Sealing Performance of Labyrinth Seals against Small Oil Particles in HDDs

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Small oil particles generated from lubricant for ball bearings in HDDs sometimes cause fatal problems such as head crashes. A labyrinth seal is often used to prevent oil particles from scattering onto the disk surfaces. In this paper, the sealing performance of labyrinth seals for small oil particles is investigated numerically and experimentally. Two types of labyrinth seal are considered, one a straight-type and the other a bent-type seal. It was found that the sealing capability of the labyrinth seal is generated by the centrifugal force exerted on oil particles due to disk rotation when they pass through the seal gap.

Key Words: Labyrinth Seal, HDD, Seal Performance, Leakage of Lubricant, Centrifugal Force

1. Introduction

Recently, hard disk drives (HDDs) have been required to achieve higher reliability with dramatic increase in capacity. Fatal problems such as head crashes have to be avoided. It is suggested that leakage of small oil particles from the ball bearing lubricant is one of the principal causes of head crashes. Magnetic fluid seals and non-contact labyrinth seals are often used to prevent oil particles from scattering on to disk surfaces. However, at speeds greater than 10,000 [rpm], the magnetic fluid cannot maintain the meniscus between the rotational and stationary parts because the centrifugal force exerted on the magnetic fluid breaks the meniscus down. For this reason, it is suggested that it is difficult to use magnetic fluid seals at high speeds. However, the sealing capability of labyrinth seals against leakage of small particles has not been clarified. In addition, the way in which labyrinth seals prevent small particles from passing through the seal gap has not been described. Accordingly, the design of labyrinth seals largely depends on the designer's experience. Therefore, in this paper, the sealing capability of labyrinth seals against leakage of small oil particles is investigated numerically and experimentally. The labyrinth seals treated in this paper have a small seal gap, of the order of tens of microns, and two shapes for the seal gap are discussed. One is a straight-seal gap and the other is a bent-seal gap. The influence of several design parameters on sealing performance is examined theoretically and an experiment is conducted to verify the theoretical predictions.

2. Nomenclature

\[ N : \text{rotational speed of disk [rpm]} \]
\[ h_0 : \text{seal gap [m]} \]
\[ l : \text{seal length [m]} \]
\[ r_0 : \text{shaft radius [m]} \]
\[ d_p : \text{diameter of oil particles [m]} \]
\[ m_p : \text{mass of oil particles [kg]} \]
\[ p_i : \text{pressure of inner seal area [Pa] (position 1: see Fig. 1)} \]
\[ p_o : \text{pressure of outer seal area [Pa] (position 2)} \]
\[ u_{a r}, u_{a \theta}, u_{a z} : \text{velocity of airflow in } r, \theta, z \text{ directions [m/s]} \]
\[ u_{p r}, u_{p \theta}, u_{p z} : \text{velocity of oil particles in } r, \theta, z \text{ directions [m/s]} \]
\[ U_{rel} : \text{relative velocity between airflow and oil particles [m/s]} \]
\[ \mu : \text{viscosity of air [Pa s]} \]
\[ \nu : \text{kinematic viscosity of air [m}^2/\text{s]} \]
\[ \omega : \text{angular velocity of rotational disk [rad/s]} \]
3. Experimental Results of Pressure Inside a HDD Housing

3.1 Experimental apparatus and methods

Figure 1 shows the experimental apparatus for measuring pressures inside a HDD housing. A spindle motor with two disks for a 3.5 inch HDD was used for this experiment. The rotational part containing the disks was supported by two ball bearings located in the HDD housing. Two types of labyrinth seal were prepared, one a straight-type and the other a bent-type seal. Geometrical cross-sectional views of these seals are shown in Fig. 2. Both consisted of an outer and an inner seal wall installed over an upper ball bearing. The outer seal wall was a press-fit to the rotational part. The inner seal wall was attached to the shaft and could be detached to allow the seal gap to be varied by changing the outer diameter of the inner seal wall. To clarify the sealing mechanism of labyrinth seals against leakage of small oil particles it is necessary to know the airflow in the HDD housing. Therefore, pressures at five positions in the housing, as shown in Fig. 2, were measured by a semiconductor pressure gauge. Pressure, $p_i$, in the small space (Position 1) between the labyrinth seal and the ball bearing was measured through a hole drilled in the shaft. In the experiment, the rotational speed and the seal gap were varied, as shown in Table 1. The distance between the upper disk and the housing roof was 3 mm, and between the lower disk and the bottom of the housing was 2 mm in this apparatus. The distance from the outer diameter of the disk to the circular shroud was 1 mm.

