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# Development of externally pressurized small-size conical-shaped gas bearings for micro rotary machines

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ABSTRACT

We have developed externally pressurized small-size conical shaped gas bearings for micro rotary machines such as gas blowers, compressors, and medical devices. The bearings are lightweight and enable machines and devices to be downsized by reducing the number of parts, since the journal and thrust loads can be supported by a pair of bearings. The conical bearings (diameters 8 mm and 10 mm) were designed and the manufacturing techniques were considered. The bearing type is a double-row inherent orifice bearing with four feed holes per row. The shape accuracy of the manufactured bearings is within the design limits, being less than 0.005 mm in both roundness and cylindricity. The surface roughness (arithmetic average roughness) of the parts is 0.0002 mm. The static characteristics were calculated and validated by testing on manufactured bearings. Comparative agreement was obtained between the measured and designed values. The proposed calculation method presents as a relatively simple approach for predicting the bearing characteristics. The test rotor exceeded 350 Hz (21,000 rpm) in the rotational test, and whirl vibration was absent. During testing, the maximum rotor vibration amplitude was 0.0125 mm (corresponding to an eccentricity ratio of 0.89).



## 1. Introduction

Externally pressurized gas bearings are commonly used in rotary machines and devices such as machine tools, high-temperature or cryogenic turbo machines, food-related machines, and medical devices. These machines require journal and thrust gas bearings for generic rotor support. In this configuration, the load in the thrust direction must be supported by flange parts. Consequently, these machines achieve higher circumferential rotor velocity than those fitted with rolling contact bearings, but the machine size is also increased.

Our research group has been developing maintenance-free air turbo blowers. The blowers will be used as cathodes of portable fuel cells and air cycle machines for aircrafts in cruise conditions, and require high rotational speeds (20,000–30,000 rpm). For spatial convenience, the maximum bearing diameter is approximately 10 mm. The assumed grade of balance of the rotor is G16 (specified in the Japanese Industrial Standard). The target rotor parameters are: unbalance = 0.003 mm, maximum speed = several tens of thousand rpm and mass = 20–40 g. The load capacity of the bearing is at least 10 N. This system is assumed to operate by gases bled from other sources. Therefore, gas bearings are appropriate. In this endeavor, we constructed a thin centrifugal blower from super duralumin. The blower features symmetric blades on both faces of the impeller, which increase the gas flow rate and the outlet pressure. Since the thrust exerted by the air flow is counterbalanced on the impeller, thrust bearing is required only to support the rotor mass in the vertical rotor, thereby minimizing the supporting force in the thrust direction. For this reason, we adopt a thrust-journal combined bearing. However, conventional use of thrust and journal bearings increases the rotor length and mass. Depending on the arrangement of the bearings, overhang is also increased, which reduces the natural frequency of the rotor. This frequency reduction is the primary technical barrier to realizing high-speed rotation. Thrust-journal combined bearings offer an effective solution to this problem. For instance, Miyatake et al. [1] researched the static characteristics of combined thrust-journal aerostatic porous bearings (inner diameter = 25 mm and outer diameter = 40 mm). While this approach is beneficial in practice, it appears inappropriately large-scale for our purpose. Furthermore, small-sized sintered porous bearings with controlled air breathability would be difficult and costly to fabricate. For this reason, the use of conical bearings could support the thrust and radial load components in both directions, reducing the number of machine parts, and hence the weight of the

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**Nomenclature**

| | |
|---|---|
| $A_e$ | effective bearing area |
| $C_{r1}$, $C_{r2}$ | bearing clearance in the radial direction |
| $c$ | flow coefficient of the feed hole |
| $D$ | average bearing diameter |
| $D_1$, $D_2$ | minimum and maximum bearing diameters, respectively |
| $d$ | feed hole diameter |
| $e_r$, $e_t$ | displacement of the rotor from the bearing center |
| $h_1$, $h_2$ | bearing clearance in the thrust direction |
| $K$ | bearing stiffness coefficient |
| $k$ | bearing stiffness |
| $L$ | bearing length |
| $L_2$ | length between feed hole and bearing edge |
| $M_{\text{in}}$ | mass inflow rate of the gas |
| $M_{\text{out}}$ | mass outflow rate of the gas |
| $n$ | number of feed holes |
| $P_s$ | nondimensional supply pressure ($p_s/p_a$) |
| $P_o$ | nondimensional outlet pressure at the feed hole ($p_o/p_a$) |
| $p_a$ | ambient pressure |
| $p_o$ | outlet pressure at the feed hole |
| $p_s$ | supply gas pressure |
| $Q$ | total flow rate of gas |
| $R$ | gas constant |
| $T$ | supply gas temperature |
| $W_r$, $W_t$ | load capacity |
| $\epsilon$ | eccentricity ratio ($e_r/C_r$) |
| $\eta$ | displacement ratio ($e_t/h$) |
| $\theta$ | circumferential angle |
| $\kappa$ | specific heat ratio |
| $\mu$ | viscosity coefficient of the gas |
| $\pi$ | circumference ratio |
| $\psi$ | Flow function of the gas |

machine. However, the characteristics of small-scale machine parts are sensitive to shape accuracy (such as the surface roughness) and the part dimensions, and must therefore be experimentally verified.

