Lubrication Characteristics Between the Vane and the Rolling Piston in a Rotary Compressor Used for Refrigeration and Air-Conditioning Systems

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The rolling piston type-rotary compressor has been widely used for refrigeration and air-conditioning systems due to its compactness and high-speed operation. The present analysis is part of a research program directed toward maximizing the advantages of refrigerant compressors. The study of lubrication characteristics in critical sliding components is essential for the design of refrigerant compressors. Therefore, theoretical investigation of the lubrication characteristics of a rotary compressor used for refrigeration and air-conditioning systems was studied. The Newton-Raphson method was used for the partial elastohydrodynamic lubrication analysis between the vane and the rolling piston of a rotary compressor. The results showed that the rotational speed of a shaft and the discharge pressure significantly influence the friction force and the energy loss between the vane and the rolling piston.

Key Words: Rotary Compressor, Vane, Rolling Piston, Lubrication Characteristics, Partial Elastohydrodynamic Lubrication Analysis, Newton-Raphson Method

Nomenclature

e : Eccentric length
E' : Equivalent Young's modulous
f : Friction force per unit length
h : Film thickness at arbitrary x
h_m : Film thickness at $dp/dx=0$
h_0 : Film thickness at $x=0$
N : Number of asperities per unit area
O : Center of the cylinder
O_p : Center of the rolling piston
O_v : Center of the vane tip
p : Pressure at arbitrary
P_b : Pressure of suction chamber
p_c : Contact pressure
P_{co} : Pressure of compression chamber
P_d : Discharge pressure
p_h : Hydrodynamic pressure
P_s : Suction pressure
R : Equivalent radius of the contact
R_c : Radius of the cylinder
r_i : Inner radius of the rolling piston
r_o : Outer radius of the rolling piston
$u=(u_1+u_2)/2$: Average sliding velocity
v : Elastic normal displacement
w : Load per unit lengh
x : Coordinates
x_a : Location where pressure is generated
x_b : Location where the film is broken
x_v : Displacement of the vane
α : Pressure-viscosity coefficient

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α_p : Eccentric angle of rolling piston center
β : Mean radius of curvature of the asperities
γ : Surface pattern parameter
ϕ_h : Pressure flow factor
η : Viscosity of lubricant
η_o : Viscosity of lubricant at ambient pressure
σ : Standard deviation of roughness amplitude
θ : Rotational angle of the eccentric shaft
ω : Angular velocity of the eccentric shaft
ω_p : Angular velocity of the rolling piston

1. Introduction

The rolling piston type rotary compressor has been widely used for refrigeration and air-conditioning systems due to its compactness, low cost and high-speed operation. In the rotary compressor used for refrigeration and air-conditioning systems, compression motion consists of mechanisms regulating compression volume by the rolling piston which rotates around the eccentric shaft and rotates on its axis, and the vane which has reciprocating motion in the cylinder slot. Therefore, many sliding components are included in the rotary compressor.

Especially, the sliding velocity between the vane and the rolling piston is very low and the normal force acting on the vane is very large. Also, the parts are lubricated with lubricant which contains a large amount of refrigerant. Thus, the lubrication characteristics between the vane and the rolling piston become a very critical issue. Therefore, lubrication characteristics between the vane and the rolling piston are the most important mechanical property deciding the performance and reliability of a rotary compressor used for refrigeration and air-conditioning systems.

The study of lubrication characteristics in the critical sliding component is essential for the design of refrigerant compressors. Therefore, in this paper, theoretical investigation of the lubrication characteristics of a rotary compressor for refrigeration and air-conditioning systems is studied. The Newton-Raphson method is used for the elastohydrodynamic lubrication analysis between the vane and the rolling piston in the rotary compressor.

