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DYNAMIC ANALYSIS OF ROLLER AND VANE OF INVERTER CONTROLLED
ROTARY COMPRESSORS

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ABSTRACT

This paper presents a complete dynamic analysis of roller and vane of a rotary compressor widely used for air-conditioners. An approach and a set of equations are developed estimating the type of motion at vane tip and roller interface and calculating the frictional force in accordance with the motion type. The forces on roller caused by relative motion between roller and cylinder wall are analyzed. The relative motion between vane tip and roller and the rotating velocity of roller of inverter rotary compressors and HFC410A rotary compressors are discussed in detail. The effect of secondary pressure pulse in compression chamber on the pursuing ability of vane as the compressors operate at high speed is shown.

INTRODUCTION

Roller (rolling piston) and vane are two important moving parts of a rolling piston rotary compressor widely used for room air conditioners and household refrigerators. The forces acting on them and their motions are extremely complex and coupled. The friction loss related to their motions is main part of total friction loss of a rotary compressor. And their wears greatly influence the life of the rotary compressor. Several papers were published which analyzed the motion of roller in the early development period of rotary compressors. Among other things, Yanagisawa [1] measured the friction coefficient at vane and roller interface and proved that positive and negative sliding between vane tip and roller occurred alternately during one revolution of shaft. Okada and Kuyama [2] emphasized the type of contact between vane tip and roller. An initial value of angular velocity of roller was assumed and the advanced Euler's way of numerical integration was adopted when equations were solved numerically. In recent years, due to increasing concern for energy saving and abolition of HCFC22, inverter controlled rotary compressors and HFC410A rotary compressors have being developed. For both types of compressors, the lubrication condition and wear at vane tip are roller interface and at vane and vane slot sides are more serious.

There are many factors that affect the wear of vane and roller. However, load and relative sliding velocity are primary in any quantitative analysis. The type of relative motion at vane tip and roller interface besides them also influences the wear of vane tip and roller outer surface [3]. Wear is severer as the direction of relative sliding velocity changes or dynamics mode is changed from rolling to sliding. However, the papers in the early period and the recent papers except the Okada's did not district the type of relative motion at vane tip and roller interface in their equilibrium equations of forces and moments acting vane and roller. It was always assumed that the friction force between vane and roller was proportional to the normal force. In fact, the relative motion mode between vane tip and roller may be pure rolling in some situation, for example, in high-
pressure difference or in low crankshaft speed. Thus neglecting the characteristic of pure rolling would make certain calculation error for the magnitude of sliding velocity, relative motion type between vane tip and roller and its change, especially for the HFC410A rotary compressors and inverter rotary compressors operating at low speed. In order to determine accurately the angular velocity of roller and the relative motion between vane tip and roller, an approach and a set of equations are developed estimating the type of motion and calculating the frictional force in accordance with the motion type in this paper. The changes of the angular velocity of roller and the relative motion at vane tip and roller interface with crankshaft speeds of two inverter rotary compressors and the relative motion between vane and roller of an HFC410A rotary compressor are shown.

The radial clearance between roller outer surface and cylinder wall is filled with oil during the normal operation. The relative motion between roller and cylinder wall causes normal and tangential forces acting the roller. Litter analysis was undertaken for the forces. Padhy \cite{padhy} assumed the tangential force acting roller was the centrifugal force times 0.001 and neglected the normal force. However, the normal force or tangential force may cease being negligible when an inverter rotary compressor operates at high speed. This paper analyze the normal force caused by rolling and sliding of roller relative to cylinder wall and by squeezing oil film in the roller radial clearance and the frictional force due to viscous drag.

The variations of the magnitude and the direction of load on eccentric journal bearing by roller are great. This makes that the variation of oil film thickness of eccentric bearing is also great and within certain crankshaft rotating angle range the oil film is so thin that the bearing is at boundary lubrication. In this paper, the frictional torque acting on roller by the bearing is evaluated according to the lubrication type of the bearing at each rotating angle of crankshaft that is judged from oil film thickness of the bearing at the right moment, which can be acquired from the calculated results of loci of shaft center of the bearing.

