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Influence of Clearance Volume on the Performance of the Rolling Piston Type Rotary Compressor*

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This paper is concerned with the rolling piston type rotary compressor for the air conditioner. The rotary compressor, as well as the reciprocating compressor, is one of the positive displacement compressors and has clearance volume of discharge port. But the influence of clearance volume on the performance of the rotary compressor is not equal to that of the reciprocating compressor. We analysed a reverse flow of suction gas through a suction opening caused by the re-expansion of high pressure gas in clearance volume taking account of throttling effects of suction and discharge openings. We investigated influences of operating conditions and dimensional parameters on volumetric efficiency. Also we estimated compressor work using p - V diagram. It was made clear that clearance volume of the rotary compressor hardly decreases volumetric efficiency but increases compressor work in contrast with that of reciprocating compressor. Theoretical results are in good agreement with experimental results.

Key Words: Fluid Machine, Thermal Engineering Equipment, Refrigeration, Rotary Compressor, Clearance Volume, Volumetric Efficiency

1. Introduction

The rolling piston type rotary compressor is one of the positive displacement compressors. The rolling piston rotates in the cylinder eccentrically, which causes suction, compression and discharge works of the compressor. At the end of discharge process, there exists some space under discharge valve, namely clearance volume, which cannot be swept away by the revolution of rolling piston.

In the past, the influence of such clearance volume on the performance of the compressor has been considered to decrease volumetric efficiency by the effect of re-expansion⁽¹⁾⁻⁽⁴⁾, in the same manner as reciprocating compressor. This may be the case with the rotascope type rotary compressor which works basically in the same way as the rolling piston type rotary compressor and rotates rather low speed with large size⁽⁵⁾⁽⁶⁾. But it is doubtful that this can be applied to the high speed, small size rolling piston type rotary compressor for household air conditioner.

Therefore we try to analyse the influence of clearance volume on the performance of such a compressor taking account of geometrical shapes of suction and dis-

charge ports. Theoretical results are compared with experimental results which are obtained from practical compressors with different clearance volumes.

2. Nomenclature

A : restriction area
 c : flow coefficient
 d : port diameter
 $f(\varphi)$: radial distance {see Eq.(5)}
 m : mass
 $\dot{m}$: mass flow rate
 n : rotational frequency
 P : electric power input
 p : pressure (abs.)
 R, r : radii of cylinder and rolling piston
 V : volume
 V_0 : swept volume
 $\dot{W}$: compressor work
 β : angular distance ($=\theta_{s2}-\theta_{d1}$)
 ϵ : clearance volume ratio
 η_v : volumetric efficiency
 θ : rotational angle
 θ_{d1}, θ_{d2} : front and rear angles of discharge opening
 θ_{s1}, θ_{s2} : front and rear angles of suction opening
 κ : adiabatic exponent
 λ : pressure ratio
 ρ : density
 ω : angular velocity

Subscripts

b : suction chamber
 d : discharge
 s : suction
 v : clearance volume

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3. Theoretical Analysis

Figure 1 shows a schematic view of the rolling piston type rotary compressor. A rolling piston mounted on a shaft eccentric divides the cylinder room into two chambers, i.e., suction and compression chambers, associated with a vane. As the shaft rotates, the rolling piston rotates with the eccentric in the cylinder, which causes suction work in the suction chamber and compression and discharge work in the compression chamber. When the rolling piston passes by the discharge port, as shown in Fig.2, high pressure gas in the clearance volume which exists under the discharge valve goes into the suction chamber.

By the way, in the case of a reciprocating compressor with similar clearance volume, volumetric efficiency of the compressor decreases inevitably because suction flow into the cylinder doesn't occur till the pressure in the cylinder becomes lower than the suction pressure of the compressor, but re-expansion work in the cylinder is recovered as the re-expanding gas pushes the piston⁽⁷⁾.

