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1-D CYCLE SIMULATION OFAREFRIGERANTCOMPRESSORWITHREED VALVE USING EFFECTIVE FLOW AND FORCEAREAS BY CFD

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ABSTRACT

The refrigerant compressor mainly uses the reed valve in the suction process. The behavior of such a suction valve can be well described through a three-dimensional FSI analysis. However, one-dimensional analysis is still actively used due to excessive computation time of the 3D analysis. For this one-dimensional analysis, effective flow area and effective force area are required. Valve prototypes and valve flow experiments are required for these areas. Therefore, in this paper, to reduce the time and expense, we derived approximate expressions by using CFD instead of experiment. Next, one - dimensional compressor cycle analysis was carried out and the numerical results were compared with the experiment. In addition, we also examined the performance of the compressor according to the suction valve stiffness.

Keywords: refrigerant compressors, PV diagram, refrigeration cycle, effective flow area, effective force area.

INTRODUCTION

Household refrigerators require a lot of electrical energy, and among refrigerator components, compressors consume about 80% of the total refrigerator power [1]. Therefore, raising the efficiency of compressors is very important in terms of environmental conservation and energy consumption reduction.

Various studies have been conducted to improve the efficiency of refrigerant compressors. Relatively simple mathematical models have been developed since 1950 to describe the state of the refrigerant and the motion of the valve during operation [2]. A study on the partial modeling of the cylinder was performed to more accurately describe the actual operating state [3]. Lee *et al*. [4] used piston stroke rather than frequency to control the cooling capacity of the refrigerator compressor. In addition, studies on the optimization of compression chambers and mufflers are currently underway, applying gas pulsation theory [5].

The reed valve used in refrigerator compressors is one of the important factors determining the efficiency of compressors. Therefore, various studies have been conducted to optimize the reed valve. Ferreira and Drjesson [6] investigated the effect of geometric shape changes on effective flow area and force area. Schwerzler and Hamilton [7] presented an analytical method for obtaining effective areas and compared the effective areas with the experiments.

To use the reed valve of the compressor for one-dimensional analysis, the effective flow area and effective force area should be determined in advance. Much time and effort are required to obtain these areas through experiment. Figure-1(a) indicates 1-d approximations of the compressor valves and Figure-1(b) shows the suction valve considered in this study. Hwang [8] performed the 1-D analysis after calculating effective flow area and effective force area by CFD using bilinear interpolation.

In this study, CFD analysis was performed to utilize approximate formulas for effective areas proposed by Schwerzler and Hamilton [7]. Also, one - dimensional compressor cycle analysis was performed to compare PV diagram with experiment. Finally, the performance of the compressor according to the stiffness of the suction valve was also examined.

NUMERICAL METHODS

To obtain effective force area and effective flow area, three-dimensional, steady-state, compressible, and turbulent flow are considered. The turbulent flow is modeled by a realizable k-ε turbulence model proposed by Shinh *et al.* [9]. The governing equations used RANS (Reynolds averaged Navier-Stokes) equations as Equation. (1) [10].

$$\frac{\partial(\rho\overline{u_i u_j})}{\partial x_j} = -\frac{\partial(\overline{p})}{\partial x_i} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial\overline{u_i}}{\partial x_j} + \frac{\partial\overline{u_i}}{\partial x_j} - \frac{2}{3}\delta_{ij}\frac{\partial\overline{u_m}}{\partial x_m}\right)\right] + \frac{\partial}{\partial x_i}(-\rho\overline{u_i' u_j'}) \quad (1)$$

In addition, energy equation for one-dimensional, compressible, and unsteady state flows is used to describe the cycle of the compressor, as Equation. (2) [10].

