Research on motion and friction of rolling piston in rotary compressor

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Abstract. Based on the Reynolds equation of mixed lubrication, the dynamic model of rolling piston was established, and the axis locus, rotation velocity, sliding velocity at vane tip, and friction loss of the piston were numerically solved. The analysis results indicated that the numeric results of sliding velocity at vane tip are in good agreement with the experimental results. In addition, the influences of crankshaft rotation speed, working condition, piston density and compression structure parameters on motion and friction were analyzed. Taking the ratio of piston rotation velocity and revolution velocity ($\lambda$) as evaluation index, whether $\lambda$ conforms to cosine curve can judge the motion state between piston and vane. The results show that there is not only sliding state but also rolling state between piston and vane tip, and keeping rolling state is helpful to reduce friction loss. The design schemes of lighter piston, thicker vane and larger height-diameter ratio cylinder are proposed to keep rolling state in a wider rotation speed range and smaller pressure difference condition, thereby improving the annual operating efficiency of R32 rotary compressor.

1. Introduction

At present, rotary compressor is widely used in refrigeration and air conditioning fields. Therefore, improving rotary compressor efficiency is very important for the energy-saving operation of refrigeration equipment [1]. The rolling piston is a key component of rotary compressor, so it is always the key research object of compressor developers. Yanagisawa and Shimizu analyzed the motion of rolling pistons by approximating the bearing pressure, and rolling piston sliding positive and
negative on the vane tip is proved by experiment [2]. Further studies on friction losses indicated that
the short length / diameter configuration of the cylinder is favorable to decrease total loss [3]. Wu
developed an approach and a set of equations to estimate the type of motion at vane tip and roller
interface, and discussed the piston rotating velocity and the relative motion at vane tip in R410A
inverter rotary compressor [4]. Cho studied the partial elastohydrodynamic lubrication characteristics
between vane and rolling piston, and the results showed that the shaft rotational speed and the
discharge pressure significantly influence the friction force and the energy loss [5]. Further research
showed that the friction between rolling piston and vane has the largest influence on the piston motion,
and the sliding velocity at vane tip can be reduced by reducing the weight of rolling piston [6].

As we all know, APF has been the mainstream evaluation method of air conditioning efficiency
level, and it is very significant to improve compressor efficiency under wide operating frequency
range and multiple variable conditions, especially at low operating frequency ($f \leq 40$Hz). However, the
operating conditions and frequency range are not wide enough in previous studies. In this paper, based
on the mixed lubrication model, the motion and friction of rolling piston were solved synchronously
by finite difference method, and several conclusions were obtained to improve the operating efficiency
R32 compressor. This paper will provide some new ideas for compressor product developers.

2. Mathematical model and Calculation program

2.1 Mathematical model

Figure 1 shows the forces and motions of rolling piston. The rolling piston revolves around axis $o_{x}$ with
angular velocity $\omega_{o}$ and rotates around axis $o_{y}$ with angular velocity $\omega_{r}$. The forces and moments
acting on the rolling piston include: (a) the force $F_{s}$ caused by pressure difference; (b) the
hydrodynamic force $F_{cn}$ and viscous force $F_{vf}$ between rolling piston and cylinder; (c) the pressing
force $F_{vn}$ and friction force $F_{vf}$ between rolling piston and vane tip; (d) the centrifugal force $F_{m}$ of
rolling piston; (e) the friction force $F_{t}$ of the both ends of rolling piston; (f) the friction moment $M_{h}$
between rolling piston and crankshaft; (g) the friction moment $M_{i}$ of the both ends of rolling piston.