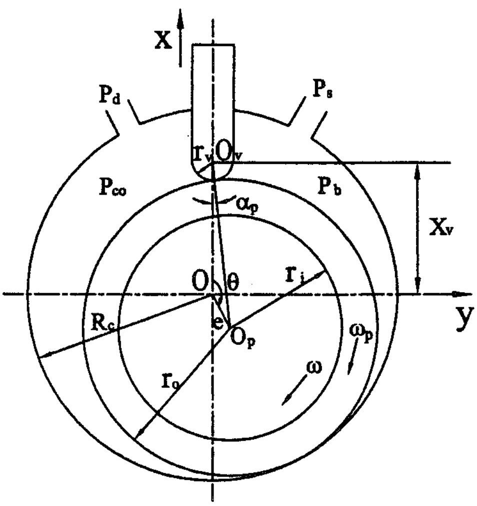

Fig. 1 Schematic diagram of cylinder part

2. Theoretical Analysis

In the rotary compressor for refrigeration and air-conditioning systems, the film thickness between the vane and the rolling piston is very thin because overall lubrication performance is controlled by the lubricant which includes a significant amount of refrigerant.

The asperities of the vane and the rolling piston cause contacts. Contact pressure supports some of the total load and hydrodynamic pressure supports the rest.

Therefore, in critical lubrication conditions, we must use the Partial EHL Analysis method so that accurate lubrication analysis is performed. Figure 1 shows the coordinates of the rotary compressor for analysis of the lubrication characteristics.

2.1 Basic equations

The contact between the vane and the rolling piston in Fig. 1 can be represented by equivalent cylinders as shown in Fig. 2. Where, u_1, u_2 are the surface velocities of two sliding components in the x-direction.

The equation for hydrodynamic pressure of rough surfaces with the same structure and rms roughness is given by

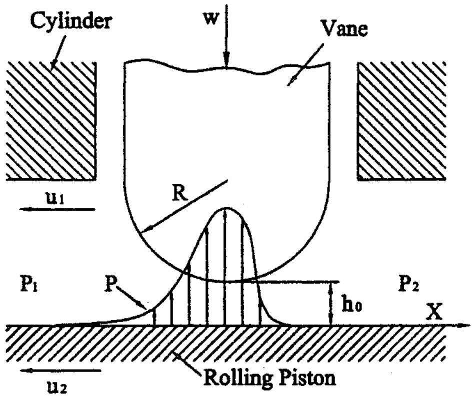

Fig. 2 The model of line contact

$$\psi_h \frac{h^3}{12\eta}\frac{dp_h}{dx} = u(\overline{h}_T - \overline{h}_{Tm}) \quad (1)$$

where ψ_h is the average flow factor given by

$$\psi_h = 1 - C\exp\left(-r\frac{h}{\sigma}\right) \; ; \gamma \leq 1$$

$$\psi_h = 1 + C\left(\frac{h}{\sigma}\right)^{-r} \; ; \gamma > 1 \quad (2)$$

C and r are constants depending on γ and can be found in Ref.

$\overline{h}_T$ is the average gap height given by

$$\overline{h}_T = \int_{-h}^{\infty}(h+\varepsilon)f(\varepsilon)d\varepsilon \quad (3)$$

Here $f(\varepsilon)$is the probability density function of combined roughness ε. Assuming this to be Gaussian, it is given by

$$f(\varepsilon) = \frac{1}{\sigma\sqrt{2\pi}}\exp(-\varepsilon^2/2\sigma^2) \quad (4)$$

A polynomial density function which closely approximates Eq. (4) is given by

$$f(\varepsilon) = \begin{cases} \frac{35}{96\sigma}\left[1-\left(\frac{\varepsilon}{3\sigma}\right)^2\right]^3 & ; |\varepsilon| \leq 3\sigma \\ 0 & ; |\varepsilon| > 3\sigma \end{cases} \quad (5)$$

