Efficiency analysis of the novel twin chamber rotary compressor

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Abstract. The efficiency of the novel twin chamber rotary compressor was analyzed by comparing with conventional one. The increasing of the compression volume is the main advantage of the novel twin chamber compressor comparing with conventional one. To make this additional volume, newly geometric concept like vane bush was applied in the novel twin chamber rotary compressor. The losses occurred in these newly parts are different compared with the conventional one. So, the efficiency analysis of the novel rotary compressor was conducted about compressor loss aspect. To analyze these efficiencies, the simulation program was developed using numerical analysis. This program based on the conventional rotary compressor one. That means all the theoretical functions and concept of the conventional rotary compressor program was applied to novel compressor simulation program. To verify the numerical simulation method of the novel compressor program, experimental and simulation results were compared. Error between the simulation and experimental results were expressed in 4.2 to 8.6%. So, the simulation program showed a good agreement with experiments. Cooling capacity of the novel compressor is higher 34.68% compared conventional compressor. The input power of the novel rotary compressor is increased about 24.94% in ARI and 22.06% in CHEER conditions compared conventional one. Energy efficiency was evaluated with EER and the novel rotary compressor is improved about 8.06% in ARI and 10.76% in CHEER condition compared with conventional one.

1. Introduction

Recently considerations of the compressor development are applying the alternative refrigerant and improve the efficiency of the compressor. Since the many international environment regulation was published, interest of the alternative refrigerants is growing as time passes. According to the regulation, conventional refrigerants should be replaced by alternative refrigerants gradually. Figure 1 shows the change of refrigerants involved in HVAC system announced by ministry of economy in Japan. The most considered alternative refrigerants are HFC-134a, HFC-407c, and HFC-410a [1-5]. The properties of these refrigerants are different compared with conventional refrigerants [6]. And these property values are changed as compression process progresses. So, to apply these alternative refrigerants in refrigerator and air-conditioning systems, the compressor must be modified properly. In addition, to obtain the high efficiency systems, optimum design of the compressor is essential because the compressor consumes the power over 80% in the refrigerator and the air-conditioning system [7]. For these reasons, many studies about the compressor optimization were conducted. Mathison et al study the novel spool compressor with multiple vapor refrigerant injection ports [8]. They used the simulation analysis based on numerical analysis and uncertainty of mass flow rate and input power was 10% and 11%, respectively. The feature of novel spool compressor is addition of vapor injection mechanism to conventional spool compressor. This mechanism evaluated that position of injection port was not affected the compressor performance, but number of injection port was affected positively COP. Yap et al was analyzed the novel cross vane expander compressor used mathematical
modelling and experiments [9]. This novel compressor used rotating vane mechanism and pressure uncertainty between numerical prediction and experiments is about 10.5%. In general vane compressor, the mechanical loss is occurred in vane part. So, if reduce the loss of the vane, like rotating vane mechanism, the compressor efficiency can improve. The cross-vane expander compressor was evaluated that it can improved COP 36.6% compare the general vane compressor. Teh et al conducted the theoretical study of a novel revolving vane compressor [10]. They proposed the rotating cylinder mechanism that can used in the vane compressor. This mechanism has some advantage like lower friction loss, enhance the smoothness of the compressor operation, and so on. So, the mechanical efficiency of this compressor was evaluated 95%. Like these, there are many studies on the optimization of the cylinder shape or the cylinder analyze method. In conventional rotary compressor, loss occurred at the vane takes most of the mechanical losses because the vane side area contacts with cylinder as rotor rotating. In this study, novel rotary compressor was suggested that used bush and twin chamber mechanisms instead of conventional vane part and one chamber.

