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Prediction of Refrigerant Leakage for Discharge Valve System in a Rolling Piston Compressor

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ABSTRACT

The flow coefficient of a discharge valve system with lift height was functionalized by experiments and Computational Fluid Dynamics(CFD) simulation to predict the mass flow rate through a discharge valve system in a rolling piston compressor with different compressor frequency and operating conditions. The flow coefficients of the discharge valve system were determined by both experiments and CFD simulation for specifically configured condition with varying discharge port diameter, valve lift height and valve shape to consider operating and geometric conditions. The experiment and CFD simulation were conducted under the incompressible flow and steady state. Ambient air was used as a working fluid. In order to verify the reliability of CFD simulation, the computational results were compared with those of experiments. The flow coefficient maps for each discharge valve system could be obtained from the computational results. The functional flow coefficient model was derived from the maps. It was applied to compressor performance simulation in order to calculate the mass flow rate at the discharge valve system as a function of diameter of discharge port and lift height of the valve. Energy Efficiency Ratio(EER) obtained from the functional flow coefficient model with varying compressor frequency showed good agreement with experimental data. The Functionalization of flow coefficient may improve the precision of compressor performance simulation.

1. INTRODUCTION

The compressor is a core component in refrigeration and air-conditioning systems, and consumes most of the energy in the system. The replacement of anti-environmental refrigerants and energy-saving demand have recently caused changes in the components and operation of vapor compression plants; in particular, compressors have been experiencing upgrades and modifications in recent years. Thus, the focus is being put on the improvement of compressor performance in recent years.

The usage of rolling piston compressors has been increasing over the years due to the good properties of the rolling piston compressor such as silent and smooth operating, good reliability and efficiency(Al-Hawaj, 2009).

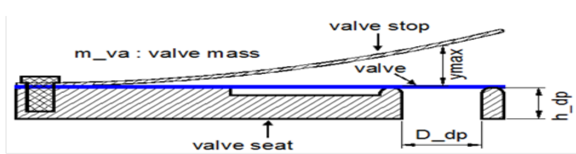


Figure 1: Schemetic diagram of the discharge valve

In order to improve the efficiency of the compressor, many techniques have been applied to the compressor such as the inverter technique to save the energy by controlling rotational speed of the compressor according to the cooling load and new concept compressors have been developed such as a swing rotary compressor and a combined roller and vane compressor.

In the stage of developing a new compressor, performance and operating properties of the compressor can be obtained by numerical compressor simulation. It can cut the cost and reduce the time to develop products (Ahn et al., 2003). Yanagisawa and Shimisu(1984) carried out the researches on predicting leakage losses of clearances inside a rolling piston compressor. Ooi and Wang(1997) proposed a mathematical model for computational simulation of a rolling piston compressor.

Figure 1 shows the discharge valve system of a rolling piston compressor. Analysis of the discharge valve system is very important because the valve behavior is closely related to over-compression loss and reliability problems. Ooi and Chai(1992), Soedel(1985) and so on have conducted researches about the discharge valve system of a rolling piston compressor.

In predicting the performance of a compressor, effective flow area of the discharge valve system should be considered because the motion of the discharge valve varies according to changes of the operating condition and compressor frequency in order to exactly calculate mass flow rate at a discharge valve system. In this study, the flow coefficient of the discharge valve system in a rolling piston compressor was functionalized from experiments and CFD simulation. The functionalized flow coefficient model was applied to the numerical compressor simulation. It could predict the performance of a compressor for different speeds and operating conditions.

2. NUMERICAL ANALYSIS

2.1 Dynamics analysis of a discharge valve

In this study, the motion of a discharge valve is assumed to be one-dimensional and characterized by a lift, and the influence of a retainer is neglected (Ma and Seok, 1996). The valve lift height can be obtained from Eq. (1)

$$F(t) = m_{eff}\ddot{y} + C_{damping}\dot{y} + k(y - \delta_0) = A_{eff}\delta p \tag{1}$$

$$C_{damping} = 2\zeta\sqrt{m_{eff}k} \tag{2}$$

The damping value $C_{damping}$ is obtained by the mass (m_{eff}) and the stiffness (k) in the single-degree-of-freedom system and expressed like Eq. (2). ζ is damping ratio and varies according to change of a bolting method or contact characteristic between the discharge valve and valve seat. In this study, damping ratio was 0.05. It was obtained empirically(Yang *et al*., 2013).