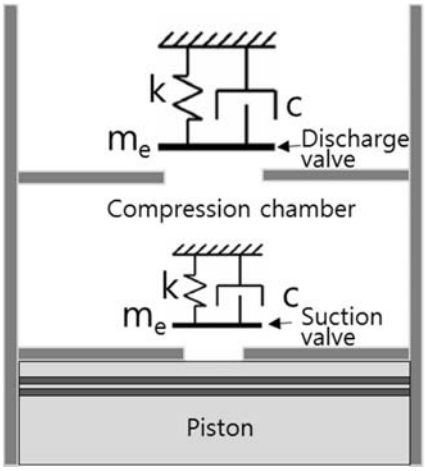
$$\frac{dT_G}{dt} = \frac{1}{m_G c_v}[H_G A_W T_w - h_G \dot{m}_G + \dot{m}_{sv} h_{suc} - \dot{m}_{dv} h_{dis} - \dot{m}_{leak} h_{leak}] - \frac{1}{m_G C_v}\left[H_G A_W + \left.\frac{\partial P_G}{\partial T_G}\right|_v \dot{V}_G - \left.\frac{\partial P_G}{\partial T_G}\right|_v v_G \dot{m}_G\right] T_G \quad (2)$$

In addition, an equation of state such as equation (3) is required.

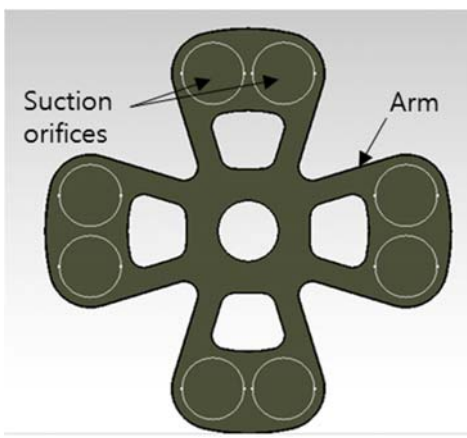
$$P_G = f(T_G, v_G) \quad (3)$$

Figure-2 shows the mesh system of the analytical model for calculating the effective force area and effective flow area. The flow inlet and outlet were located far enough from the valve, more than six times the valve diameter. The suction valve flow considered in this study is three-dimensional flow, but the number of cells is

minimized by using 1/8 model to shorten the analysis time. The number of cells used in the analysis is about 100,000. In addition, a hybrid mesh in which hexahedral cells and tetrahedral cells



(a) 1-D approximation of valves



(b) Suction reed valve

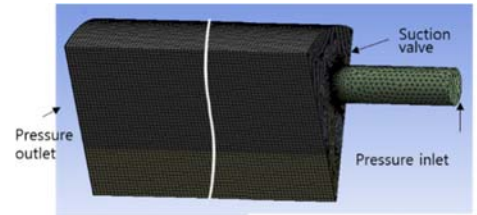
Figure-1. Schematic of the compressor valves.

are mixed is used as the mesh system of the suction valve flow. The discharge valve was assumed to be an axisymmetric model and analyzed in 2D. The mesh uses quadrilateral cells and the number of cells is about 80,000. The definitions of effective flow area and effective force area are shown in equations (4) and (5). Detailed symbol descriptions can be found elsewhere [8].

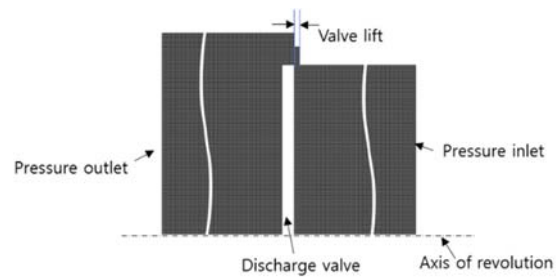
$$(KA) = \frac{(KA)_1}{\left[1 + \left(\frac{(KA)_1}{K_e A_e}\right)^2\right]^{1/2}} \tag{4}$$

$$(FA) = \frac{\mathrm{F}}{\Delta P_V} = (KA)^2\left[\frac{A_p}{(KA)_1^2} + \frac{1}{A_e}\right] \tag{5}$$

In this study, numerical analysis is performed using the commercial program, Fluent [11]. The physical properties of isobutane (R600a) are assumed to be real gas models. The ratio of evaporator pressure and condenser pressure is assumed to be about 9 and the frequency is assumed to be 100 Hz.



(a) Suction valve



(b) Discharge valve

Figure-2. Mesh systems for effective areas.

Table-1. Opening and closing conditions for reed valves.

