The Performance Analysis of R32 Rotary Compressor Used for Room Air Conditioners

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Abstract. R32 has been used as an alternative refrigerant of R410A in room air conditioners for its zero ODP, low GWP, and high performance. In this study, performance of R32 hermetic rotary compressor with high pressure shell was analyzed with a more comprehensive theoretical model. The simulation results showed good agreement with the experimental results. Besides, in order to improve the accuracy, in the theoretical model, the two phase leakage model was chosen in the leakage passage of radial clearance, and the gas leakage was considered in the side of sliding vane when the height of oil level is lower than the upper end face of cylinder. The analysis results show that the main factors that influence the cooling capacity are heating loss and leakage loss, and the main factors that influence the input power are discharge pressure loss, suction heating loss, clearance volume loss, bearing friction loss and sliding vane friction loss. The theoretical model would have great useful for designation and performance optimization of rotary compressor.

1. Introduction
In recent years, in order to respond to the environmental and energy safe policies, the speed has improved to the substitution of refrigerant and the research of high efficiency rotary compressor in room air conditioners. R32 has gradually entered the market of room air conditioners for its zero ODP, low GWP, and advantageous thermo-physical properties. While, the leakage loss and heating loss in R32 rotary compressor would be much obvious because of its big pressure ratio and high discharge temperature. And thus bring in lower compressor performance and reliability. So, it’s significant to study the performance of rotary compressor and thus to improve the efficiency. The main study aspects are size optimization, oil option et al. Takashi Matsuzaka [1] found the rotary compressor efficiency can reach 74% based on the highest designation level and manufacturing level. H. Wakabayashi et.al [2] found that the motor loss account about 50% of the total loss, and heating loss 20%, and decreasing heat transfer loss and leakage loss were very important for improving rotary compressor performance. Keiju Sakaino [3] designed a high efficiency rotary compressor by the comparison of round valve and flat valve. Y. Youn [4] optimized the discharge system of rotary compressor by P-V diagram, the valve and cutout angle were mainly optimized. Besides, many researchers have studied the oil used in the R32 rotary compressor [5], [6-11]. In the past, most researchers studied the performance of the hermetic rotary compressor in a certain aspect, such as the discharge valve study, the leakage study, the heat transfer study, P-V diagram analysis, or the oil selection research et.al [12], [1-4, 13-15], and the R32 rotary compressor is the same situation.

In this paper, the performance of R32 rotary compressor (large amount of them have been used in the room air conditioners) will be analyzed with a more comprehensive theoretical model, and the
simulation accuracy will be verified by the experiment conducted in the performance test platform of R32 rotary compressor. The performance under different condensing temperature and loss distribution (cooling capacity loss, leakage loss, and input power loss et.al) will be analyzed. The theoretical model would have great useful for designation and performance optimization of rotary compressor.

2. Model
The theoretical simulation model is composed of dynamic part and thermodynamic part. It can output cooling capacity, input power, COP, discharge temperature, efficiencies, losses distribution, oil film thickness, valve working curve et.al. In this paper, the performance will be analyzed from cooling capacity, input power, COP, discharge temperature, efficiencies, and losses distribution aspects.

2.1. Conservation Equation
Energy:

\[
\frac{dT}{d\theta} = \frac{1}{c_i m_i + c_l m_l} \left( \frac{1}{\omega_i} \left( \sum_i \dot{Q}_i + \sum_i h_i \dot{m}_i + c_i \sum_i T_i \dot{m}_i \right) - \left[ h - v \left( \frac{\partial h}{\partial v} \right)_T \right] + \frac{\partial P}{\partial v} \right) \frac{dm}{d\theta} - c_i T \frac{dm_i}{d\theta} \left( \frac{\partial h}{\partial v} \right)_T - v \frac{\partial P}{\partial v} \frac{dV}{d\theta} \right]
\]

Oil mass:

\[
\frac{dm}{d\theta} = \sum_i \dot{m}_i \tag{2}
\]

Refrigerant mass:

\[
\frac{dm_i}{d\theta} = \sum \dot{m}_i \tag{3}
\]

Where, \(m_i, m_l, Q_i\) respectively responds to the gas mass, the oil mass, the heat flux in the controlled volume, \(c_i\) is gas specific volume, \(c_l\) is liquid specific volume, \(\omega_i\) is angle speed, \(h\) is enthalpy, \(T_d\) is oil temperature.