3.2 Experimental results

Figure 3 shows the variations of pressure at measuring positions from 1 to 5 when the disks rotate. In this figure, all pressures are indicated by subtracting the pressure at Position 4. As seen in Fig. 3, the pressure at each measuring position becomes constant about 7 s after the disks start rotating. In addition, the pressures at measuring positions decrease in the order of Positions 3, 4, 5, 1 and 2, with the pressure at Position 2 becoming the minimum. Therefore, it is concluded that air in the HDD housing flows in the direction of Positions 3, 4, 5, 1 and 2, and is circulated in the housing.

Figure 4 shows the pressure difference, $\Delta p$, between the inside (Position 1) and the outside (Position 2) of the labyrinth seal when the seal gap and the rotational speed are changed. $\Delta p$ increased parabolically with increase in rotational speed. $\Delta p$ becomes larger as the seal gap is de-
creased because the viscous resistance of the seal gap is increased. At a rotational speed of \( N = 13000 \text{ [rpm]} \) and a seal gap of \( h_0 = 20 \text{ [\mu m]} \), the pressure difference, \( \Delta p \), reaches 120 \text{ [Pa]}.

4. Measurement of the Number of Small Oil Particles Flowing out from the Labyrinth Seal

From the measured pressures in the HDD housing, it was found that air in the housing flows through the ball bearings and then flows from the inside to the outside of the labyrinth seal. Therefore, it is considered that small oil particles generated in ball bearings are carried out from the ball bearings and the seal with this airflow. In the experiment, the number of small oil particles passing through the seal was counted by a laser particle counter. The size of particles measured was 0.3, 0.5, 1.0 and 3.0 [\mu m] diameter.

4.1 Experimental apparatus and the measuring method

Figure 5 shows the experimental apparatus for measuring the number of small oil particles flowing out from the seal. The HDD housing and the particle counter were set in a class 100 clean bench. Measurement of the number of small oil particles by the counter was carried out by sucking air from the top of the housing while replacement clean air was fed from a small hole formed at the bottom of the housing. Measurements were repeated three times for the same conditions, and the average number of small particles in a one cubic foot volume was obtained. Experimental conditions are shown in Table 2.

4.2 Experimental results

Figure 6 shows the relationship between the rotational speed and the number of particles when the labyrinth seal is detached. The number of particles increases with rotational speed for diameters of 0.3, 0.5 and 1.0 [\mu m]. For example, the number of particles of 0.3 [\mu m] diameter is about 10 million per cubic foot at a rotational speed of 13000 [rpm]. However, the number of particles of 3.0 [\mu m] diameter decreases with increasing rotational speed. This is because the influence of the centrifugal force due to disk rotation on the particle locus becomes larger for larger particles, causing the particles to tend to hit against the outer seal wall and become attached to it.

Figure 7 shows the relationship between the pressure difference and the number of particles with a diameter of 0.3 [\mu m] passing through the labyrinth seal when the seal gap is changed between the straight-type and the bent-type seal. Figure 7 shows that the number of particles is greatly reduced as the seal gap becomes smaller. There is only a small difference in the number of particles between the two seal types. It can be concluded that reducing the seal gap in a straight-type seal is more effective for preventing small particles from flowing out from the seal than using a bent-type seal.

5. Numerical Calculations

5.1 Numerical calculation method

As mentioned above, it is considered that small oil particles are carried out from the labyrinth seal by airflow generated by disk rotation. Therefore, to theoretically estimate the sealing performance, the airflow stream lines near the labyrinth seal need to be obtained. In this
paper, the airflow stream lines are numerically obtained by using CFD (Computational Fluid Dynamics). Figure 8 shows the calculated area near the labyrinth seal. We treat only a straight-type seal because there was little difference between the experimental performances of the two seal types. We term the square space, of 2 [mm] length and 1.5 [mm] height, the inner seal region.

In the numerical calculations, the following assumptions were made:

1. Airflow in the seal gap and the inner seal region is incompressible laminar flow. In addition, airflow in the seal gap is Poiseuille flow in the axial direction and Couette flow in the circumferential direction;

2. The shape of small oil particles is spherical and the force acting on a particle by the airflow is subjected to Stokes drag law. In addition, the centrifugal force in the radial direction and the Coriolis force in the circumferential direction also act on the particle;

3. The shape of the labyrinth seal is axially symmetric and a pressure gradient does not exist in the circumferential direction.