Using the polynomial frequency density function given in Eq. (5), $\overline{h}_T$ becomes

$$\overline{h}_T = \begin{cases} h & ; h \geq 3\sigma \\ \frac{3\sigma}{256}[35 + z(128 + z(140 + z^2(-70 + z^2 (28-5z^2))))] & ; h < 3\sigma (z = h/3\sigma) \end{cases} \quad (6)$$

The coefficient of viscosity η is taken to vary with pressure by the relation

$$\eta = \eta_0 \exp(\alpha p) \quad (7)$$

The nominal film thickness h for the parabolic approximation of the cylinder pair is

$$h = h_0 + \frac{x^2}{2R} + v \quad (8)$$

where v is the elastic deformation of two cylinders and is given by

$$v = -\frac{2}{\pi E'}\int_{s_1}^{s_2} p_h \ln(x-s)^2 ds - \frac{2}{\pi E'}\int_{s_1}^{s_2} p_c \ln(x-s)^2 ds + C_0 \quad (9)$$

The contact pressure p_c is calculated by the mean contact pressure-compliance relationship and is given by

$$p_c = k_c E' F_{2.5}\left(\frac{h}{\sigma}\right) \quad (10)$$

where,

$$k_c = \left(\frac{8\sqrt{2}}{15}\right)\pi(N\beta\sigma)^2\sqrt{\frac{\sigma}{\beta}}$$

and

$$F_{2.5}\left(\frac{h}{\sigma}\right) = \int_{\frac{h}{\alpha}}^{\infty}\left(\zeta - \frac{h}{\sigma}\right)^{2.5} f^*(\zeta)d\zeta$$

The function $F_{2.5}(h/\sigma)$ can be closely approximated by

$$F_{2.5}\left(\frac{h}{\sigma}\right) = \begin{cases} 4.4086\times 10^{-5}\left(4-\frac{h}{\sigma}\right)^{6.804} & ; h < 4\sigma \\ 0 & ; h \geq 4\sigma \end{cases}$$

After obtaining asperity contact pressure p_c, the hydrodynamic pressure p_h can be calcuated with the load relationship given by

$$w = \int_{x_a}^{x_b} p_h(x)dx + \int_{x_a}^{x_b} p_c(x)dx \quad (11)$$

The boundary conditions to analyze the lubrication charateristics are given by

$$p_h = p_1 \qquad at\ x = x_a$$
$$p_h = p_2,\ \frac{\partial p_h}{\partial x} = 0 \ at\ x = x_b$$

After obtaining both hydrodynamic pressure p_h and asperities pressure p_c, the friction force between the vane and the rolling piston can be calculated by the followiry equation :

$$f = \int_{x_a}^{x_b}\frac{1}{2}\frac{\partial p_h}{\partial x}dx - \int_{x_a}^{x_b}\eta\frac{u_1}{h}dx - \mu\int_{x_a}^{x_b} p_c dx \quad (12)$$

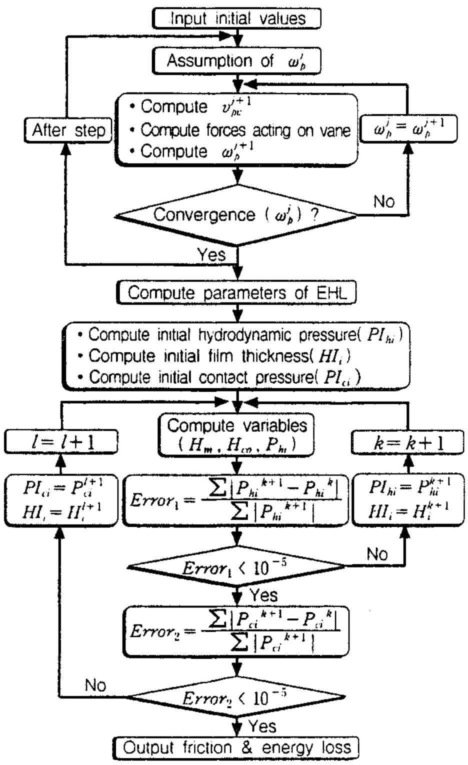

Fig. 3 Flow chart

3. Numerical Analysis

In this paper, the partial EHL analysis method is used so that accurate lubrication analysis can be performed. The Newton-Raphson method is applied for the partial EHL analysis. Also, the Runge-Kutta method is applied to analyze the motions of the vane and the rolling piston.

