The Influence of the Vane on the Lubrication Characteristics Between the Vane and the Rolling Piston of a Rotary Compressor

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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 the critical sliding component is essential for the design of refrigerant compressors. Therefore, theoretical investigation of the lubrication characteristics of a rotary compressor being used for refrigeration and air-conditioning systems was investigated. The Newton-Raphson method was used for a partial elastohydrodynamic lubrication analysis between the vane and the rolling piston of a rotary compressor. The results demonstrated that the vane thickness and the center line position of the vane significantly influenced the friction force and the energy loss between the vane and the rolling piston.

Key Words : Rotary Compressor, Vane, Rolling Piston, Lubrication Characteristics, Vane Thickness, Vane Center Line

Nomenclature

e : Eccentric length
E' : Equivalent Young's modulus
f : Friction force per unit length
h : Film thickness at arbitrary x
h_m : Film thickness at $dp/dx=0$
h_o : 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 x
P_b : Pressure of suction chamber
p_c : Contact pressure
P_∞ : 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 length
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
α_p : Eccentric angle of rolling piston center
β : Mean radius of curvature of the asperities
γ : Surface pattern parameter
ψ_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

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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 being used for refrigeration and air-conditioning systems, compression motion consists of mechanisms that regulate compression volume by the rolling piston, which rotates around the eccentric shaft and rotates on its axis, and the vane which has a reciprocating motion in the cylinder slot. Therefore, many sliding components are included in this system. (Cho et al., 1996)

In particular, 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 solids are lubricated with lubricant which includes a large amount of refrigerant. Thus, the lubrication characteristics between the vane and the rolling piston comprise a very critical condition. Therefore, lubrication characteristics between the vane and the rolling piston are the most important mechanical component affecting 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 was conducted. The Newton-Raphson method was used for the elastohydrodynamic lubrication analysis between the vane and the rolling piston in the rotary compressor.

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.

Lubrication characteristics are critical to system performance. 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, under critical lubrication conditions, we must use the Partial EHL Analysis method to ensure that accurate lubrication analysis is performed. Fig. 1 presents the coordinates of the rotary compressor for analysis of the lubrication characteristics. (Cho et al., 2001)

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 denoted as

$$\psi_h \frac{h^3}{12\eta} \frac{dp_h}{dx} = u(\bar{h}_T - \bar{h}_{T_m}) \tag{1}$$

where ψ_h is the average flow factor given by

$$\begin{aligned} \psi_h &= 1 - C\exp\left(-r\frac{h}{\sigma}\right) \quad ; \gamma \leq 1 \\ \psi_h &= 1 + C\left(\frac{h}{\sigma}\right)^{-r} \quad ; \gamma > 1 \end{aligned} \tag{2}$$

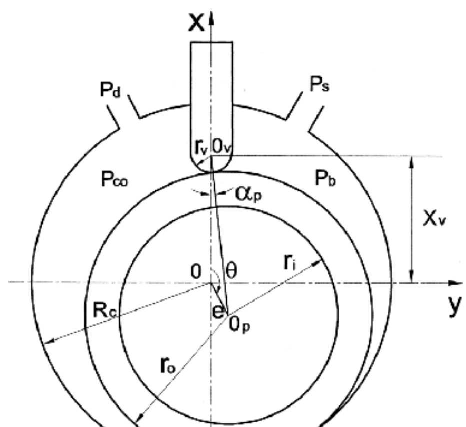

Fig. 1 Schematic diagram of the cylinder

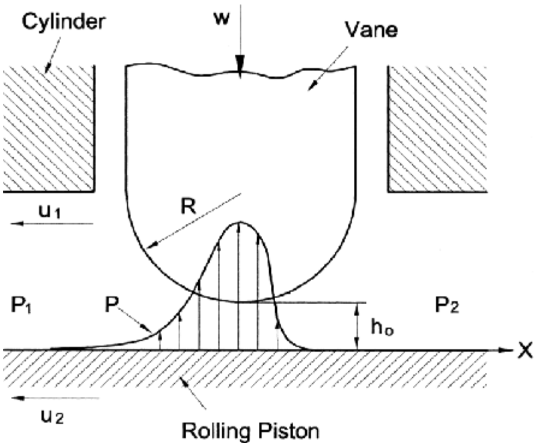

Fig. 2 Model of a line contact

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

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

$$\bar{h}_T = \int_{-h}^{\infty} (h+\epsilon) f(\epsilon)\, d\epsilon \tag{3}$$

Here $f(\epsilon)$ is the probability density function of combined roughness ϵ. Assuming this to be Gaussian, it is denoted as

