Design of Gas-Magnetic Hybrid Bearing for Gyro Motors

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Abstract

A kind of gas-magnetic hybrid bearing for the application of gyro motor is proposed to solve the problem of friction and wear,when the gas bearing gyro motor is starting and stopping. The bearing is combined with the advantages of two pairs of symmetrical permanent magnetic bearings and the H-type gas bearings. It can effectively improve the reliability and service life of the gyro motor, which can be realized in the non contact state. An analytical formula for the gas-magnetic hybrid bearing is presented for calculating the axial and radial stiffness. The single bearing capacity of the permanent magnetic bearing is compared with the gas bearing on the mixed bearing. The result shows that, it can be used as an auxiliary supporting to force, because the supporting force providedby the permanent magnet bearings is less the aerodynamic bearingson the high-speed operation state. In the stop state or the state of low speed, the supporting force providedby the permanent magnet bearings can overcome its own gravity, and the problem of the dynamic pressure bearing on inthe dry friction conditions is solved.

Keywords

Gas-magnetic hybrid bearing, gyro motor, non-contact bearing.

1. Introduction

The bearing is the core component of the gyro motor.The precision and life of the gyro motor depend on the high-speed rotating bearing. In order to realize the gyro motor rotating aroundhigh-speed and reliable, it is necessary to solve the problem of friction and wear of bearing support for the reliability of the gyro motor. At present, the bearingson the gyro motor are the ball bearing and the gas bearing. The former area kind of contact bearing, which is usually used in the gyroscope motor with low speed. The latter a the first choice of high speed motor, but these bearings have not enough to support force at low speed, especially when the motor starts and stops, which will cause dry friction. This friction will destroy the accuracy of the bearing surface, meanwhile bring pollutants to lead its performance and life is affected.

Permanent magnetic bearing is a kind of non-contact bearing. It is more noticed because of its simple structure and the function of non-contact supports at any time. However, limited by the performance of permanent magnetic materials, the permanent magnet bearing system has poor stability and stiffness, and it can not meet the requirements of gyro motor. At present, it is no practical engineering application and it is still at the experimental stage.

In view of the shortage of the bearings used in the gyro motor, a kind of gas-magnetic hybrid bearing applied to the gyro motor is proposed to this paper. The bearing combines the advantages of gas bearing and permanent magnetic bearings, and two pairs of permanent magnetic bearings are installed on the basis of the gas bearing. It solves the problem of friction and wear in the process of starting and stopping of gyro motor. At the same time, the permanent magnetic bearing and the gas bearingswork together to improve the stability of the systemin high-speed operation. Besides,this article applied the analytical designmethod to derivethe mathematical model of radial support and axial support of the bearing of gas and magnetic mixed. The parameters of the model were then analyzed and the proper parameter design of non-contact start-stop was provided. Finally, the model was applied to compare and validate with the example calculation.

2. Basic Structure and Principle

The gas-magnetic hybrid bearing consists of two parts:the gas bearing and the permanent magnetic bearing. The structural representationis shown in figure 1. The working principle and medium of the two kinds of bearings is different, and they can work independently and jointly.

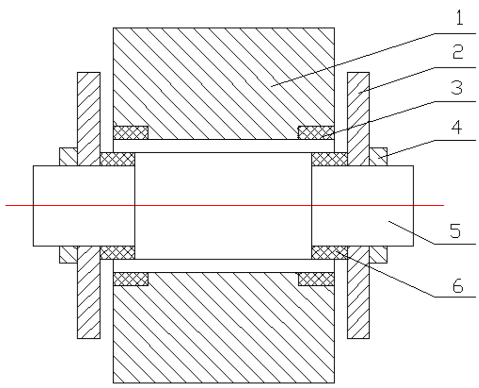

Fig 1. The structural representation of gas-magnetic mixed bearings

1. Sleeve, 2. Thrust plate, 3. Permanent magnetic ring, 4. Nuts, 5. Axes, 6. Permanent magnetic ring

On the basis of the gas bearing, two pairs of identical permanent magnets include inner and outer are installed at both ends of the shaft and the shaft sleeve, respectively. The inner and outer permanent magnetic rings adopt the permanent magnet material. Both the outer surface of the radial directionand the inner permanent magnetic ring permanent magnet rang inner circle surface has the same polarity. The inner and outer permanent magnetic rings are staggered in the axial direction, so they will produce the radial force while the axial force is produced. Due to the bearing structure is complete symmetry, the axis and shaft sleeve will reach the equilibrium position under the action of radial force and axial force. The shaft and the axle sleeve are not contact. When the motor is starting or stopping, the friction and wear of the gas bearing will happen. When the bearing in the high-speed, the function of its structure is equivalent to thepneumatic bearing.

