January, 2005

Design and development of orifice-type aerostatic thrust bearing

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Abstract – Air bearings are extensively used in precision machines and equipment in recent years. Unlike conventional bearings, there is no physical contact between the sliding surfaces. The moving surface can glide smoothly on the other surface, minimising the control effort to achieve high accuracy and precision. Analytical study was carried out avoiding complexities in the formulation. The governing parameters were kept at a minimum. This helped in easy understanding of the dynamics and performance behavior of the air bearings. A simple design methodology was also developed to assist in the design or the selection process of the bearing. Finally, a three-dimensional (3D) model was built for Computational Fluid Dynamics (CFD) analysis to verify the theoretical result. A test set-up was designed to measure and study the static and dynamic performance of different kind of air bearings for future air bearing development. Moreover, the experimental data were also taken and compared with the analytical results.

Keywords: Stiffness, Load capacity, Analytical study, CFD analysis, Experimental set-up

1 BACKGROUND

Air Bearings have been employed mostly in precision machine design, like Coordinate Measurement Machines (CMM) and wafer inspection machines, to achieve ultra-precision and high accuracy. Unlike conventional ball or roller bearings, there is zero friction between the bearing surfaces. The engineers, therefore, can minimise the control effort required to achieve high accuracy and precision.

In air bearings, the load is supported by a thin film of pressurised air. Pressurised air is continuously supplied to the bearing surface and escapes through the bearing gap. The continuous flow of pressurised air through the bearing gap is enough to support the load. Depending on how the air is supplied to the bearing surface, they are typically classified as: orifice or porous-type. In orifice-type, the pressurised air is fed through one or a number of precise orifices. In porous-type, the air is supplied through the entire surface. The surface is porous and it contains microscopic holes, which are inherent in the bearing material. The air diffuses in the microscopic holes. As a result, there is a uniform pressure on the bearing surface.

Air bearings have obvious advantages over contact-type bearings. However, because of their low viscosity and nature of flow, they have some limitations. They may exhibit self-excited vibrations, called pneumatic hammer, if not designed properly. The damping of the air bearing is also not as good as that of the oil bearing. The load capacity and stiffness are also relatively poor. Nevertheless, the advantages of using air bearings always excited the researchers and obliged them to look for new designs.

1.1 Related works

Boffey et al. [1-3] have done experimental investigation on steady-state performance characteristics like stiffness because of orifice restrictor size and pocket dimensions. However, they only focused on the grooved type pocket of the orifice bearing. Stout et al. [4-8] have presented design charts to enable the design of rectangular type of aerostatic thrust bearings using pocketed and annular orifice restrictors. They also exploited the principles developed for the analysis of aerostatic journal bearings to the area of flat pad bearings. Fourka and Bonis [9] have developed a simulation program to investigate the influence of feeding system type on the performance of externally pressurised gas bearings. They used different kinds of multiple inlets specifically designed with orifices or porous compensation. Roblee and Mote [10] presented a design method for flat, circular thrust bearings.

2 OBJECTIVES

There are countless numbers of works already acknowledged in various research and journal papers, but they are of a little practical importance because of the complexities involved in the formulation and analysis. The authors’ objectives were to develop high performance orifice-type air bearings with low cost, building test facilities, and developing new and simple tools for analysis.

Moreover, SIMTech wanted to embark on building capabilities in non-contact type of bearings. Design and Development of air bearings can be a stepping-stone in this direction. The work comprises a multitude of tasks to be started simultaneously. They are as follows:

- Mathematical formulation and analysis
- Development of design methodology
- Simulation of air bearings
- Design of test set-up
Experimental studies
In section 3, mathematical formulation was carried out for a single orifice, circular pocket air thrust bearing. The bearing performance is analysed for different bearing parameters in section 4. Based on the results of the analytical study, a simple design methodology was proposed in section 5. Analytical results were validated using simulation techniques and experimental results. Section 6 talks about CFD simulation. Design of experimental set-up and testing were illustrated in section 7.

3 THEORETICAL ANALYSIS

Single orifice-type, circular-pocket air bearing is considered for the analysis. The entire flow can be divided into two parts – flow through the orifice and the bearing surface, as shown in Fig. 1.

3.1 Flow Through the Orifice
In considering the flow through the orifice, following assumptions have been made [11]:
- There are no pressure losses upstream of the throat of the jet, i.e. the pressure at the entry to the jet is the supply pressure \( P_o \).
- The pressure immediately downstream of the jet is the static pressure in the throat of the jet.

