2021

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ABSTRACT

Based on the average Reynolds' equation of hydrodynamic lubrication, the axis locus and the minimum oil film thickness of the journal bearings under the dynamic load were numerically solved in this paper, and the movement and bearing characteristics of the compressor bearings at different rotation speeds were analyzed to evaluate the influence of the friction loss of the journal bearings on the performance of the rotary compressor. The simulation results indicated that the minimum oil film thickness of the sub bearing is smaller than the critical oil film thickness at low rotation speed ($n \leq 1800$ rpm), therefore, the friction power increases significantly with the decrease of rotation speed. The effects of width-diameter ratio, clearance and viscosity of lubricating oil on the improvement of minimum oil film thickness were further analyzed. By optimizing the width-diameter ratio of the bearing, the load carrying capacity of the oil film is improved, and the friction power of sub bearing reduces by more than 80%. The experimental results showed that the performance of rotary compressor can be improved by more than 1% at low rotation speed.

1. INTRODUCTION

The main and sub bearings are the supporting mechanism of the crankshaft of the rotary compressor. Their lubrication and friction have a critical impact on the performance and reliability of the compressor. Therefore, many scholars and engineers have conducted a lot of research on them. Okada and kuyama (1982) solved the motion and lubrication of rolling piston based on the simplified Reynolds' equation, and compared the influence of structural parameters of eccentric bearing on friction power. Hirayama et al. (2006) coupled the modified Reynolds' equation and the elastic contact equation in consideration of surface roughness and elastic deformation of the bearing of the rotary compressor, and analyzed the influence of the surface roughness on the lubrication characteristics of the bearing. Ito et al. (2010) conducted a numerical analysis of the rolling piston rotation based on four kinds of friction forces acting on the rolling piston, in which the oil film viscous force between the piston and the crankshaft was solved according to the Reynolds' equation. Matsui et al. (2010) established an analysis model of mixed lubrication of reciprocating compressor bearings, calculated the loss caused by oil viscosity and solid contact pressure, and designed a new bearing which can reduce its friction loss by 20%. Mi and Meng (2016) numerically analyzed the thermodynamic lubrication of rolling piston and two journal bearings in rotary compressors based on the transient energy equation of viscous fluids, and compared the results with isothermal analysis. The results show that the difference of lubrication performance between the thermal and isothermal cases is relative small, due to the mild temperature rise in the bearing system. Fu et al. (2016) evaluated the lubrication state of the bearings by measuring the electrical resistance between lubricated surfaces. The research shows that the lubrication state in the compressor using high viscosity oil is better than that using low viscosity. Kong et al. (2016) proposed to use contact stress to evaluate the wear condition of the bearings in rotary compressor, and analyzed the influence of bearing height, diameter and clearance on the wear through finite element simulation and accelerated life test.

The performance of compressor at low speed (n ≤ 1800 rpm) has a great influence on the APF of air conditioner, while the operation at low speed often results in poor lubrication of the bearing and high friction power of the bearings. Based on the average Reynolds' equation of hydrodynamic lubrication, the axis locus and the minimum oil film thickness of the compressor bearings under dynamic load are numerically solved. The movement and lubrication characteristics of the bearings under different rotating speeds are analyzed. An improvement scheme is proposed to reduce the friction power of the compressor.

2. ANALYSIS MODELS AND CONDITIONS

2.1 average Reynolds' equation of hydrodynamic lubrication

Based on the Reynolds' equation of average flow model (Pair and Cheng, 1978) and the contact factor (Wu and Cheng, 1989), the basic control equation of hydrodynamic lubrication in journal bearing is expressed as:

$$\frac{\partial}{r^2\partial\theta}\left(\varphi_\theta h^3\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial y}\left(\varphi_y h^3\frac{\partial p}{\partial y}\right)=\frac{6\mu(U_\mathrm{j}+U_\mathrm{b})\varphi_\mathrm{c}}{r}\frac{\partial h}{\partial\theta}+\frac{6\mu(U_\mathrm{j}-U_\mathrm{b})\sigma}{r}\frac{\partial\varphi_\mathrm{s}}{\partial\theta}+12\mu\varphi_\mathrm{c}\frac{\partial h}{\partial t} \tag{1}$$

Where, h is the nominal oil film thickness (smooth surface), r is the radius of the bearing, p is the oil film pressure, μ is the lubricant viscosity, U_j and U_b are the surface velocities of journal and bearing respectively, φ_θ and φ_y are the pressure flow factors in tangential and axial directions respectively, φ_s is the shear flow factor, φ_c is the contact factor, and σ is the standard deviation of combined roughness.