3. The Mathematical Model of Radial Stiffness

3.1 A Mathematical Model of Radial Stiffness for Gas Bearing

According to the calculation method of Osman and Wildman [1], the following a number of steps can be calculated.

(1) Calculating the load W

$$W = \sum_{i=1}^{4} W_i \tag{1}$$

Where W_1 --Rotor gravity.

W_2 --Environmental shock additional force.

W_3 --General demand for additional force.

W_4 --Centrifugal additional force caused by dynamic imbalances.

(2) Calculating bearing parameters Λ

$$\Lambda = \frac{6\mu\omega}{P_a}\left(\frac{R_i}{C}\right)^2 \tag{2}$$

Where P_a --Environmental pressure of bearing.

ω --Speed.

μ --Viscosity of lubricants.

C --Radial unilateral mean gap.

R_i --Journal bearing radius.

(3) According to the literature [1], the end flow coefficient is obtained to calculate the load bearing of the unprovoked leakage axis

$$W_i = \frac{W}{A} \tag{3}$$

Where W_i --Unloaded leaky shaft bearing.

(4) Calculating the load ratio

$$a = \frac{W_{\mathrm{i}}}{P_{\mathrm{a}} l d} \tag{4}$$

Where l --Bearing length

d --Bearing diameter

(5) The eccentricity ξ and the attitude angle ϕ are obtained according to the literature [1].

(6) Calculating radial stiffness K_r

$$K_r = \frac{W_i}{e\cos\phi} \tag{5}$$

Where $e = C\xi$.

3.2 A Mathematical Model of Radial Stiffness for Permanent magnetic Bearings

According to the simplified mathematical model of permanent magnetic bearing established by Yonnet [2], a single pair of permanent magnetic bearing structure schematic diagram is shown in figure 2.

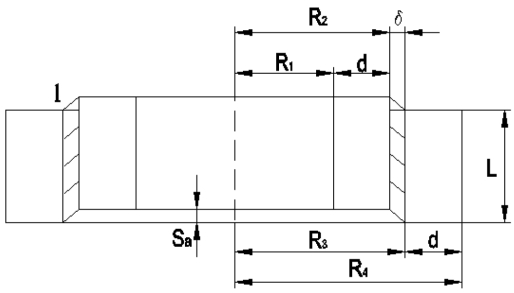

Fig 2. Schematic diagram of single side permanent magnet bearings

A single of permanent magnetic bearing stiffness:

$$K_t^{'} = -\frac{J^2 P}{8\pi\mu_0}\left[2\rho\left(S_a\right) - \rho\left(S_a + L\right) - \rho\left(S_a - L\right)\right] \tag{6}$$

Where

$$\rho\left(z\right) = \ln\frac{\left[\left(2d+\delta\right)^2 + z^2\right]\left(\delta^2 + z^2\right)}{\left[\left(d+\delta\right)^2 + z^2\right]} \tag{7}$$

J --Magnetization of internal and external magnetic rings.

d --Radial thickness of internal and external magnetic rings.

μ_0 --Vacuum permeability.

L --Permanent magnet ring width.

$P = 2\pi\left(R_2 + 0.5\delta\right)$

When S_a is very small, it can be approximated $S_a = 0$.

The two pairs of permanent magnetic bearing stiffness:

$$K_t = -\frac{J^2 P}{2\pi\mu_0}\ln\frac{\delta^2\left(2d+\delta\right)^2\left[\left(d+\delta\right)^2 + L^2\right]}{\left(d+\delta\right)^4\left(\delta^2 + L^2\right)\left[\left(2d+\delta\right)^2 + L^2\right]} \tag{8}$$

3.3 Example

Radial stiffness of gas-magnetic hybrid bearing

$$K = K_r + K_t \tag{9}$$

Where K_t --The radial stiffness of a magnetic bearing

The selection of dynamic pressure bearing design parameters is shown in Table 1.

Table 1. Parameters of dynamic pressure bearing of gas

R_i (mm)	l (mm)	ω ($rad/\min$)	C (mm)	μ ($kg \cdot s/cm^2$)	W (g)	P_a
4	16	300	0.0025	1.85×10^{-10}	700	$1.8P_0$

P_0 --Atmospheric pressure

Calculating: $K_r = 1.5kg/\mu m$

The relationship between the various parameters is discussed by the (10):

$$K_t = -\frac{J^2 P}{2\pi\mu_0}\ln\frac{\delta^2\left(2+\frac{\delta}{d}\right)^2\left[1+\left(\frac{\delta}{d}\right)^2+\left(\frac{L}{d}\right)^2\right]^2}{\left(1+\frac{\delta}{d}\right)^4\left(\delta^2 + L^2\right)\left[\left(2d+\delta\right)^2 + L^2\right]} \tag{10}$$