$R_{c}$ - Radius of cylinder
$R_{1}$ - Inner radius of rolling piston
$R_{2}$ - Outer radius of rolling piston
$e$ - Eccentricity of crankshaft
$R_{v}$ - Radius of vane tip
$B_{c}$ - breadth of vane
$L_{v}$ - Length of vane
$o$ - Axis of cylinder
$o_{x}$ - Axis of rolling piston
$o_{y}$ - Axis of vane tip

Not shown in the left picture:
$H_{c}$ - Height of cylinder
$H_{e}$ - Height of rolling piston bearing
$C_{e}$ - Radial clearance at rolling piston bearing

Figure 1. Forces acting on rolling piston and main dimensions of compressor
The motion equation of rolling piston is as follows:

\[
I_r \frac{d\omega}{dt} = M_b - M_r - R_2(F_{ct} + F_{ct})
\]  

(1)

Where, \(M_b = R_1F_{si} \), \(F_{bi} \) is the friction force between rolling piston and crankshaft, \(I_r \) is the rotational inertia of rolling piston, and the formula of \(M_r \) can be obtained by previous paper [6].

The total load \(F_{xy} \) acting on rolling piston is as follows:

\[
F_{xy} = F_g + F_r + F_m + F_{vn} + F_{vf} + F_{cn} + F_{cf}
\]  

(2)

Where, the first six items on the right side of “=” can be obtained through the previous papers [2-4].

Both the inner and outer wall frictions of rolling piston can be regarded as bearing systems, and figure 2 shows schematic diagram of journal bearing.

**Figure 2.** Journal bearing

**Figure 3.** Actual measurement result of \(\eta_1 \)

Based on the Reynolds equation of average flow model and the contact factor [7-8], the basic control equation of mixed lubrication in journal bearing is expressed as:

\[
\frac{\partial}{r^3 \partial \beta} \left( \phi h^3 \frac{\partial p}{\partial r} \right) + \frac{\partial}{\partial z} \left( \phi h^3 \frac{\partial p}{\partial z} \right) = \frac{6 \eta (U_1 + U_b) \varphi_s \frac{\partial h}{\partial r}}{r} + \frac{6 \eta (U_1 - U_b) \sigma \frac{\partial \varphi_s}{\partial \beta}}{r} + 12 \eta \varphi_s \frac{\partial h}{\partial t}
\]  

(3)

Where, \(h = C_e(1 + \epsilon \cos \beta) \), \(\epsilon = \partial C_e, r \) is the radius of the bearing, \(p \) is the oil film pressure, \(U_1 \approx R_1 \omega_1 \), \(U_b = R_1 \omega_2, \varphi_s \) and \(\varphi_s \) are the pressure flow factors in tangential and axial directions respectively, \(\varphi_s \) is the shear flow factor, \(\varphi_s \) is the contact factor, and \(\sigma \) is the standard deviation of combined roughness.

According to the elastic contact model proposed by Greenwood and Tripp [9], and further combined with the research conclusion of Song [10], the formula for calculating the contact pressure is as follows:

\[
p = \frac{32}{15} (0.075)^2 \sqrt{\frac{\pi \eta_1 \sigma^2}{0.075}} \int s (s - H)^{3/2} e^{-\frac{H^2}{2}} ds
\]  

(4)

Where, \(H = h/\sigma, \eta_1 \) is the areal density of asperities (the number of rough acoustic peaks in unit area), \(E_1, E_2 \) are the Young's modulus of the bearing and the journal respectively, and \(v_1, v_2 \) are the Poisson's ratios. The actual measurement results of \(\eta_1 \) are shown in Figure 3. As can be seen from
Figure 3, there is a strong correlation between $\eta_1$ and $\sigma$ in the cast iron crankshaft surface, so a fit formula is given for accurately predicting contact stress at different roughness.

In the mixed lubrication state, the bearing carrying force $W$ and the friction force $F_T$ are composed of two parts: hydrodynamic pressure and metal contact. The calculation formulas are:

$$W = \int_0^{2\pi} \int_0^B p r d\theta dz + \int_0^{2\pi} \int_0^B p r d\theta dz$$  \hspace{1cm} (5)

$$F_T = \int_0^{2\pi} \int_0^B \left[ \frac{\eta(U_j - U_b)}{h} (\varphi_i + \varphi_b) + \frac{h}{2} \frac{\partial \varphi}{\partial \varphi} + f_0 p \right] r d\theta dz$$  \hspace{1cm} (6)

Where, $B$ is the width of journal bearing, $f_0$ is the contact friction coefficient between the bearing and the journal, and the factors $\varphi_i$, $\varphi_b$, and $\varphi_p$ can be obtained by previous paper [11].