When an inverter rotary compressor operates at a high speed, the pursing ability of its vane decreases. It is showed in this paper that the pursing ability of the vane could be effectively improved by means of reducing the secondary pressure pulse in compression chamber.

**THEORETICAL ANALYSIS**

Vane dynamics

The constraint forces on the vane sides by vane slot are defined as two forces applied by the outer edge or the inner edge of slot \( R_{n1} \) and \( R_{n2} \) instead of defining the reaction forces on suction side and on discharge side as Phday \cite{phday}. If the \( R_{n1} \) points to suction side, it is defined as positive. And if the \( R_{n2} \) points to discharge side, it is positive. Thus the combinations of their sign show the declining direction of vane and then reflect the slapping motion of vane in slot.

By introducing \( \delta_1 = \begin{cases} 1, & 0 \leq \theta \leq \pi; \\ -1, & \pi \leq \theta \leq 2\pi, \end{cases} \)

\( \delta_2 = \frac{R_{n1}}{|R_{n1}|}, \) \hspace{1cm} \( \delta_3 = \frac{R_{n2}}{|R_{n2}|}\)

the friction forces corresponding to \( R_{n1} \) and \( R_{n2} \) become:

\[ R_{n1} = \delta_1 \delta_2 \mu_w R_{n1}, \quad R_{n2} = \delta_3 \delta_2 \mu_w R_{n2} \quad (1) \]
\( \mu_{sc} \) is the frictional coefficient.

When the roller slides against vane the friction force between vane tip and roller is given by:

\[
F_{sr} = \delta_s \mu_{sc} F_{vm}
\]

\[\delta_s = \frac{v_s}{|v_s|}\]

in which \( v_s \) is the sliding velocity of roller relative to vane tip. But when the relative motion between the roller and vane is pure rolling the previous formula is not valid. \( F_{sr} \) need be solved from the motion equation of roller.

The lengthways and transverse gas forces exerted on vane \( F_{vg} \) and \( F_{vg} \), the counterclockwise moments about vane nose center \( O_v \) caused by the gas forces \( M_{vg} \) and \( M_{vg} \), spring force \( F_s \), inertia force \( F_i \) and viscous force \( F_f \) can be calculated by the gas pressure in working chambers and rotating speed of crankshaft. So considering equation (1) and (2), when the roller slides against the vane, the constraint forces or unknown forces are given by:

\[
\begin{bmatrix}
\delta_s \mu_{sc} & \delta_t \mu_{sc} & \cos \alpha + \delta_t \mu_{sc} \sin \alpha \\
1 & -1 & -\sin \alpha + \delta_t \mu_{sc} \cos \alpha \\
c_t + L_{rv} & c_t & 0
\end{bmatrix}
\begin{bmatrix}
R_{s1} \\
R_{s2} \\
F_{vm}
\end{bmatrix}
= 
\begin{bmatrix}
c_f \\
-R_{vg} \\
-M_{vg} - M_{vg}
\end{bmatrix}
\]

(3a)

When the relative motion between the roller and vane is pure rolling \( F_{sr} \) is thought as a known force. So we have:

\[
\begin{bmatrix}
\delta_s \mu_{sc} & \delta_t \mu_{sc} & \cos \alpha \\
1 & -1 & -\sin \alpha \\
c_t + L_{rv} & c_t & 0
\end{bmatrix}
\begin{bmatrix}
R_{s1} \\
R_{s2} \\
F_{vm}
\end{bmatrix}
= 
\begin{bmatrix}
c_f - F_{sr} \sin \alpha \\
-F_{vg} - F_{i} \cos \alpha \\
-M_{vg} + F_{sr} + F_{r} R_{v}
\end{bmatrix}
\]

(3b)

in which \( c_f = F_{vg} + F_i - F_{vm} - F_{r} \) and \( c_t = R_c - X_v + \delta_t \mu_{sc} \delta_s / 2 \).