According to relevant literatures (pair and Cheng, 1978; pair and Cheng, 1979; Wu and Cheng, 1989), these factors are expressed as follows:

$$\varphi_\theta=\varphi_y=1-0.90e^{-0.56H} \tag{2}$$

$$\varphi_\mathrm{s}=\begin{cases}1.899H^{0.98}e^{-0.92H+0.05H^2} & H\le 5\\ 1.126e^{-0.25H} & H>5\end{cases} \tag{3}$$

$$\varphi_\mathrm{c}=\begin{cases}e^{-0.6912+0.782H-0.304H^2+0.0401H^3} & 0\le H<3\\ 1 & H\ge 3\end{cases} \tag{4}$$

Where, $H = h / \sigma$.

2.2 Metal contact pressure

According to the elastic contact model proposed by Greenwood and Tripp (1970), the formula for calculating the contact pressure is as follows:

$$\begin{cases}p_\mathrm{t}=\dfrac{16\sqrt{2}}{15}\pi(\eta\beta\sigma)^2\sqrt{\dfrac{\sigma}{\beta}}EF_{2.5}(H)\\ F_{2.5}(H)=\dfrac{1}{\sqrt{2\pi}}\displaystyle\int_H^\infty (s-H)^{2.5}e^{-\frac{s^2}{2}}ds\\ \dfrac{1}{E}=\dfrac{1}{2}\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)\end{cases} \tag{5}$$

Where, β is the radius of asperity, η is the areal density of asperities, E is the equivalent modulus of elasticity, $E_{1,2}$ are the Young's modulus of the bearing and the journal respectively, and $\nu_{1,2}$ are the Poisson's ratios.

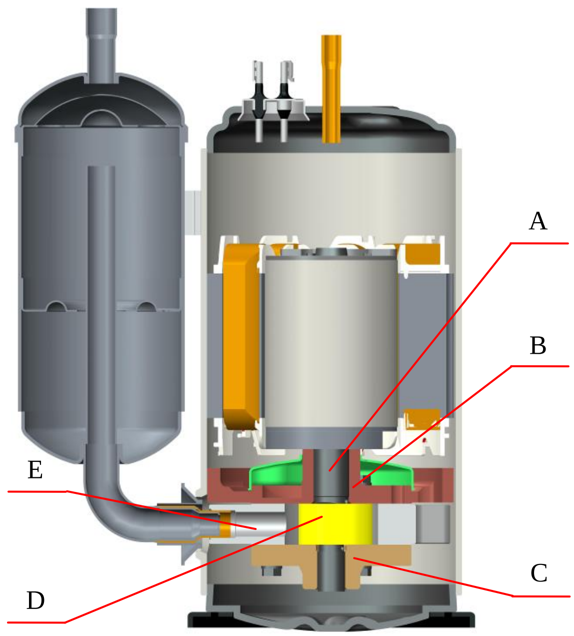
2.3 Carrying capacity for mixed lubrication

The carrying capacity of the bearing in the mixed lubrication state includes two parts: the load carrying capacity of the oil film generated by the fluid dynamic pressure and the load carrying capacity of the metal contact. The calculation formula is:

$$W = \iint p ds + \iint p_t ds \tag{6}$$

2.4 Compressor and conditions

The research object is the main bearing and sub bearing of a single cylinder rotary compressor, and the three-dimensional model of the compressor is shown in Figure 1.