![Figure 1](Image)

**Figure 1.** Changing the refrigerants involved in HVAC system in japan

2. Numerical Analysis
In generally, ratio between the rotor and the cylinder diameter of the conventional rotary compressor is above 0.75 [11]. So, the working volume is small in comparison with the size of the compressor. Proposed novel twin chamber rotary compressor is overcome this problem through secure the additional working volume with the rotor inside. Figure 2 shows the cross-sectional diagram of the side cylinder view with the conventional rotary compressor and the novel twin chamber rotary compressor.

![Figure 2](Image)

**Figure 2.** Cross-sectional diagram of the side cylinder view

Difference between this cross-sectional view is the power transmission part acting on the rotor. In conventional one, figure 2 (a), the shaft power acting on whole of the inner rotor area. In figure 2(b),
the shaft power of the novel twin chamber rotary compressor acting on the inner rotor area partially. Figure 3 shows the cross-sectional diagram of the upper cylinder view. The vane, figure 3 (a), have upper and down motion as compressor is working. In this process, the large mechanical loss occurs in the vane tip and side. To solve this problem, the novel twin chamber rotary compressor adopts the bush mechanism and fixed vane geometry that assembled with the cylinder and the axial shaft as figure 3 (b). These two compressors have different geometry and loss characteristics. So, the numerical approach method was used for accurate losses analysis and compare with two compressors. Detail of the numerical analysis explanation with the conventional rotary compressor refer to advanced research [12].

Figure 3. Cross-sectional diagram of the upper cylinder view

2.1. Suction and compression volume
To calculate suction and compression volume, the roller must be defined with shaft rotating angle because these volumes are changing as the compressor is rotating. The position angle of the roller is expressed equation (1) ~ (3).

\[
\angle O_1AO_2 = \alpha_{oc} \sin^{-1}\left(\frac{e \sin \theta}{r_{oci}}\right) \\
\angle O_2BO_2 = \alpha \sin^{-1}\left(\frac{e \sin \theta}{r_{ico} + r_{bush}}\right) \\
\angle O_1CO_2 = \alpha_{ic} \sin^{-1}\left(\frac{e \sin \theta}{r_{ico}}\right)
\]

\(O_1\) is central point of the inner cylinder, \(O_2\) is central point of the outer cylinder, A is contact point between outer surface of the roller and the vane, B is contact point between center line of the roller and the vane, C is contact point between the inner surface of the roller and the vane, \(e\) is eccentric length, \(\theta\) is rotating angle, \(r_{oci}\) is inner diameter of outside cylinder, \(r_{ico}\) is outer diameter of inner cylinder, and \(r_{bush}\) is radius of the bush. Suction and compression volume can calculate using these relations of the position angle and trigonometrical function. Volume of inner part is calculated by Equation (4) ~ (7).

\[
V_{isc}(\theta) = \frac{1}{2} H_{ic} [r_{ico}^2 \theta - \alpha_{ic} - r_{icl}^2 \theta - e \sin \theta (r_{ico} \cos \alpha_{ic} - e \cos \theta) - B_{vane} y_{ic}]
\]

\[
y_{ic} = r_{ico} \cos \alpha_{ic} - e \cos \theta - r_{icl}
\]

\[
V_{icc}(\theta) = V_{icm} - V_{isc}(\theta)
\]
\[ \begin{align*}
V_{icm} &= \frac{\pi}{4} (r_{ico}^2 - r_{ici}^2) \\
V_{isc}(\theta) &= \frac{1}{2} H_{oc} [(e + r_{oc})^2 \theta - r_{oc}^2 (\theta + \alpha_{oc}) - \varepsilon \sin(\theta)(r_{oc} \cos \alpha_{oc} + e \cos \theta) - B_{vane} y_{oc}] \\
y_{oc} &= r_{oco} - (r_{oco} \cos \alpha_{oc} + e \cos \theta) \\
V_{oc}(\theta) &= V_{ocm} - V_{osc}(\theta) \\
V_{ocm} &= \frac{\pi}{4} (r_{oco}^2 - r_{ocl}^2) 
\end{align*} \] (8) 

\[ \begin{align*}
\sum F_x &= F_1 + F_2 = 0 \\
\sum F_y &= f(F_{t1} + F_{t2}) = -m_{bush} \ddot{y} \\
\sum M &= r_{bush}(F_1 + F_2) = -M_{bush}
\end{align*} \] (12) 