The equation group is different from general liner equation group in that its coefficient matrix includes the properties that are relevant to the sign of unknown variables.

**Roller dynamics.**

The resultant normal force of fluid dynamic pressure caused by rolling and sliding of roller outer surface relative to cylinder wall is evaluated by Matin's equation. The average velocity causing fluid dynamic pressure effect is obtained by the absolute velocity of roller and cylinder at the minimal radial clearance between roller and cylinder wall and the absolute velocity of the minimal radial clearance itself:

\[
u = \left[ (u_t - u_c) + (u_r - u_c) \right] / 2 = (2 \omega_r R_c - \omega_t e - \omega_r R_{ro}) / 2
\]

in which \( R_c \) is radius of cylinder, \( R_{ro} \) is radius of roller outer surface, \( e \) is eccentricity and \( \omega_t \) and \( \omega_r \) are angular speeds of crankshaft and roller. The resultant normal force are given by:

\[
F_{n1} = 4.9 \mu_{sc} H_{r} R_c R_{ro} \left[ (R_c - R_{ro}) [\omega_r R_c - (\omega_t e + \omega_r R_{ro}) / 2] / \delta_{sc} \right]
\]
\( \mu_{orc} \) is the dynamic viscosity of oil within the clearance, \( \delta_{rc} \) is the minimum of the clearance and \( H_c \) is the height of cylinder.

The normal force exerted by roller radial squeezing oil film in the clearance is evaluated by:

\[
F_{on2} = -12\pi\mu_{orc}R_c^2H_c\left[\varepsilon^3(1+\varepsilon^2)^2\right]\frac{d\delta_{rc}}{dt}
\]

in which \( \varepsilon = 1 - \delta_{rc}/\varepsilon \).

Neglecting the change of oil viscosity along with the clearance, viscous drag on roller outer surface by oil in the radial clearance between roller and cylinder wall is evaluated by:

\[
F_{on} = \mu_{orc}H_cL_\alpha V_\alpha/\delta_{rc}
\]

\[
L_\alpha = 2R_c(\pi-\gamma)\delta_{rc}/\sqrt{\delta_{rc}(2\varepsilon-\delta_{rc})}
\]

The friction moment on the inside surface of roller by eccentric shaft is calculated by the attitude \( \varepsilon_{eb} \) and attitude angle of eccentric journal bearing \( \varphi_{eb} \):

\[
m_{eb} = 2\pi\mu_{eb}R_{eb}^3(\omega_y-\omega_x)/[\varepsilon_{eb}\sqrt{1-\varepsilon_{eb}^2} - F_{eb}\varepsilon_{eb}\sin\varphi_{eb}]/2
\]

The \( \varepsilon_{eb} \) and \( \varphi_{eb} \) are given from the differential equation of the loci of shaft center of the bearing \([5]\). Since magnitude and direction of load of the bearing are great, the variations of \( \varepsilon_{eb} \) and oil film thickness of the bearing are also great. If the oil film thickness is less than a certain value at some rotating angle the bearing is in mixed or boundary lubrication condition and the frictional torque is as follows:

\[
m_{eb} = \mu_{eb}F_{eb}R_{eb}\cos\varphi_{eb}
\]

So we have the motion equation for the roller:

\[
I_r\frac{d\omega_r}{dt} = m_{eb} - m_a - R_{ro}(F_a + F_{on})
\]

in which \( m_a \) is the moment at roller faces.

