Study on the capacity performance of journal bearings with different spiral groove structures by CFD method

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Abstract. This paper uses the computational fluid dynamics (CFD) method to evaluate the capacity performance of journal bearings with different spiral groove shapes. A detail discussion is conducted to compare the capacity performance of spiral groove journal bearings with different spiral groove depths, widths, numbers, and pitches. For journal bearings with different spiral groove structures, the variation of the load carrying capacity and the oil film pressure with different attitude angles is illustrated. The influence of the spiral groove structures on the pressure distribution of journal bearings is investigated. The present research can provide technical support and references to the design and the capacity performance of journal bearings with the spiral grooves.

Keywords: Journal bearings; Spiral groove; Load carrying capacity; Structure design

1. Introduction

Journal bearings are widely used in rotating machinery due to their stable operation, good vibration resistance, high bearing capacity. Generally, the bearing structure affects the bearing capacity performance. With the development of industry, the demand for journal bearings with high capacity performance urgently needs to be met. It is necessary to design reasonable structures to improved the bearing capacity performance.

Sing et al. [1] found that a reasonable groove shape can greatly improve the bearing performance. Sharm et al. [2-3] compared the capacity performance of bearings with different groove shapes and the results showed that the bearing with the spiral groove has better capacity performance. C.M. Rodkiewick et al. [4] conducted experiments on the influence of the groove structure on the bearing performance and concluded that the inclined groove structure can improve the bearing capacity. Yoshio et al. [5-6] compared the pressure distribution of bearings with the groove structure and smooth-surface bearings under the same working conditions. It is concluded that the comprehensive performance of bearings with the groove structure is better than that of smooth-surface bearings. In
addition, V.O. Dzyura et al. [7] found that the manufacturing accuracy and surface roughness between the groove structures will also affect the carrying loading capacity and the service life of journal bearings.

In this paper, the capacity performance of journal bearings with the spiral groove is studied. The influence of the spiral groove shape on the bearing capacity is discussed by designing different spiral groove depths, widths, numbers, and pitches. This can provide technical support for the reasonable selection of the structure parameters of the spiral groove bearings.

2. Theory
2.1. Governing equations
The basic equations are used to solve all flow problems as follows:

Mass conservation equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0$$ (1)

where \( \rho \) indicates the fluid density and \( \mathbf{v} \) is the fluid velocity vector.

Momentum conservation equations

$$\frac{\partial (\rho \mathbf{v})}{\partial t} + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla p + \nabla \cdot (\tau) + \rho \mathbf{g} + \mathbf{F}$$ (2)

where \( p \) means the static pressure, \( \tau \) is the stress tensor, \( \rho \mathbf{g} \) and \( \mathbf{F} \) are the gravitational body force and external body forces respectively. The stress tensor \( \tau \) is defined as follows:

$$\tau = \mu \left[ (\nabla \mathbf{v} + (\nabla \mathbf{v})^T) - 2/3 \nabla \cdot \mathbf{v} \mathbf{I} \right]$$ (3)

where \( \mu \) is the fluid viscosity, \( \mathbf{I} \) represents the unit tensor.

2.2. The load carrying capacity
The load carrying capacity can be evaluated by integrating the pressure over the journal surface, which is defined as follows:

$$F_x = -\int_{-L/2}^{L/2} \int_{-z/2}^{z/2} p \cos \phi R \mathbf{d} \phi \mathbf{d} z$$ (4)

$$F_y = -\int_{-L/2}^{L/2} \int_{-z/2}^{z/2} p \sin \phi R \mathbf{d} \phi \mathbf{d} z$$ (5)

$$F = \sqrt{F_x^2 + F_y^2}$$ (6)

where \( F \) is the load carrying capacity, \( L \) is the bearing length, \( R \) is the radius of the journal. Then, the attitude angle is calculated as:

$$\theta = \arctan \left( \frac{F_y}{F_x} \right)$$ (7)