Based on the above model, the total load should be equal to the bearing capacity, that is, formula (2) is equal to formula (5), so that the variation of oil film thickness can be obtained. The bearing friction moment can be obtained by formula (6), then the piston motion can be obtained by formula (1). Based on the variation of oil film thickness and piston motion, formulas (3) and (4) can be solved. Therefore, formulas (1) - (6) constitute a closed system of equations for solving piston motion and friction.

2.2 Calculation program

The mixed Lubrication equation and the axis locus are solved by finite difference method (hahn method), which requires the assumption of the initial value of the axis locus of the rolling piston bearing [12]. It can be known from the above model that the rolling piston rotation angular velocity $\omega_r$ is a key parameter that links various forces and moments, so the dynamic model should be solved by assuming the initial value of $\omega_r$.

![Flow chart of calculation program](chart.png)
Figure 4 shows the main flow of the calculation program: (a) set initial values; (b) solve for the angular acceleration \( (d\omega_r/dt) \) and the axis movement speed of rolling piston bearing \( (d\varphi/dt) \); (c) obtain the new \( \omega_r, \delta, \) and \( \varphi; \) (d) continue the calculation of the next step until \( \omega_r, \varepsilon \) and \( \varphi \) all meet the above convergence conditions (7). According to this program, the motion and friction of rolling piston were solved synchronously at every step, so the characteristics can be analyzed in a wide range of frequencies and working conditions.

The convergence condition of the calculation program is that the piston rotation velocity and axis locus meet the relative accuracy requirements:

\[
\left| \frac{\omega_r(i+1) - \omega_r(0)}{\omega_r(0)} \right| < 0.01 \quad \text{and} \quad \left| \frac{\varepsilon(i+1) - \varepsilon(0)}{\varepsilon(0)} \right| < 0.01 \quad \text{and} \quad \left| \frac{\varphi(i+1) - \varphi(0)}{\varphi(0)} \right| < 0.01 \tag{7}
\]

3. Conditions and Model validation

Table 3 and 4 respectively show the main design parameters and working conditions of the compressor, in which the design parameters of compressor 1, condition 1 and condition 2 are the same as those in the literature [2], so as to compare the analysis results.

| Compressor | \( R_1 \) | \( H_1 \) | \( R_2 \) | \( R_1 \) | \( R_1' \) | \( H_1' \) | \( L_1 \) | \( B_1 \) | \( C_1 \) |
|------------|-------|-------|-------|-------|-------|-------|-------|-------|-------|
| 1          | 27    | 23.8  | 24.3  | 14.4  | 6     | 14    | 24    | 4.7   | 0.013 |
| 2          | 18    | 25    | 14.2  | 10.8  | 3.2   | 16    | 20    | 3.2   | 0.010 |
| 3          | 22    | 18.5  | 16.6  | 11.5  | 3.2   | 11    | 22.5  | 3.2   | 0.010 |

| Condition | Refrigerant | Oil type | Suction temperature(\(^{\circ}\)C) | Suction pressure(MPa) | Discharge pressure(MPa) |
|-----------|-------------|----------|---------------------------------|----------------------|------------------------|
| 1         | R22         | 4GS      | 10                              | 0.58                 | 2.03                   |
| 2         | R22         | 4GS      | 18                              | 0.74                 | 2.83                   |
| 3         | R32         | POE68    | 18.3                            | 1.02                 | 2.86                   |
| 4         | R32         | POE68    | 18.3                            | 1.02                 | 3.47                   |
| 5         | R32         | POE68    | 25                              | 1.47                 | 2.48                   |

The comparisons between the simulation results of compressor 1 in this paper and the simulation results and experimental results in the literature [2] are shown in Figures 5 and 6. It can be seen that the simulated values of the sliding velocity \( v_s \) at vane tip in this paper is more consistent with the experimental values. Under the working condition 1, the coefficient of determination \( R^2 \) between the simulated values of the new model and experimental values is 0.94, while the original is 0.9. Under the working condition 2, the coefficient of determination \( R^2 \) between the simulated values of the new model and experimental values is 0.84, while the original is 0.68. Therefore, the new calculation program in this paper can more accurately simulate the motion and friction characteristics of rolling piston.
4. Simulation results and Discussions