The relative motion and frictional force between vane and roller

The relative sliding velocity between vane and roller which is positive as the friction force exerted on vane by roller points to suction side from discharge side is given by the tangential velocities of vane and roller at contact point:

\[
V_s = [R_{ro}\omega_r + e\omega_x, \cos(\theta + \alpha)] - [-e\omega_y, (\cos\theta g \alpha + \sin \theta) \sin \alpha] = R_{ro}\omega_r + e\omega_x, \cos \theta / \cos \alpha
\]

If \( V_s = 0 \) at some rotational angle of crankshaft, the relative motion and frictional force between vane and roller is pure rolling at the right moment. If the derivative of \( V_s \) with respect to time is zero, the type of motion will continue. By equation (4) and the following equation:

\[
\frac{dV_s}{dt} = R_{ro}\frac{d\omega_r}{dt} + e\frac{\cos \theta d\omega_x}{\cos \alpha} - e\omega_x^2 \frac{\sin \theta}{\cos \alpha} + \lambda e\omega_x^2 \frac{\cos^2 \theta \sin \alpha}{\cos^2 \alpha} = 0
\]

we get:
\[ F_v = (m_{zh} - m_a - l) \left( \frac{d\omega}{dt} \right) R_{ro} - F_a \]  

\[ \frac{d\omega}{dt} = \frac{e}{R_{ro} \cos \alpha} \left[ (\cos \theta \frac{d\omega}{dt} - \omega^2 \sin \theta + \lambda \omega^2 \cos^2 \theta \cos \alpha) \right] \]

When \( V_r = 0 \), it is assumed at first that the pure rolling continues. Then \( F_v \) is obtained by solving equation (5) and (6) using iteration. If the \( F_v \) satisfies the following condition:

\[ -\mu_{ny} F_{ny} \leq F_v \leq \mu_{ny} F_{ny} \]

it is indicated that the pure rolling can continue. Otherwise the pure rolling can not continue and thus we should use equation (3a) to calculate \( F_v \), but \( \delta = \text{sign} \left( \frac{dV_r}{dt} \right) \frac{dV_r}{dt} \left( F_v - \mu_{ny} F_{ny} \right) \).

RESULTS AND DISCUSSIONS

Figure 1 and Figure 2 show the relative sliding velocity between vane tip and roller and the rotating velocity of roller of the inverter rotary compressor which has cylinder diameter of 44mm, crankshaft eccentric of 4.2mm and roller thickness of 4.5mm. When the current frequencies output by inverter are 45, 50, 60, 90 and 120Hz the maximal sliding velocities are 0.3, 0.5, 1.1, 2.8 and 4m/s respectively. Obviously the sliding velocity increases more quickly. Figure 3 and Figure 4 show those of another compressor, which has cylinder diameter of 57mm, and same eccentric and roller thickness. Its sliding velocity at lower current frequency is higher than that of the previous compressor and the fluctuation of rotating velocity of roller is lower since the outer diameter and mass of its roller is greater. For the smaller compressor, the relative pure rolling between vane tip and roller outer surface appears at 60Hz as shown Figure 1. At 50Hz and 45Hz, the range of pure rolling becomes larger. The relative motion of vane and roller is pure rolling almost for one revolution at 30Hz, which is not shown in the figure. Above calculations are taken at the condition that the pressure difference between suction and discharge is 1.5MPa. When the pressure difference decreases to 1.0MPa, the relative motion becomes sliding at some rotating angle of crankshaft, as shown Figure 5. When the frictional coefficient at vane and roll interface decreases from 0.15 to 0.11 the range of sliding extends.

Dynamics of R410A rotary compressor is analyzed by applying the size of previous smaller compressor. However, the dynamic viscosity of oil is taken as \( 4 \times 10^{-3} \) Pa s so that the oil film thickness of eccentric bearing calculated could be comparable with that of R22 compressor. The relative motion between vane and roller at 50Hz and 60Hz is shown in Figure 6. It is seen that the relative motion modes at common shaft speed only include positive sliding and pure rolling. This is beneficial to decreasing wear of vane and roller[1].