Figure 3 shows the processes of the above partial EHL analysis and the rolling piston behavior analysis.

In Fig. 3, the angular velocity (ω_P) of the rolling piston is computed by the upper process. Using the data, the intial values for the lower process are calculated and the final outputs are computed by the lower process.

4. Results

The operating conditions and the geometrical shapes of rotary compressor are summarized in Table 1.

Table 1 Geometrical shapes & operating conditions of rotary compressor(R22)

Item	Values	Unit
Suction/Discharge pressure	5.34/20.86	kgf/cm²
Rotational speed of a shaft	3386	rpm
Oil viscosity (at 120℃)	3	cP
Vane spring coefficient	1.39	kgf/cm
Vane tip radius	0.4	cm
Vane thickness	0.4	cm
Vane mass	20.48	g
Rolling piston outer radius	1.95	cm
Rolling piston inner radius	1.315	cm
Rolling piston mass	127.92	g
Cylinder radius	2.4	cm
Cylinder height	2.78	cm

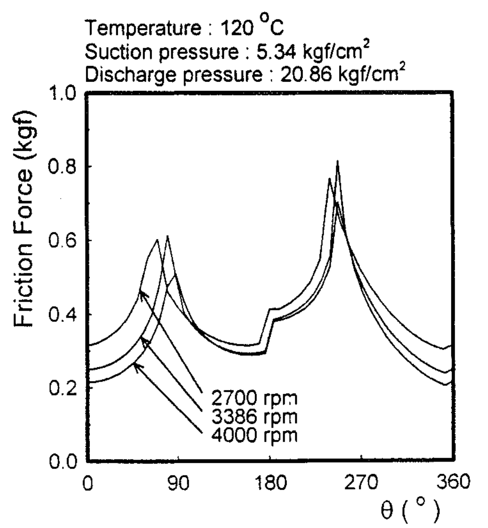

Fig. 4 Friction forces between vane and rolling piston to the variation of rpm

4.1 Effect of the rotational speed of an eccentric shaft

When the suction pressure is 5. 34kgf/cm² and the discharge pressure is 20.86kgf/cm², the effect of the roational speed of the eccentric shaft on friction force between the vane and the rolling piston is shown in Fig. 4.

In Fig. 4, the friction force between the vane and the rolling piston decreases with an increase in the rotational speed of the eccentric shaft, because the film thickness increases and the contact pressure decreases accordingly. The reason is that the sliding velocity between the vane and the