Events	Events conditions	BC change
Opening of discharge valve	Compression chamber pressure ≥ Discharge pressure	Wall → Pressure outlet
Closing of discharge valve	1) Compression chamber pressure ≤ Discharge pressure 2) piston is at its original position	Pressure outlet → Wall
Opening of suction valve	Pressure on compression-chamber side surface of suction valve ≤ Pressure on compression-chamber side surface of suction valve	Wall → Interior
Closing of suction valve	1)Pressure on compression-chamber side surface of suction valve ≥ Pressure on compression-chamber side surface of suction valve 2) Piston is at its original position	Interior → Wall

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The initial position of the piston at the beginning of the analysis is assumed to be bottom dead center. As the compression progresses, the discharge valve is opened when the pressure of the cylinder reaches the discharge pressure, and the discharge valve is closed when the piston reaches the top dead center. When the piston is moving from the top dead center to the bottom dead center, the pressure on the both sides surfaces of the suction valve are compared. When the pressure on the piston side becomes greater than the pressure on the compression chamber side, the suction valve is opened. Then, when the piston side pressure becomes greater than the pressure on the compression chamber side, the suction valve is closed. Since the behavior of the valve in the entire cycle depends on the effective areas, the areas should be accurately determined. Table-1 summarizes the sequence of processes during a cycle.

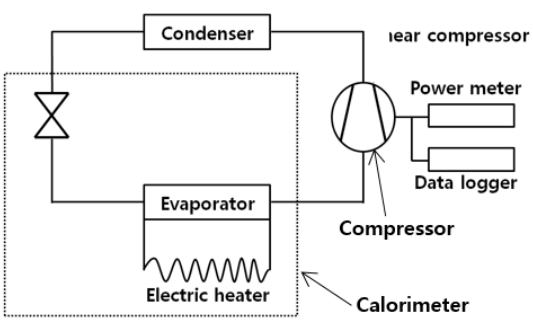
EXPERIMENTAL METHOD

The calorimeter as shown in Figure-3 (a) was used for measuring compressor performance. Refrigerant temperatures at the compressor inlet and the condenser outlet are 25℃. For compression chamber pressure and valve lift, pressure and gap sensors are mounted as shown in Figure-3 (b). The more detailed information is available elsewhere [12].

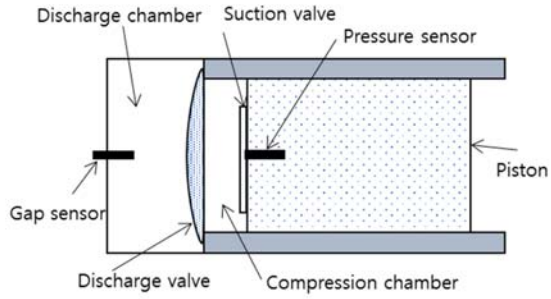
RESULTS AND DISCUSSIONS

Effective areas for suction and discharge valves

Figure-4 shows the variation of effective flow area with valve lift. The considered maximum pressure ratio is about 1.04 for the suction valve and about 1.12 for the discharge valve. In the case of the effective flow area of the suction valve, the valve lift increases linearly up to about 0.5 mm, and then gradually becomes constant. The effective flow area of the discharge valve increases linearly with the valve lift. This flow area determines the valve flow rate in 1-d analysis.

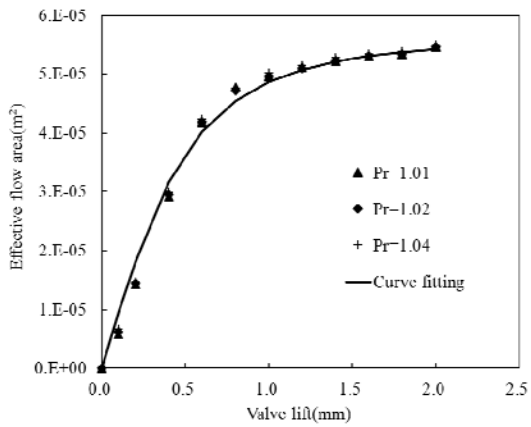


(a) Schematic of the experimental equipment

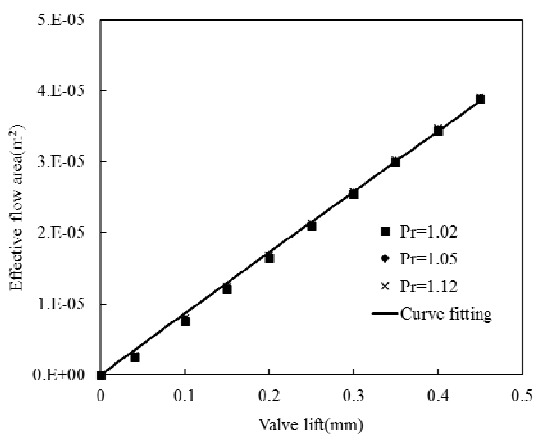


(b) Position of sensors

Figure-3. Schematic of the experimental equipment.