2.2. The leakage mass flow rate model

1) The leakage rate from radial clearance of rolling piston and cylinder:
When the oil level is lower than the up end face of cylinder, there will be gas leaking from the radial clearance, and the two phases flow is chosen in this model:

\[
\dot{m}_1 = C_1 \frac{H \delta l \Delta P}{12 v_1 L_e} + C_2 \frac{H \delta l \Delta P}{12 v_2 L_e} \tag{4}
\]

Where, \(\Delta P\) - the pressure difference \((P_r - P_e)\), \(H\) - the height of cylinder, \(v_1\) - specific volume of liquid oil, \(v_2\) - specific volume of refrigerant gas, \(L_e\) - the equivalent length of leakage path; \(\delta\) - the clearance between rolling piston and cylinder; \(C_1\) - the height coefficient of liquid leakage; \(C_2\) - the height coefficient of gas leakage. Commonly, \(C_1, C_2\) are determined by the height of oil level, in this study, \(C_1=0.8, C_2=0.2\).

2) The leakage rate from end clearance of rolling piston:
The leakage from this leakage path can be consumed as the viscous laminar flow between two flats, and the leakage mass flow \((\dot{m}_2)\) is:

\[
\dot{m}_2 = \frac{1}{2} \left( \frac{\delta^2 \Delta P}{6 v L} + U \rho \right) H \delta \tag{5}
\]
Where $\delta$ is the leakage clearance; $\Delta P$ is pressure difference ($P_c - P_e$), $U$ is internal energy, $\rho$ is density, $H_c$ is the height of cylinder, $v$ is specific volume, $L$ is the length of leakage path.

3) The leakage rate from end clearance of sliding vane:
Considering that the thickness/the end clearance of sliding is very large (about 200), and the flow velocity is very large, so the Fanno flow model is selected to calculate the leakage from it.

4) The leakage rate from side clearance of sliding vane:
Similarly, the leakage from this path also can be consumed as the viscous laminar flow between two flats, and the leakage mass flow ($\dot{m}_3$) is:

$$\dot{m}_3 = \frac{1}{2} C_H \delta \left( \frac{\delta^2 \Delta P}{6v_1 L} \right) + \frac{1}{2} C_H \delta \left( \frac{\delta^2 \Delta P}{6v_2 L} \right) + U \rho H \delta$$  \hspace{1cm} (6)

Where $\delta$ is the leakage clearance, $U$ is internal energy, $\rho$ is density, $L$ is the length of leakage path; $H$, $\Delta P$, $v_1$, $v_2$, $C_1$, $C_2$ are same with the Eq.(5).

2.3. The heat transfer model

2.3.1. The heat transfer between suction tube and suction gas.

1) The inner wall temperature of suction tube:
$$T_{sp1} = \frac{2T_s + T_a}{3}, \; \; \; T_{sp2} = \frac{2T_s + T_{sp1} + T_{sp3}}{4}, \; \; \; T_{sp3} = T_{cw}(\theta_{am}, T_a)$$  \hspace{1cm} (7)

Where, the suction tube is divided into three sections, $T_{sp1}$ is the gas temperature in the accumulator, $T_{sp2}$ is the gas temperature between the accumulator and compressor shell, $T_{sp3}$ is the gas temperature in the inner of compressor shell, $T_{cw}$ is the temperature of cylinder inner wall, $T_a$ is the ambient temperature.

2) The convection heat transfer coefficient of suction tube inner wall ($\alpha_{sp}$):
$$\alpha_{sp} = 0.023 \frac{\lambda}{D} Re^{0.8} Pr^{0.4}$$  \hspace{1cm} (8)

Where, $Re = U D / \nu$, $Pr = C_p \mu / \lambda$, $D$ is inner diameter of suction tube, $X$, $\nu$, $\mu$, $C_p$, $\lambda$ are properties of fluid in the suction tube.

