Experimental verification of externally pressurized gas journal bearings with asymmetric gas supply (Supply gas pressure control operation using a small size test rig)

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Abstract
Externally pressurized gas journal bearings with an asymmetric gas supply mechanism have been developed by one of the authors. This bearing has a large load capacity compared with conventional symmetric gas supply bearings because low and high pressurized gases are supplied to loaded and counter-loaded side bearing surfaces, respectively. It has been proposed that this type of bearing has advantageous characteristics applicable for use in a general purpose X-ray computed tomography scanner gantry. The adaptation of this gas bearing to the device can conceivably contribute to an improved performance by increasing the rotational speed and decreasing the operating noise. This is effective for higher precision scanning and for reduction of the level of stress on a patient. Numerical calculations of this bearing were conducted and the resulting characteristics were compared with those of a conventional symmetric supply gas journal bearing. The effectiveness of the bearing for this application was demonstrated by conducting rotation tests using a small size test rig. The bearing diameter and length were 60 and 120 mm, respectively. The test bearing was operated under supply gas pressure controlled conditions. The rotor vibration amplitude decreased under the controlled pressure supply conditions, although the amplitude increased under conventional symmetric gas supply conditions with an increase in rotational frequency. The gas flow rate decreased by 21.4% under controlled supply pressure conditions compared with conventional supply pressure conditions. The rotor of the test rig was safely supported by this bearing, and effective data for the practical operation was obtained.

Key words: Externally pressurized gas bearings, Large load rotor, Load capacity, Gas flow rate, Pressure control, X-ray CT gantry

1. Introduction

Externally pressurized gas journal bearings have advantages such as high precision rotation, low frictional loss, and high rotational speeds. Previously, these bearings were developed and studied for high precision and high-speed rotating machine tools and cryogenic turbo machines, etc. Recently, externally pressurized gas journal bearings with asymmetric gas supply have been proposed and studied (Ise, et al., 2012), (Ise, 2011). This bearing was developed to support large load rotors such as large eccentric rotors similar to those used in vibration exciters. These exciters were developed to detect subtle changes in the physical state of the Earth’s structure using a precise and continuous sinusoidal signal (Kumazawa, et al., 2000), (Ikuta, et al., 2002), (Yamaoka, et al., 2001). Because these devices are set underground, the use of an asymmetric supply type bearing is suitable for ease of maintenance. In this bearing, high pressure and low pressure gases are supplied to the loading and a counter-loading side bearing surfaces, respectively. This supply pressure difference provides a large load capacity and small gas flow rate compared with conventional gas journal bearings. The static characteristics of these asymmetric gas supply bearings have already been studied. This bearing has been demonstrated as...
being more useful in terms of support of the load compared with the previously developed bearings (Okano, et al., 2006), (Ise, et al., 2007a), (Ise, et al., 2010), (Ise, et al., 2007b).

We propose that this type of bearing is applicable for use in an X-ray computed tomography (hereafter called as X-ray CT) scanner gantry (Tokumiya, et al., 2013). The rotary drum of the gantry has electrical units to computed tomography scanning, as shown in Fig.1. The drum is supported by rolling contact bearing. The dynamic unbalance is roughly canceled by the arrangement of these devices. The rotary drum inner diameter is 800 or 1,000 mm and the rotating speed is 100 or 220 rpm. In present devices, rolling contact bearings are used to support the rotor. The maximum circumferential speed exceeds 11.5 m/s in present conditions. Rotation produces a somewhat unbalanced centrifugal force since the rotary drum has a small eccentricity although the unbalance is cancelled; therefore, the force due to residual unbalance with the drum mass gives to the bearings as the load. For these reasons, a large torque is required in the operation. A greater rotational speed is presently demanded of the gantry because realization of high speed rotation will result in higher precision scanning. However, a higher rotational speed results in a shorter bearing lifetime and an increase of noise, which tends to create higher patient anxiety. Radiation exposure of the patient also can be reduced by this mechanism because examination time decreases by high speed rotation. The use of the proposed gas bearings can potentially solve these problems.

The final purpose is application of this bearing for the X-ray CT gantry. For this purpose, verification of the proposed bearing mechanism under the supply gas pressure control condition is first necessary. This report presents rotational tests of the proposed hydrostatic gas journal bearings with asymmetric gas supply using a small sized preliminary test rig to verify the effectiveness of the proposed mechanism.