3. Simulation
Due to the complex structure of the spiral groove bearing, the full fluid field simulation calculation is applied to solve the complex fluid movement by ANSYS FLUENT. The simulation model includes three parts: circular fluid field, spiral groove fluid field, and oil inlet. The top of the oil inlet is set as the fluid inlet. Two end faces of the circular fluid field and spiral groove fluid field are the fluid outlets. The inner surface of the circular fluid field is set as the rotating surface. The rotational speed is set as 16103 rpm. In the simulation model, different spiral groove structure parameters including the depth, width, number and pitch are chosen to compare the capacity performance of journal bearings. The range of the spiral groove depth is 0.2-0.4 mm. The spiral groove width is designed in the range of 2-3 mm. The spiral groove numbers are set to 1, 2, 3 respectively. Single spiral groove pitch and 0.5 times the pitch are applied in the geometry model. Then, the geometry model is meshed by the hexahedral grid. The grid number is 4732994 and the node number is 6918282. With this grid density, the fluid
field calculation can achieve stable convergence. The simulation model is shown in Figure 1. The simulation parameters are listed in Table 1.

For the numerical calculation, cavitation phenomenon is modelled by Rayleigh-Plesset model for its stability and liable to convergence. For the cavitation of oil-lubricated journal bearing is mostly caused by the escape of dissolved air. The cavitation pressure of lubricating oil is set to 0 Pa. The bearing material is aluminum.

### Table 1. Simulation parameters.

| Parameter                  | Value         |
|----------------------------|---------------|
| Bearing diameter (mm)      | 35            |
| Bearing length (mm)        | 108           |
| Spiral groove depth (mm)   | 0.2-0.4       |
| Spiral groove width (mm)   | 2-3           |
| Spiral groove number       | 1-3           |
| Spiral groove pitch (mm)   | 45            |
| Relative clearance         | 1.5‰          |
| Rotational speed (rpm)     | 16103         |

Figure 1. The simulation model

4. Results and discussion

4.1. The influence of the spiral groove depth on the bearing capacity

The influence of the spiral groove depth on the bearing capacity is studied with the minimum oil film thickness of 5 μm. The spiral groove depth is set to 0.2 mm, 0.25 mm, 0.3 mm, 0.35 mm, and 0.4 mm. As shown in Figure 2(a), the results show that with the increase of the spiral groove depth, the bearing capacity gradually decreases. The maximum carrying force decreases from 16731 N to 16532 N, which declines by 1.2%. The maximum carrying force occurs when the attitude angle is 42° or 45°. The relationship between the oil film pressure, the spiral groove depth, and the attitude angle is shown in Figure 2(b). The oil film pressure shows an uptrend when the attitude angle is in the range of 33°-51°. Besides, the oil film pressure decreases with the increase of the spiral groove depth. When the attitude angle is 51° and the spiral groove depth is 0.2 mm, the maximum oil film pressure is obtained. When the attitude angle is 33° and the spiral groove depth is 0.4 mm, the oil film pressure reaches the minimum value. Therefore, when the spiral groove depth is 0.2 mm, the bearing at the 42° attitude angle shows good capacity performance.
4.2. The influence of the spiral groove width on the bearing capacity

Figure 3(a) shows the relationship between the attitude angle and the bearing carrying force under different spiral groove widths. The spiral groove width is set to 2 mm, 2.25 mm, 2.5 mm, 2.75 mm, and 3 mm. It can be found that the bearing capacity gradually decreases with the increase of the spiral groove width. The maximum carrying force decreases from 16532 N to 16301 N, which declines by 1.4%. The maximum carrying force occurs when the attitude angle is 42° or 45°. Figure 3(b) shows the oil film pressure in the range of 33°-51° for the bearing attitude angle when the spiral groove width is 2 mm, 2.25 mm, 2.5 mm, 2.75 mm, and 3 mm respectively. The oil film pressure indicates a rising trend when the attitude angle is in the range of 33°-51°. The oil film pressure decreases with the increase of the spiral groove width. When the attitude angle is 51° and the spiral groove width is 2 mm, the oil film pressure reaches the maximum value. When the attitude angle is 33° and the spiral groove depth is 3 mm, the minimum oil film pressure is obtained. The results show that when the spiral groove width is 2 mm and the attitude angle is 45°, better bearing capacity performance is obtained.

