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Lubrication characteristics of a rotary compressor used for refrigeration and air-conditioning systems (the influence of alternative refrigerants) †

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Abstract

The rolling piston type rotary compressor has been widely used for refrigeration and air-conditioning systems due to its compactness and high-speed characteristics. However, it is necessary to develop alternative refrigerants that can guarantee environmental protection. In addition, advanced refrigerant compressors must be further developed to overcome the compatibility problems inherent in refrigeration and air-conditioning systems. The refrigerant compressor is the most important mechanical component, which determines the performances of refrigeration and air-conditioning systems. Therefore, we theoretically investigated the lubrication characteristics of the rotary compressor currently used in both refrigeration and air-conditioning systems with an alternative refrigerant. In the theoretical investigation, the Runge-Kutta method is used to analyze the behavior of a rolling piston in the rotary compressor. Subsequently, the Newton-Raphson method is used, which provided good performance in the analysis of the elastohydrodynamic lubrication of the line contacts between a rolling piston and a vane in the rotary compressor. The results demonstrate that the alternative refrigerants influence the friction force and the energy loss between the vane and the rolling piston.

Keywords: Alternative refrigerant; Lubrication characteristics; Rolling piston; Rotary compressor; Vane

1. Introduction

The Montreal Protocol on Substances that Deplete the Ozone Layer is a landmark international agreement designed to protect the stratospheric ozone layer. In compliance with this agreement, alternative refrigerants that can guarantee environmental protection must be developed.

Therefore, refrigerant compressor industries have placed much effort in the development of a refrigerant compressor compatible with alternative refrigerants, but which will not show a drop in performance.

Table 1 presents restricted refrigerants and alternative refrigerants for refrigeration and air conditioning systems.

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 operability. In the rotary compressor used for refrigeration and air-conditioning systems, compression motion consists of mechanisms that regulate the compression volume by use of the rolling piston, which rotates around an eccentric shaft and rotates on its axis, and the vane, which undergoes a reciprocating motion in the cylinder slot. Therefore, there are many sliding components in this system [1].

2. Theoretical analysis

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

Lubrication characteristics are critical to system performance, particularly because the asperities of the vane and the rolling piston can cause contacts. Contact pressure supports a portion of the total load and hydrodynamic pressure supports the rest.

Partial EHL analysis is used to ensure an accurate lubrication analysis for critical lubrication conditions. Fig. 1 presents the coordinates of the cylinder part in the rotary compressor for the analysis of lubrication characteristics [2].

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 and u_2 are the surface velocities of two sliding

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Table 1. Restricted refrigerants and alternative refrigerants.

Uses	Refrigerants with restriction	Alternative refrigerants
Car air-conditioner	CFC-12	HFC-134a
Room air-conditioner	HCFC-22 (R22)	R-407C (HFC-32/125/134a=23/25/52%) R-410A (HFC-32/125=50/50%)
Electric refrigerator	CFC-12 R-502	HFC-134a

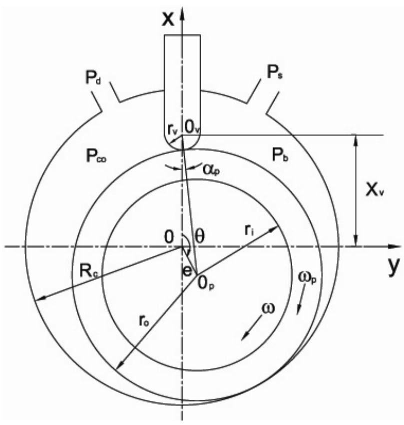

Fig. 1. Schematic diagram of the cylinder part in a rotary compressor.

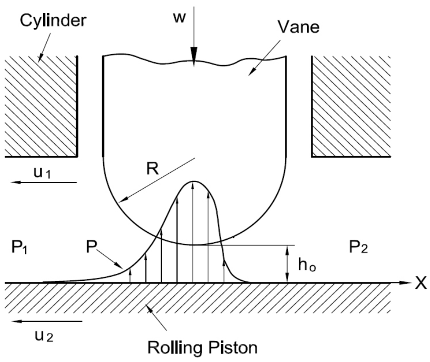

Fig. 2. Model of a line contact.

components in the x -direction.

The equation for the hydrodynamic pressure of rough surfaces with the condition and roughness is denoted as

$$\psi_h \frac{h^3}{12\eta}\frac{dp_h}{dx} = u\left(\overline{h_T} - \overline{h_{Tm}}\right) \tag{1}$$

where ψ_h , u , and p_h are average flow factor, average velocity, and hydrodynamic pressure, respectively. Then, $\overline{h_{Tm}}$ is the average gap height at $dp_h/dx = 0$, while $\overline{h_T}$ is the average gap height given by

$$\overline{h_T} = \int_{-h}^{\infty}\left(h+\varepsilon\right) f(\varepsilon) d\varepsilon \tag{2}$$

where $f(\varepsilon)$ is the probability density function of combined

Table 2. Geometrical shapes and properties of a rotary compressor.