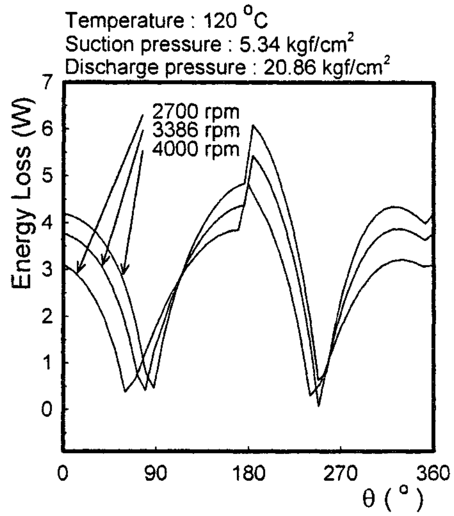

Fig. 5 Energy losses between vane and rolling piston to the variation of rpm

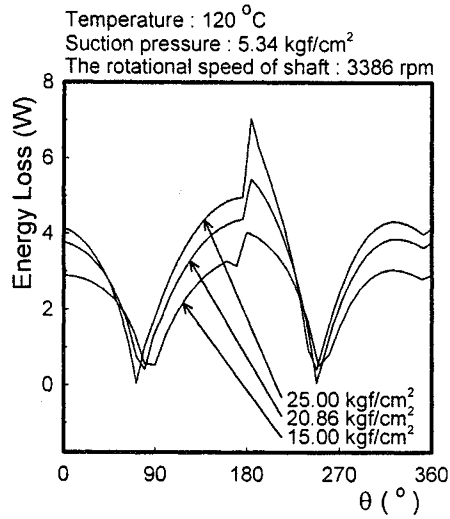

Fig. 7 Energy losses between vane and rolling piston to the variation of discharge pressure

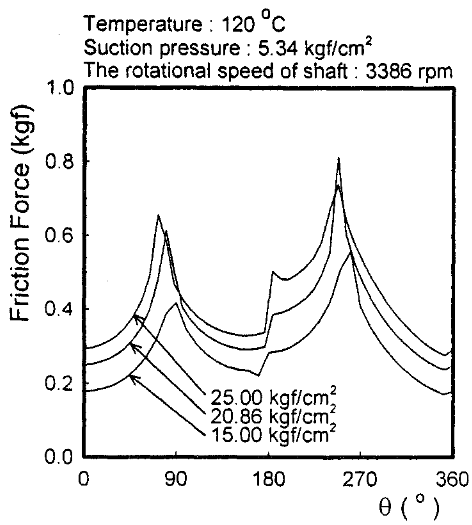

Fig. 6 Friction forces between vane and rolling piston to the variation of discharge pressure

rolling piston decreases with an increase in the rotational speed of the eccentric shaft.

When the suction pressure is 5.34kgf/cm^2 and the discharge pressure is 20.86kgf/cm^2, the effect of the rotational speed of the eccentric shaft on energy loss between the vane and the rolling piston is shown in Fig. 5.

In Fig. 5, the energy loss between the vane and the rolling piston increases with an increase in the rotational speed of the eccentric shaft.

4.2 Effect of the discharge pressure

When the suction pressure is 5.34kgf/cm^2 and the rotational speed of the eccentric shaft is 3386rpm, the effect of the discharge pressure on friction force between the vane and the rolling piston is shown in Fig. 6.

In Fig. 6, the friction force between the vane and the rolling piston increases with an increase in the discharge pressure. The reason for this is that the normal force acting on the vane increases with an increase in the discharge pressure.

When the suction pressure is 5.34kgf/cm^2 and the rotational speed of the eccentric shaft is 3386rpm, the effect of the discharge pressure on friction force between the vane and the rolling piston is shown in Fig. 7.

In Fig. 7, the energy loss between the vane and the rolling piston increases with an increase in the discharge pressure. The reason for this is that the increase in the friction force is more than that of the sliding velocity between the vane and the rolling piston, because the influence of the two parameters(friction force, sliding velocity) for the energy loss is opposite.

4.3 Effect of the solubility

Because the compressor is lubricated with two phases, the solubility of the refrigerant and the

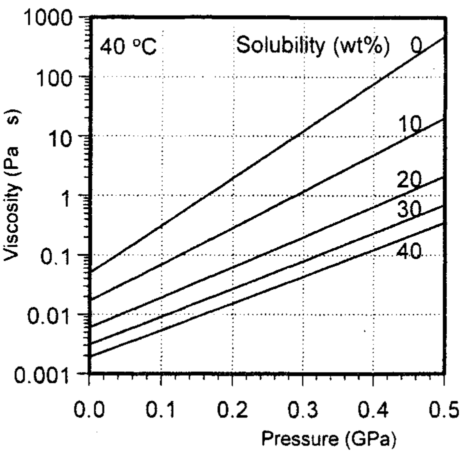

Fig. 8 Viscosity-pressure solubility compliance of refrigerant(R22) and lubricant