On the contrary, in the case of a rolling piston type rotary compressor, suction flow into the cylinder is almost finished when re-expansion occurs and

Fig.1 Schematic view of the rolling piston type rotary compressor

re-expansion work is consumed as friction loss at the discharge opening. Therefore, the influence of clearance volume on the performance of the rotary compressor is different from that of the reciprocating compressor.

3.1 Simple analysis of reverse flow rate

In the case of a rolling piston type rotary compressor, as mentioned above, the gas in the clearance volume flows into the crescent-shaped cylinder room through discharge opening and mixes with previously suctioned gas in the room. The mixed gas flows reversely through suction opening and suction port from inside to outside of the cylinder. We model the reverse flow channel as a channel shown in Fig.3. The model channel is developed along the cylinder inner circumference and includes two restrictions, namely suction and discharge openings, and cylinder volume between clearance volume and suction port. The reverse flow rate must be calculated as an unsteady and compressible flow strictly, but the calculation is very complicated. Therefore we assume in this study a steady and compressible flow to calculate instantaneous reverse flow rate with emphasis on the simplicity rather than on the strictness of the analysis of the clearance volume influence on the compressor performance. Also we assume the gas in control volume mixes with reverse flow and becomes uniform immediately.

Fig.2 Explanation of reverse flow channel

Fig.3 Model of reverse flow channel

Expressing the pressure in clearance volume as p_v , pressure in cylinder volume as p_b and pressure in suction port as p_s which is the constant compressor suction pressure, instantaneous mass flow rates $\dot{m}_d$ through discharge opening and $\dot{m}_s$ through suction opening are expressed as follows.

$$\dot{m}_d = c_d A_d \sqrt{2 \frac{\kappa}{\kappa-1} p_v \rho_v \{ \lambda^{2/\kappa} - \lambda^{(\kappa+1)/\kappa} \}} \quad \dots\dots(1)$$

$$\text{Here } \lambda = p_b/p_v \quad (p_b/p_v \geq \lambda_{cr})$$

$$\lambda = \lambda_{cr} \quad (p_b/p_v < \lambda_{cr})$$

$$\dot{m}_s = c_s A_s \sqrt{2 \frac{\kappa}{\kappa-1} p_s \rho_s \{ \lambda^{2/\kappa} - \lambda^{(\kappa+1)/\kappa} \}} \quad \dots\dots(2)$$

$$\text{Here } \lambda = p_s/p_b \quad (p_s/p_b \geq \lambda_{cr})$$

$$\lambda = \lambda_{cr} \quad (p_s/p_b < \lambda_{cr})$$

Where κ is adiabatic exponent and λ_{cr} is critical pressure ratio equal to $\{2/(\kappa+1)\}^{\kappa/(\kappa-1)}$. Then discharge opening area A_d and suction opening area A_s change with shaft revolution. As shown in Fig.4, when discharge opening is machined to semi-conical shape, with bottom diameter d_d and height $d_d/2$, area A_d of restriction between V-shaped edge of discharge opening and outer circumference of rolling piston is expressed by Eq.(3). And cylindrical restriction area A_s between suction opening on cylinder inner circumference and rolling piston outer circumference is expressed by Eq.(4).

$$A_d = \begin{cases} \sqrt{2} R (\theta - \theta_{d1}) f\left(\frac{\theta - \theta_{d1}}{2}\right) & (\theta_{d1} \leq \theta \leq \theta_{d2}) \\ \sqrt{2} d_d f\left(\theta - \frac{\theta_{d1} + \theta_{d2}}{2}\right) & (\theta_{d2} < \theta \leq \theta_{s2}) \end{cases} \quad \dots\dots(3)$$

Fig.4 Explanation of restriction area

$$A_s = \begin{cases} \pi d_s f\left(\frac{\theta_{s1} + \theta_{s2}}{2} - \theta\right) & (\theta_{d1} \leq \theta \leq \theta_{s1}) \\ (\pi - \alpha) d_s f\left(\frac{\theta_{s2} - \theta}{2}\right) & (\theta_{s1} < \theta \leq \theta_{s2}) \end{cases} \quad \dots\dots(4)$$