2.3.2. The heat transfer between cylinder wall and refrigerant.

1) The temperature of cylinder inner wall ($T_{cw}$)
The temperature of cylinder inner wall ($T_{cw}$) is influenced by oil sump temperature and suction gas temperature, When $0 \leq \theta < \theta_{am}$

$$T_{cw} = T_a - 12 - 8 \frac{\theta}{\theta_{am}}$$  \hspace{1cm} (9)

When $\theta_{am} \leq \theta < \theta_{dm1}$

$$T_{cw} = 0.2504(T_s - 273.15) + 0.7371(T_a - 273.15) + 13.6 \frac{\theta}{\theta_{dm1} - \theta_{am}} - 4$$  \hspace{1cm} (10)

When $\theta_{dm1} \leq \theta < 2\pi$

$$T_{cw} = T_a - 20$$  \hspace{1cm} (11)
Where, \( T_o \) — the oil sump temperature \( /K \), \( \theta \) — circumferential angle of cylinder/ rad, \( \theta_{sm} \) — place angle of suction port, \( \theta_{dm2} \) — place angle of discharge port.

2) The convection heat transfer coefficient of cylinder inner wall:
The formula summarized by Liu\cite{22} is chosen to calculate the heat transfer coefficient of cylinder wall.

2.4. The friction loss model
The friction loss model in different place of the compressor is shown in Tab.1.

| Friction place                      | Friction power model                           |
|-------------------------------------|------------------------------------------------|
| Main bearing and sub bearing        | \( L_{s1} = m_{s1b} \omega_s \)               |
| Eccentric bearing                   | \( L_{s2} = m_s (\omega_e - \omega_s) \)       |
| Side of sliding vane                | \( L_{s3} = u_v [V_e | |R_{s1}| + |R_{s2}|] \) |
| The top of sliding vane             | \( L_{s4} = [F_e V_e] \)                        |
| Viscous shear loss in the outface of sliding vane | \( L_{s5} = [F_e V_e] \) |
| End face of rolling piston          | \( L_{s6} = F_{e} e \omega_s + m_e \omega_e \) |
| Between rolling piston and cylinder | \( L_{s7} = [F_{i} (e \omega_e + R_s \omega_r)] \) |
| Thrust bearing                      | \( L_{s8} = m_{b} \omega_e \)                  |
| Counterweight                       | \( L_{s9} = C_d \frac{P}{2} \omega_e \) \( R_{s1} A_{b1} + R_{s2} A_{b2} \) |

Note: \( \omega_s, \omega_e \) - rotation speed of eccentric shaft and rolling piston respectively; \( m_{s1b} \) - friction torque of main and sub bearing to the eccentric shaft; \( m_s \) - friction torque of main and sub bearing to the eccentric shaft; \( u_v \) - friction coefficient in the side of sliding vane; \( V_e \) - velocity of sliding vane; \( R_{s1}, R_{s2} \) - reaction force of sliding vane at outer and inner of slot respectively; \( F_{e} \) - tangential force between sliding vane and rolling piston; \( V_s \) - relative sliding velocity between sliding vane and rolling piston; \( F_{i} \) - viscous shear force; \( F_e, m_e \) - friction force and friction moment respectively in the end of rolling piston and cylinder; \( e \) - eccentric distance; \( F_{i} \) - tangential friction force of rolling piston; \( R_{o} \) - external radius of rolling piston; \( m_{b} \) - friction torque of thrust bearing to the eccentric bearing; \( C_d \) - balance coefficient; \( \rho \) - density of counterweight; \( R_{s1}, R_{s2} \) - radius of main and sub counterweight respectively; \( A_{b1}, A_{b2} \) - section areas of main and sub counterweight respectively.

2.5. Properties of oil and refrigerant mixture
The properties of oil and refrigerant mixture in the simulation model were obtained by the fitted formulas which were fitted according to the experiment curves.