2.2 Discharge mass flow rate

The flow of the discharge valve system is assumed to be isentropic and one-dimension compressible flow through an orifice. Mass flow rate at the discharge valve system can be calculated by Eq. (3).

$$\dot{m} = C_v P_u A\sqrt{\frac{2n}{n-1}RT_u\left[P_r^{2/n} - P_r^{n+2/n}\right]} \tag{3}$$

Where, C_v is the flow coefficient affected by flow area of the discharge valve system. The flow area is determined by diameter of discharge port and lift height of discharge valve.

2. EXPERIMENTAL SET-UP

In order to investigate influences of the valve lift height and diameter of the discharge port on the flow coefficient of the discharge valve system, an experimental device similar to an experimental one of orifice flow was designed because it is difficult to measure the differential pressure and the motion of the discharge valve inside the real compressor at different conditions. Figure 2 and Figure 3 show the schematic diagram of the experimental

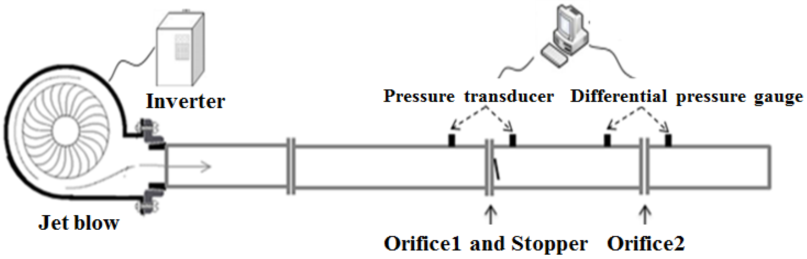


Figure 2: Schmetic diagram of experimental set-up

Figure 3: The orifice plate1

set-up and an orifice plate respectively. The length of pipes of the experimental device was sufficiently long to make the flow fully developed. Stoppers shown in Figure 4 that were similar to shape of the discharge valve but have fixed lift height were used instead of the discharge valve in order to conduct the experiment under steady state. Three types of the discharge valve system were used in the experiments. These types have different length(L), width(W) and radius(R) as shown in Figure 4.

Two orifice plates were installed inside the pipes, one was designed to measure actual flow rate, and the other one had a flow path like one of the real discharge valve system in a rolling piston compressor as shown in Figure3. Ambient air was used as an operating fluid. Also, the experiment was conducted under an assumption of incompressible flow because an expansibility factor which means difference between experimental results of compressible and incompressible flow on the flow coefficient of orifices is approximating 1 (CEN, 2003).

The flow rate was controlled varying rpm of the motor of the blow fan with an inverter. The actual mass flow rate was calculated by Eq. (4).

$$\dot{m}_{act} = C\varepsilon A\sqrt{\frac{2\Delta P_2 \rho}{1-\beta^4}} \tag{4}$$

Where, ρ is the density of ambient air, ΔP_2 is the differential pressure at the orifice2, β is the ratio of diameter of the orifice to pipe duct, C is a flow coefficient of the orifice2, ε is expansibility factor and A is the area of orifiec2. The flow coefficient can be calculated by the Reader-Harris/Gallagher(1998) equation.

The flow coefficient of the discharge valve system was determined by Eq. (5)

$$C_v = \frac{\dot{m}_{act}}{\dot{m}_{th}} \qquad \text{where,} \qquad \dot{m}_{th} = \frac{\pi D_{dp}^2}{4}\sqrt{\frac{2\rho\Delta P_1}{1-\beta^4}} \tag{5}$$

4. CFD SIMULATION

There were experimentally limits due to the capacity of the blow fan and low sensitivity of the differential pressure gauge. Therefore, CFD simulation was conducted using Fluent 14.0 to obtain extra data for small flow area. The CFD solutions were validated with the experimental results. Table 1 shows CFD simulation conditions.

Figure 5 shows geometry for the CFD simulation. The orifice plate2 measuring the actual mass flow rate was omitted from the geometry. In order to make the flow fully developed, however, the length of upstream and downstream was sufficiently long.