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

In the paper of Patir and Cheng (1979), a polynomial density function which closely approximates equation (4) is given by

$$f(\epsilon) = \begin{cases} \dfrac{35}{96\sigma}\left[1-\left(\dfrac{\epsilon}{3\sigma}\right)^2\right]^3 & ; |\epsilon| \le 3\sigma \\ 0 & ; |\epsilon| > 3\sigma \end{cases} \tag{5}$$

Using the polynomial frequency density function given in equation (5), $\bar{h}_T$ becomes

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

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

$$\eta = \eta_o \exp(\alpha p) \tag{7}$$

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

$$h = h_o + \frac{x^2}{2R} + v \tag{8}$$

where v is the elastic deformation of two cylinders, which is denoted as

$$\begin{aligned} v = &-\frac{2}{\pi E'}\int_{s_1}^{s_2} p_h \ln(x-s)^2 ds \\ &-\frac{2}{\pi E'}\int_{s_1}^{s_2} p_c \ln(x-s)^2 ds + C_o \end{aligned} \tag{9}$$

In the paper of Greenwood and Tripp (1971), the contact pressure p_c is calculated by the mean contact pressure-compliance relationship and is given by

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

where,

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

and

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

In the paper of Prakash and Czichos (1983), The function $F_{2.5}(h/\sigma)$ can be closely approximated by

$$F_{2.5}\left(\frac{h}{\sigma}\right) = \begin{cases} 4.4086\times10^{-5}\left(4-\dfrac{h}{\sigma}\right)^{6.804} & ; h<4\sigma \\ 0 & ; h \ge 4\sigma \end{cases}$$

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

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

The boundary conditions to analyze the lubrication characteristics are given by

$$\begin{aligned} p_h &= p_1 && \text{at } x = x_a \\ p_h &= p_2,\ \frac{\partial p_h}{\partial x} = 0 && \text{at } x = x_b \end{aligned}$$

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 and defined as

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

3. Numerical Analysis

In this paper, The partial EHL analysis method was used in order to analyze lubrication characteristics accurately. The Newton-Raphson method was applied for the partial EHL analysis. Also, the Runge-Kutta method was applied to analyze the motions of the vane and the rolling piston.

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

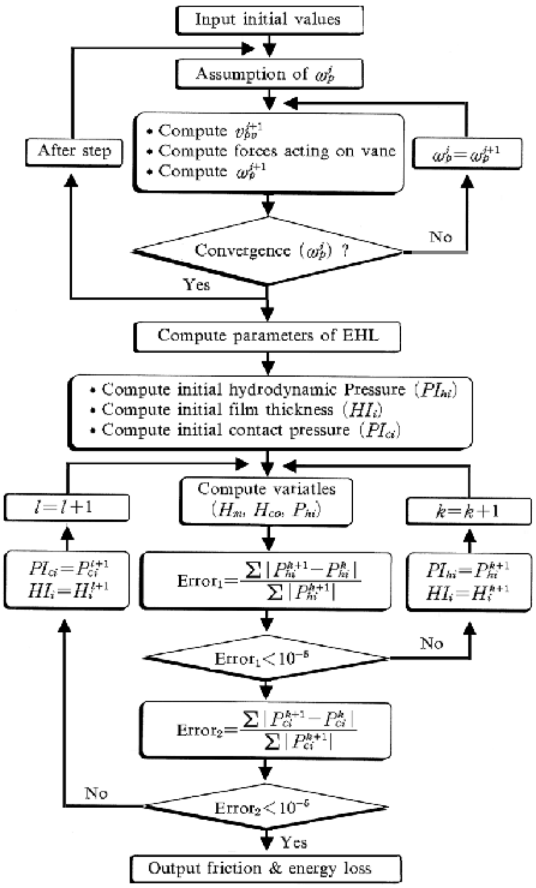

Fig. 3 Flow chart

In Fig. 3, the angular velocity (ω_p) of the rolling piston is computed by the upper process. Utilizing the data to calculate the intial values for the lower process, 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.

4.1 Effect of the vane thickness

When the operating temperature is 120℃, the effect of the vane thickness on the normal force acting on the vane is shown in Fig. 4.