Items	Values	Unit
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

roughness ε , as what has been presented in reference [3]. Viscosity η is assumed to vary with pressure (p) through the relation

$$\eta = \eta_o \exp(\alpha p) \tag{3}$$

where η_o and α are ambient viscosity and pressure-viscosity coefficient, respectively.

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

$$h = h_o + \frac{x^2}{2R} + v \tag{4}$$

where v and h_o are the elastic deformation of two cylinders and film thickness at $x = 0$, respectively.

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

$$w = \int_{x_a}^{x_b} p_h(x)dx + \int_{x_a}^{x_b} p_c(x)dx \tag{5}$$

In reference [4], the contact pressure p_c is calculated by the mean contact pressure-compliance relationship, which will not be repeated here.

The boundary conditions used to analyze the lubrication characteristics are denoted by

$$p_h = p_1$$

at $x = x_a$ (location where the pressure is generated)

$$p_h = p_2 , \quad \frac{\partial p_h}{\partial x} = 0$$

at $x = x_b$ (location where the film is broken).

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

$$f = \int_{x_a}^{x_b} \frac{1}{2}\frac{\partial p_h}{\partial x}dx - \int_{x_a}^{x_b} \eta \frac{u_1}{h} dx - \mu \int_{x_a}^{x_b} p_c dx \tag{6}$$

where μ is the coefficient of friction between the interacting

Table 3. Operating conditions of a rotary compressor with refrigerant (R22, R407C, and R410A) and lubricant.

Items	R22	R407C	R410A	Unit
Suction/Discharge pressure	0.54/2.0	0.54/2.41	0.55/3.18	*MPa*
Rotational speed of shaft	3386	3386	3460	*rpm*
Oil viscosity (at 40℃)	0.050 (SUNISO 4GS)	0.053 (POE VG46)	0.053 (POE VG46)	$Pa \cdot s$

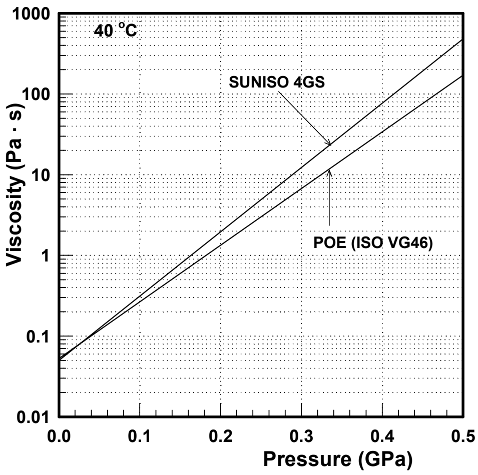

Fig. 3. Viscosity-pressure compliance of the lubricants.

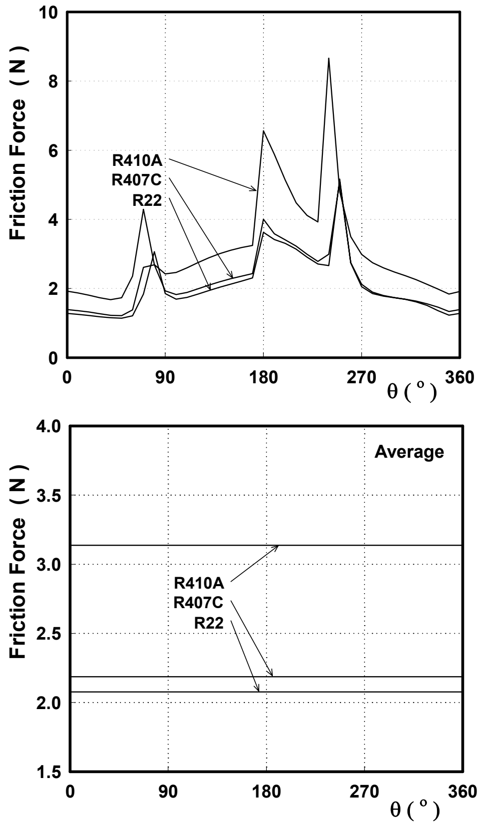

Fig. 4. Friction forces between a vane and a rolling piston versus the type of refrigerants.

asperities.

3. Results

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

The geometrical shapes and properties of the cylinder part in the rotary compressor are summarized in Table 2.