4.1 Effect of crankshaft rotation speed on movement and friction of rolling piston

Taking compressor 2 and condition 3 as the analysis object and working condition, respectively, the variations of the axis locus, rotation velocity, sliding velocity and friction loss of the rolling piston with the crankshaft rotation speed are simulated. The figure 7 shows the variations of the loaded force and axis locus with the crankshaft rotation speed (operating frequency f). When $F_{xy}=1500\text{N}$, define the relative force as 100%.

![Figure 7. The relative loaded force and axis locus of rolling piston](image)

It can be seen from Figure 7 that the eccentricity $\varepsilon$ decreases as $f$ increases. The reason is that the bearing capacity of the oil film increases with the increase of $f$, but the loaded force of the bearing changes little with $f$ under the same working condition. On the other hand, the axis locus of the rolling piston bearing is in the angular range of $-80^\circ$~$40^\circ$, while the loaded force is in the range of $80^\circ$~$180^\circ$. The reason is that the loaded force acts on the piston, and the piston should also be in the second and third quadrants under the force of the oil film, but there is a $180^\circ$ phase difference between the journal axis locus and the bearing axis locus, so the axis locus of crankshaft is in the fourth and first quadrants.
The piston rotation velocity $\omega_r$, the ratio of rotation to revolution angular velocity $\lambda$, the sliding velocity at vane tip $v_{sv}$, and the friction loss of each surface are respectively shown in Figures 8–11. It can be seen from Figure 8 that $\omega_r$ increases with the increase of $f$ but the regularity of the change is not significant. Taking the ratio of piston rotation angular velocity and revolution angular velocity ($\lambda$) as evaluation index, as shown in Figure 9, 40Hz is the inflection point of $\omega_r$ at which the $\lambda$ can maintain the “cosine” shape.

Compared with Figures 9 and 10, the sliding velocity is always zero when the operating frequency is lower than 40Hz of inflection point, which indicates that the motion state between rolling piston and vane is relative rolling. When the operating frequency is higher than 40Hz, $\lambda$ presents a “broken line” shape, and the sliding speed increases with the increase of operating frequency $f$. The inflection points of the “broken line” all fall on the “cosine” line. The reason is that when the rotation velocity changes to the rolling state between piston and vane tip, the direction of relative sliding changes reversely. So the angular acceleration of piston rotation also changes positively and negatively.

It can be seen from Figure 11, the loss of each surface of the rolling piston increases with the increase of the operating frequency $f$. In particular, the friction loss of outer wall also takes 40Hz as the inflection point. The friction loss is very small when $f \leq 40Hz$, and the friction loss increases rapidly when $f$ is in 40Hz–80Hz. Therefore, the motion state of rolling piston and vane tip is the key factor affecting the friction loss of rolling piston. It is helpful to reduce the friction loss to keep the relative rolling state.
4.2 Effect of working condition on movement and friction of rolling piston

In order to optimize the performance of compressor under different working conditions, the effect of pressure difference ($\Delta p = p_2-p_1$) is analyzed. Figure 12 shows the variation of $\lambda$ under working condition 5. Figure 13 shows the variation of friction loss of outer wall at different $\Delta p$.

![Figure 12. The variety of $\lambda$ under condition 5](image1.png)

![Figure 13. Friction loss of outer wall](image2.png)

It can be seen from Figure 12 that even if the operating frequency $f$ is as low as 20Hz, $\lambda$ can't keep “cosine” shape all the time, and the rolling piston and the vane tip always slide relative to each other in the range of $150^\circ$-$200^\circ$ of crankshaft angle. Because the pressing force $F_{vn}$ and friction force $F_{vf}$ decrease greatly with the decrease of $\Delta p$, the relative rolling state between rolling piston and vane tip cannot be maintained. A short time of relative sliding may lead to the separation of rolling piston and vane, which may cause the impact noise.