Considering the flow through a nozzle of the same throat diameter as the jet, the relationship between supply pressure and static pressure at the throat is given by [11]:

\[
\frac{P_o}{P_d} = \left[ 1 - \frac{\gamma-1}{2} \left( \frac{v}{a_o} \right)^2 \right]^{\gamma/\gamma-1} \tag{1}
\]

where \( P_d \) is the static pressure at the throat, \( P_o \) is the supply pressure, \( v \) is the velocity at the throat, \( a_o \) is the speed of sound, \( \gamma \) is the ratio of specific heats for the air.

Assuming isentropic expansion, the mass flow through a jet is given by:

\[
m = C_D A_0 \sqrt{\frac{2RT_o}{\gamma-1} \left( \frac{P_o}{P_d} \right)^{\frac{\gamma}{\gamma-1}} - \left( \frac{P_o}{P_d} \right)^{\frac{2\gamma}{\gamma-1}}} \tag{2}
\]

where \( C_D \) is the coefficient of discharge, \( A \) is the area of the throat, \( \rho_o \) is the air density at supply condition, \( R \) is the gas constant, \( T_o \) is the temperature.

3.2 Flow Through the Bearing Surface
In almost all practical bearings, the flow in the bearing clearance is laminar and pressure losses occur due to viscous shearing in the gas film. Following assumptions have been made to simplify the problem:
- Inertia forces due to acceleration can be neglected compared to frictional forces due to viscous shearing.
- Laminar flow conditions exist at all points in the gas film.
- Pressure is constant over any section normal to the direction of flow.
- There is no slip at the boundaries between the fluid and the plates.

Under these assumptions, and assuming isothermal conditions in the gas film, the mass flow rate through the bearing surface can be given by:

\[
m = \frac{\pi b h (P_o - P_a)}{12 \mu R T_o \log \left( \frac{b}{a} \right)} \tag{3}
\]

where \( h \) is the air gap, \( P_a \) is the ambient pressure, \( \mu \) is the viscosity of air, \( b \) is the bearing radius, \( a \) is the pocket radius.

3.3 Load Capacity
The load capacity can be defined as the total load supported by the bearing surfaces. It can be obtained by integrating the pressure over the whole of the bearing surface. The total load capacity can be given by:

\[
W = (P_o - P_a) \frac{\pi (b^2 - a^2)}{2 \log \left( \frac{b}{a} \right)} \tag{4}
\]

3.4 Stiffness
Stiffness can be defined as the rate of change of load capacity with respect to change in air gap.
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\[ K = \frac{dW}{dh} \]  \hspace{1cm} (5)

Higher stiffness means less compliance. This means there will be small change in air gap corresponding to large variation in load.

4 PERFORMANCE ANALYSIS

4.1 Effect of Orifice Diameter

Load and stiffness plots are taken for different size of orifice diameter. The orifice diameter is varied from 0.1 to 1.0 mm. The other parameters, such as bearing radius, pocket radius, and supply pressure, were kept constant. Figs. 2 and 3 show the plot of load capacity and stiffness with respect to the air gap for different values of the orifice diameter, respectively.

![Fig. 2. Plot of load capacity with air gap.](image)

![Fig. 3. Plot of stiffness with air gap.](image)

The maximum load capacity is same for all values of orifice diameter. However, as the orifice diameter decreases, the peak value of stiffness increases. At the same time, the corresponding value of air gap is also decreasing.

4.2 Effect of Supply Pressure

All bearing parameters were kept constant except the supply pressure. The orifice diameter was kept at 1.0 mm. The supply pressure was varied from 6 to 10 bar. Figs. 4 and 5 show the plot of load capacity and stiffness versus air-gap for different values of the supply pressure, respectively.

![Fig. 4. Load versus air gap plot for different P₀.](image)

![Fig. 5. Stiffness versus air gap plot for different P₀.](image)

As the supply pressure increases, the maximum load capacity increases. The peak value of stiffness also increases with the supply pressure.

4.3 Effect of Bearing Diameter

The bearing diameter was varied from 30 to 80 mm. The other parameters were kept constant. Figs. 6 and 7 show the plot for load and stiffness for different values of the bearing diameter, respectively.

![Fig. 6. Load plot for different bearing diameter.](image)
From the load plot, it was observed that the load capacity increases with the bearing diameter. The peak value of stiffness also increases with the bearing diameter.

From the analysis, it has been observed that the maximum load capacity is mainly decided by the bearing dimensions and supply pressure conditions. It does not depend on the orifice diameter. However, orifice diameter has greater impact on the stiffness. As the orifice diameter decreases, the stiffness increases. Stiffness can be also increased by increasing the supply pressure and the bearing diameter. This shows that smaller orifice diameter can result into higher stiffness at smaller air-gap.