A – Crankshaft B – Main bearing C – Sub bearing D – Rolling piston E – Cylinder

Figure 1: 3D model of rotary compressor

The basic parameters and operation conditions of the compressor are given in Table 1 and Table 2 respectively. The bearing motion and friction characteristics of the compressor at three kinds of rotation speeds (1800rpm, 3600rpm and 5400rpm) are calculated. The effects of wide-diameter ratio, clearance and lubricant viscosity on improving the minimum oil film thickness are further analyzed.

Table 1: Main dimensions of compressor

Main dimensions	Displacement, V_c	Diameter of main journal, D_m	Width of main bearing, B_m	Diameter of sub journal, D_s	Width of sub bearing, B_s	clearance, C
Parameter	10.2 cm^3	13 mm	35 mm	11 mm	14 mm	20μm

Table 2: Operation conditions

Temperature Points	Evaporation	Condensation	Superheat	Subcooling
Parameter	7.2℃	46℃	11.1K	8.3K

3. RESULTS AND DISCUSSIONS

3.1 Variations of Axis locus and friction power with rotation speed

As a variable-speed compressor, it is necessary to evaluate the bearing lubrication within the allowable operating speed range. Fig. 2 and Fig. 3 respectively show the variations of the main bearing and the sub bearing's axis locus with the crankshaft speed. It can be seen from the figure that the relative eccentricity always decreases with the increase of speed, because the bearing capacity of oil film increases with the increase of frequency, but the load of bearing changes very little with the speed.

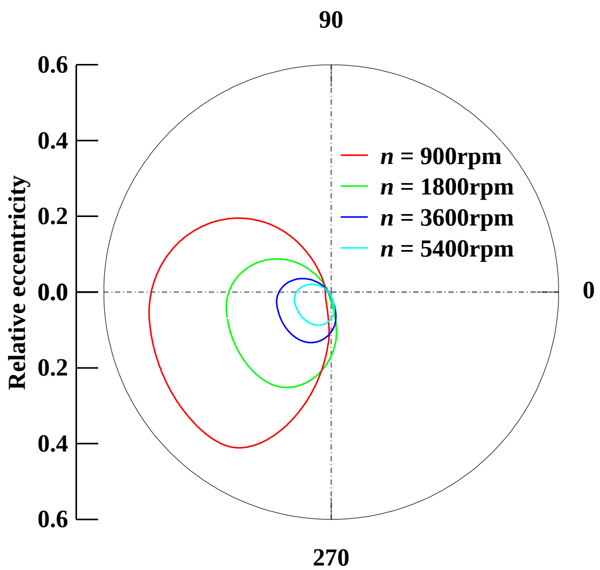


Figure 2: Axis locus of the main bearing

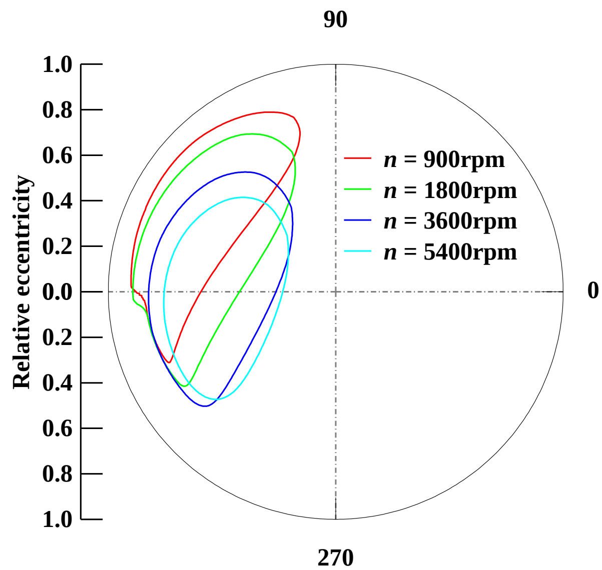


Figure 3: Axis locus of the sub bearing

Compared with Fig. 2 and Fig. 3, the axis loci of main bearing are mostly in the range of 180 °~ 270 °, but the axis loci of sub bearing are mostly in the range of 90 °~ 180 °. The reason for this difference is that the diameter and width of the sub bearing are smaller than that of the main bearing. The lubrication state of the sub bearing is mixed lubrication while that of the main bearing is fluid lubrication. In mixed lubrication, both metal contact and oil film play a role of bearing, but their directions are different, so the direction of resultant force is deflected compared with the direction of oil film force.