3.1 Effect of the refrigerants

Table 3 shows the actual operating conditions of a rotary compressor at the operating temperature of 40 ℃.

In Table 3, the difference of the operating conditions is caused by a different vapor pressure of the refrigerants. If the lubrication characteristics are analyzed to the operating conditions in the established compressor, the lubrication characteristics of the compressor with the alternative refrigerants will become unreliable. Therefore, it seems proper that the lubrication characteristics of the compressor are analyzed in each actual operating condition, and the analysis results can be used to modify the compressors with alternative refrigerants.

Fig. 3 shows the viscosity-pressure compliance of the lubricants between the vane and the rolling piston of a rotary compressor when the operating temperature is 40 ℃.

The effect of the refrigerants on the friction force between the vane and the rolling piston when the operating temperature is 40 ℃ is demonstrated in Fig. 4.

In Fig. 4, the friction forces have two peak points (near 80° and 250°). 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 forming between the vane and the rolling piston becomes thin, the friction forces become extremely high. One reason may be the influence of the third term of Eq. (6), which is relatively larger than those of the other terms. Therefore, it seems at that these points at almost all regions of the vane tip are in dry contact. 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. One reason may be the influence of the second term of Eq. (6), which is relatively larger compared with other terms. Moreover, the friction force increases as the step shape draws near 180° because of the vane load increasing dramatically near the point.

When the established refrigerant, the R22 with mineral oil SUNISO 4GS, is used, the friction force between the vane and the rolling piston is reduced to the smallest value. Meanwhile, when the alternative refrigerant, the R410A with synthetic oil POE (Polyol Ester) VG46, is used, the friction force attains the largest value. However, since R22 is a restricted refriger-

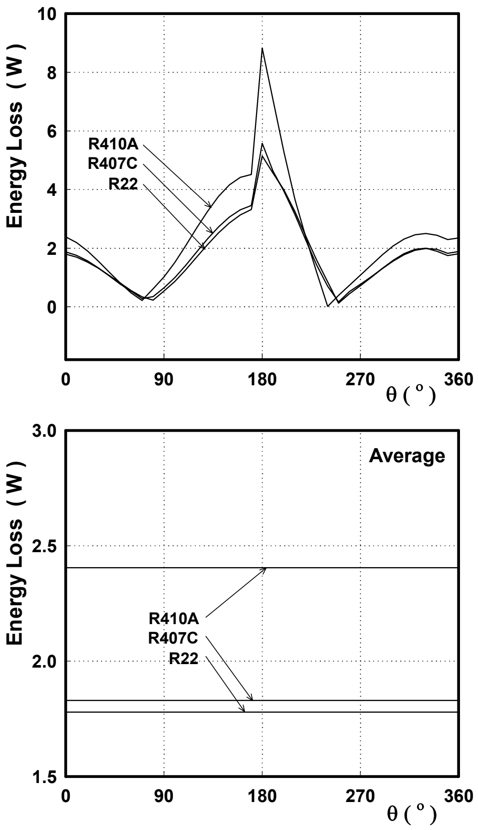

Fig. 5. Energy losses between a vane and a rolling piston versus the type of the refrigerant.

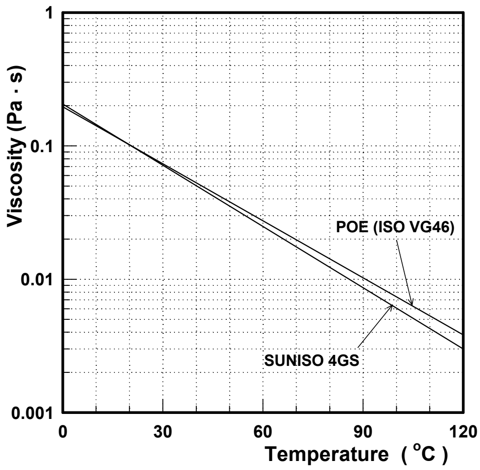

Fig. 6. Viscosity-temperature compliance of the lubricants.

ant, an alternative refrigerant must be used.

The R407C with POE VG46 has a slightly larger friction force than the R22 with SUNISO 4GS, but the R410A with POE VG46 has a much larger friction force than the R22 with SUNISO 4GS. The difference is due to the effect of the specifications of the refrigerant with lubricant and the effect of the normal force, particularly by the discharge pressure, acting on the vane under the operating conditions.

When the operating temperature is 40 ℃, the effect of the refrigerants on the energy loss between the vane and the rolling piston is shown in Fig. 5.

In Fig. 5, when R22 with SUNISO 4GS is used, the energy loss between the vane and the rolling piston becomes the smallest. Additionally, when the alternative refrigerant, the R410A with POE VG46, is used, the energy loss becomes the largest.

