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A STUDY ON THE PERFORMANCE IMPROVEMENT OF COMPRESSORS USING THE WASTE VIBRATION OF SPRINGS

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ABSTRACT

Many studies have been performed to increase the efficiency of the refrigerant compressor. In this study, we tried to improve the cooling power and efficiency by preventing the rise of refrigerant temperature through the thermal management of the compressor. In particular, the thermal efficiency was improved by heat transfer enhancement using the waste spring vibration. To this end, a CFD model considering valve behavior was developed to analyze the whole cycle including suction and compression processes. A thermal management analysis of the compressor was performed through this model. The results show that using the waste vibration of the dynamic part, the heat transfer of the internal heat of the compressor to the outside air was improved by up to 4% resulting in increasing cooling power and EER up to 0.5% and 0.5% respectively.

Keywords: refrigerant compressor, waste vibration, thermal management, EER, CFD.

INTRODUCTION

Since environmental and energy regulations for the compressor used in the refrigeration and air conditioning area are strengthened, it is very important to improve the compressor efficiency. Generally, a reciprocating compressor or a scroll compressor is widely used for a refrigerant compressor depending on a compression ratio or a cooling capacity. Screw compressors are used in automotive air conditioners and package air conditioners where a big cooling capacity and a low compression ratio are used. Reciprocating compressors are used for a home refrigerator having a small capacity and a high compression ratio. The efficiency of the scroll compressor becomes low when the compression ratio is high or the cooling power is low. This is because the effect of the internal refrigerant leakage on the efficiency becomes much greater.

Although most home refrigerator compressors are reciprocating compressors, some refrigerators use linear compressors using linear motors [1]. Since the linear compressor uses a linear motor with high magnetic force, the crank structure necessary for the reciprocating compressor is not required. Therefore, the linear compressor not only has low friction loss and wear, but also shows a 20-30 % higher efficiency compared to a reciprocating compressor. This is because the suction and discharge valves of the linear compressor are aligned in one direction and the flow resistance is therefore low [2]. Kim and Jeong [3] have studied the performance characteristics of a capacity-modulated linear compressor used in a typical refrigerator; Kim *et al*. [4] have studied the performance characteristics of inherent capacity-modulated linear compressors for household refrigerators through numerical analysis and experiments. Kim *et al*. [5] studied the dynamic characteristics of a linear compressor through numerical analysis and experiments. Bradshaw *et al*. [6] studied a comprehensive model of a compact linear compressor for cooling electronics, and Chen *et al*. [7] studied the magnetic field distribution of linear motors through finite element analysis. Tsai and Chiang [8] studied a magnetically levitated linear actuator of a linear compressor through numerical analysis and experiments. Yang and Huang [9] studied a dual fuzzy controller for the phase or stroke of a linear compressor used in a Stirling cryogenic freezer.

As described above, the efficiency and performance of the linear compressor have been greatly improved. However, in order to obtain a comparative advantage compared with the reciprocating type compressor, it is necessary to further improve the efficiency. In particular, the increase of the refrigerant temperature should be minimized while the refrigerant passes through the suction system of the compressor. For this, a thermal structure to effectively release the heat generated inside the compressor to the outside air is required. There are two major thermal resistances for the internal heat to be released to the outside. One is the thermal resistance from the internal flow to the compressor shell and the other is the thermal resistance from the compressor shell to the outside air.

In this study, in order to improve the efficiency of the linear compressor, a new heat dissipation structure that effectively removes the internal heat of the compressor was considered. Since springs are used in the compressor to move the dynamic part like the piston, we intended to use the waste vibration of the springs to reduce the thermal resistance between the compressor internal flow and the compressor shell. The vibration of the springs is used to form a fast flow velocity area inside the compressor, thereby enhancing the convective heat transfer coefficient on the inner surface of the compressor shell. When the internal heat of the compressor is effectively released, the

superheat of the refrigerant in the suction port is reduced, thereby increasing the cooling capacity of the compressor. Figure-1 shows a schematic view of the compressor considered in this study.