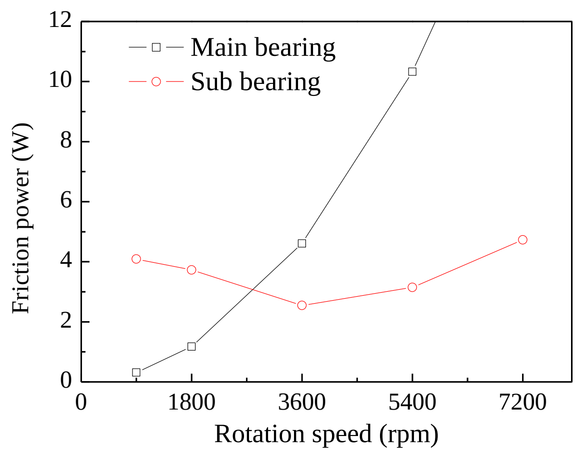


Figure 4: Friction power of the main and sub bearings

3.2 Influences of width-diameter ratio, oil viscosity and bearing clearance

In order to improve the lubrication performance of the sub bearing, the influences of the key parameters such as width-diameter ratio B/D, oil viscosity μ, clearance C on the minimum oil film thickness and carrying capacity are analyzed. Fig. 5 shows the variations of the minimum oil film thickness with the crankshaft angle under different widths, and Fig. 6 shows the variations of oil film carrying capacity C_O and metal contact carrying capacity C_C with the crankshaft angle under different widths. Fig. 7 shows the variations of minimum oil film thickness with crankshaft angle under different oil viscosities and Fig. 8 shows the variations of minimum oil film thickness with crankshaft angle under different clearance.

