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STUDY ON THE COMPRESSOR PERFORMANCE CHANGE ACCORDING TO THE MACHINE ROOM ENVIRONMENT IN A REFRIGERATOR

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ABSTRACT

The performance of the compressor varies depending on the ambient environment in which the compressor is located in the refrigerator. The performance of the compressor varies depending on the presence or absence of the cooling fan, in particular. In this study, the convective heat transfer coefficient of the compressor shell surface is calculated numerically with varying the speed of the fan in the refrigerator machine room. The results show that the convective heat transfer coefficient of the compressor surface changes to about 4 ~ 10 W / m^2K depending on the presence or absence of the cooling fan. This change of the convective heat transfer coefficient can reduce the compressor superheat up to about 7 K.

Keywords: compressor, fan, convective heat transfer coefficient, superheat, refrigerator.

INTRODUCTION

Today, refrigerators adopt mostly vapor compression cycle. It consists of four basic components a compressor, a condenser, an expansion valve and an evaporator. The compressor compresses the gas through an electric motor and transfers the high pressure to the condenser. In the condenser, the high temperature and high pressure gas delivered from the compressor is cooled and converted into liquid. The expansion valve decompresses condensed liquid refrigerant in the condenser to a pressure that can cause evaporation by throttling. The liquid refrigerant moves to the evaporator inside the refrigerator, takes the heat away from it, turns into gas, and goes back to the compressor. This is how a refrigeration cycle works [1].

Shang and Fang [2] optimized the minimal stable superheat for a variable speed refrigeration system to increase the refrigerator efficiency. Recently refrigerators use natural convection or forced convection to cool the condenser [3]. Castro *et al.* [4] showed that the compressor running time becomes higher with poor cooling of a machine room, which implies more energy consumption. The convective heat transfer coefficient on the outer surface of the compressor shell changes depending on the cooling conditions, which may affect compressor performance. However, few studies have been performed on the change of the convective heat transfer coefficient by the cooling conditions.

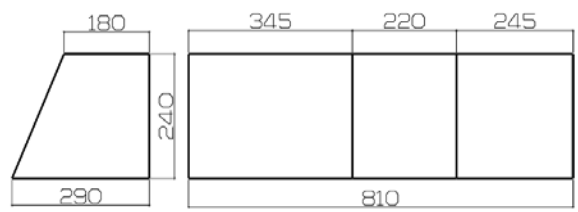
Therefore, in this study, the velocity distribution and the temperature distribution of the interior of the refrigerator machine room in the natural convection and the forced convection were examined. The heat flux and the convective heat transfer coefficient at the compressor surface were then calculated. Also, we predicted the change of superheat of the compressor according to convective heat transfer coefficient.

NUMERICAL METHODS

The heat flux of the compressor surface, q", was obtained by CFD with varying the rpm of the fan in order to know the performance change of the compressor according to the convective heat transfer coefficient. The convective heat transfer coefficient of the compressor surface was calculated using Equation (1).

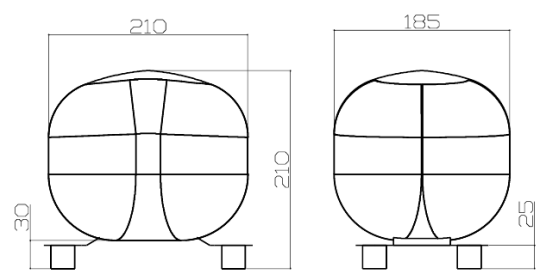
$$q'' = h(T_s - T_\infty) \quad (1)$$

The machine room of the refrigerator was modeled using Catia V5 [5]. Figure-1 shows a schematic of a side view and a front view of the machine room. In Figure-1 (b), the left region is for the compressor, the middle region is for the condenser, and the right region is for the fan and shroud.



(a) Side view (b) Front view

Figure-1. Schematic of the machine room (unit: mm).



(a) Front view (b) Side view

Figure-2. Schematic of the compressor (unit: mm).

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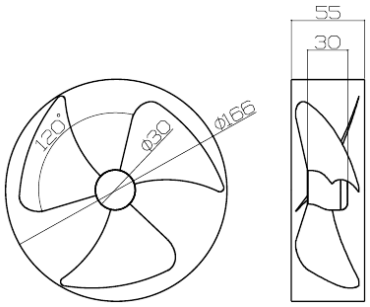


(a) Front view(b) Side view

Figure-3. Schematic of the blade (unit: mm).

Figure-2 shows a schematic of the compressor. The shape of the compressor supports is simplified to avoid excessive cell number when generating meshes.

Figure-3 shows a schematic view of the condenser fan. The fan has three blades and a diameter of 166 mm. We used the MRF (moving reference frame) technique provided by Fluent [6] to describe the rotation of the fan. For grid generation, ICEM CFD [7] was used.

Figure-4 shows the tetrahedral mesh system. The mesh used tetrahedral cells and the number of cells generated is about 9.3 million. The use of such an excessive cell number has the disadvantage of too long analysis time. Therefore, to reduce the analysis time, tetrahedral cells were converted into polyhedral cells [8] and the number of cells was reduced to about 3.8 million. Figure-5 shows the final polyhedral mesh system.

The machine room is assumed to be a closed space and the internal fluid is air. The outside air temperature of the machine room is assumed to be 25 ºC, the compressor surface temperature to 40 ºC, the condenser surface temperature to 31 ºC, and the external convective heat transfer coefficient of the machine room to be 5 W/m^{2}K.