In Fig. 4, the normal force acting on the vane increases with an increase in the vane thickness. The reason for this increase is the effect of the discharge pressure acting on the vane, which increases with an increase in the vane thickness. Compared with the effect of the discharge pressure acting on the vane, other terms very slightly influence the normal force, relatively.

When the operating temperature is 120℃, the effect of the vane thickness on the friction force between the vane and the rolling piston is outlined in Fig. 5.

Table 1 Geometrical shapes and operating conditions of a rotary compressor

Items	Values	Unit
Suction/Discharge pressure	0.52/2.0	MPa
Rotational speed of shaft	3386	rpm
Oil viscosity at 40℃	0.050	Pa·S
Oil viscosity at 120℃	0.003	Pa·S
Spring constant	13.6	N/cm
Vane tip radius	0.4	cm
Vane thickness	0.4	cm
Vane mass	2.1	g
Rolling piston outer radius	1.95	cm
Rolling piston inner radius	1.315	cm
Rolling piston mass	13	g
Cylinder radius	2.4	cm
Cylinder height	2.78	cm

In Fig. 5, the friction forces have two peak points. The positions of these points are close to the axis angles where the relative sliding velocity between the vane and the rolling piston is zero, and because the film that formed between the vane and the rolling piston became thin, the friction forces become extremely high. Therefore, it seems, at these points, almost all region of the vane tip are in dry contact. And the friction forces are relatively small, except for the two peak points, because the friction between the lubricated surfaces under relative motion results from the shear stress of the oil film. And, the friction force increased as the step shape near 180° because of the load increased dramatically near the point.

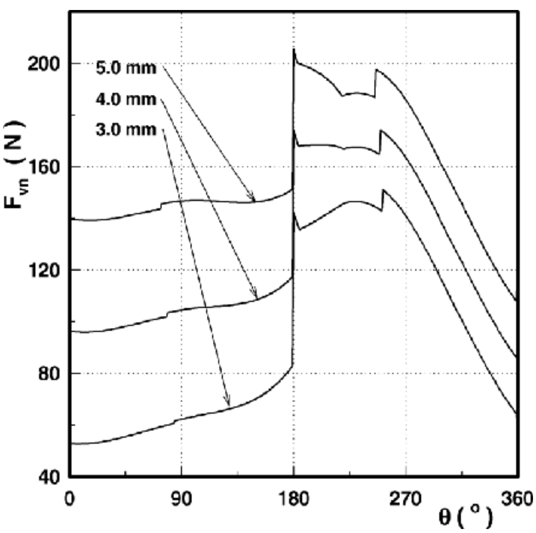

Fig. 4 Normal forces between the vane and the rolling piston to the variation of the vane thickness

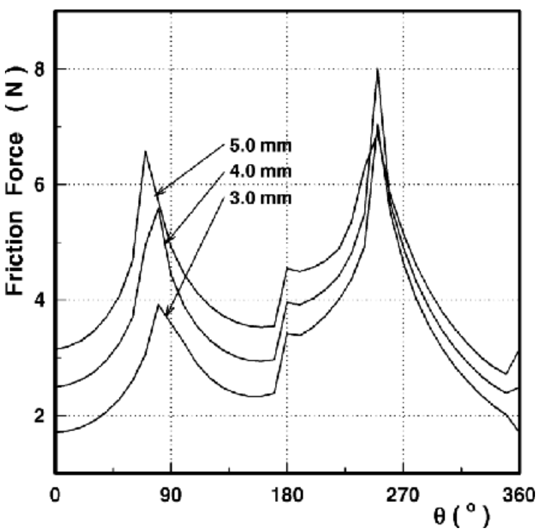

Fig. 5 Friction forces to the variation of the vane thickness

Also, the friction force between the vane and the rolling piston increases with an increase in the vane thickness. The source of this increase is that the normal force acting on the vane increases with an increase in the vane thickness.

When the operating temperature is 120℃, the effect of the vane thickness on the energy loss between the vane and the rolling piston is shown in Fig. 6.