Where the δ is far less than the d, and the extinction coefficient is equal to:

$$K_t = -\frac{J^2 P}{2\pi\mu_0}\ln\frac{4\delta^2\left[1+\left(\frac{L}{d}\right)^2\right]^2}{\left(\delta^2 + L^2\right)\left[\left(2d+\delta\right)^2 + L^2\right]} \tag{11}$$

By the form (11), the K_t is proportional to the J^2, the relationship of the δ and the K_t is inverse ratio, meanwhile the d and the K_t is direct ratio. So it proves that the overall mixed gas magnetic bearing radial stiffness can be increasedby increasing the radial stiffness of permanent magnet bearings. The method includes increasing the diameter of the magnetic bearing, reducing the gap between the internal and external magnetic rings and selecting the permanent magnetic material with high magnetization. The selection of design parameters for permanent magnetic bearings is shown in table 2.

Tab 2. Parameters of permanent magnetic bearing

$d\,(mm)$	$L\,(mm)$	$\mu_0\,(H/m)$	$\delta\,(mm)$	$J\,(H)$
2	5	$4\pi\times10^{-7}$	0.002	1

Calculating: $K_t = 0.04kg/\mu m$

From this, when the inner and outer sleeve of the bearing is not touched, the maximum magnetic force can support the load of 80grams.If the load is changed, the different parameters are selected to meet the requirements by d , J and δ . According to the actual condition, d and δ will not be large or small. At present, magnetization of DfES about 1H, so the value of K_t is far less than the radial stiffness of a gas dynamic bearing.

4. The Mathematical Model of Axial Stiffness

4.1 A Mathematical Model of Axial Stiffness for Permanent Magnetic Bearings

It can be divided into the following 3 steps for calculation [5].

(1) The gap magnetic conductivity of double cylinder permanent magnetic bearings Λg and the axial deviation $\partial\Lambda g/\partial Sa$.

Inside, the axial length of the magnetic cylinder is L, the radial thickness is d,the radial average gap is δ , and the axial relative displacement of inner and outer cylinder is S_a , the outer radius of the inner magnetic cylinder is R_2 , the inner radius is R_3 ,the outer radius of the outer magnetic cylinder is R_4 ,the inner radius is R_s , then the magnetic line length of the gap between the inner and outer magnetic cylinder is $l=\sqrt{{S_a}^2+\delta^2}$.

The equivalent section of magnetic flux perpendicular to the magnetic line of force

$$S=\frac{2\pi\left(R_2+\delta/2\right)L\delta}{\sqrt{{S_a}^2+\delta^2}} \tag{12}$$

Calculating the Λ by magnetic conductance formula

$$\Lambda=\frac{\mu S}{l} \tag{13}$$

Where the μ is the magnetic permeability.

The gap magnetic conductance is obtained:

$$\Lambda_g=\frac{2\mu_0\pi\left(R_2+\delta/2\right)L\delta}{{S_a}^2+\delta^2} \tag{14}$$

$$\frac{\partial\Lambda_g}{\partial S_a}=\frac{4\mu_0\pi\left(R_2+\delta/2\right)L\delta{S_a}^2}{\left({S_a}^2+\delta^2\right)^2} \tag{15}$$

(2) Magnetic gap magnetic flux ϕ_g

It is assumed that there is no magnetic flux in the gap of magnetic flux.

$$\begin{cases} B_m S_m = B_g S_g \\ H_m L_m = K_r H_g \delta \end{cases} \tag{16}$$

Where: B_m ——Magnetic flux density at the working point of permanent magnet.

H_m ——The magnetic field strength of the permanent magnet working point.

B_g ——Gap flux density.

H_g ——Gap magnetic field strength.

S_m ——The average area of magnetic flux in permanent magnets perpendicular to the direction of the magnetic ring.

S_g ——the gap flux surface.

L_m ——the equivalent length of permanent magnet ring direction.

K_r ——the magnetoresistance coefficient.

The equation [3] form of the air gap load line:

$$\tan\alpha = \frac{B_m}{H_m} = \frac{L_m \Lambda_t}{S_m} \tag{17}$$

$$B_m = \frac{B_r H_c \tan\alpha}{B_r + H_c \tan\alpha} = \frac{B_r H_c L_m \Lambda_t}{B_r S_m + H_c L_m \Lambda_t} \tag{18}$$

According to the principle of flux continuity

$$\Phi_g = B_m H_m = 2\pi R_{pj} L B_m = \frac{2\pi R_{pj} L B_r H_c L_m \Lambda_t}{2\pi R_{pj} L B_r + H_c L_m \Lambda_t} \tag{19}$$

Where $R_{pj} = \frac{(R_2 + R_3)}{2}$

Λ_t ——General magnetic conductance of magnetic path.