**Nomenclature**

\[ C_r : \text{Radial clearance} \]
\[ D : \text{Bearing diameter} \]
\[ e : \text{Rotor displacement from bearing center} \]
\[ f : \text{Rotational speed in Hz} \]
\[ h_{sl} : \text{Slot clearance} \]
\[ L : \text{Bearing length} \]
\[ L_2 : \text{Length between slot and bearing edge} \]
\[ L_{sl} : \text{Slot length} \]
2. Working mechanism of the proposed bearing

Experimental verification of the capacity of externally pressurized gas journal bearings with asymmetric gas supply (called AGS bearings hereafter) to support large load rotors was conducted. The working mechanism and the characteristics of the AGS bearing were examined by comparison with the conventional type of externally pressurized gas journal bearing. A schematic of the proposed AGS bearing and its gauge pressure distribution are shown in Fig.2 (b), and for comparison, that of a conventional symmetric bearing is shown in Fig.2 (a). In this study, it is assumed that the notched segments of the rotor cause eccentricity rather than the X-ray devices affixed to the rotary drum. The rotary drum mass of the actual X-ray CT is 1,200 kg approximately, i.e., 12,000 N is loaded to the journal bearing. In addition, it is estimated that the magnitude of unbalance is 11.1 kg·m in maximum depending on the arrangement of the internal devices. Therefore, the centrifugal force of 5,900 N occurs at maximum speed of 220 rpm (3.67 Hz), this is loaded to the journal bearing. A notched, unbalanced rotor creates an unbalanced centrifugal force by its rotation, and this force adds as well as subtracts from the load on the bearing. The notched segments are located at the outside of the bearing, as shown in Fig.2. The bearing supports the full cylindrical section of the rotor. Now, let us assume that the rotor displaces slightly toward the lower side from the bearing center. The net load capacity of a journal gas bearing approximately depends on the difference of the sum of the lower side (loading side) gauge pressure and that of the upper side (counter-loading side) in the bearing clearance. In a conventional symmetric bearing (Fig.2 (a)), a constant gas pressure $p_s$ is supplied to the lower side (loading side) and upper side (counter-loading side). This bearing should support the unbalanced centrifugal force added to the counter-loading force generated by the bearings themselves. Therefore, this bearing requires a high gross load capacity. If the rotor mass is very large, the rotor initially displaces to the lower side. In the AGS bearing shown in Fig.2 (b), different pressures $p_l$ and $p_u$ are supplied to the loading and counter-loading side bearing surfaces, respectively. High pressure $p_l$ is supplied to the loading side and low pressure $p_u$ is supplied to the counter-loading side. This supply mechanism results in a large pressure difference at the bearing surface, which produces a large load capacity compared with conventional type bearings. Consequently, the rotor can be positioned at the bearing center. Centrifugal force is generated by the rotation of the drum. For this reason, the supply gas pressure control is required according to the phase of the unbalance of the rotor in case of use of this bearing. This bearing mechanism is new approach. If the load is small compared with rotor stiffness, the rotor displacement can still be maintained at the same position. However, if the load is large, the supply pressure should be changed in correspondence to the rotational frequency since the direction and magnitude of the load changes during the rotation of this bearing.

![Fig. 2 Working mechanism of the proposed bearing in comparison with the conventional type](image-url)
3. Configuration of the bearing

A schematic of the proposed AGS bearing configuration is shown in Fig. 3. In this study, a rectangular slot restriction type bearing is considered. The bearing is installed in a casing, which has gas supply ports set within it. Let us assume that the rotor loads the bearing in the downward direction, i.e., the rotor displacements toward the loading side from the bearing center are the same as discussed previously. High and low gas pressure are supplied to the loading and counter-loading sides from the supply ports to the bearing surface, respectively, as shown by the red and blue arrows in Fig. 3. Separation components are installed at the boundaries between the loading and counter-loading domains to divide each pressure region halfway around the groove. Supplied gases flow to the bearing surfaces through circumferential grooves and rectangular slot restrictors. The gases flow into the clearance between the inner periphery surface of the bearing and outer periphery surface of the rotor, and then flow out of the bearing. A large load capacity is produced by this structure and gas flow mechanism.