In Fig. 6, the energy loss between the vane and the rolling piston increases with an increase in

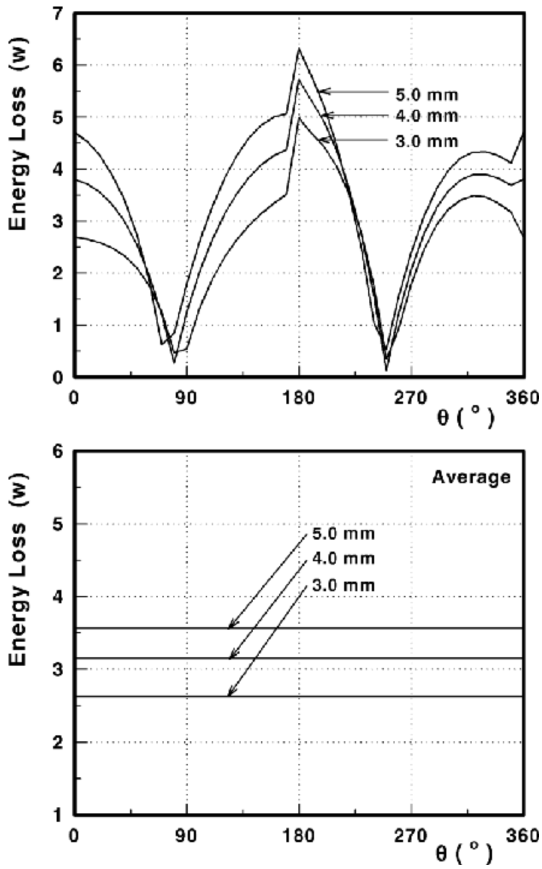

Fig. 6 Energy losses to the variation of the vane thickness

the vane thickness, because the friction force between the vane and the rolling piston increases with an increase in the vane thickness.

4.2 Effect of the center line position of the vane

When the center line position of the vane is altered as shown in Fig. 7, the lubrication characteristics are shown in Figs. 8～10.

When the operating temperature is 120℃, the effect of the center line position of the vane on the normal force acting on the vane is shown in Fig. 8.

In Fig. 8, if the center line position of the vane is moved to the suction chamber, the normal force acting on the vane increases. The reason for this increase is the effect of the discharge pressure acting on the back of the vane, which increases when the center line position of the vane moves towards the suction chamber. Also, the effect of the compression pressure acting on the vane tip increases with a movement of the center line position of the vane to the compression chamber.

When the operating temperature is 120℃, the effect of the center line position of the vane on the friction force between the vane and the rolling piston is shown in Fig. 9.

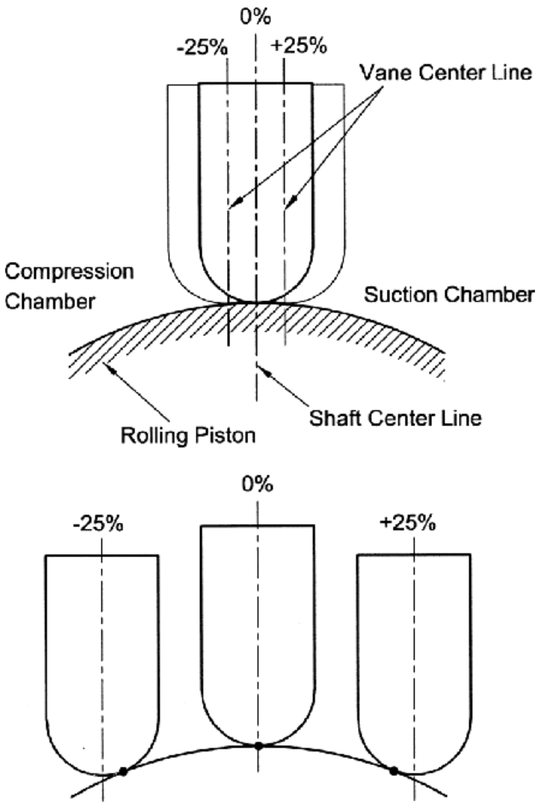

Fig. 7 Model of the variation of a vane center line

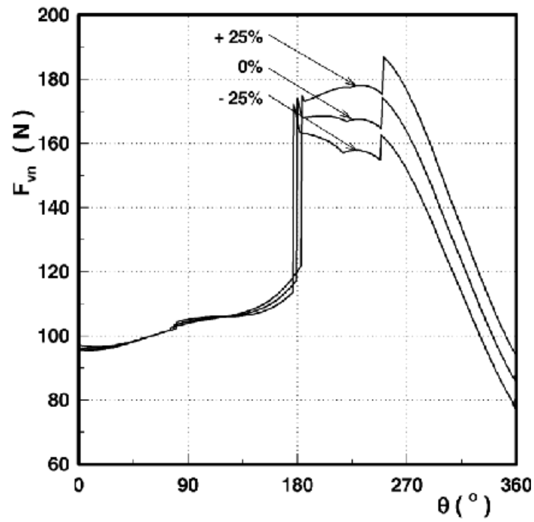

Fig. 8 Normal forces between the vane and the rolling piston to the variation of the vane center line