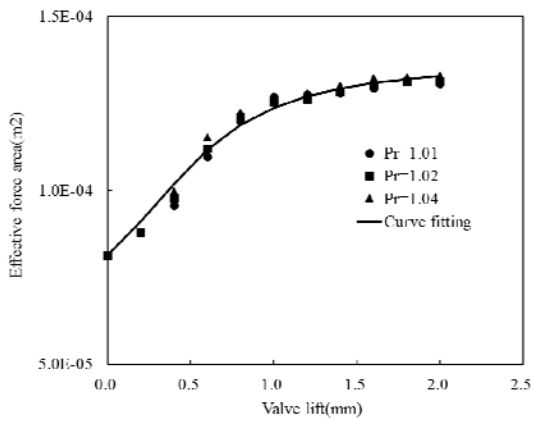


(a) Suction valve

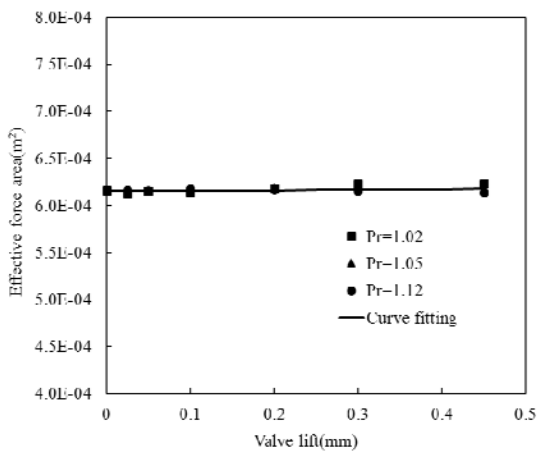


(b) Discharge valve

Figure-4. Variation of effective flow area with valve lift for suction valve and discharge valve.



(a) Suction valve



(b) Discharge valve

Figure-5. Variation of effective flow area with valve lift for suction valve and discharge valve.

Figure-5 shows the variation of effective force area with valve lift. The considered maximum pressure ratio is about 1.04 for the suction valve and about 1.12 for the discharge valve. First, in case of the effective force area of the suction valve, the valve lift increases linearly up to about 0.5 mm, then gradually becomes constant. The effective force area of the discharge valve is constant according to the valve lift. This force area determines the force acting on the valve in the 1D analysis.

Comparison of PV diagrams

Figure-6 shows the PV diagram where compression chamber volume is numerically calculated for both cases. The spring constant, effective mass and damping coefficient are 120 N/m, $1.97X10^{-5}$ kg, and $9.72X10^{-3}$. Numerical and experimental PV diagrams showed significant differences in compression and expansion processes. This difference is due to various causes, mainly due to the refrigerant entering the suction system from inside the compressor housing, the three-dimensional behavior of valves, and internal heat transfer etc. The experimental compression work was about 36W, but the numerical compression work was estimated to be about 43W.

Compressor performance according to stiffness of suction valve

As shown in Table-2, the variation in the performance of the compressor while changing the stiffness of suction valve is shown in Figure-7. The stiffness of suction valve depends on the valve shape, suction orifice position, and so on.