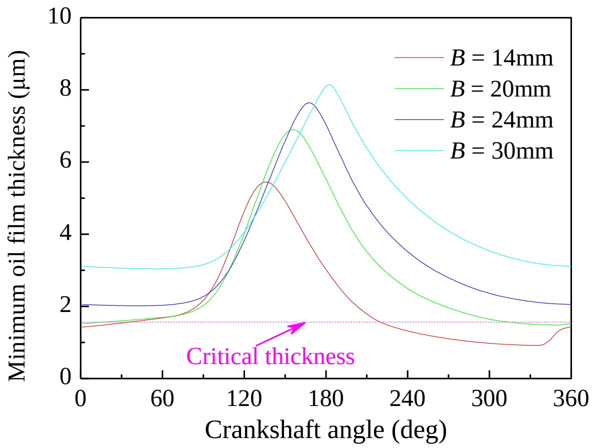


Figure 5: Effect of width on minimum oil film thickness

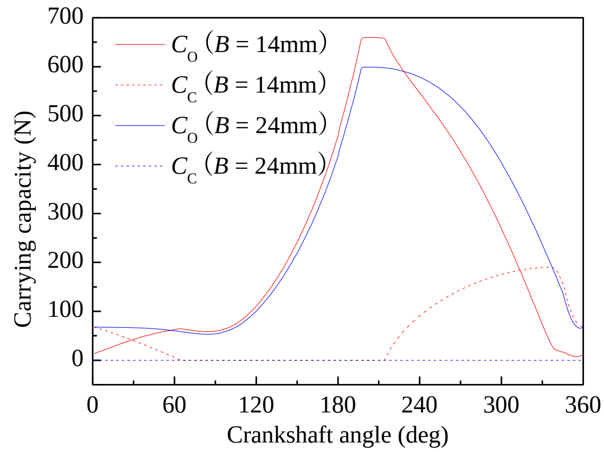


Figure 6: Effect of width on carrying capacity

It can be seen from Fig. 5 that the minimum oil film thickness increases as the bearing width B increases. When the width is 14 mm or 20 mm, the minimum oil film thickness is less than the critical oil film thickness (h = 3σ), so the bearing is in the mixed lubrication state. In a running cycle, the smaller the bearing width is, the greater the proportion of the time in the mixed lubrication state is. It can be seen from Fig. 6 that the oil film and metal contact both play a bearing role when the bearing width is 14mm (B/D= 1.27), and the angle range of metal contact bearing is the same as that of the minimum oil film thickness in Fig. 5 which is lower than the critical oil film thickness. When the bearing width is 24mm (B/D = 2.18), the bearing is in the fluid lubrication state, so the metal contact carrying capacity is zero.

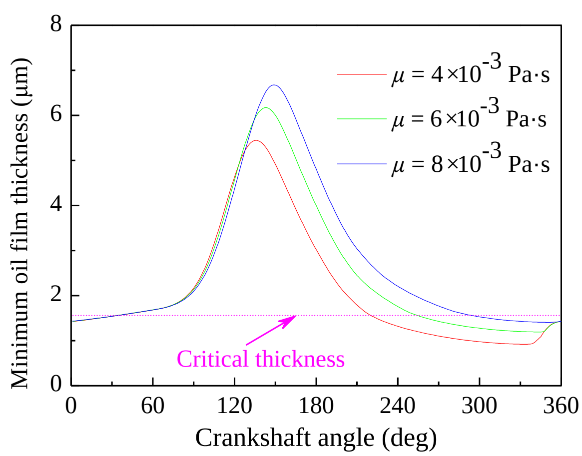


Figure 7: Effect of oil viscosity on minimum oil film thickness

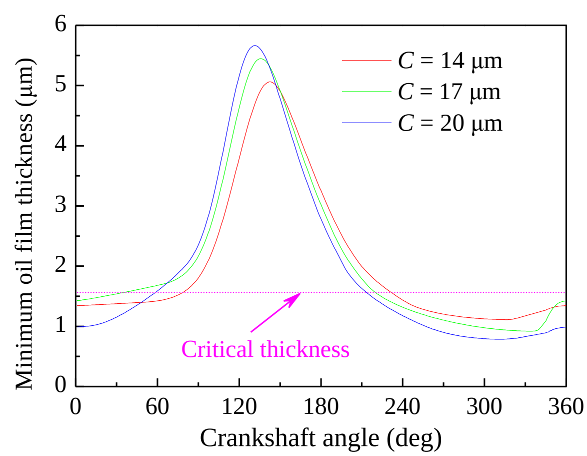


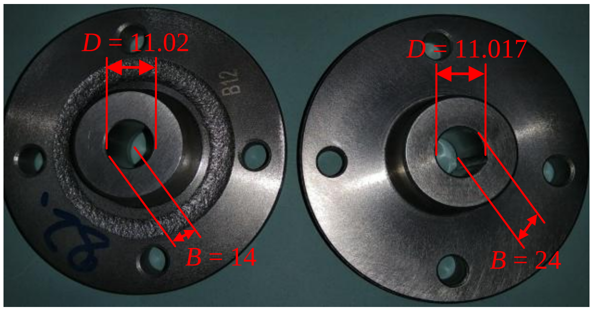
Figure 8: Effect of bearing clearance on minimum oil film thickness

As can be seen from Fig. 7, with the increase of the oil viscosity, the minimum oil film thickness increases, and the crankshaft angle range below the critical thickness decreases. Even if the viscosity of lubricating oil is doubled, metal contact cannot be avoided. In addition, considering the adverse effect of high viscosity oil in low temperature environment, we think that choosing higher viscosity lubricating oil is not a feasible measure.

It can be seen from Fig. 8 that there is an optimal clearance to make the minimum oil film thickness better. Based on the Reynolds' boundary condition, the hydrodynamic effect (the first item on the right side of formula (1)) will be weakened when the clearance increases, and the squeeze effect (the last item on the right side of formula (1)) will play a major role under the dynamic load conditions. If the clearance is too small, the crankshaft angle range where the minimum oil film thickness is below the critical oil film thickness will increase.