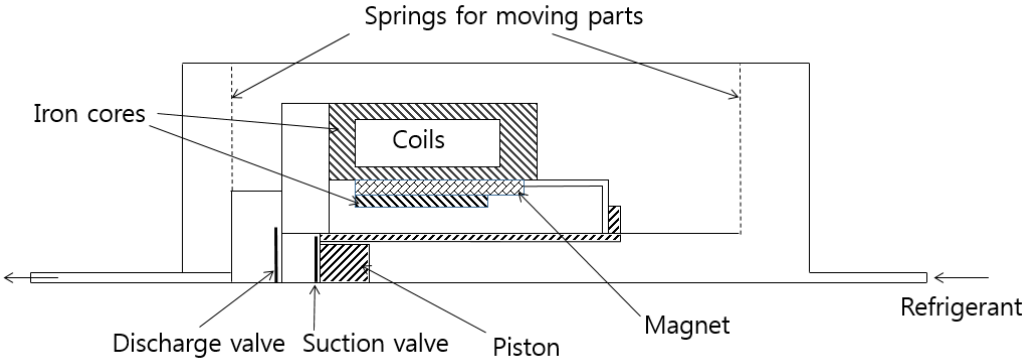

Figure-1. Schematic of a compressor with internal springs.

Numerical method

The fluid flow considered in this study is a three-dimensional, compressible, steady-state turbulent flow. We use the realizable k-e turbulence model proposed by Shinh *et al.* [10]. The governing equations used in this study are continuity equations; Reynolds averaged Navier-Stokes equations and energy equations [11].

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left(\rho \vec{V}\right) = 0 \tag{1}$$

$$\rho \frac{D\vec{V}}{Dt} = -\nabla p + \nabla \cdot \bar{\bar{\tau}} + \rho \vec{f} \tag{2}$$

$$\rho \left[\frac{\partial h}{\partial t} + \nabla \cdot \left(h\vec{V}\right)\right] = -\frac{Dp}{Dt} + \nabla \cdot (k\, \nabla T) + \phi \tag{3}$$

In addition, since the cylinder and piston are inherently unsteady, compression and expansion processes are analysed through a 1-d transient model. This 1-d transient model is coupled with the steady-state CFD model at the suction valve and the discharge valve. The detailed coupling method between the transient 1-d model and the steady CFD model is described in Oliveira *et al.* [12].

The number of grids used in the analysis is about 35,000. A hybrid grid was used in which rectangular and triangular cells are mixed. In this study, a numerical analysis was carried out using the commercial program Fluent [13]. A real gas model for the physical properties of isobutane (R600a) is used. Also, the ratio of evaporation pressure to condensation pressure was based on the general household refrigerator.

The initial position of the piston for the CFD analysis is the bottom dead center. The behavior of the valve in a cycle is predicted with a rigid body model and details of the rigid body model for reed valves are described in Oliveira *et al.* [14]. When the cylinder pressure becomes higher than the spring force in the compression process, the discharge valve opens. When the cylinder pressure becomes lower than the spring force in the expansion process, the discharge valve is closed. Then, when the pressure on the cylinder side of the suction valve becomes larger than that on the piston-port side, the suction valve opens. The suction valve closes when the pressure difference across the suction valve reaches zero. Table-1 shows a series of events for valves during a cycle. In order to improve the flow inside the compressor using the vibration of the spring, the flow passage inside the compressor was modified as shown in Figure-2. The convective heat transfer coefficient was increased by enhanced flow velocity which resulted from narrowing the flow path. In the numerical analysis, the mass flow rate of the gap was changed applying momentum sources rather than changing the shape of the compressor shell.

RESULTS AND DISCUSSIONS

Flow distribution with varying gap

In order to investigate the effect of the vibration of the dynamic part of the compressor on the refrigerant flow according to the gap size, the mean flow rate in the gap was determined according to gap size. The refrigerant was assumed to be incompressible. Figure-3 shows the variation of mean velocity of the refrigerant with gap size. As the refrigerant velocity increases in the gap, the convective heat transfer coefficient between the dynamic parts of the compressor and the compressor shell increases, thereby allowing the internal heat to be released more effectively.

Figure-4 shows the steady-state results where the transient effects of the vibration of the dynamic part are incorporated with momentum sources. The refrigerant flow in the gap changes its direction with time, but many recirculation flows in the gap are generated since a steady-

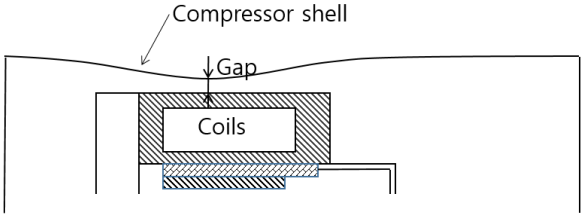

Figure-2. Schematic of a modified compressor shell.

Table-1. Cycle events modeling.