The viscous force between roller outside surface and cylinder wall is negligible even for large compressor operating at high speed because for previous larger compressor its maximum is just about 0.5N at 150Hz while the friction force between vane and roller is about 23N. The normal force caused by roller and cylinder squeezing oil film in the radial clearance between roller and cylinder is also negligible. However, the normal force caused by sliding and rolling of roller relative to cylinder wall is not negligible. For the previous large compressor operating at 150Hz, the maximum of normal force is about 700N whereas the maximum of component of gas force in crankshaft eccentric direction is 1200N. When the normal force is considered the
calculated value of minimal roller radial clearance is 2.5 μ while when the normal force is neglected that is 3.8 μ. At 120Hz the maximal normal force is 460N and the calculated error of minimal radial clearance is 1.1 μ. Since it is very important for guaranteeing the performance and reliability of rotary compressors to accurately estimate the minimum of the roller radial clearance, the normal force caused by relative motion between roller and cylinder is not negligible for the rotary compressor which has large cylinder diameter or operates at high speed.

Figure 7 and Figure 8 show the normal and tangential forces on vane by roller and vane slot at 60Hz and 120Hz. The sign of tangential force by roller, which represents its direction, is consistent with the sign of relative sliding velocity between vane and roller as shown Figure 1. The cause of that the normal force by roller jumps at 180° is that the friction forces by vane slot change their directions at the rotating angle. The vane slapping motion in slot can be made out from the variation of the signs of normal forces by vane slot.

When an inverter rotary compressor operates at a high speed, the vane inertia force increases and the normal force between vane and roller rises in the vicinity of 180° but falls about at 360°. The rise of normal force increases friction loss. The fall of normal force decreases the seal force between vane tip and roller and even causes vane to separate from roller. The effect of vane density on the pursuing ability of vane is great. However, the effect of the lengthways gas force on vane is also important. Figure 9 and Figure 10 show the pressures in compression chambers of compressors with three different structures, the corresponding lengthways gas forces on vane and the normal forces between vane tip and roller. The angle included between notch and cylinder axis of Compressor I is 45°. After shaft angular displacement passes 330° the pressure in compression rises quickly to form a secondary pressure pulse and becomes 3.5MPa about at the rotating angle of 350° because the flow area between compression chamber and notch is very small at this time. Thus about at the angle of 350° the lengthways gas force on vane is so small that the vane is separated from the outer surface of roller. When the notch angle is 35° as Compressor II, the pressure in compression chamber falls and the discharge loss decreases from 488W to 462W. But the amplitude of secondary pressure pulse is not small enough to avoid vane separating from roller. If the notch is connected with a large chamfer at discharge side of vane slot as Compressor III, the flow area between compression chamber and clearance volume increases to a great extent near 350° and the secondary pressure pulse obviously decreases as shown in Figure 9. The minimum of gas force on vane increases from 18N to 68N. The minimal normal force between vane and roller becomes 30N and the vane presses the roller all the time.

CONCLUSIONS

This paper presents a complete dynamic analysis of roller and vane of rotary compressors. An approach is developed estimating the type of motion and calculating the friction force at vane tip and roller interface in order to determine accurately the relative motion between vane and roller. The relative motion between vane and roller and the angular velocity of roller of inverter rotary compressors and HFC410A rotary compressors are discussed. The calculated results show that the viscous force and the radial squeeze force of oil film between roller and cylinder wall are negligible, but the normal force caused by relative rolling and sliding motion between roller and cylinder wall should not be neglected because it has significant effect on the calculated result of roller radial clearance. It is shown that the pursuing ability of vane of inverter rotary compressors could be effectively improved by reducing the secondary pressure pulse in compression chamber.
REFERENCES

[1] Yanagisawa T, etc., Motion analysis of rolling piston in rotary compressor, 82ICECP, pp185–192

Fig.1 Sliding velocity between vane and roller for small comp.
Fig.2 Rotating speed of roller for small comp.

Fig.3 Sliding velocity between vane and roller for large comp.
Fig.4 Rotating speed of roller for large comp.
Fig. 5 Sliding velocity between vane and roller at 30 Hz.

Fig. 6 Sliding velocity between vane and roller for R410A.

Fig. 7 Forces on vane by roller and slot at 60Hz.

Fig. 8 Forces on vane by roller and slot at 120Hz.

Fig. 9 Pressures in compression chambers and gas forces on vane.

Fig. 10 Normal forces between vane tip and roller.