1) Solubility (S)
i) \( T \geq 70^\circ C \):

\[
S(P,T) = x \cdot S(T) + \frac{x \cdot (1-x)}{F(x,y)} \quad (12)
\]

\[
x = \frac{P}{4.5}, y = \frac{T}{T_{cr}}
\]

\[
F(x,y) = (a_1 + a_2 \cdot y + a_3 \cdot y^2 + a_4 \cdot y^3 + a_5 \cdot y^4) \cdot (a_6 + a_7 \cdot x + a_8 \cdot x^2 + a_9 \cdot x^3 + a_{10} \cdot x^4)
\]

\[
S(T) = 1305.64972 - 14.06377T + 0.0568596T^2 - 0.000102225T^3 + 0.000000689408T^4 \quad (14)
\]
ii) \( t<70^\circ C \) and \( P/P_{sat}\leq0.95 \)

\[
S(P,T) = (1-x) + \frac{x\cdot(1-x)}{F(x,y)}
\]

\[
x = 1 - \frac{P}{P_{sat}}, y = \frac{T}{T_C}
\]

\[
F(x,y) = b_1 + b_2 \cdot x + b_3 \cdot x^2 + b_4 \cdot y + b_5 \cdot y^2 + b_6 \cdot xy + b_7 \cdot x^2 y + b_8 \cdot x^2 y^2
\]

\[
P_{sat} = 1.7900 \times 10^3 - 2.8175 \times 10^{-1} T + 1.7447 \times 10^{-2} T^2 - 5.1493 \times 10^{-5} T^3 + 6.2225 \times 10^{-8} T^4
\]  

(15)

(16)

(17)

iii) \( t<70^\circ C \) and \( P/P_{sat}>0.95 \)

\[
S(P,T) = S_{sys} + \left(\frac{20P-19P_{sat}}{P_{sat}}\right) \cdot (1 - S_{sys})
\]

\[
S_{sys} = S(0.95P_{sat}, T)
\]

Where, \( t—temperature/^\circ C, T—temperature/K, P—pressure/Pa. \( P_{sat}—saturation temperature of pure \) \( R32, a_i~a_9, b_i~b_9 \) are constant parameters of Eq.(13), Eq.(16) respectively.

2) Viscosity

\[
\ln(v_{max}) = c_1 + c_2 \cdot S + c_3 \cdot S^2 + c_4 \cdot t + c_5 \cdot t^2 + c_6 \cdot S \cdot t + c_7 \cdot S^2 \cdot t + c_8 \cdot S \cdot t^2 + c_9 \cdot S^2 t^2
\]

(19)

Where, \( v_{max} \) is kinematic viscosity/(mm²/s), \( t \) is temperature/°C, \( S \) is solubility, \( c_1~c_9 \) are constant parameters.

3) Density

\[
\rho_{mix} = \frac{1}{2} \left[ S \cdot \rho_{ref} + (1-S) \rho_{oil} + \frac{1}{S \cdot \rho_{ref} + \frac{1}{(1-S) \rho_{oil}}} \right]
\]

(20)

Where, \( \rho_{mix}, \rho_{ref}, \rho_{oil} \) are density of oil and refrigerant mixture/(kg/m³), pure refrigerant and pure oil respectively.

3. Results Analysis

3.1. Experimental verification

In order to verify the accuracy of the simulated results, the experiment was conducted on an R32 compressor performance test platform. The basic parameters of hermetic rotary compressor with high pressure shell studied in this experiment are shown in Tab.2. The experiment was conducted under 5 conditions (shown in Tab.3), of which the GX condition is the high efficiency condition designed to match the air conditioner with COP of 3.4–3.6 [16].