2.2. Dynamics of vane bush

The vane bush helps the smoothly movement when the roller rotates. This mechanical part is mainly different compared conventional rotary compressor. Figure 4 shows the schematic diagram of the bush and Equation (12) ~ (14) shows the force and moment equation.
$F_x$ is resultant force acting on x-axial with $F_1$ and $F_2$. $F_y$ is resultant force acting on y-axial with $F_{t1}$ and $F_{t2}$. $f$ is friction coefficient, $m_{bush}$ is mass of the bush, $y$ is acceleration when bush is moved, $M_{bush}$ is resultant moment of the bush. The loss of the bush is calculated with Equation (12) ~ (14) and expressed Equation (15) and (16).

\[ L_s = (F_{t1} + F_{t2}) \dot{y} \]  
(15)

\[ L_{bush} = M_{bush} \dot{\alpha} \]  
(16)

$L_s$ is loss occurred in the bush side, $L_{bush}$ is loss of the bush, $\dot{y}$ is velocity of the bush, and $\dot{\alpha}$ is differential value as time in equation (2).

2.3. Dynamics of the roller

The force acting on the roller consist of gas pressure force, centrifugal force occurred roller rotating, and shaft rotating force. First, gas pressure force is occurred with different between compression and suction chamber pressure in outer and inner cylinder. It works perpendicular to the contact point between the roller and the cylinder and the contact point between the roller and the vane as Equation (17) and (18).

\[ F_{ic} = 2(P_{icc} - P_{isc})r_{ico}\sin\left[\left(\pi - \theta - \alpha_{ic}\right)/2\right]H_{ic} \]  
(17)

\[ F_{oc} = 2(P_{occ} - P_{ocs})r_{oc}\sin\left[\left(\theta + \alpha_{oc}\right)/2\right]H_{oc} \]  
(18)

$P_{icc}$ is pressure in inner compression part, $P_{isc}$ is pressure in inner suction part, $P_{occ}$ is pressure in outer compression part, and $P_{ocs}$ is pressure in outer suction part. Second, centrifugal force ($F_{cen2}$) and the Coriolis force ($F_{cori}$) are calculated by Equation (19) and (20).

\[ F_{cen2} = m_{roller}r_g \dot{\alpha}^2 \]  
(19)

\[ F_{cori} = m_{roller}(2\dot{\alpha} \dot{y} \cos \alpha + r_g \ddot{\alpha}) \]  
(20)

$m_{roller}$ is mass of the roller, $r_g$ is center of roller mass, $\ddot{y}$ is differential with time in equation (16). Third, the centrifugal force occurred when the crankshaft rotates and the vertical drag for the linear motion of the rollers are calculated by Equation (21) and (22).

\[ F_{cen1} = m_{roller} \omega^2 \]  
(21)

\[ F_m = m_{roller} \dot{y} \cos \alpha \]  
(22)

$\omega$ is angular velocity of the compressor. Force acting on the roller is calculated resultant force that radial direction force ($F_r$) and tangential direction force ($F_t$) as Equation (23) ~ (25).

\[ F_r = F_{cen1} + F_{oc} \cos \left[\left(\theta + \alpha_{oc}\right)/2\right] + \cdots + F_m \cos(\theta + \alpha) \]  
(23)

\[ F_t = -F_{oc} \sin \left[\left(\theta + \alpha_{oc}\right)/2\right] + \cdots - F_m \sin(\theta + \alpha) \]  
(24)

\[ F_{roller} = (F_r^2 + F_t^2)^{1/2} \]  
(25)

