono Mine ACROSS Station-, *Proceedings of the 1st International Workshop on Active Monitoring in the Solid Earth Geophysics*, No.S4-P03 (2004-7), pp.218-222.

(4) Yamaoka, K. et al., A trial monitoring temporal variation of seismic velocity using an ACROSS system, *The Island Arc*, Vol.10, (2001), pp.336-347.

(5) Ikuta, R. et al., Continuous monitoring of propagation velocity of seismic wave using ACROSS, *Geophysical Research Letters*, Vol.29, No.13, 1029/2001GL013974, (2002).

(6) Ise, T. et al., Hydrostatic Asymmetric Journal Gas Bearings for Largely Unbalanced Rotors of Seismic ACROSS Transmitters, *Journal of Advanced Mechanical Design, Systems, and Manufacturing*, Vol.1, No.1, (2007), pp.93-101.

(7) Ise, T. et al., Application of Hydrostatic Air Bearings to ACROSS, *Proceedings of the 1st International Workshop on Active Monitoring in the Solid Earth Geophysics*, No.S4-P01 (2004-7), pp.208-211.

(8) JSME Research Group for Gas Bearings, *Theory, Design and Application of Gas Bearings*, Text book for the course of gas bearings, (1973), p.80, JSME (in Japanese).

(9) Togo, S., *Gas Bearing Design Guide Book*, (2002), p.31, Kyoritsu Shuppan Co., Ltd. (in Japanese).

(10) Ise, T. et al., Hydrostatic Asymmetric Journal Gas Bearings for ACROSS Transmitters, *Proceedings of the IFToMM 7th International Conference on Rotor Dynamics*, No.185, (2006-9).

(11) Ise, T. et al., Development of Hydrostatic Air Bearings supported ACROSS, *Science and engineering reports of Tohoku-Gakuin University*, Vol.39, No.1, (2005), pp.31-36, (in Japanese with English abstract).