Conical bearing shapes have been developed and investigated by several researchers. Lester et al. [2] researched the behavior of externally pressurized bearings in an incompressible fluid. They reported on the configuration, the design method, and the calculated characteristics. Srinivasan et al. [3] numerically analyzed the characteristics of externally pressurized gas bearings with a conical bearing shape. Sinha et al. [4] analyzed a non-constant gap externally pressurized conical bearing subjected to different temperature- and pressure-dependent viscosities. Ingle and Ahuja [5] experimentally investigated the characteristics of externally pressurized conical bearings with a bearing diameter and length of 42 mm. Fukuyama and Someya. [7] analyzed the characteristics of conically shaped spiral groove bearings. However, as demonstrated by the above citations, studies of conical bearings have been restricted to numerical analysis, incompressible fluids, hydrodynamic bearings, and rotor diameters of tens of millimeters.

To reduce the size and weight of turbo machines, we have developed small-size conically shaped externally pressurized gas bearings. The maximum diameter of the bearings is 10 mm, and they are designed for several tens of thousand rpm rotation. This paper presents the design and configuration of the bearings and experimentally verifies their static and rotational characteristics to assess their suitability for practical use.

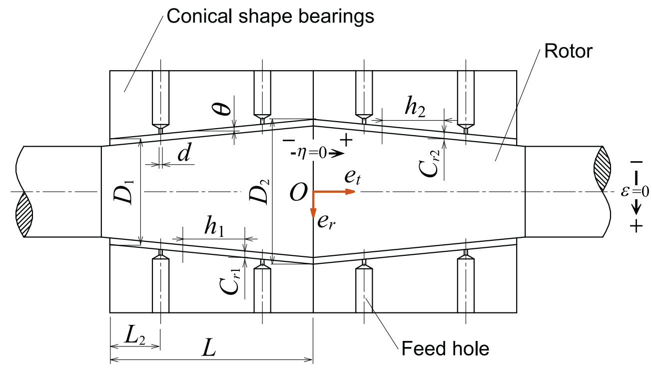


**Fig. 1.** Configuration of the bearing-rotor system.

## 2. Configuration of bearing-rotor system

The bearing-rotor system in this study is configured as shown in Fig. 1. The rotor is supported by a pair of conical shaped gas bearings installed in a casing. The dimensions of the bearings are: minimum diameter $D_1$ = 8 mm, maximum diameter $D_2$ = 10 mm, bearing length $L$ = 15 mm, flow-out length $L_2$ = 5 mm, and feed hole diameter $d$ = 0.4 mm. The bearing type is a double-row inherent orifice bearing, with four feed holes per row. In practical applications, impellers are installed at the end of the rotor. This bearing is used for vertical axis in the rotational tests. The bearing is loaded in the thrust and radial directions by the rotor mass and the centrifugal force caused by the static and dynamic imbalance, respectively. Since the rotor rotates at several tens of thousand rpm, the taper angle $\theta$ from the centerline of the bearings is small ($\theta$ = 3.8 degrees), implying that load capacity is much larger in the radial direction than in the thrust direction. An outer view of the manufactured bearings and test rotor is shown in Fig. 2. The rotor length is 94 mm, and the bearing parts are positioned at the center. The nominal radial clearance of the bearings $C_r$ = 0.010 mm (Because of manufacturing errors, the actual radial clearances are $C_{r1}$ = 0.014 mm and $C_{r2}$ = 0.012 mm, respectively). These clearances satisfy the specified tolerance indicated in the diagram, namely, 0.005 mm. The bearing clearances in the thrust direction, $h_1$ and $h_2$, are calculated from the radial clearances $C_{r1}$ and $C_{r2}$, respectively, as $h = C_r/\tan\theta$. Therefore, the actual thrust clearances are $h_1$ = 0.180 mm and $h_2$ = 0.210 mm in the neutral position, and increasingly separate as the taper angle increases. The rotor is constructed from titanium alloy (Ti–6Al–4V, ISO 5832-3), which yields high tensile (radial) strength. The bearings are fabricated from martensitic stainless steel (SUS420J2, ISO X30Cr13)

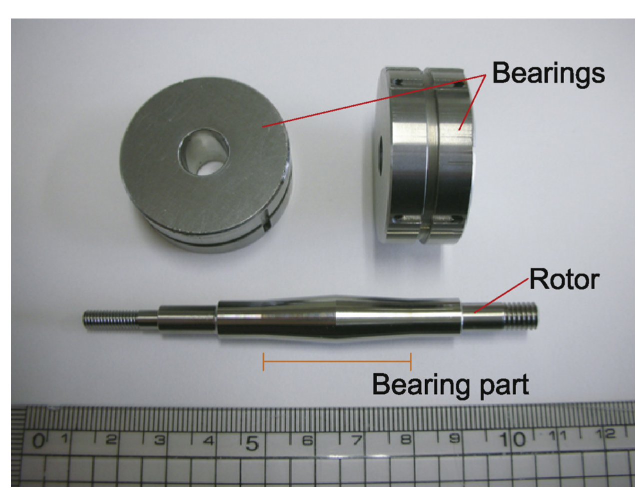


**Fig. 2.** Photograph of the manufactured bearings and test rotor.