The loss of the roller occurs in the upper and lower surfaces of the roller, the bearing, and between the roller and the eccentric part. And these losses are calculated by Equation (26) ~ (29).
\[ L_b = \dot{\alpha} M_b \]  
(26)

\[ M_b = \frac{\pi \mu \dot{\alpha}}{\delta_{side}} \left[ (r_{oct}^4 - r_{ecc}^4) + (r_{oct}^4 - r_{ico}^4) \right] \]  
(27)

\[ L_c = (\omega - \dot{\alpha}) M_c \]  
(28)

\[ M_c = f_c r_{ecc} F_{roller} \]  
(29)

\[ \mu \text{ is viscosity, } \delta_{side} \text{ is gap between the upper and lower surfaces of the roller and bearing, and } f_c \text{ is friction coefficient calculated by Sommerfeld number and slenderness ratio [10].} \]

2.4. Dynamics of the bearing

As the crankshaft rotates, centrifugal forces occur not only in the forces acting on the rollers and eccentric portions but also in the balanced weight for the stability of the compressor. Force acting on the main journal bearing and the sub journal bearing are calculated by Equation (30) ~ (35).

\[ \sum F_r = F_{roller, r} + F_{ecc} - F_{uw} + F_{iw} - F_{fjr} + F_{mjr} = 0 \]  
(30)

\[ \sum M = -F_{mjr} l_{mj} - F_{sjr} l_{sj} - F_{lw} l_{iw} + F_{uw}(l_{uw} + l_{mj}) + F_{ecc}\left(\frac{H_{oc}}{2} - \frac{H_{ecc}}{2}\right) = 0 \]  
(31)

\[ \sum F_t = F_{roller, t} + F_{mjt} - F_{sjt} = 0 \]  
(32)

\[ \sum M = -F_{mjt} l_{mj} - F_{sjt} l_{sj} = 0 \]  
(33)

\[ F_{mj} = (F_{mj}^2 + F_{mjt}^2)^{1/2} \]  
(34)

\[ F_{sj} = (F_{sjr}^2 + F_{sjt}^2)^{1/2} \]  
(35)

\[ \mu_{mj} \text{ and } \mu_{sj} \text{ are the friction coefficient of the main journal bearing and sub journal bearing, respectively. These coefficients are calculated by Sommerfeld number and slenderness ratio.} \]

3. Compressor model and verification

3.1. Compressor model

Compressor analysis between the conventional and novel one was carried out using simulation program based on numerical analysis. This program considers the mass flow rate, leakage, loss occurred in mechanical part, lubricant oil, and so on. The refrigerant adopted in this simulation program was used R410a. Operating condition was applied the ARI and CHEER. In ARI conditions, condensing pressure is 34.38 kgf/cm\(^2\), condensing temperature is 54.5°C, evaporating pressure is 10.15 kgf/cm\(^2\), evaporating temperature is 7.2°C, pressure ratio is 3.4, sub-cooled temperature is 8.3°C, and suction temperature is 18.3°C. In case of CHEER conditions, condensing pressure is 23.37 kgf/cm\(^2\), condensing temperature is 37.8°C, and pressure ratio is 2.3. Other operating conditions in CHEER are same with ARI conditions. The energy balance of the compressor simulation applied based on 1st law of thermodynamics and mass conservation equation as equation (36) and (37). The kinetic and potential energies are neglected.
\[
\frac{dE}{dt} = Q - W + \sum m_{in}h_{in} - \sum m_{out}h_{out} \tag{36}
\]
\[
\frac{dm}{dt} = \sum m_{in} - \sum m_{out} \tag{37}
\]

Cooling capacity \( \dot{Q}_c \), input power \( \dot{W}_{\text{comp}} \), energy efficiency ratio (EER) was used to evaluate the compressor performance as Equation (38) ~ (40).

\[
\dot{Q}_c = m_{\text{dis}} \Delta h \tag{38}
\]
\[
\dot{W}_{\text{comp}} = \frac{(\dot{W}_{\text{ind}} + L_{\text{mech}})}{\eta_{\text{motor}}} \tag{39}
\]
\[
\text{EER} = \frac{\dot{Q}_c}{\dot{W}_{\text{comp}}} \tag{40}
\]

\( m_{\text{dis}} \) is average flow rate of discharge refrigerant, \( \dot{W}_{\text{ind}} \) is indicated power, \( L_{\text{mech}} \) is mechanical loss, and \( \eta_{\text{motor}} \) is motor efficiency. Table 1 shows the leakage model occurred in the cylinder of the novel twin chamber rotary compressor.