Where $\alpha = \cos^{-1}\{1 - 2R(\theta - \theta_{s1})/d_s\}$ and function $f(\varphi)$ is radial distance between cylinder inner circumference and rolling piston outer circumference at the angle φ from the direction of rolling piston center as shown in Fig.4.

$$f(\varphi) = R - r \cos\left\{\sin^{-1}\left(\frac{R-r}{r} \sin \varphi\right)\right\} - (R-r) \cos \varphi \quad \dots\dots(5)$$

Pressures in clearance volume and cylinder volume are expressed as follows assuming adiabatic change.

$$p_v = p_d (\rho_v/\rho_d)^\kappa = p_s (\rho_v/\rho_s)^\kappa \quad \dots\dots(6)$$

$$p_b = p_s (\rho_b/\rho_s)^\kappa \quad \dots\dots(7)$$

Then gas density in each volume is approximated by next equation.

$$\rho_v = \left(m_v - \frac{1}{\omega} \int_{\theta_{d1}}^{\theta} \dot{m}_d d\theta\right) / V_v \quad \dots\dots(8)$$

$$\rho_b = \left(m_b + \frac{1}{\omega} \int_{\theta_{d1}}^{\theta} \dot{m}_d d\theta - \frac{1}{\omega} \int_{\theta}^{\theta_{s2}} \dot{m}_s d\theta\right) / V_b \quad \dots\dots(9)$$

Where m_v and m_b are initial ($\theta = \theta_{d1}$) masses of gas in clearance volume V_v and cylinder volume V_b , namely $m_v = \rho_d V_v$ and $m_b = \rho_s V_b$. Though V_v and V_b change during rolling piston revolution from θ_{d1} to θ_{s2} , their values are represented by the values at $\theta = 0$ because their changing rates are so small.

Reverse mass flow rate Δm_s through suction opening during rolling piston revolution from θ_{d1} to θ_{s2} is calculated as follows.

$$\Delta m_s = \frac{1}{\omega} \int_{\theta_{d1}}^{\theta_{s2}} \dot{m}_s d\theta \quad \dots\dots(10)$$

Then volumetric efficiency drop $\Delta \eta_v$ due to reverse flow from clearance volume is expressed by Eq.(11) and overall volumetric efficiency η_v is expressed by Eq.(12) using η_{v0} which is the volumetric efficiency excluding the influence of clearance volume.

$$\Delta \eta_v = \Delta m_s / (\rho_s V_0) \quad \dots\dots(11)$$

$$\eta_v = \eta_{v0} - \Delta \eta_v \quad \dots\dots(12)$$

3.2 Analysis of compressor work

Influence of clearance volume on the compressor work will be analysed using pressure volume (p - V) diagram in suction and compression chambers. Figure 5 shows a simplified p - V diagram when the compressor with swept volume V_0 and clearance volume

V_V receives gas at pressure p_s and compresses and discharges it at pressure p_d . In suction chamber, gas volume V_0 is received during one revolution of the compressor and p - V relation is expressed by the horizontal line EH in Fig.5 neglecting suction loss. On the other hand, in compression chamber, compression starts at point A' with volume (V_0+V_V) and pressure p_s' which is the pressure in suction chamber at the end of suction expressed by next equation.

$$p_s' = p_s \left(\frac{V_0}{V_0+V_V} \right)^K \left(\frac{m_b+m_V-\Delta m_s}{m_b} \right)^K \quad (13)$$

From Point A' to point B' (pressure p_d) gas in compression chamber is compressed adiabatically and it is discharged at pressure p_d during the period expressed by the line B'C. Therefore, compressor work during one revolution is evaluated by the area (A'B'CFA) obtained by the subtraction of suction work area (HEOK) from compression and discharge work area (A'B'CIL) in p - V diagram. Mathematically it is expressed by next equation.