4. Proposed AGS bearing characteristics

The proposed AGS bearing characteristics were evaluated using the results of calculations. The detailed bearing characteristics have been calculated in a previous study (Ise, et al., 2012), where the divergence formulation method was applied (Yoshimoto, 1986), (JSME, 1973), (Togo, 2002). This method uses the Reynolds equation, which (in the study) was differentiated under conditions of laminar, viscous, and isothermal flow; solved numerically. These calculation conditions are similar to those used in the previous studies of gas bearing by many researchers. It is demonstrated that the experimental results are well agreed with the calculated results. These static characteristics were also verified experimentally. Figure 4 shows the calculation model. A single admission type journal gas bearing with a rectangular slot restrictor was taken up for the example. In Fig. 4, the eccentricity ratio $\epsilon$ is defined such that the downward direction from the geometrical center is positive. The primary dimensions and supply gas pressure for the calculations are listed in Table

![Fig. 3 Configuration of the proposed AGS type hydrostatic journal gas bearing](image)

![Fig. 4 Calculation model of the proposed AGS bearing](image)
I as an example. All calculations used air as the working gas.

Figure 5(a) shows the calculated results of the dimensionless load capacity of the bearing relative to the eccentricity ratio for various counter loading side pressures $P_u$ also expressed in dimensionless values. Under the conditions of the calculations, the black line in the figure shows the performance under the conventional symmetric gas supply condition, while the other lines are indicative of asymmetric gas flow conditions. Under conventional symmetric conditions, the load capacity is symmetrical about at the point $\epsilon = 0$ (Point A). The load capacity increases with increasing pressure asymmetry since the counter-loading force decreases. Eccentricity ratio $\epsilon$ is defined by $C_r / e$, where $e$ is rotor displacement from bearing center. Therefore, $\epsilon = 1$ means touchdown of the rotor and the bearing. When $W = 0.5$, the rotor displaces to $\epsilon = 0.4$ at $P_u = 6$ (Point B). For the same load capacity, the rotor position is at $\epsilon = 0.1$ for $P_u = 2$ (Point C), i.e., the rotor position is nearly at the bearing center. The $P_u$ and $P_l$ are nondimensional values of the supply gas pressures divided by atmospheric pressure $p_a$. This indicates that asymmetrical gas flow can decrease the practical risk of collision between bearing and rotor because the rotor approaches the bearing center under load. Consequently, the use of the AGS bearing for the large load X-ray CT gantry is expected to increase the safety margin during operation.

Figure 5(b) shows the gas flow rate $Q$ relative to the eccentricity ratio $\epsilon$ for various counter loading side pressures $P_u$ expressed in dimensionless values. The flow rate is observed to decrease with increasing pressure asymmetry. The flow rate is compared relative to the eccentricity ratio at a load capacity of $\bar{W} = 0.5$ (for example, points B and C). The flow rates under asymmetric supply conditions are less than that under the symmetric condition despite the fact that the same load capacity is obtained. In fact, the value is nearly half, and the AGS bearing is capable of supporting a large load with a low gas flow rate. This contributes for saving the driving power in operation.

| Table 1 Bearing dimensions and parameters used for the calculations |
|-----------------|---------------|
| Bearing diameter | $D$ [ mm ] | 60 |
| Bearing length | $L$ [ mm ] | 120 |
| Slot length | $L_s$ [ mm ] | 10 |
| Radial clearance | $C_r$ [ mm ] | 0.020 |
| Length between slot and bearing edge | $L_2$ [ mm ] | 60 |
| Slot clearance | $h_s$ [ mm ] | 0.010 |
| Supply pressure | $P_u$ [-] | 2–6 |

![Image of Fig. 5](image_url)

**Fig. 5** Influence of counter-loading side supply gas pressure, where the black line represents symmetric gas supply conditions.
5. Experimental verification of the bearing characteristics

5.1. Experimental setup and measurement system

A schematic of the experimental setup and the measurement system of the AGS bearing is shown in Fig. 6. A photographic outer view of the experimental apparatus is shown in Fig. 7. The AGS bearing and notched rotor were placed horizontally. The test bearing was set in a bearing casing. Pressurized air was supplied to the bearing from a gas compressor through a mist separator, an air filter, a regulator, a flow meter, and a pressure meter. The rotor was driven by a servo motor connected with a coupling. The motor rotational speed was controlled using a direct current power supply source. Displacement of the notched rotor was measured by two noncontact type displacement meters set along the horizontal (x direction) and vertical (y direction) axes, as shown in Fig. 6. The rotor included notches outside of the bearing to generate an unbalanced centrifugal force, and the magnitude of unbalance \(M_r\) was \(32.5 \times 10^{-3} \text{ kg-m}\). The centrifugal force \(F\) is expressed as \(F = M_r(2\pi f)^2\), i.e., the rotor would generate 3,200 N at a rotational speed of 50 Hz. The shaft mass was 4.56 kg. In this study, verification and confirmation of the bearing mechanism is purpose. Although these values are so small compared with the load of the actual X-ray CT drum caused by the mass and centrifugal force, the bearing diameter was set at 60 mm in accordance with the previous reports (Ise, et al., 2007), (Ise, et al., 2012). Magnitude of unbalance was set as large as possible in this experimental setup configuration. The test was performed up to 10 Hz in 1 Hz steps from 1 Hz to verify the pressure delay, and the speed was controlled under quasi-static conditions. In the case where the centrifugal force acted in the downward direction, the sum of the rotor mass and centrifugal force loaded on the bearing was 107.3 N at 7 Hz. Conversely, in the case where the centrifugal force acted in the upward direction, the difference of the rotor mass and centrifugal force loaded on the bearing was -18.2 N at the same rotational frequency. Since the load was small as mentioned above, the supply pressure to the lower and upper domain was set at \(P_l = P_u = 4.2\) in order to clarify the change of the rotational vibration amplitude. In upper domain, the minimum pressure was set at \(P_u = 2\) to accommodate the load direction on the bearing, given by the total of the rotor mass and centrifugal force.