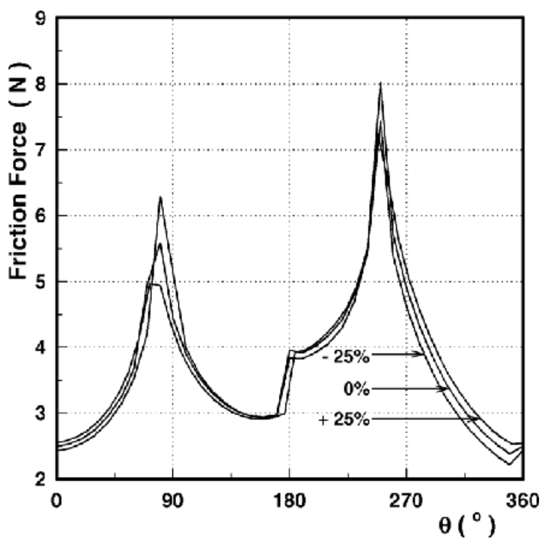

Fig. 9 Friction forces to the variation of the vane center line

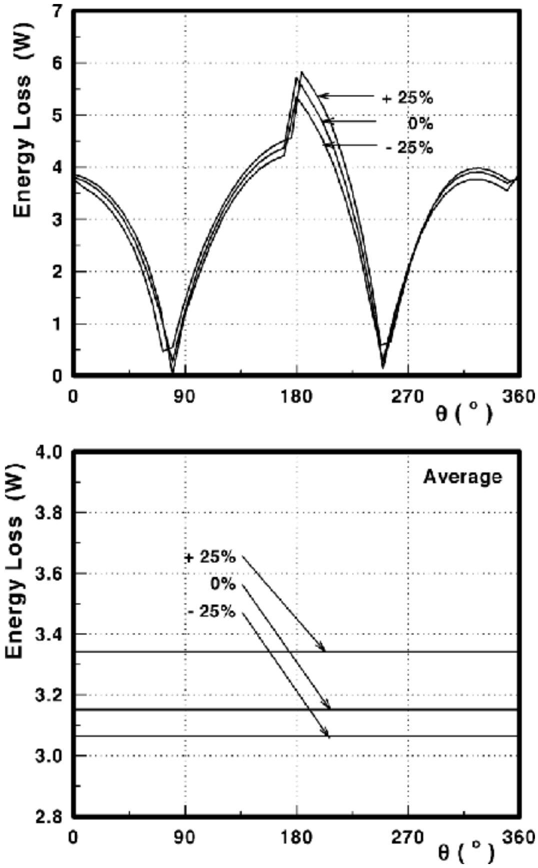

Fig. 10 Energy losses to the variation of the vane center line

In Fig. 9, if the center line position of the vane is moved to the suction chamber, the friction force between the vane and the rolling piston increases. The reason for this increase is that the effect of the discharge pressure acting on the back of the vane increases the position of the vane center line moves towards the suction chamber.

When the operating temperature is 120℃, the effect of the center line position of the vane on the energy loss between the vane and the rolling piston is shown in Fig. 10.

In Fig. 10, if the position of the vane center line is moved to the suction chamber, the energy loss between the vane and the rolling piston increases. The reason for this increase is that the friction force between the vane and the rolling piston increases with the movement of the position of the vane center line to the suction chamber.

5. Conclusions

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

The results are as follows :

(1) The vane thickness significantly influence the friction force and the energy loss between the vane and the rolling piston. The friction force and the energy loss between the vane and the rolling piston increase as the vane thickness increases, because the normal force acting on the vane increases.

(2) The position of the vane significantly influence the friction force and the energy loss between the vane and the rolling piston. The friction force and the energy loss between the vane and the rolling piston increase as the position of the vane thickness moves to the suction chamber, because the normal force acting on the vane increases.

Therefore, the above results show that the modification of the design of a refrigerant compressor is required, and we have to find the optimum conditions to minimize the friction force and the energy loss in a rotary compressor.

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