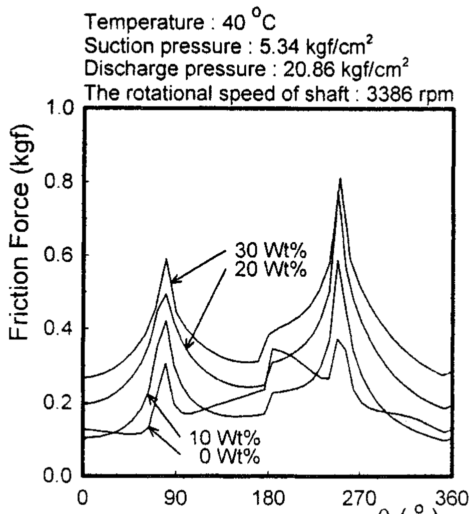

Fig. 9 Friction forces between vane and rolling piston to the variation of solubility

lubricant is an important parameter to analyze the lubrication characteristics of the compressor.

The viscosity-pressure-solubility compliance of the refrigerant(R22) and lubricant is shown in Fig. 8.

When the suction pressure is 5.34kgf/cm^2, the discharge pressure is 20.86kgf/cm^2 and the rotational speed of the eccentric shaft is 3386rpm, the effect of the solubility on the friction force between the vane and the rolling piston is shown in Fig. 9.

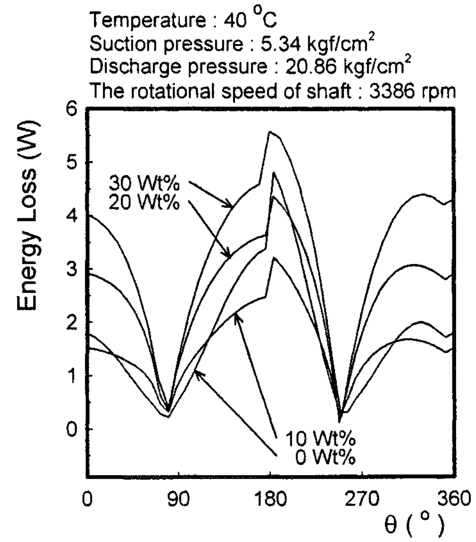

Fig. 10 Energy losses between vane and rolling piston to the variation of solubility

In Fig. 9, if the solubility of the refrigerant is more than 10wt%, it is known that the friction force between the vane and the rolling piston increases. The reason for this is that the film thickness becomes thinner due to the decrease in viscosity, and the contact pressure increases as the solubility increases. However, if the solubility is less than 10wt%, the friction force increases. This is because the film thickness increases and the fluid pressure increases according to the increase in the viscosity.

When the suction pressure is 5.34kgf/cm^2, the discharge pressure is 20.86kgf/cm^2 and the rotational speed of the eccentric shaft is 3386rpm, the effect of the solubility on energy loss between the vane and the rolling piston is shown in Fig. 10.

In Fig. 10, for the range of solubility of refrigerant between 10~30wt%, the energy loss increases with the increase in the solubility. Howerev, if the solubility is less than 10wt%, the energy loss decreases with the increase in the solubility. The reason for this is that the variation of fluid friction force is greater than that of the sliding velocity between the vane and the rolling piston, when the solubility of the refrigerant increases.

5. Conclusions

In this paper, the partial EHL characteristics is analyzed between the vane and the rolling piston in a rotary compressor used for refrigeration and air-conditioning systems.

The results are as follows :

(1) The solubility of the refrigerant significantly influence the friction force and the energy loss between the vane and the rolling piston.

(2) The rotational speed of the shaft and the discharge pressure of the compressor significantly influence the friction force and the energy loss between the vane and the rolling piston.

Therefore, the above three parameters must be considered to analyze the lubrication characteristics between the vane and the rolling piston in a rotary compressor used for refrigeration and air-conditioning systems.

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