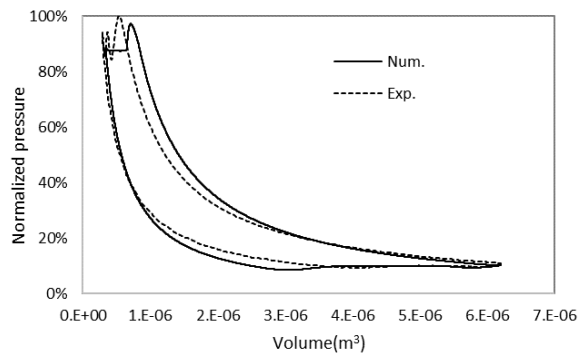
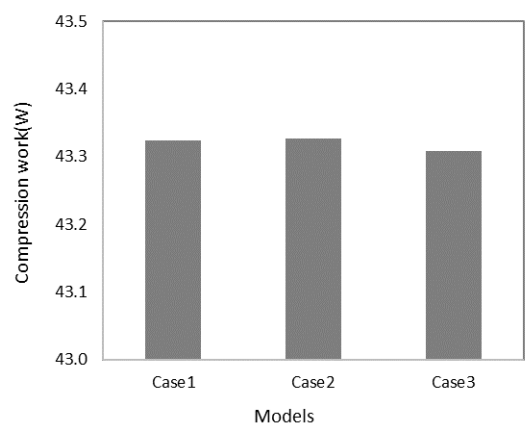


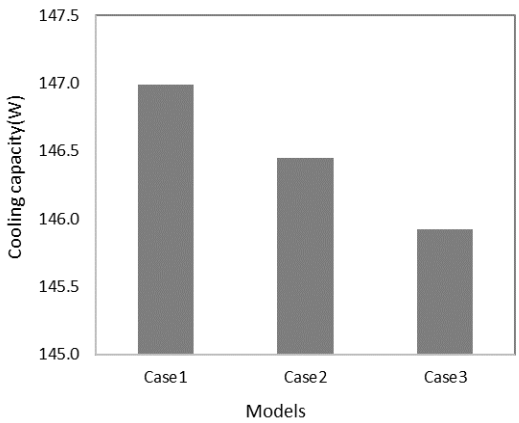
Figure-6. Comparison of PV diagram of the numerical model with the experiment.

Table 2. Variation of spring constant, effective mass and damping coefficient.

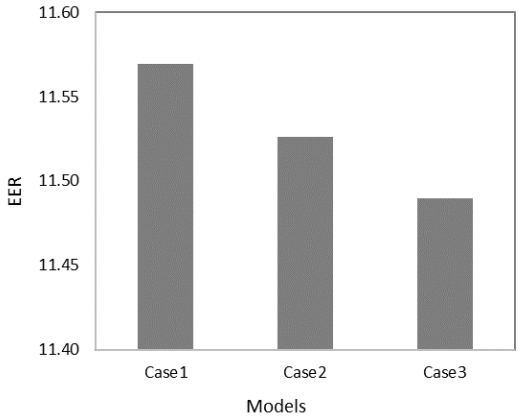
	Case1	Case2	Case3
k(N/m)	**80**	**120**	**160**
m_e(kg)	1.31E-05	1.97E-05	2.63E-05
C	6.48E-03	9.72E-03	1.30E-02



(a) Compression work



(b) Cooling capacity



(c) EER

Figure-7. Variation of compression work, cooling capacity and EER with case.

Figure-7 shows the variation of compression work, cooling capacity and EER with case. As the spring constant increases, the maximum lift of the valve decreases and the amount of refrigerant flowing into the compression chamber decreases. As a result, the cooling capacity is reduced. Also, the compression work does not change much but slightly increases and then decreases again. However, the compression work remains almost constant as the variation is only 0.4%. Therefore, as the spring coefficient increases, the compression work remains constant, and the cooling capacity decreases, eventually reducing the EER. When the spring constant is increased from 80 N/m to 160 N/m, the EER decreases by about 0.7%.

CONCLUSIONS

In this paper, 1-d lumped analysis is performed to calculate the performance of a refrigeration compressor. Also, CFD was performed to derive an approximate expression for the effective areas. The following conclusions were drawn from this study.

a) The effective areas of the suction and discharge valves are hardly affected by the pressure ratio. Therefore, it is possible to derive an approximate expression from the geometric information of the valves.

b) PV diagram from 1-d numerical analysis qualitatively agrees with experiment. However, there are noticeable differences in compression and expansion processes. Therefore, it is necessary to consider refrigerant inflow to the suction system from inside the compressor housing.

c) If the stiffness of the suction valve is reduced, the compression work remains almost constant, and the cooling capacity increases, eventually increasing the EER.

It is necessary to consider the inflow of the refrigerant from inside the compressor housing for the 1-d model in the future.

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