In numerical calculations, two-dimensional Navier-Stokes equations and the equation of continuity in the cylindrical coordinate system are used. These equations are solved by the MAC (Marker and Cell) method, which is one of the calculation methods for incompressible laminar flow.

5.2 Equations of motion for small oil particles

By applying Stokes Law, the drag force against small oil particles with diameter \(d_p\) acted on by the airflow can be given as follows:

\[
D = |D_r, D_\theta, D_z|^T = -6\pi \cdot \mu \cdot (U_{rel}) \frac{d_p}{2}
\]  

Since the centrifugal force acts in the radial direction, and the Coriolis force in the circumferential direction, we get,

\[
F_r = m_p \frac{v_p^2}{d_p/2}, \quad F_\theta = -2m_pu_p \frac{v_p}{d_p/2}
\]

After the velocity distribution of airflow in the inner seal region is determined, the loci of oil particles can be obtained by integrating the equations of motion twice.

5.3 Results of numerical calculations

Figure 9 (a), (b) and (c) show calculated results obtained by solving CFD and the equations of motion. Figure 9 (a) shows the velocity distribution in the inner seal region at \(N = 10000 \text{ [rpm]}\) and \(h_0 = 60 \text{ [\mu m]}\). From these results, there exist two eddies near the upper and the lower walls, which are generated by the radial airflow due to disk rotation.

Figure 9 (b) shows the loci of oil particles with a diameter of 0.3 [\mu m]. The number of each locus of oil particles indicates the position of particles at the end seal of a ball bearing, from where they flow out as shown in Fig. 8. The end seal gap of the ball bearing is divided into 10 equal parts with numbers ranging from 1 to 9. The calculated results show that oil particles flowing out from Positions 1, 2, 3 and 9 are trapped in the inner seal region and

Fig. 8 Schematic views of labyrinth seal

Fig. 9 Calculated results (\(h_0 = 60 [\mu m], 10000 \text{ [rpm]}\))
cannot enter the labyrinth seal gap.

Figure 9 (c) shows the relationship between the positions of particles flowing out from the end seal gap of a ball bearing and flowing into the labyrinth seal gap. The labyrinth seal gap is also divided into 10 equal parts. Particles flowing out from Positions 6 and 7 of the end seal enter the labyrinth seal gap at almost the same positions. In addition, particles at Positions 4 and 5 of the end seal enter the labyrinth seal gap at lower position numbers, and particles at Position 8 enter it at higher position numbers.

Figure 10 shows the positions of the end seal gap where particles enter the labyrinth seal gap when the labyrinth seal gap and the rotational speed are changed. Particles in the shaded area can enter the labyrinth seal gap. The extent of positions where particles can enter the labyrinth seal gap becomes narrower as the labyrinth seal gap is decreased, but it is insensitive to rotational speed.

In actual labyrinth seals, particles move in three-dimensional directions but if we view particles at some plane fixed in the circumferential direction, the loci of particles can be expressed as shown in Fig. 11. Figure 11 shows two-dimensional loci of oil particles in the labyrinth seal gap when particles enter the gap from different positions. The vertical axis is the axial coordinate of the seal gap and the horizontal axis is the radial coordinate. Particles move from the bottom to the top.

As mentioned previously, centrifugal force acts on the oil particles due to disk rotation. As a result, particles gradually move to the right hand side and are trapped by the outer seal wall. Particles at Positions 5 to 7 are trapped by the outer seal wall. However, particles at Positions 1 to 4 pass through the seal gap. Consequently, Position 5 is the boundary position for trapping particles by the labyrinth seal.

Figures 12, 13 and 14 show the relationship between
the pressure difference and the boundary position for trapping particles when the rotational speed, the diameter of particles and the labyrinth seal gap are changed, respectively. The sealing performance becomes higher as the line shifts to the right and in a downward direction. These figures show that the sealing performance of labyrinth seals against leakage of small particles can be improved by increasing the rotational speed and decreasing the labyrinth seal gap.