$$W = \frac{K}{K-1} p_s' (V_0+V_V) \left[\left(\frac{p_d}{p_s'} \right)^{(K-1)/K} - 1 \right] + p_s' (V_0+V_V) - p_s V_0 - p_d V_V \quad (14)$$

In the case of momentary closing of suction opening which causes no reverse flow, compressor work is designated by the area (A''B''CFA) in Fig.5 and in the case of slow closing of suction opening which causes whole reverse flow of high pressure gas in clearance volume, it is expressed by the area (ABCF). By the way, in the case of a reciprocating compressor with same swept volume V_0 and clearance volume V_V , compressor work is expressed by the area (ABCG)⁽⁷⁾. As compared with this area, rotary compressor work area (A'B'CFA) is larger by the area (CGF) of re-expansion work. And additional area (AA'B'B) is compression work of the re-expanded gas which has not flowed reversely through suction opening and has not decreases volumetric efficiency.

Fig.5 Pressure volume (p - V) diagram

4. Experiment

In order to investigate the influence of clearance volume on the performance of the rolling piston type rotary compressor experimentally, we measured compressor flow rate and electric power input while changing clearance volume of the compressor. Two changing methods of clearance volume are shown in Fig.6. The first one, shown by Fig.6(a), utilizes a drilled hole on the cylinder. The hole is connected to the discharge port by the milled channel with rather wide cross section. Clearance volume is changed by the height of the spacer inserted into the hole. In the second method, shown by Fig.6(b), milling of discharge port on the cylinder head changes clearance volume and front angle of discharge opening.

(a)

(b)

Fig.6 Changing method of clearance volume in experiment

The compressor is enclosed in a semi-hermetic shell and joined to the refrigerating cycle using refrigerant 22 (CHClF_2) as working fluid. Under steady state operating condition, flow rate of liquid refrigerant is measured with rotameter and electric power input is measured with watt-meter. Temperatures of compressor and piping line are measured with copper-constantan thermocouples and pressures are measured with precise Bourdon type pressure gauges. In the experiment, in order to minimize thermal influence of clearance volume variation, environment of the compressor is controlled to keep constant oil temperature under the same operating condition. Operating pressures are suction pressure $p_s = 0.583 \text{ MPa}$ $\{5.95 \text{ kgf/cm}^2\}$ (equivalent to saturation temperature 5°C) and discharge pressure $p_d = 1.73, 2.03, 2.43 \text{ MPa}$ $\{17.6, 20.7, 24.8 \text{ kgf/cm}^2\}$ (equivalent to saturation temperature $45, 52, 60^\circ\text{C}$). Superheat of suction gas is 10°C and sub-cool of condensed liquid is 5°C . The compressor is operated with power source of $100\text{V } 60\text{Hz}$, and it rotates at about 3400

rpm. Table 1 shows main dimensions of the compressor used for experiment.

5. Results and Discussions

5.1 Volumetric efficiency

Figure 7 shows experimental volumetric efficiency when clearance volume is changed as shown in Fig.6(a). Its horizontal axis is expressed by the clearance volume ratio $\epsilon (=V_v/V_0)$. Volumetric efficiency η_v is obtained as the ratio of measured flow rate to ideal flow rate of the compressor. The change of η_v is less than 1% in the range of ϵ from 1.5% to 9% though its level is different for each operating pressure condition.

Also theoretical results derived from section 3.1 are illustrated in Fig.7 referring to experimental data for $\epsilon = 1.5\%$. In the calculation, flow coefficients c_d and c_s in Eqs.(1) and (2) are both assumed to be 0.7 which is decided from flow measuring experiments executed specially. In the same manner as experimental results, theoretical results hardly decrease with increase of ϵ except in the range of very small ϵ . This tendency is very different from steep decrease of theoretical volumetric efficiency $\eta_v = \eta_v - \epsilon \{p_d/p_s\}^{1/K} - 1$ for reciprocating compressor⁽⁷⁾ as shown in Fig.7.