As can be seen from Figure 13, a larger $\Delta p$ does not mean that the friction loss of outer wall is always greater, and it has different friction loss characteristics at different crankshaft rotation speeds. When $f = 50$Hz, the friction loss under condition 5 with the minimum $\Delta p$ is the largest, but when $f = 80$Hz, the friction loss of condition 4 with the maximum $\Delta p$ is the largest. This means that if the friction loss under the condition with smaller $\Delta p$ is to be reduced, the sliding velocity should be reduced, such as increasing the thickness of vane $B_v$ to increase the $F_{vn}$.

4.3 Effect of the material on movement and friction of rolling piston

Several new materials (cast iron, alumina ceramic and PEEK) were used in rolling piston, and the effects of different material density on motion and friction characteristics were analyzed.

![Figure 14. The variety of $\lambda$ when $f = 80$H](image3.png)

![Figure 15. Friction loss of three material pistons](image4.png)
Figure 14 shows the comparison of $\lambda$ of three kinds of material pistons in compressor 2 under condition 3. Figure 15 shows the variation of friction loss of three material pistons’ outer wall. It can be seen from Figure 14 that the PEEK piston can still keep relative rolling state when $f = 80$Hz, and the Al$_2$O$_3$ ceramic piston is in relative sliding state, but its following performance is better than that of the cast iron piston. With the decrease of material density, the relative rolling state between rolling piston and vane is more favorable. The rotational inertia of the light piston is smaller, which makes the fluctuation of the rotation velocity better, so it better matches the movement of vane tip. As shown in Figure 15, the friction loss of outer wall decreases with the decrease of roller density, which will help to improve the mechanical efficiency of rotary compressor.

4.4 Effect of cylinder structure on movement and friction of rolling piston

It is a common engineering method to improve the mechanical efficiency by optimizing the structural parameters of cylinder. The compressor 3 is proposed, which has a larger height-diameter ratio than compressor 2. Figure 16 shows the variation of $\lambda$ of compressor 3 with $f$ under condition 2. Figure 17 shows that the friction loss of outer wall in compressor 2 and compressor 3 varies with $f$.

Figure 16. The variety of $\lambda$ of compressor 3  Figure 17. Friction loss in two compressors

It can be seen from Figure 16 that the inflection point of compressor 3 is 60Hz, which is significantly higher than that of compressor 2. It can be seen from figure 17 that the friction loss of outer wall in compressor 3 is significantly lower than that in compressor 2 in the operating frequency range of 40Hz - 100Hz. The reason is that the compressor with larger height-diameter ratio has thinner piston and larger pressing force $F_{vp}$, which makes rolling piston and vane have better follow-up and smaller sliding velocity. When the operating frequency $f$ reaches 120Hz or even higher, the sliding velocity difference of two compressors is small, because the friction moment $M_S$ of rolling piston bearing plays a major role. Therefore, the friction loss in compressor 3 is higher.

5. Conclusions

Based on the mixed lubrication model, the motion and friction loss of rolling piston are numerically solved. Furthermore, the effects of crankshaft rotation velocity, working condition, piston density, and compression structure parameters were analyzed.

The simulation results of rolling piston motion are in good agreement with the experimental results. It is proposed that the ratio of piston rotation velocity and revolution velocity ($\lambda$) can be used as the
evaluation index, and the motion state between rolling piston and vane can be judged according to whether \( \lambda \) conforms to “cosine” shape.

The conclusions helpful to design high efficiency R32 rotary compressor are as follows: (1) With the increase of crankshaft rotation speed, the motion state between rolling piston and vane changes from rolling to sliding, which leads to the sharp increase of friction loss at vane tip; (2) With the decrease of pressure difference, it is more difficult to keep the rolling state between piston and vane, and the sliding velocity should be reduced by increasing the thickness of vane; (3) The smaller the density of piston material is, the better it is to keep the rolling state between piston and vane, so light material or hollow piston should be used to reduce friction loss; (4) A larger height-diameter ratio cylinder can keep rolling state in a wider range of crankshaft rotation speed, that is to say, the crankshaft rotation speed range with lower friction loss is wider.

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