3.3 Optimization of sub bearing

According to the above analysis, the width-diameter ratio and clearance of the sub bearing in the rotary compressor are not good design. We adjusted these two parameters and confirmed the improvement effect through the compressor performance test. Fig. 9 shows a comparison of the physical photos of the two schemes.



(a) Original scheme (b) Improved scheme

Figure 9: Comparison between the original scheme and the improved scheme

Fig. 10 shows the simulation results of the sub bearing friction power of the two schemes, and Fig. 11 shows the compressor COP of the two schemes at different rotation speeds. It can be seen from Figure 10 that the rotation speed of the minimum friction power is reduced from 3600 rpm to 1800 rpm. When n = 1800 rpm, the power of the improved sub bearing is 84.8% lower than that of the original scheme. It can be seen from Fig. 11 that the COP of the improved compressor are significantly improved at low speed. For example, the cop of the improved scheme is increased by 1.4% when n = 1800 rpm, but the cop of the improved scheme is reduced by 0.47% when n = 7200rpm. As we all know, the influence of low speed performance on APF is far greater than that of high speed, so the improved scheme in this paper will be beneficial to improve the comprehensive energy efficiency level of the air conditioner.

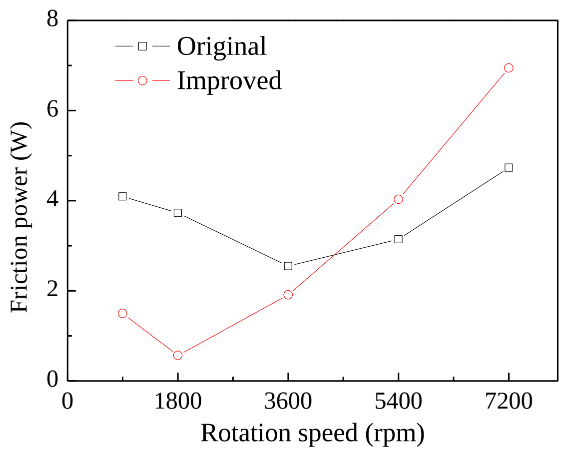


Figure 10: Comparison of the friction power

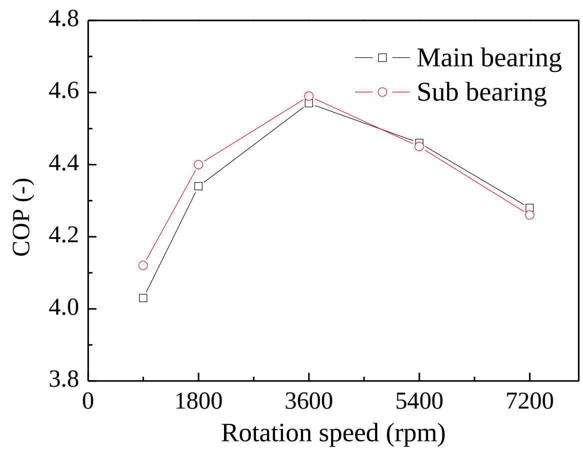


Figure 11: Comparison of the COP

4 CONCLUSIONS

Based on the average Reynolds' equation of hydrodynamic lubrication, the motion and load-carrying characteristics of the bearings of the rotary compressor are simulated. According to the simulation results, the sub bearing was optimized and the compressors were tested for performance. The following conclusions can be obtained from the analysis:

- At a lower rotation speed ($n \leq 1800$ rpm), the minimum oil film thickness of the sub bearing is smaller than the critical oil film thickness, and the friction power increases significantly with the decrease of rotating speed.

- The minimum film thickness increases with the increase of the width-diameter ratio and the oil viscosity, but increases first and then decreases with the increase of the clearance.

- By optimizing the width-diameter ratio and clearance of the bearing, the friction power of the sub bearing is reduced by more than 80%, and the experiment shows that the performance of the rotary compressor at low rotation speed is improved by more than 1%.

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