The reason can be explained with Fig.8 which shows changes of restriction

Table 1 Main dimensions of the compressor

Cylinder radius	R	mm	27.0
Rolling piston radius	r	mm	23.4
Cylinder height	l	mm	23.8
Swept volume	V_0	cm^3	13.5
Clearance volume	V_v	cm^3	0.20
Suction port diameter	d_s	mm	8.0
Discharge port diameter	d_d	mm	7.0
Front angle (suc. opening)	θ_{s1}	rad	0.31
Rear angle (suc. opening)	θ_{s2}	rad	0.61
Front angle (dis. opening)	θ_{d1}	rad	-0.38
Rear angle (dis. opening)	θ_{d2}	rad	-0.12

Fig.7 Influence of clearance volume on volumetric efficiency (1)

Fig.8 Changes of restriction areas, pressures and reverse mass flow rate

areas A_d and A_s , pressures p_v and p_b and reverse mass flow rate $\dot{m}_s$ accompanying rolling piston revolution. At first, immediately after rolling piston passes by the front angle θ_{d1} of discharge opening, discharge restriction area A_d is so small though pressure difference $(p_v - p_b)$ across discharge opening is so large that gas in the clearance volume can hardly flow into suction chamber and reverse mass flow rate $\dot{m}_s$ is just small. As the area A_d increases with the shaft revolution, gas flow into suction chamber increases and reverse mass flow rate $\dot{m}_s$ increases while pressure p_v decreases. As the shaft rotates more and approaches front angle θ_{s1} of suction opening, second restriction area A_s at suction opening becomes rather small and reverse mass flow rate $\dot{m}_s$ decreases while chamber pressure p_b increases. This means that reverse flow is limited by two restrictions of suction and discharge openings. Therefore, even if clearance volume ratio ϵ becomes larger, reverse mass flow rate $\dot{m}_s$ is not so much affected though pressure p_v delays in dropping and pressure p_b is a little high at the end of reverse flow ($\theta = \theta_{s2}$). And total reverse mass flow Δm_s evaluated by the enclosed area with the curve $\dot{m}_s$ and horizontal axis is almost unchanged.

In order to confirm the above, we measured pressure change in clearance volume using a piezo-electric pressure transducer and obtained pressure-time relation as shown in Fig.9. Pressure drop during re-expansion is rather slow and in this case it reaches minimum a little after suction restriction closing (θ_{s2}). This tendency is similar to that of calculated curve of p_v shown in Fig.8.

Now, let's examine the influence of compressor operating conditions on volumetric efficiency theoretically. Figure 10 shows the influence of compressor rotational frequency n . Its longitudinal axis is expressed by volumetric efficiency drop $\Delta\eta_v$ calculated by Eq.(11). $\Delta\eta_v$ increases with decrease of n . This is because the period between discharge restriction opening and suction restriction closing

Fig.9 Pressure-time relation in clearance volume

becomes longer and reverse flow increases. That tendency is more remarkable for larger clearance ratio ϵ . But, when compressor is operated by a two phase induction motor with commercial power source of 50 or 60 Hz, efficiency drop $\Delta\eta_v$ is fairly small, being about 1%. As n becomes closer to 0, efficiency drop $\Delta\eta_v$ approaches the value $\Delta\eta_v = \epsilon \{(p_d/p_s)^{1/k} - 1\}$ for reciprocating compressor⁽⁷⁾.

Figure 11 shows change of volumetric efficiency drop $\Delta\eta_v$ when operating discharge pressure p_d is changed while keeping suction pressure p_s constant. As pressure ratio p_d/p_s increases, $\Delta\eta_v$ increases. But the changing rate is very small in the range of common pressure ratios 3~4 for the air conditioner. And its absolute value is fairly low as compared with that for reciprocating compressor.