This was performed for the ease of measuring the change of the amplitude by decreasing the bearing stiffness. The gas pressure was supplied to the lower and upper domains by a standard hand-operated regulator and an electro-pneumatic regulator, respectively. The electro-pneumatic regulator was controlled by the output of a function generator. A sinusoidal wave profile, as shown in Fig. 6, was chosen since the centrifugal force loaded on the bearing is sinusoidal with respect to the rotation. The pressure control timing was tuned manually through a trial and error process.

Fig. 6 Configuration of the experimental setup and the measurement system
5.2. Verification of the delay time of the air supply pressure

The externally pressurized gas bearings used in this study allows for the control of the supply pressure in coincidence with the rotation of the rotor. However, the occurrence of a delay time is expected because of the compressibility of the gas. For this reason, an understanding of the delay time in the case of using an electro-pneumatic regulator is necessary, and the delay time of the bearing to be used in this study was determined before the rotational test. The distance of the gas piping between the electro-pneumatic regulator and the bearing used in these tests was 0.5 m. The pressure regulator was installed at the air inlet of the casing, i.e., the point closest to the bearing surface. Figure 8 shows the output of the electro-pneumatic regulator and of the function generator in the cases of rotation at 2, 5, 7, and 10 Hz. The control voltage was constant at every frequency. Both outputs were synchronized. No gas pressure delay appears versus the function generator output at 2 Hz, and the pressure amplitude is the same as the set value. A gas pressure delay of 0.05 s was found to occur under 5 and 7 Hz conditions. The upper limit of the pressure amplitude was not found to change, but the lower limit gradually decreased. It was found that the function generator output and the pressure amplitude were of opposite phase after 7 Hz. Furthermore, it was found that the maximum and minimum pressure amplitudes were suppressed. In 10 Hz condition, pressure amplitude markedly decreased. The amplitude is about 1.5 atmospheres. It is thought that small effect is obtained in this amplitude. This could have been caused because the discharge pressure control of the electric-pneumatic regulator can not follow the input signal by high frequency. This phenomenon might be mitigated by changing the flow channel structure inside of the bearing as another factor. Considering the above results, it was not found to be possible to conduct the tests under high-frequency pneumatic control. Therefore, it was decided that the rotational tests be performed up to 7 Hz.

5.3. Rotational test under quasi-static conditions

The actual dimensions of the manufactured test bearing are as follows: $h_{bl} = 0.020$ mm and $C_r = 0.045$ mm. All other dimensions are the same as those given in Table 1. Figure 9 shows the characteristics of the bearing load capacity. In this study, the experiments were conducted that frequency difference was set to the function generator and the motor rotational frequency. Since rotor is operated quasi-statically, the rotor amplitude slowly varies by the phase difference of each other. Minimum amplitude could be obtained by the time histories. The amplitude was verified and compared with the calculated values.

At a $f = 5$ Hz rotation, the nondimensional load capacity $\tilde{W}$ was calculated as 0.105 and 0.017 for lower and upper directions, respectively. The rotor displacements in the bearing clearance are shown by the broken green arrow in the symmetric gas supply condition in Fig. 9. It is to be expected that the shaft displaces between points D and E (shown by the broken green arrow in Fig. 9) under asymmetric gas supply conditions. The measured net supply pressure amplitude of the upper side $P_u$ was 2.5 to 4.2, as shown in Fig. 8 for $f = 5$ Hz. This means outlet pressure of the electro-pneumatic regulator...