Events conditions	Events
Force by fluid pressure = Force by spring at discharge valve during compression process	Discharge valve open
Force by fluid pressure = Force by spring at discharge valve during expansion process	Discharge valve closed
Force by fluid pressure at cylinder-side suction valve surface = Force by fluid pressure at piston-side suction valve surface during expansion process	Suction valve open
Force by fluid pressure at cylinder-side suction valve surface = Force by fluid pressure at piston-side suction valve surface	Suction valve closed

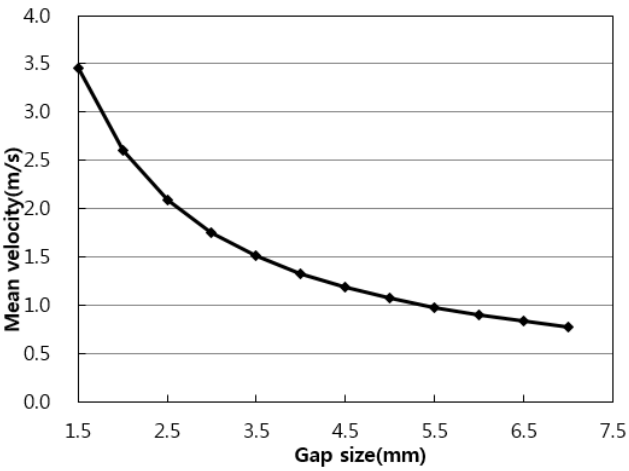

Figure-3. Variation of mean velocity on shell gap.

state model was adopted. As the gap size decreases, the flow pattern does not change but the mean velocity increases. Thus, the heat transfer across the gap becomes enhanced with decreasing gap size

Temperature distribution with varying gap

In order to investigate the effect of refrigerant velocity on heat release across the gap, the steady-state analysis using the momentum source was performed. Figure. 5 shows the variation of temperature at the measurement positions with momentum sources. The temperature was normalized based on the case of gap size 7.0 mm. As the gap size decreases, the flow velocity of the refrigerant increases and consequently the convective heat transfer coefficient increases in the gap, and the temperature tends to decrease. The refrigerant temperature at the suction port for the gap size of 1.5 mm is reduced by about 4.3% compared to that for the gap size of 7.0 mm. The refrigerant at the suction port determines the initial refrigerant temperature in the compression chamber. Therefore, as the refrigerant temperature decreases, the density of the refrigerant increases. As the mass flow rate of the refrigerant increases, the cooling power of the compressor increases, thereby improving the performance of the compressor.

Figure-6 shows the temperature contours at the piston and suction port according to the gap size. Since the refrigerant enters the compression chamber through the inside of the piston, the effect of the piston temperature on the refrigerant temperature is large. As shown in Figure-6, the piston temperature tends to decrease with decreasing gap size. The piston temperature is about 59 °C at gap size of 7 mm, about 62 °C at gap size of 2.5 mm, and about 57 °C at gap size of 1.5mm. Thus, the refrigerant temperature at the suction port also decreases, thereby increasing the performance of the compressor.

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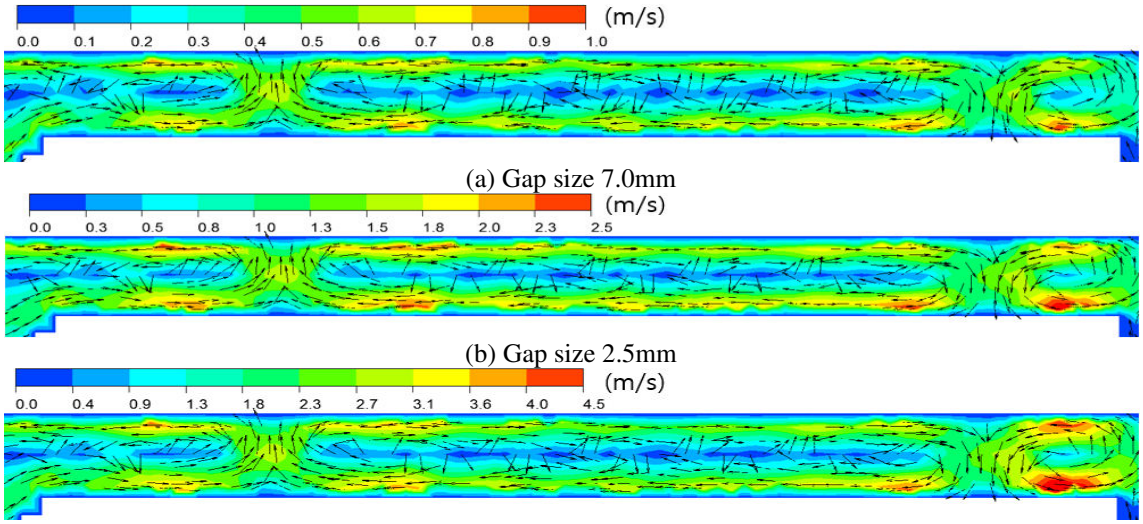


(a) Gap size 7.0mm

(b) Gap size 2.5mm

(c) Gap size 1.5mm

Figure-4. Velocity vector on shell gap.