Figure 6 shows the result of the CFD simulation. Figure 6(a) is the pressure contour at orifice1, Figure 6(b) is the velocity contour and Figure 6(c) is the velocity vector at orifice1. In Figure 6(c), a recirculation zone is observed below the discharge port. It influences the effective flow area.

Figure 4: Shape of the stopper used in the experiment

Table 1 : conditions for CFD simulation

	Incompressible steady-state
Viscous model	k-ε standard wall function
Inlet boundary	Mass flow inlet
Outlet boundary	Outflow

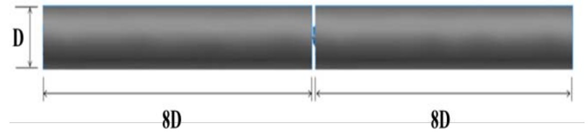
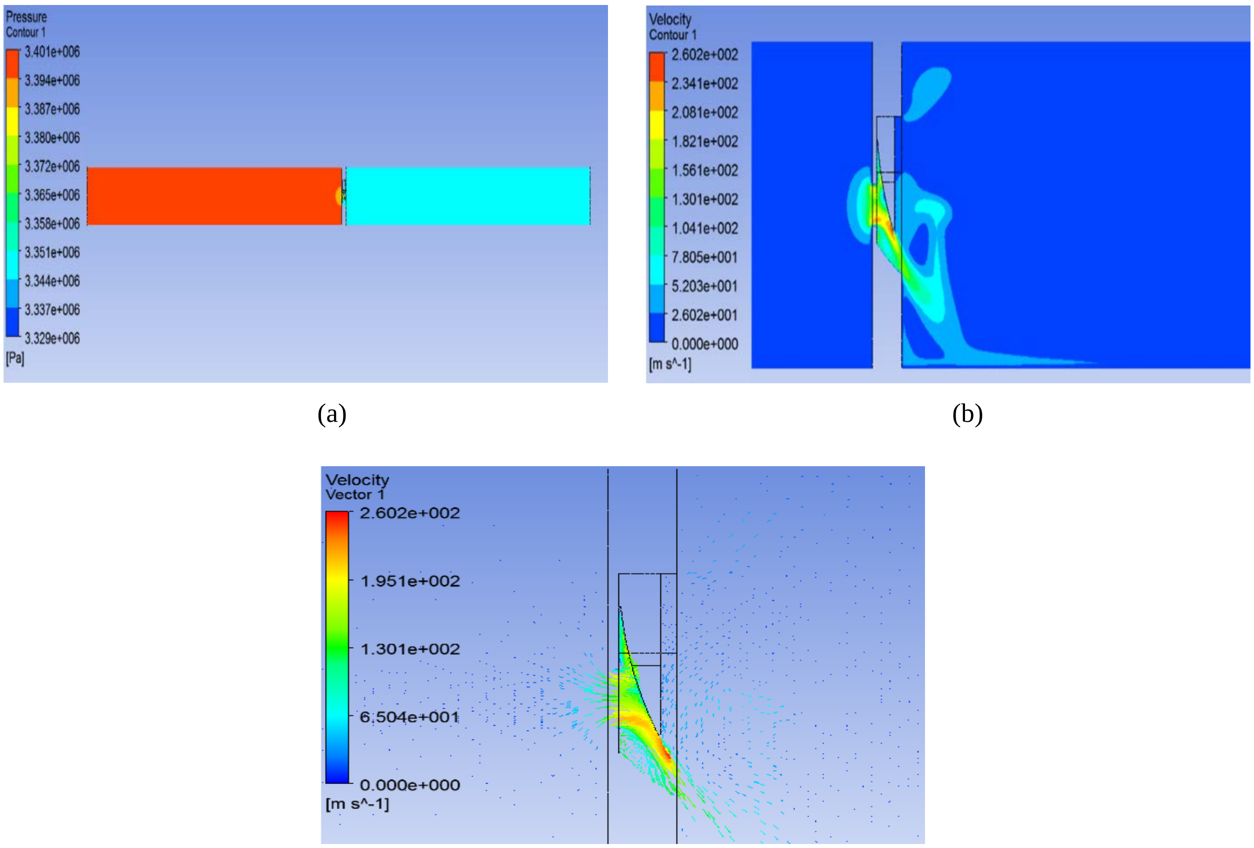