6. Comparison with Experimental Results

6.1 Passing rate of particles

In order to confirm the validity of numerical predictions, experimental results are compared with numerical ones. The new parameter of passing rate, \( P_r \), is introduced in this paper to compare these results. It is defined as the ratio of the number of particles passing through the seal to the number of all particles generated from ball bearings when the seal is detached. \( P_r \) is theoretically expressed by the following equation, assuming that particles are uniformly distributed in the airflow:

\[
P_r = \frac{q_s}{q_{s0}} \cdot \frac{P_b \cdot (R_f - R_i)}{(R_o - R_i)}
\]

where,
- \( q_{s0} \) : is the volume flow rate without a labyrinth seal
- \( q_s \) : is the volume flow rate with a labyrinth seal
- \( P_b \) : is the ratio of the number of particles entering the labyrinth seal gap to the number of particles flowing out from the ball bearing. \( P_b \) is given as follows:

\[
P_b = \frac{(B_o - B_i)}{10}
\]

- \( B_o \) : is the higher number of the boundary position of particles at the end seal of a ball bearing that can enter the labyrinth seal gap
- \( B_i \) : is the lower number of the boundary position of particles at the end seal of a ball bearing that can enter the labyrinth seal gap (see Figs. 9 (b) and 10)
- \( R_o \) : is the number of the position at the inlet of the labyrinth seal gap corresponding to the higher number of the particle position at the end seal gap where particles can enter the labyrinth seal gap
- \( R_i \) : is the number of the position at the inlet of the labyrinth seal gap corresponding to the lower number of the particle position at the end seal gap where particles can enter the labyrinth seal gap (see Fig. 10)
- \( R_f \) : is the boundary position at the inlet of the labyrinth seal gap where particles can be trapped by the outer seal wall (see Fig. 14)

The values for obtaining the passing rate, such as \( q_{s0}, q_s, \cdots \) are calculated from the numerical results.

Figure 15 shows the relationship between the pressure difference and the passing rate of particles with a diameter of 0.3 [\( \mu \text{m} \)] when the seal gap is changed. This figure shows that the passing rate is dramatically decreased with the reduction of the seal gap. The numerical passing rates for \( h_0 = 60 [\mu \text{m}] \) and \( h_0 = 30 [\mu \text{m}] \) show good qualitative and quantitative agreement with the experimental ones. However, the numerical passing rate for \( h_0 = 20 [\mu \text{m}] \) cannot be obtained because no particle passes through the seal in numerical calculations although an experimental passing rate was measured. The reason for this discrepancy may be that airflow in the housing is assumed to be steady state in numerical calculations, but in practice non-steady state flow is generated by the whirling motion of rotating parts.

6.2 Analysis of chemical composition of particles attached to the seal wall

The above discussion predicts that the sealing capability of the labyrinth seal for small particles is generated by the centrifugal force exerted on the particles when they pass through the seal gap. In order to confirm the validity of this prediction, the chemical composition of particles attached to the inner and outer seal walls was measured using a Gas Chromatograph-Mass Spectrometer (GC-MS). Before taking the measurement, the inner and outer labyrinth seal walls were carefully cleaned with acetone. The seal gap was 30 [\( \mu \text{m} \)] and the chemical composition of particles attached to the walls was measured after the spindle motor was rotated for 10 minutes.

Figure 16 (a) and (b) show the results for the chemical composition attached to the inner seal wall surface before and after the spindle motor was rotated. There is no difference between the results, which means that no particles were attached to the inner seal wall even after the spindle motor was rotated.

Figure 17 (a) and (b) show the measured results for the outer seal wall. Figure 17 (c) shows the chemical composition of the lubricant for ball bearings for reference. Figure 17 (b) shows different peaks to those in Fig. 17 (a). However, peaks in Fig. 17 (b) are very similar to those in Fig. 17 (c). This means that small oil particles are attached to the outer seal wall. Thus, the sealing capability for particles is generated by the effect of the centrifugal force exerted on particles due to disk rotation.
7. Conclusions

In this paper, the sealing performance of labyrinth seals for small oil particles is investigated numerically and experimentally. The following conclusions are drawn:

1. From the results of pressure measurements in the HDD housing, air in the housing is circulated in a direction from the inside of the seal to the outside. The pressure difference between the inside and the outside of the seal is of the order of tens of Pa.

2. The sealing capability of a labyrinth seal is generated by the centrifugal force exerted on oil particles when they pass through the seal gap.

3. The numerical passing rate for \( h_0 = 60 \mu m \) and \( h_0 = 30 \mu m \) show good agreement with experimental results. Therefore, the calculation method presented in this paper is effective for estimating the sealing performance of labyrinth seals against small oil particles.

References