Figure 12 shows the results of experiments in which clearance volume is changed as shown in Fig.6(b). Horizontal axis of the figure indicates the angular distance β ($=\theta_{s2}-\theta_{d1}$). At each operating condition, η_v gradually decreases

Fig.10 Relationship between volumetric efficiency drop and rotational frequency

Fig.11 Relationship between volumetric efficiency drop and pressure ratio

with increase of β . This tendency agrees well with that of theoretical results which are illustrated in Fig.12 referring to the experimental data for $\beta = 0.99$ rad. Concerning the changing method of clearance volume, the reason for the decrease of η_v may be considered to be (1) increase of absolute value of clearance volume V_v (2) decrease of front angle θ_{d1} . In these experiments, the range of clearance volume ratios ϵ is 1.5~2.5 % and it is considered that the influence of absolute clearance volume change on η_v is very small referring to the results of Fig.7. The second factor, namely change of θ_{d1} , means change of discharge opening area. Figure 13 shows theoretical volumetric efficiency drop $\Delta\eta_v$ when discharge port diameter d_d ($=R(\theta_{s2}-\theta_{d1})$) is changed. The figure includes two cases of changing d_d : (1) rear angle θ_{d2} is changed while keeping front angle θ_{d1} constant and (2) front angle θ_{d1}

Fig.12 Influence of clearance volume on volumetric efficiency (2)

Fig.13 Relationship between volumetric efficiency drop and discharge port diameter

is changed while keeping rear angle θ_{d2} constant. $\Delta\eta_v$ hardly changes in the case of (1) but it increases with increase of d_d in the case of (2). This indicates that location of discharge opening, namely front angle θ_{d1} , is more influential on volumetric efficiency than port diameter d_d , namely distance between front and rear edges of discharge opening. The same thing can be said about the suction port and rear angle θ_{s2} of suction opening becomes an important factor. From the above, it becomes clear that the tendency of η_v decreasing with increasing β in Fig.12 is caused by the change of front angle θ_{d1} .

5.2 Compressor power

Figure 14 shows changes of compressor electric power input P when clearance volume is changed as shown in Fig.6(a). Power input P is non-dimensionalized by the power P_0 at $\epsilon=0$ which is estimated by the method of least squares of experimental results. As clearance ratio ϵ increases, P/P_0 increases almost linearly and

Fig.14 Influence of clearance volume on compressor electric power input

Fig.15 Pressure-time relations in suction and compression chambers

its increasing rate becomes large as discharge pressure p_d increases. This increasing tendency is different from decreasing tendency for reciprocating compressor power. Also ratio of theoretical power estimated by the work of Eq.(14) is illustrated in Fig.14. This is in good agreement with experimental results and it is confirmed that influence of clearance volume on the compressor power can be evaluated using p - V diagram shown in Fig.5 and Eq.(14).

By the way, the increase of power as mentioned above is considered to be caused by the increase of pressure rise during compression process as shown in p - V diagram. Figure 15 shows experimental pressure-time relations in suction and compression chambers of the operating compressor. Those pressures were measured with piezo-electric transducers mounted at $\theta = 1.13$ and 5.78 rad on the cylinder wall. For larger ϵ , pressure is higher during compression process. This means it is more reasonable to consider that most of the gas in clearance volume doesn't flow reversely through suction port but increases compression chamber pressure and compressor work.

In addition, when clearance volume is changed as shown in Fig.6(b), measured compressor power decreases slightly with increase of clearance volume. This may be because of decrease of effective compressor mass flow rate.

6. Conclusions

In this paper the influence of clearance volume on the performance of the rolling piston type rotary compressor was analysed theoretically and experimentally. The results are summarized as follows.

(1) Influence of clearance volume on volumetric efficiency is fairly weak as compared with that for reciprocating compressor.

(2) Reverse flow from clearance volume to suction port is restricted by the suction and discharge openings between them.

(3) Influence of clearance volume on volumetric efficiency is much affected by the locations of front angle of discharge opening and rear angle of suction opening.

(4) As clearance volume increases, compressor work increases because pressure in compression chamber rises earlier.

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