Figure 5: Geometry for the CFD simulation



(a) (b)

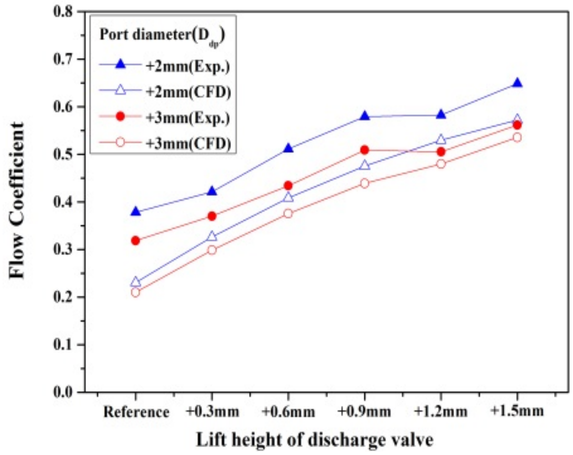
(c)

Figure 6: The results of the CFD simulation (a) Pressure contour, (b) Velocity contour, (c) Velocity vector

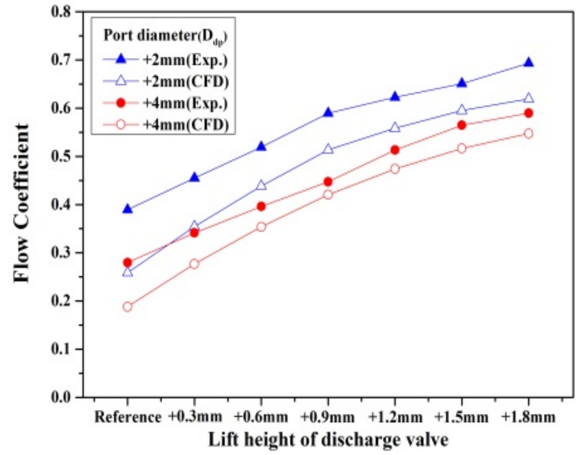
5. RESULTS AND DISCUSSION

5.1 Validation of CFD simulation

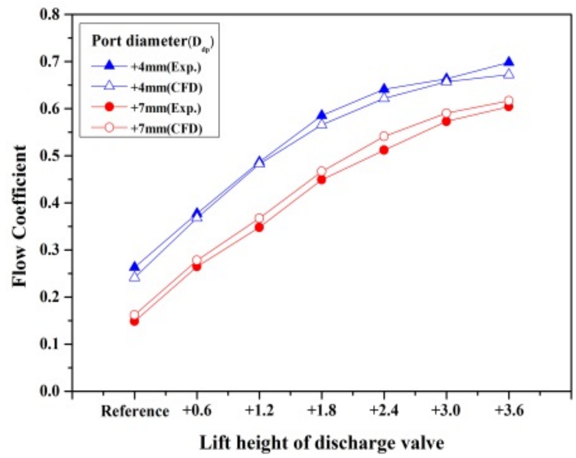
In order to check reliability of CFD simulation results, validation was conducted by comparison of the result of simulation with the experimental result. The experiment for the discharge valve system that has a relatively large discharge port was only carried out. Figure 7 shows comparison of the experiment with CFD simulation for flow coefficient. It can be seen that result of the CFD simulation for the type A which has relatively small discharge port shows substantial difference with the experimental one but the result of the CFD simulation of the type C which has relatively large discharge port shows good agreement with result of the experiment. It is because that the precision of the differential pressure gauge is sharply reduced when flow rate through an orifice is very small due to small discharge area. Therefor there might be many errors in the experimental results for a small discharge port. The maps for the flow coefficient of each discharge valve system were obtained by the CFD simulation



(a) : Type A



(b) : Type B



(c): Type C

Figure 7: Comparison of flow coefficient for experimental and CFD simulation results