**Tab.2 Basic parameters of R32 hermetic rotary compressor**

| Structure type | Refrigerant | Oil | Cylinder diameter \((D_{cy})/mm\) | Cylinder height \((H)/mm\) | Displacement volume \((V_h)/cm^3\) | Eccentric distance \((e)/mm\) | Motor efficiency /% |
|---------------|-------------|-----|-------------------------------|----------------|-----------------------------|----------------|------------------|
| Single cylinder | R32 | POE-VG74 | 50.0 | 19.00 | 13.35 | 4.96 | 82 |
The cooling capacity ($q_0$), input power ($P$), COP, discharge temperature ($t_d$), mass flow rate ($m$), were mainly tested shown in Tab.4. The comparisons show that most deviations of the simulated results to the experimental results are less than 3%. In addition, $t_d$ has been higher than the motor allowable temperature (130ºC) under ASHARE/T1, which is adverse to the compressor reliability. While, the GX working condition can not only obtain a lower $t_d$ but a higher COP. So the GX condition is reasonable to set as a normal condition in R32 rotary compressor.

### Tab.4 Test results

| Items          | Condition | Cond1   | Cond2   | Cond3   | Cond4   | Cond5   |
|----------------|-----------|---------|---------|---------|---------|---------|
| $q_0$/W        | Exp       | 4441.4  | 3806.7  | 4143.9  | 3415.6  | 3421.9  |
|                | Sim       | 4441.5  | 3742.8  | 4085.3  | 3367.9  | 3368.8  |
| Ab error/%     | Exp       | 0       | 1.68    | 1.41    | 1.4     | 1.6     |
|                |           | 833.6   | 1120.8  | 966.1   | 1173.1  | 1160.2  |
| $P$/W          | Sim       | 819.4   | 1104.9  | 957.3   | 1171.2  | 1154.3  |
| Ab error/%     | Exp       | 1.7     | 1.42    | 0.91    | 0.16    | 0.51    |
|                |           | 532.8   | 339.7   | 428.9   | 291.2   | 294.9   |
| COP/W          | Sim       | 542     | 339     | 427     | 288     | 289.5   |
| Ab error/%     | Exp       | 1.73    | 0.21    | 0.44    | 1.1     | 2.91    |
|                |           | 61      | 58.5    | 60      | 51.3    | 46.6    |
| $m$/kg/h       | Sim       | 61      | 56.9    | 58.9    | 50      | 45.6    |
| Ab error/%     | Exp       | 0       | 2.74    | 1.83    | 2.46    | 2.12    |
|                |           | 78.0    | 102.7   | 89.3    | 115.6   | 134.4   |
| $t_d$          | Sim       | 78.6    | 103.1   | 91.0    | 114.1   | 133.5   |
| Ab error/%     | Exp       | 0.77    | 0.39    | 1.90    | 1.30    | 0.67    |

### 3.2. The compressor performance under different condensing temperature

In order to have a comparison, the performances of the same rotary compressor with R410A &POE-VG74 oil were also simulated. The comparisons (in Fig.1) indicated that under any condition, the cooling capacity ($q_0$), COP, indicated efficiency ($\eta_i$), adiabatic efficiency ($\eta_{ad}$), input power ($P$) of R32 rotary compressor is higher than R410A rotary compressor, and the mass flow rate ($m$), mechanical efficiency ($\eta_{me}$), volumetric efficiency ($\eta_v$) are lower. In the whole, the performance of R32 rotary compressor is superior to R410A rotary compressor. Besides, the R32 mass flow rate in the air conditioner system is lower than R410A, which means that the charge amount in the R32 air conditioner is much lower than in R410A air conditioner.
As ambient temperature \( (t_a) \) is not constant in the practical ambient, the air conditioner would work under part cooling load condition or over cooling load condition. The performance under different condensing temperature \( (t_c) \) i.e. different cooling load is shown in Fig.1 (Cond1, Cond2, Cond3), which shows that the compressor efficiency is higher under part cooling load condition i.e. \( t_c \) is 40°C than under GX condition, and lower under over cooling load condition i.e. \( t_c \) is 50°C.

![Fig.1](image)

**Fig.1** The performance parameters under different condensing temperature
3.3. The loss distribution of rotary compressor

The performance losses mainly include cooling capacity loss and power loss, as the losses are influenced by many factors, it’s impossible to calculate these losses through experiment. The loss distribution was analyzed by the simulation model under GX condition, and then the main factors that influence the compressor performance would be found out, which is a prerequisites for the optimization work.