### Table 1. Leakage model of the novel compressor

| Type                      | Location                                      |
|---------------------------|-----------------------------------------------|
| Leakage caused by torque  | Outer suction port ↔ Outer compression chamber |
|                           | Outer compression chamber ↔ Inner suction port |
|                           | Inner suction port ↔ Inner compression chamber |
| Leakage between the vane and the bush | Outer suction chamber ↔ Inner suction chamber |
|                           | Outer compression chamber ↔ Inner compression chamber |
|                           | Outer suction chamber ↔ Inner compression chamber |
|                           | Outer compression chamber ↔ Inner compression chamber |
| Leakage between the cylinder and the roller surface | Outer compression chamber ↔ Inner suction chamber |
|                           | Outer compression chamber ↔ Inner compression chamber |
|                           | Outer suction chamber ↔ Inner compression chamber |
| Shafts                    | Shaft ↔ Inner suction chamber                 |
|                           | Shaft ↔ Inner compression chamber             |
|                           | Shaft ↔ Outer suction chamber                 |
|                           | Shaft ↔ Outer compression chamber             |
| Radial clearance leakage  | Outer Compression chamber ↔ Outer suction chamber |
|                           | Inner compression chamber ↔ Inner suction chamber |

### 3.2. Verification

The novel compressor simulation program is developed based on conventional compressor simulation program. The difference between two simulation programs are only numerical analysis of the bush geometry and calculate the suction and compression volume. In this paper, to verify the novel compressor simulation program, the conventional simulation program was verified under the pressure ratio condition of 1.93 ~ 4.30. Refrigerant and lubricant oil was used same with novel compressor conditions. Operating speed is 45Hz and 90Hz, sub-cooled and super-heated temperature are 8.3°C and 11.1°C, respectively. Condenser temperatures were applied 40 to 55 Celsius. In 45Hz conditions, maximum error of cooling capacity and input power are 5.9% and 5.1%. In case of 90Hz, maximum error of cooling capacity and input power are 8.6% and 4.2%. Figure 5 shows the verification results of the conventional rotary compressor simulation program.
4. Result and Discussions

4.1. Cooling Capacity

Cooling capacity is determined the discharge mass flow rate and enthalpy difference between inlet and outlet port of the compressor as shown equation (41). In the conventional and the novel compressors, the enthalpy difference is same because the inlet and outlet port conditions of the compressors are same. The cooling capacity difference between the conventional and the novel compressors are determined the discharge mass flow rate as shown equation (42). \( \rho \) is the density of the refrigerant, \( A \) is discharge area, and \( V \) is discharge velocity. This equation can be expressed pressure ratio \( (P_r) \), pressure \( (P_u) \) in upstream area, flow coefficient \( (C) \), and specific heat ratio \( (k) \) as equation (42).

\[
\dot{m} = \rho AV \tag{41}
\]

\[
\dot{m} = CP_uA[2k/(k - 1)]RT_u \left( P_r^{2/n} + P_r^{(n+2)/n} \right) \tag{42}
\]

The conventional rotary compressor has one suction port and one discharge port. But the novel twin chamber rotary compressor has two suction and discharge port because there are two compressor chambers. The suction and compression volume of the novel compressor is bigger than 24.56%. It is why the cooling capacity of the novel compressor is higher than conventional one as shown in figure 6.
In ARI and CHEER conditions, the cooling capacity of the novel compressor is 34.68% higher than conventional compressor, respectively.