4.3. The influence of the spiral groove number on the bearing capacity

To study the influence of the spiral groove number on the bearing capacity performance, the spiral groove number N is set to 1, 2, 3. Figure 4(a) shows that when the spiral groove number increases from 1 to 2 to 3, the corresponding maximum carrying force decreases from 16478 N to 8934 N to 5133 N. The maximum carrying force of double spiral groove is decreased by 45.8% compared with that of single spiral groove. The maximum carrying force of three spiral groove is decreased by 68.8% compared with that of single spiral groove. As shown in Figure 4(b), when the spiral groove number is 1, 2, 3, the maximum oil film pressure is 31.4MPa, 21.3MPa, 13.7 MPa respectively. The maximum oil film pressure of double spiral grooves is decreased by 32.2% compared with that of...
single spiral groove. While the maximum oil film pressure of three spiral grooves is decreased by 56.4% compared with that of single spiral groove. Therefore, the bearing capacity can be improved by decreasing the spiral groove number. The oil film pressure distribution under different spiral groove numbers is shown in Figure 5. It can be investigated that after the fluid field is divided by the spiral grooves, the oil film pressure in each bearing carrying load region is evenly distributed, and the low pressure region is fully expanded.

**Figure 4.** The influence of the spiral groove number on the bearing capacity: (a) the bearing carrying force; (b) the oil film pressure.

**Figure 5.** The oil film pressure distribution under different spiral groove numbers: (a) single spiral groove; (b) double spiral grooves; (c) three spiral grooves.

4.4. **The influence of the spiral groove pitch on the bearing capacity**

Three spiral groove bearing structures are established to discuss the influence of the spiral groove pitch on the bearing capacity performance. Three bearing structures are single spiral groove with single pitch, double spiral groove with single pitch, and single spiral groove with 0.5 times the pitch. Figure 6(a) shows the relationship between the carrying force and the offset angle under different spiral groove pitches. It can be observed that when the spiral groove number is same, the carrying force decreases with the decrease of the pitch. The single spiral groove structure with single pitch at 42° attitude angle has the maximum carrying force of 16478N. While the double spiral groove structure with single pitch at 30° attitude angle has the maximum carrying force of 8934 N. When the pitch is decreased by 0.5 times, the carrying force of single spiral groove bearing is 10777 N, which is decreased by 34.6% compared with that of the single pitch. As shown in Figure 6(b), the maximum oil film pressure of 31.4MPa is obtained when the bearing structure is the single spiral groove with single pitch. The maximum oil film pressure of the double spiral groove with single pitch is 21.3MPa. The maximum oil film pressure of the single spiral groove with 0.5 times the pitch is 22.9MPa. Compared with the single spiral groove with single pitch, the maximum oil film pressure of the double spiral groove with single pitch and the single spiral groove with 0.5 times the pitch is declined by 32.2% and
27.1%, respectively.

Figure 6. The influence of the spiral groove pitch on the bearing capacity:
(a) the bearing carrying force; (b) the oil film pressure.

The oil film pressure distribution under different spiral groove pitches is shown in Figure 7. The spiral groove divides the high oil film pressure zone obviously. Because different pitches correspond to different spiral angles, therefore the spread form of the high oil film pressure zone is significantly affected by the spiral angles of the spiral grooves. For the single spiral groove with 0.5 times the pitch, the area of the high oil film pressure zone is larger than that of the single spiral groove with single pitch.

Figure 7. The oil film pressure distribution under different spiral groove pitches:
(a) single spiral groove with 0.5 times the pitch; (b) single spiral groove with single pitch.

5. Conclusion
In this paper, the influence of the spiral groove bearing structure on the bearing capacity performance is discussed. The main conclusions are as follows:

(1) With the increase of the spiral groove depth, the bearing capacity gradually decreases. When the spiral groove depth is 0.2 mm, the bearing at the 42° attitude angle shows good capacity performance. Besides, with the increase of the spiral groove width, the bearing capacity gradually decreases. When the spiral groove width is 2 mm and the attitude angle is 45°, better bearing capacity performance is obtained.

(2) As the spiral groove numbers increase, the maximum carrying force and maximum oil film pressure decreases and the bearing capacity becomes worse. The oil film pressure distribution is obviously divided by the spiral grooves. The area of the high oil film pressure zone shrinks significantly.

(3) As the spiral groove pitches increase, the maximum carrying force and maximum oil film pressure increases, which shows good bearing capacity performance. The area of the high oil film pressure zone is also expanded.
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