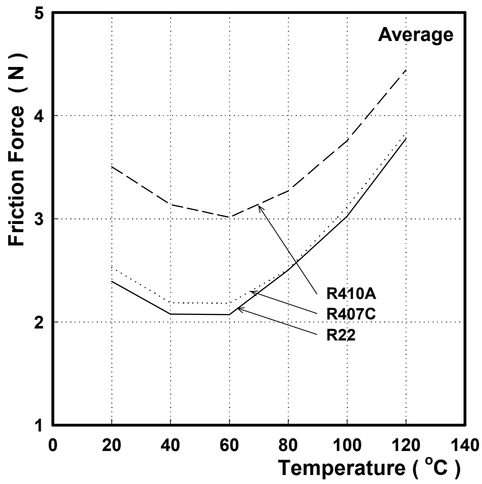

Fig. 7. Friction-temperature characteristics between a vane and a rolling piston for various refrigerants.

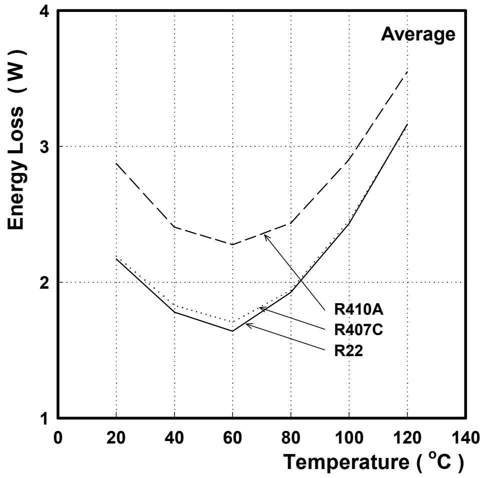

Fig. 8. Energy losses between a vane and a rolling piston versus the temperature.

3.2 Effect of the operating temperature

Fig. 6 shows the viscosity-temperature compliance of the lubricants between the vane and the rolling piston of a rotary compressor.

Fig. 7 presents the effect of refrigerants on the average friction force between the vane and the rolling piston for various operating temperatures.

The R407C with POE VG46 has a slightly larger friction force than the R22 with SUNISO 4GS, but the R410A with

POE VG46 has a much larger friction force than the R22 with SUNISO 4GS. This difference is due to a reason similar to that shown in Fig. 4.

Likewise, if operating temperature rises, the average fluid friction force between the vane and the rolling piston decreases while the average contact friction force between the vane and the rolling piston increases. Therefore, there is a point where the average friction force attains the smallest value, which results from the effect of the increasing contact ratio caused by the decrease of viscosity as the operating temperature increases.

For the variables of operating temperature, the effect of the refrigerants on the average energy loss between the vane and the rolling piston is shown in Fig. 8.

The R407C with POE VG46 has a slightly larger energy loss than the R22 with SUNISO 4GS, but the R410A with POE VG46 has a much larger energy loss than the R22 with SUNISO 4GS. This difference is due to a reason similar to the process outlined in Fig. 7. This result shows that there are certain optimum temperature conditions in which the energy loss between the vane and the rolling piston is minimized in a rotary compressor.

4. Conclusions

In this paper, partial EHL characteristics were 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 refrigerants and the lubricants significantly influence the friction force and the energy loss between the vane and the rolling piston, and the friction force and the energy loss for alternative refrigerants with POE VG46 are larger than those for R22 with SUNISO 4GS.

(2) With regards to temperature, if the operating temperature rises, the average fluid friction force between the vane and the rolling piston decreases while the average contact friction force between the vane and the rolling piston increases. Thus, there are optimum conditions in which the friction force and energy loss between the vane and the rolling piston are minimized in a rotary compressor.

The abovementioned results show the need to develop an alternative refrigerant compressor because friction force and energy loss for alternative refrigerants are larger than those of established refrigerants. Moreover, results show that there are certain optimum conditions in which the friction force and energy loss between the vane and the rolling piston are minimized in a rotary compressor.

Nomenclature

e : Eccentric length
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 $x = 0$
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_{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 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
ψ_h : Pressure flow factor
η : Viscosity of lubricant
η_o : Viscosity of lubricant at ambient pressure
θ : Rotational angle of the eccentric shaft
μ : Friction coefficient between the interacting asperities
ϖ : Angular velocity of the eccentric shaft
ϖ_p : Angular velocity of the rolling piston

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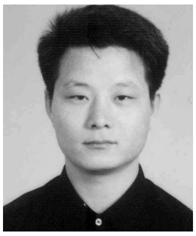

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