3.3.1. Analysis of cooling capacity loss distribution

The cooling capacity losses mainly include leakage loss, heating loss, and clearance volume loss. As shown in Fig.2, among these losses the leakage loss occupies about 50%, cylinder heating loss occupies about 40%, and these two losses occupy the most. Comparatively, the clearance volume loss and suction tube heating loss occupy less. This is because R32 compressor has higher pressure difference and higher temperature difference between discharge side and suction side, and there are more refrigerants with higher temperature leaking off from discharge side to suction side, which would heat the refrigerant in the suction chamber and then the less quality of refrigerant would be compressed.

As leakage loss is a main factor that influences the cooling capacity, it’s essential to know its influence mechanism. The main leakage paths are radial clearance; end face of rolling piston, side face of sliding vane, end face of sliding vane (shown in Fig.3). Among these leakage paths, radial clearance leaks the most, and then is the end face of sliding vane and end face of rolling piston, and the side face of sliding vane leaks the least. It’s efficient to decrease leakage to decrease the areas of radial clearance, the end face of sliding vane and end face of rolling piston.

As shown in Fig.2 and Fig.3, apart from leakage loss from side face of sliding vane, all the cooling capacity losses of rotary compressor with R32 used are higher than with R410A used.

![Fig.2 Cooling capacity loss distribution](image1)

![Fig.3 Leakage loss distribution](image2)

3.3.2. Analysis of input power loss distribution

The factors that influence the input power are mainly motor loss, mechanical power loss, and indicated power loss, shown in Fig.4. Among these input power losses, the motor loss occupies about 50%, indicated power loss occupies about 30%, and mechanical power loss occupies about 20%.

The indicated power loss is presented in Fig.5. Still, leakage loss occupies the most about 50%, and then is heating loss about 10%, and discharge pressure loss and clearance volume loss about 20% respectively.

The mechanical power loss is presented in Fig.6. The eccentric bearing loss occupies the most, about 40% of mechanical power loss, and then is top sliding vane, side sliding vane, and main bearing, all about 15% of mechanical power loss respectively. The relatively lower is thrust bearing loss, sub bearing loss, and balancer, which are all less than 10% of mechanical power loss.
As shown in Fig.4, among the indicated power losses of R32 compressor, apart from heating power loss, the pressure loss and leakage power loss are lower than of R410A compressor. As shown in Fig.5, among the mechanical power losses of R32 compressor, the main bearing and balance friction losses are lower than of R410A compressor, other friction losses all higher than of R410A compressor. In the whole, the motor loss and mechanical loss are higher and indicated power loss is lower in rotary compressor with R32 used than with R410A used.

4. Conclusions
The performance of R32 rotary compressor was simulated using the dynamic and thermodynamic model of rotary compressor. In order to improve the accuracy of the theoretical model, the two phase leakage model was chosen in the leakage passage of radial clearance, and the gas leakage was considered in the side of sliding vane when the height of oil level is lower than the upper end face of cylinder. The accuracy was verified by the experiment, and the deviation of the simulation model to the experiment is less than 2%. The cooling capacity loss and input power loss were also calculated by the simulation model. The main conclusions in this paper are as follows:

1) Under the studied conditions, the cooling capacity ($q_0$), COP, indicated efficiency ($\eta_i$), adiabatic efficiency ($\eta_{ad}$), input power ($P$) of R32 rotary compressor is higher than R410A rotary compressor, and the mass flow rate ($m$), mechanical efficiency ($\eta_{me}$), volumetric efficiency ($\eta_v$) are lower. In the whole, the performance of R32 rotary compressor is superior to R410A rotary compressor.

2) Among the cooling capacity loss, the leakage loss occupies about 50% (in which the leakage from radial clearance, the end face of sliding vane and end face of rolling piston occupy the most); the cylinder heating loss occupies about 40%, and these two losses occupy the most. This is because R32 compressor has higher pressure difference and higher temperature difference.
3) Among the input power losses, the motor loss occupies about 50% of input power loss, indicated power loss occupies about 30% of input power loss, mechanical power loss occupies about 20% of input power loss.

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