![Diagram showing cooling capacity comparison between conventional and novel compressors under ARI and CHEER conditions.](image)

**Figure 6.** Comparing cooling capacity with Conventional and Novel compressors

4.2. **Input Power**

Compressor input power is expressed shaft power and motor efficiency as equation (43). And shaft power is calculated indicated power and mechanical loss as equation (44). Figure 7 and 8 shows the PV-diagram and mechanical loss, respectively.

![PV-diagram and mechanical loss for conventional and novel compressors.](image)

**Figure 7.** PV-Diagram of the compressors

![Mechanical losses for conventional and novel compressors at variable rotating speeds.](image)

**Figure 8.** Mechanical losses with variable rotating speed
In PV-diagram, inner area of the dot means adiabatic work and inner area of the line means indicated work. In novel compressor, indicated loss is bigger than conventional compressor because additional volume, in the roller, is exist as figure 7 (b). Figure 8 show the mechanical losses of two compressors. Loss of the novel compressor is lower than conventional compressor because the vane loss is eliminated. It is also advantage in mechanical efficiency when the compressor speed is increased. In conventional compressor, the mechanical efficiency is decreased as compressor speed is increased because the vane loss is increased as compressor speed is increased. But the novel compressor has bush mechanism instead of the vane, mechanical efficiency did not affected significantly by compressor speed.

\[
W_{\text{comp}} = \frac{W_{\text{shaft}}}{\eta_{\text{motor}}} (43)
\]

\[
W_{\text{shaft}} = W_{\text{mdt}} + L_{\text{mech}} (44)
\]

Motor efficiency of two compressors used same value. Finally, compressor input power of the novel compressor is higher than conventional compressor as figure 9. In ARI conditions, input power of the novel compressor is 24.94% higher than conventional compressor. In case of CHEER conditions, input power of the novel compressor is 22.06% higher than the conventional compressor.

![Bar chart comparison of input power with Conventional and Novel compressors](image)

**Figure 9.** Comparing input power with Conventional and Novel compressors

### 4.3. Energy Efficiency Ratio

EER(Energy Efficiency Ratio) is factor that evaluate the compressor performance. It is expressed cooling capacity and input power as equation (40). In ARI conditions, EER of the novel compressor is 8.06% higher than conventional compressor. In case of CHEER conditions, EER of the novel compressor is 10.76% higher than the conventional compressor. It is expressed in figure 10.
5. Conclusions
In this paper, the novel twin chamber rotary compressor was evaluated compared the conventional rotary compressor. To reduce the mechanical loss, the vane mechanism is replaced by bush mechanism. And additional compressions volume in novel compressor is leaded the increase the cooling capacity. Summary of evaluations are as follows.
- The compression volume of the novel compressor is increased 24.56% compared the conventional compressor when the cylinder size is same. It leaded the increasing amount of the discharge refrigerant mass of the novel compressor. So, cooling capacity of the novel compressor is increased 34.68% both ARI and CHEER conditions.
- To reduce the mechanical loss, the bush mechanism was applied the novel compressor. This mechanism advantage in decreasing the vane loss. Despite of that, input power of the novel compressor is increased 24.94% and 22.06% in ARI and CHEER conditions. It is because the indicated loss occurred in the additional compression volume was increased.
- Overall compressor performance was evaluated EER that expressed the ratio of cooling capacity and input power. In ARI condition, EER of the novel compressor is 8.06% higher than novel compressor one. In case CHEER condition, EER of the novel compressor is 10.76% higher than novel compressor one.

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ACKNOWLEDGEMENT
This work was supported by “Human Resources Program in Energy Technology” of the Korea Institute of Energy Technology Evaluation and Planning (KETEP), granted financial resource from the Ministry of Trade, Industry & Energy, Republic of Korea. (No. 20164010201000)