5.2 Flow coefficient of the discharge valve system

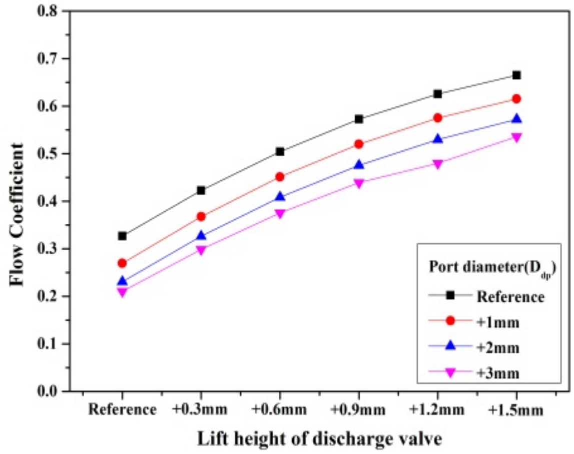
Figure 8 shows the flow coefficient of discharge valve systems with valve lift height and diameter of discharge port. It can be seen that the flow coefficient increases with an increase of lift height of the valve and decrease of diameter of the discharge port. Eq. (6) was derived from the result of CFD simulation as a function of lift height of the valve and diameter of the discharge port. Table 2 shows constants for Eq. (6) of each type of discharge valve system.

This equation was applied to numerical compressor simulation in order to calculate the mass flow rate at the discharge valve system. The lift height was calculated by Eq. (1) and then it was used as a variable of Eq. (6) for every crank angle. Where, y_v is the lift height of the discharge valve and D_{dp} is the diameter of the discharge valve system.

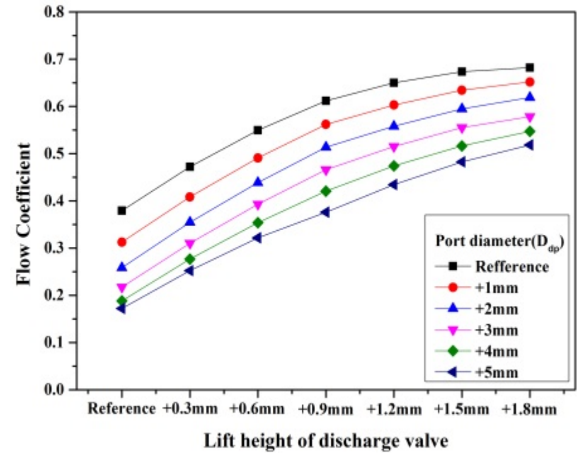
$$C_v = a + by_v + cy_v^2 + dy_v^3 + eD_{dp} + fD_{dp}^2 + gD_{dp}^3 + hy_vD_{dp} + iy_v^2D_{dp} + jy_vD_{dp}^2 + ky_v^2D_{dp}^2 \quad (6)$$

Table 2 : Constants for Eq. (6)

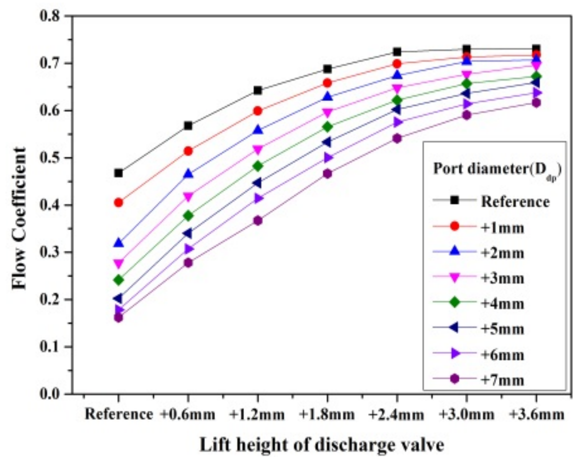
	a	b	c	d	e	f	g	h	i	j	k
Type A	2.78194	0.32304	0.03215	-0.02749	-0.69211	0.06356	-0.00201	-0.01492	-0.00115	0.00183	0.00055
Type B	1.32125	-0.28783	0.01616	-0.00007	-0.37204	0.10356	-0.00699	0.21514	-0.04838	-0.01418	0.00344
Type C	1.19661	-0.18895	0.00813	-0.00005	-0.13464	0.00277	0.00081	0.08624	-0.00999	-0.00432	0.00058