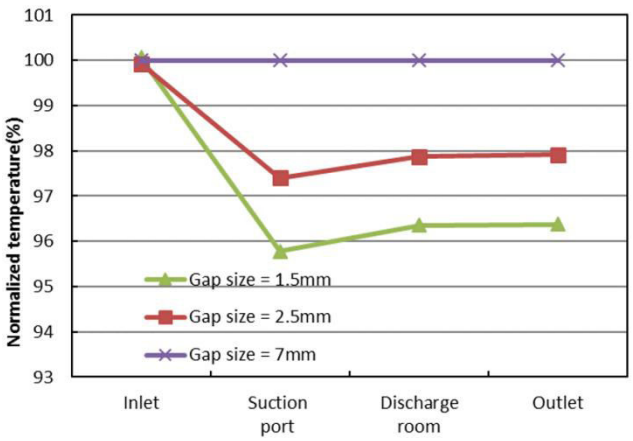
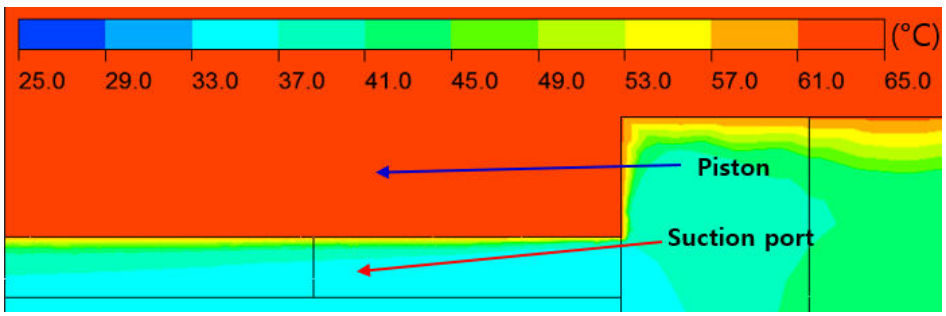


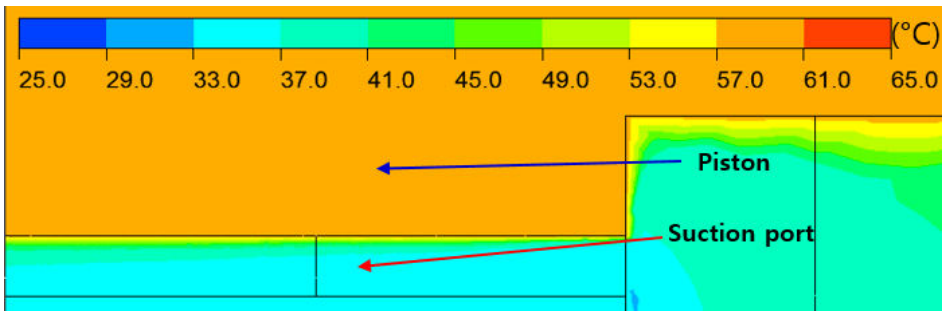
Figure-5. Variation of refrigerant temperature.

Compressor performance with varying gap

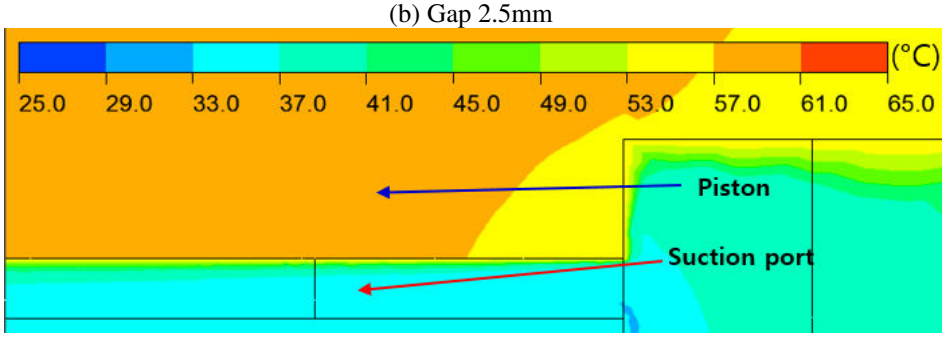
The effect of temperature variation on gap performance was investigated. Cooling power, compression work, and EER were normalized with the results at a gap size of 7.0 mm. Figure-7 shows the cooling capacity of the compressor according to the gap size. The compressor cooling power increases as the gap size decreases. This is because as the gap size decreases, the flow velocity of the refrigerant increases and the convective heat transfer coefficient also increases, thereby decreasing the internal temperature of the compressor. As the internal temperature decreases, the temperature of the refrigerant entering the compression chamber decreases, which increases the density of the refrigerant and consequently increases the cooling power. When the