regulator. However, actual effective pressure amplitude narrows further to 2.8 and 4.2. This was verified before the rotation test by measuring the displacement of the shaft in the static condition. This condition is expressed by the broken brown arrow in Fig. 9. It is to be expected that the shaft displaces between points E and F. Figure 11 shows the time history of the rotor vibration at $f = 5$ Hz. The plots represents the amplitude of vibration in the $y$ direction. The total amplitude exceeds 0.018 mm at 5 Hz. This amplitude includes roundness of the shaft outer surface of 0.005 mm. Figure 10 shows rotor amplitude of $f = 1$ Hz. The total amplitude coincides with the designed values. Actual amplitude could be obtained 0.013 mm by subtracting the roundness of the shaft from the amplitude of Fig. 11. This value is equivalent with $\epsilon = 0.14$. Conversely, the amplitude decreases to 0.0045 mm at 5 Hz under pressure controlled conditions considering roundness same as above mentioned. These values well coincide with Fig. 9. Consequently, the effect of supply gas pressure control is verified.

At a $f = 7$ Hz rotation, the nondimensional load capacity $\bar{W}$ was calculated as 0.148 and -0.025 for upper and lower directions, respectively. The rotor displacements in the bearing clearance are shown by the solid green arrow in the symmetric gas supply condition in Fig. 9. It is to be expected that the shaft displaces between points G and H (shown by the solid green arrow in Fig. 9) under asymmetric gas supply conditions. The measured net supply pressure amplitude (outlet pressure of the electro-pneumatic regulator) was 3.0 to 4.2, as shown in Fig. 8 for $f = 7$ Hz. However, actual effective pressure amplitude narrows further to 3.4 and 4.2. This was verified using same method as mentioned above. This condition is expressed by the solid brown arrow in Fig. 9. It is to be expected that the shaft displaces between points H and I. Figure 12 shows the time history of the rotor vibration at $f = 7$ Hz. In the symmetrical gas supply pressure condition, the rotor vibration amplitude increase monotonically with the rotational frequency. The total amplitude exceeds 0.019 mm at 7 Hz considering roundness of the rotor outer surface. This value is equivalent with $\epsilon = 0.22$. This amplitude agreed with calculated value as shown in Fig. 9. Conversely, the amplitude decreases to 0.015 mm at 7 Hz under pressure controlled conditions. These values well coincide with Fig. 9. However, effect of supply gas pressure control is small comparison with the $f = 5$ Hz. From the above results, It was clarified that the optimum conditions for minimizing the amplitude is present.

The gas flow rate $Q$ also decreases due to pressure control. In the symmetrical gas supply pressure condition, 23.3 L/min was expended. Under pressure controlled conditions, the flow rate decreased to 18.3 Liter/min (21.4% reduction).

Fig. 8 Measured time histories of the lower supply pressure amplitude relative to the function generator output
This value was obtained by integrating over a one minute section using the discrete time history data. This value coincides with the calculated value using the pressure amplitude, as shown in Fig. 8.

Fig. 9   Load capacity characteristics of the test bearing

Fig. 10  Measured time history of the rotor vibration ($f = 1$ Hz)

6. Conclusion

To solve the problems associated with rotary machines such as those which occur for the notched unbalanced rotor of an X-ray CT scanner gantry, this study investigated the basic rotational characteristics of an externally pressurized journal gas bearing with asymmetric gas supply. A small sized trial test rig was designed and constructed to support a notched unbalance rotor that simulated the X-ray CT scanner gantry. Quasi-static tests were conducted under low gas supply pressure conditions. The effectiveness of the AGS bearing and the nature of the supplied gas delay were verified.

The obtained results are listed as follows.

(1) The pressure delay increases with increasing control frequency. The pressure amplitude decreases with increasing rotational frequency. A useful frequency of 7 Hz was chosen for this test rig under the prevalent conditions.

(2) The vibration amplitude of the notched unbalanced rotor decreased in comparison with the symmetrical supply condition. The rotor was safely supported by this bearing even under low gas supply pressure conditions.

(3) The gas flow rate decreased in comparison with the symmetrical supply condition; a 21.4% reduction in gas flow rate was obtained in this experiment.

In this study, the pressure control timing was tuned manually through a trial and error process. A rotation test is presently planned using the shaft displacement as a feedback control of the air pressure. In addition, we are considering means of improving the high-speed response of the pressure control by incorporating an inner gas flow channel within the bearing. These enhancements will be reported in the near future.
Fig. 11 Measured time history of the rotor vibration ($f = 5$ Hz)

Fig. 12 Measured time history of the rotor vibration ($f = 7$ Hz)

References


English abstract).