(a): Type A



(b): Type



(c): Type C

Figure 8: Flow coefficient of the discharge valve system for each valve type

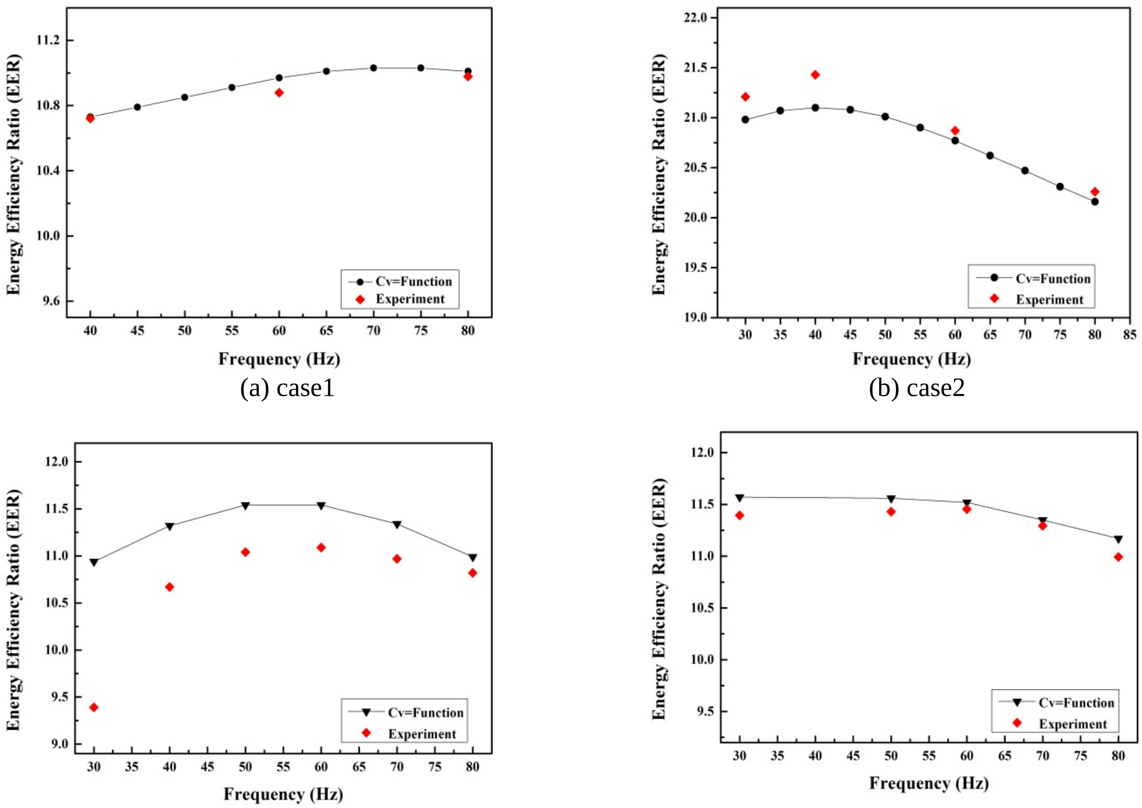
5.3 Numerical compressor simulation

The compressor performance simulation to which the functional flow coefficient model was applied was carried out under the several operating conditions with varying compressor frequency. Table 3 shows the operating conditions and specification of the compressors for the numerical simulation. The compressor performance simulation for the compressors using the type A and B of the discharge valve system which have been widely used was conducted. The result of the simulation was compared with one of calorimeter experiments. Figure 9 shows the change of EER(Energy Efficiency Ratio) that is expressed as Eq. (7) with varying compressor frequency. There is maximum difference of 16.1% in Case3. Although there is substantial difference at low frequency in case3, the difference between the result of the simulation and the calorimeter experiment decrease with increase of compressor frequency and tendency of the simulation result is similar to data of the calorimeter experiment. Except for case3, the result of the simulation for three cases shows good agreement with result of the calorimeter experiment. The precision of the compressor performance simulation can be improved by using the functional flow coefficient model because the mass flow rate at the discharge valve system related to cooling capacity of a refrigeration system is more exactly predicted.