B_r ——Residual magnetic induction intensity of permanent magnets.

H_c ——The coercivity of permanent magnets.

$L_m = 2d$

(3) The radial magnetized permanent magnet bearings for two-way axial magnetic force F:

According to the theory of electromagnetic field, the magnetic energy of the gap between the magnetic cylinders is:

$$W = \frac{{\Phi_g}^2}{2\Lambda_g} \tag{20}$$

Base on the method of virtual displacement, the axial magnetic force is:

$$F = \frac{\partial W_g}{\partial Z} = -\frac{{\Phi_g}^2}{2{\Lambda_g}^2} \times \frac{\partial \Lambda_g}{\partial Z} \tag{21}$$

Reduction of (14) (15) (19) into (21)

$$F = \frac{\left[\pi\left(R_2 + R_3\right) \times 2d\Lambda_t\right]^2 S_a {B_r}^2 L}{\left[\pi\mu_r\mu_0\left(R_2 + R_3\right)L + 2d\Lambda_t\right]^2 \cdot 2\pi\mu_0\left(R_2 + 0.5\delta\right)\delta} \tag{22}$$

Axial stiffness:

$$K_A = F/S_a = \frac{\left[\pi\left(R_2 + R_3\right) \times 2d\Lambda_t\right]^2 {B_r}^2 L}{\left[\pi\mu_r\mu_0\left(R_2 + R_3\right)L + 2d\Lambda_t\right]^2 \cdot 2\pi\mu_0\left(R_2 + \delta/2\right)\delta} \tag{23}$$

4.2 Axial Stiffness of Gas Bearing

According to the reference, the axial stiffness of gaseous dynamic pressure bearing will be calculated by using the method and data proposed in it [4]

$$\frac{r_0 - r_b}{r_0 - r_i} = 0.72 \tag{24}$$

Where r_0 ——The outer diameter of the thrust plate.

r_b ——The radius of the base circle.

r_i ——Inner diameter of thrust plate.

Calculate the load-caring ability of thrust plate bearing:

$$W = 0.75\pi\mu\omega\left(r_0 - r_i\right)^2\left(r_0 + r_i\right)^2 E_f S_a^{-2} \tag{25}$$

Axial stiffness

$$K_B = 0.75\pi\mu\omega\left(r_0 - r_i\right)^2\left(r_0 + r_i\right)^2 D_f S_a^{-3} \tag{26}$$

The method and experimental data provided by the document [4] are obtained D_f and E_f .

4.3 Example

The axial stiffness of a magnetic bearing can be expressed in an analytical manner. The axial stiffness of the gas bearing should be solved by Wheatley parameters E_f and D_f the simple analytical formula is shown as expression (26). Considering the limit condition, the stiffness of the gas and magnetic hybrid bearing can be simplified as the expression:

$$K = K_A + K_B \tag{27}$$

When One side of the permanent magnetic bearing is $S_a = 0$, the other side has the maximum stiffness, and the comprehensive stiffness is K_A. At this time, the dynamic bearing can reach the maximum stiffness.

Citing the data in the document [5] for calculation:

$R_1 = 10.7mm$, $d = 2.5mm$, $L = 8.1mm$, $S_a = 0.8mm$, $B_r = 0.8T$, $\mu_r = 1.05$

$$K_A = \frac{6718.6}{\left(2329.56 + S_a^{\ 2}\right)^2} = 1.24Kg/\mu m$$

Citing the data in the document [4] for calculation:

Thrust plate bearing radius $r_0 = 18$, $r_i = 11$, the speed $\omega = 30000r/\min$, the maximum stiffness $D_f = 0.198$,and the maximum load $E_f = 0.088$, $K_B = 7.17Kg/\mu m$

The calculation can be obtained K_B is about 6 times K_A in the limit case, the support provided by the permanent magnetic bearing is 4.96kg.

5. Conclusion

The conclusions of this paper are provided as follows:

1. The radial support of the permanent magnetic bearing are far smaller than that of the acting gas bearing, which indicates that the permanent magnetic force is more suitable as the auxiliary radial supports.

2. The axial stiffness of the permanent magnetic bearing is smaller than that of the acting gas bearing, thus the acting gas bearing should be mainly considered in design at the same order of magnitude.

3. The radial and axial stiffness of the bearing of gas and magnetic mixed is superior to that of a similar size of acting gas bearing, particularly the advantage of the axial stiffness is more obvious.

4. The bearing of gas and magnetic mixed could support the rotor weight radially and axially when the acting gas bearing is not working, which resolves the dry friction problem of the starting and stopping process of the acting gas bearing effectively.

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Industrialization and Application of High Pressure Piston Pump for Seawater Desalination.

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