(a) Gap size 7.0mm



(b) Gap 2.5mm



(c) Gap 1.5mm

Figure-6. Temperature contour on piston and suction port.

cooling power increases, the compressor efficiency can also increase if the same compression work is applied.

Figure-8 shows the variation of compression work with the gap size. The compression work decreases as the gap size decreases. Generally, as the cooling power increases, the compression work increases. However, the compression work is decreased due to the valve opening and closing timings. As the gap size decreases, more refrigerant is sucked and compressed, so the refrigerant pressure rises more rapidly, thereby increasing the discharge valve lift and delaying the closing time of the discharge valve. If the discharge valve closing time is delayed, the expansion process is delayed, which reduces the compression work.

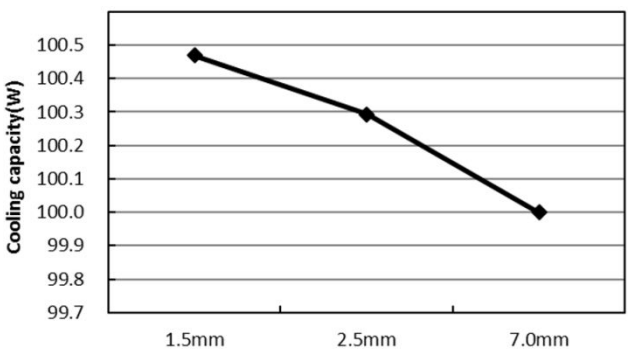


Figure-7. Variation of cooling capacity with gap size.

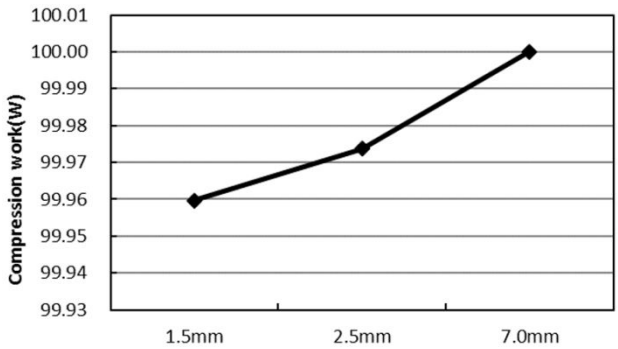


Figure-8. Variation of compression work with gap size.

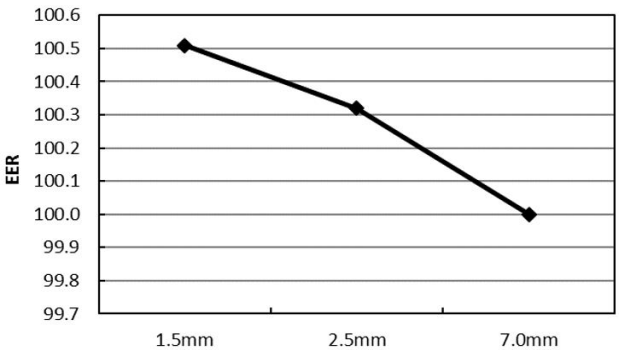


Figure-9. Variation of EER with gap size.

Figure-9 shows the variation of EER(energy efficiency ratio) of the compressor with gap size. As the gap size decreases, the cooling power increases, the compression work decreases and the EER therefore increases. The EER increases up to 0.5 % with varying gap size.

CONCLUSIONS

In this study, the effect of the gap flow due to the spring vibration on the compressor performance was analyzed with varying gap size. The conclusions of this study are as follows:

a) As the gap size decreases, the internal flow velocity increases, and consequently the convective heat transfer coefficient increases, and the internal temperature of the compressor decreases up to about 4%.
b) Both cooling power and EER increase up to about 0.5% since the compressor internal temperature and the refrigerant suction temperature decrease. Therefore it can be seen that decreasing the gap size is beneficial to improve compressor efficiency by enhancing heat transfer.
c) The transient effect of the dynamic part vibration was successfully incorporated to the steady state model using the momentum sources.

In the future, it is necessary to perform an experiment to verify the effects of the gap flows.

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