$$EER = \frac{\text{Cooling capacity}\ (BTU)}{\text{Input work}(KW)} \tag{7}$$

Table 3: Operating conditions for numerical compressor

	Case1	Case2	Case3	Case4
Valve type	Type A	Type B	Type B	Type B
Pressure ratio	3.4	3.4	3.4	2.3
Suction pressure	9.95(bar, A)	9.95(bar, A)	9.95(bar, A)	9.95(bar, A)
Discharge pressure	33.8(bar, A)	33.8(bar, A)	33.8(bar, A)	23.9(bar, A)
Capacity of Comp.	10.2cc	12.8cc	12.8cc	12.8cc
Comp. type	Single rotary	2stage rotary	Twin rotary	Twin rotary



(a) case1 (b) case2

(c) case3 (d) case4

Figure 9: The results of numerical compressor simulation

6. Conclusion

In this study, the flow coefficient of the discharge valve system in a rolling piston compressor was functionalized by using result of CFD simulation. The following conclusions can be obtained.

The functionalized flow coefficient model can improve the precision of numerical compressor simulation for variable speed and different operating conditions of a compressor. However, there is maximum difference of 16.1% in a case of numerical compressor simulation. In order to reduce difference between the numerical compressor simulation and calorimeter experiment, researches on a flow coefficient of clearances and a friction coefficient inside a compressor should be conducted, because these factors are closely related to cooling capacity and mechanical loss.

The developed simulator for predicting compressor performance can be very useful to design the rolling piston compressor in the product development stage.

NOMENCLATURE

A_{eff}	Effective force area of the discharge valve	(m^2)
$C_{damping}$	Damping value	(-)
C_v	Flow coefficient of the discharge valve system	(-)
D_{dp}	Diameter of discharge port	(mm)
K	Stiffness of the discharge valve	(-)
$\dot{m}_{act}$	Actual mass flow rate	(kg/s)
$\dot{m}_{th}$	Theoretical mass flow rate	(kg/s)
m_{eff}	Effective mass of the discharge valve	(kg)
n	Specific heat ratio	(-)
P_u	Pressure of upstream of the discharge valve system	(Pa)
P_r	Pressure ratio of upstream and downstream of the discharge system	(-)
R	Gas constant	(-)
T_u	Temperature at upstream of the discharge valve system	(K)
y	Lift height of the discharge valve	(mm)
β	Ratio of diameter of orifice to pipe duct	(-)
δ_0	Initial displacement	(mm)
ε	Expansibility factor	(-)
ρ	Density	(kg/m^3)
δp	Differential pressure	(Pa)
ζ	Damping ratio	(-)

REFERENCES

Al-Hawaj, O., 2009, Theoretical modeling of sliding vane compressor with leakage. Int. J. Refrigeration 32 (7), 1555-1562.

Ahn, J. M., Cho, K. M., Kim, H.J., 2003. Analytical study on the performance of a twin rotary compressor. In: Proceedings of the SAREK Summer Annual Conference, Korea, P. 123

Yanagisawa, T., Shimisu, T. 1985. Leakage losses with a rolling piston type rotary compressor. I. Radial clearance on the rolling piston. Int. J. Refrigeration 8 (2), 75-84.

Ooi, K. T., Wang, T .N., 1997, A computer simulation of a rotary compressor for household refrigerators. Applied Thermal engineering, Vol. 17, No. 1, p,65-78.

Soedel, W., 1984, Design and Mechanics of Compressor Valves, Perdue University, U.S.A, 1984

Ooi, K. T., Chai, G.B., 1992, A Simple Valve Model to Study the Performance of a Small Compressor, International Compressor Engineering Conference. Paper 803.

Ma, Y. C., Seok, J. W., 1996, Dynamic response behavior on rotary compressor valve system. In: Proceedings of the SAREK Spring Annual Conference. Korea, pp. 76-81.

Yang, J. S., Mei, L., Noh, K. Y., Sa, B. D., Choi, G. M., 2013, A sensitivity study of size parameters in twin-type rolling piston compressor, Int. J. Refrigeration 36 786-794.

CEN, 2003, Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full - Part 1: General principles and requirements(ISO 5167-1:2003), BRITISH STANDARD ISTITUTION.

READER-HARRIS, M.J. and SATTARY, J.A., 1996, The orifice plate discharge coefficient equation – the equation for ISO 5167-1. In *Proc*. of 14th North Sea Flow Measurement Workshop, Peebles, Scotland, East Kilbride, Glasgow, National Engineering Laboratory, October 1996, p. 24

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