5 DESIGN METHODOLOGY

The basic understanding of air bearing performance was utilised to develop a simple methodology to help the design and selection process. Design is an iterative process, but the number of iterations can be minimised by adopting a careful strategy. Fig. 8 shows the flow chart of the strategy developed.

Based on the load and stiffness requirements, the bearing dimensions, orifice size, and supply pressure can be suitably selected using this approach. Optimum air-gap and correspondingly the required pre-loading can be also obtained.

6 SIMULATION OF AIR BEARINGS

ANSYS-FLOTRAN was used for CFD simulation. A quarter model of the air bearing was made and mesh was created using mapped meshing technique. Fig. 9 shows the contour plot of pressure along the orifice and bearing surface. It is evident from the plot that the pressure inside the pocket is constant.

6.1 Comparison with Theory

Fig. 10 shows the plot of pressure variation as we move from the center of the bearing to the periphery. The theoretical and simulated plots are shown for 20 µm air gap. The theoretical plots are shown for different coefficients of discharge. Similarly, Fig. 11 shows the pressure for 30 µm air gap.
The pressure plots from the two sources show a good correlation, but there are some deviations too. For smaller air gaps, the deviation is very small. The simulation results show that there is a significant pressure drop at the interface of pocket and bearing surface. This may be because of the throttling. It is unaccounted for in the mathematical formulation.

7 DESIGN OF EXPERIMENTAL SET-UP

An experimental set-up was designed to measure both the static and dynamic parameters. Static parameters include load capacity, stiffness and mass flow rate, while the dynamic parameters include damping ratio and natural frequency. The test set-up is shown in Fig. 12. The supporting structure is securely clamped on top of the granite block. The granite block is supported on the base structure through 3-point support system. This provides high rigidity to the system.

The load is directly applied from the top via a guide shaft. The guide shaft is supported on the air bushing. It provides free movement in the vertical direction, but constrains any side movement. Any moment created due to the eccentricity of the load will be supported by the air bushing. The air bushing is mounted on the block through o-rings. This helps in automatic adjustment due to any misalignment in the axes.

One bearing adapter is used between the load and the specimen bearing. The contact between the bearing and adapter is through a ball joint. This provides extra degree of freedom to the bearing so that it can align freely to the surface.

The pressurised air is first fed to filter and drier, and then to the bearing and the bushing. In order to measure the air gap, two sets of displacement probes are used. They are the contact-type. They are placed on the top surface of the bearing and 180° apart. This can eliminate error due to tilting of the bearing. This is shown in Fig. 13.

7.1 Testing and Experimentation

Two sets of orifice-type air bearings having circular pocket at the center were used for testing. The air gap is measured for each load and load plot is obtained. Stiffness plot is obtained from the load plot. But there are a lot of uncertainties and disturbance in the data. Finding stiffness from noisy data by differentiating can give a very erroneous result. Therefore, polynomial curve fitting was done and then differentiated the resultant curve to get the stiffness plot.

7.2 Comparison with Theory

Figs. 14 and 15 show the load and stiffness plots respectively. Analytical plots for different supply pressures are also shown beside the experimental one.
The experimental curves for both load and stiffness follow the trend obtained analytically. At larger air gaps, the deviation is rather small. However, analytical results are in good agreement with the experimental one and can be utilised to further enhance understanding of bearing performance.

The discrepancy in the result may be due to variation in the pressure at the orifice. Direct pressure measurement at the orifice may not be possible. The pressure can be estimated using some indirect method. The electrical circuit equivalent to the pneumatic system can be helpful in estimating the pressure. But again it requires careful modeling of the system. Pressure distribution under the bearing surface can be helpful in understanding the problem. This can be also helpful in estimating the pressure at the orifice. But it is not an easy task. The surface requirement for air bearing itself is very stringent. Under such circumstances, any deliberate insertion of irregularity on the surface can alter the behavior.

8 CONCLUSION

This work is the result of SIMTech’s pursuit to achieve new goals and building new capabilities. Non-contact types of bearings are the critical components in the design of ultra-precision machines. In this work, the emphasis was on building a basic groundwork, so that the process of achieving a good and optimum design of air bearing with low cost can be expedited.

The analytical results gave an idea how an air bearing can perform under given conditions and parameters. CFD simulation can be helpful in further broadening our understanding. The testing facilities can help us in understanding the performance behavior of different kinds of bearings and can be used to develop new kinds of bearings with better performance, but with low cost.

9 INDUSTRIAL SIGNIFICANCE

Air bearings are critical components in any precision motion stage. The complexity in formulation and analysis and higher cost are the main deterrents in its possible industrial applications. If these problems can be overcome, they can be more confidently used in applications where friction and spillage are major concerns.

REFERENCES