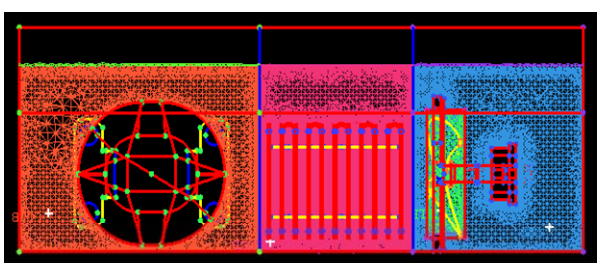


Figure-4. Tetrahedral mesh system.

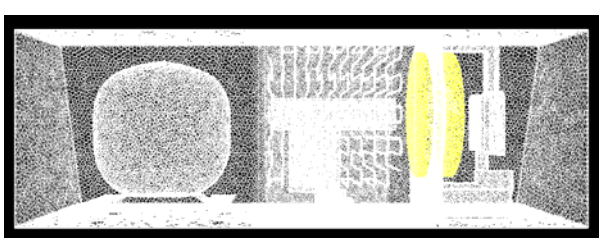


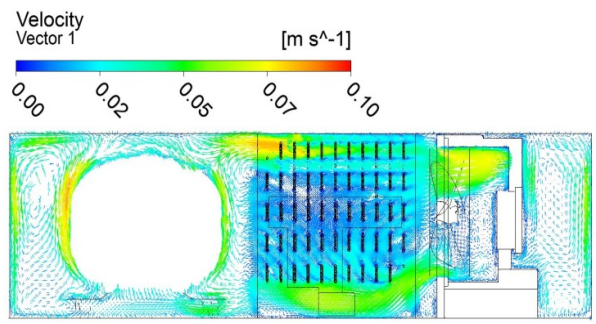
Figure-5. Polyhedral mesh system.

In order to obtain the convective heat transfer coefficient of the compressor surface, the case where there is no fan in the machine room was considered. In this case, the rpm of the fan is 1340. In addition, the variation in the convective heat transfer coefficient of the compressor surface was also examined by varying the rotational speed of the fan. We use the Boussinesq approximation for natural convection analysis.

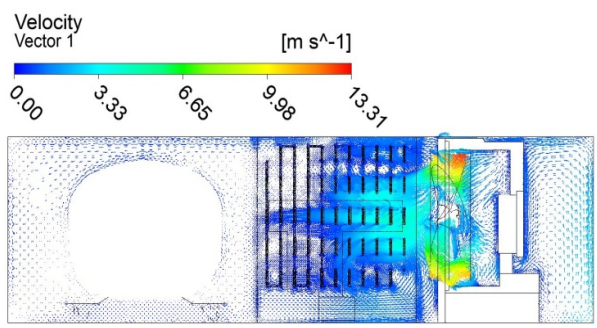
RESULTS AND DISCUSSIONS

Flow distribution in machine room according to presence or absence of a fan

Figure-6 shows the velocity vectors for the natural convection and the forced convection. For the natural convection, the maximum velocity occurs at the top of the condenser and at the back of the compressor. For the forced convection occurs at the end of the fan blades. The maximum flow velocity and the average flow velocity at the compressor surface are summarized in



(a) Natural convection



(b) Forced convection

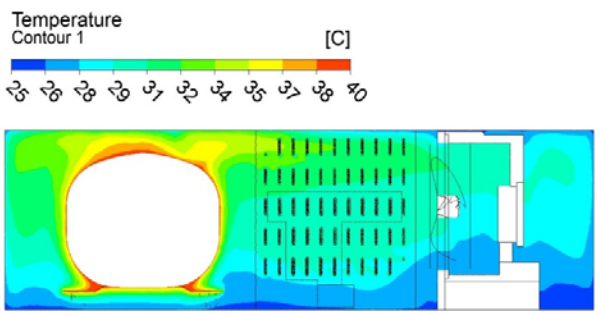
Figure-6. Velocity vectors in the machine room.

Table-1. Velocity of natural convection and forced convection.

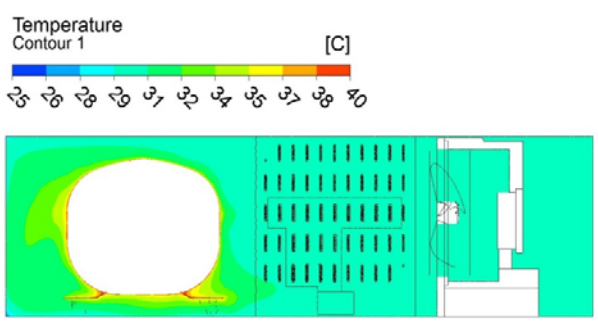
Cases	Maximum velocity	Compressor surface average velocity
Natural convection	0.1m/s	0.05m/s
Forced convection	13.3m/s	0.24m/s

Table-1 the average flow velocity at the compressor surface is about 5 times greater for the forced convection than for the natural convection. The greater the flow velocity on the compressor surface, the more efficiently the heat generated by the compressor is released, which can contribute to reducing the compressor superheat.

Figure-7 (a) shows the temperature contour in the machine room with no cooling fan. In the case of natural convection, the heat is stagnated around the compressor due to a very low flow velocity, and a high temperature region is formed. Also, it can be seen that the lower part of the machine room has a relatively low temperature due to no heat transfer. Figure-7 (b) shows the temperature contour in the machine room with a cooling fan of 1340 rpm. In the case of forced convection, heat is efficiently transferred from the compressor surface, so that the compressor ambient temperature distribution is relatively low compared to the natural convection case. In addition, it can be seen that the temperature distribution inside the machine room is relatively uniform due to high flow velocity. This means that the heat inside the machine room can be released to the outside more effectively.



(a) Natural convection



(b) Forced convection

Figure-7. Temperature contours in the machine room.

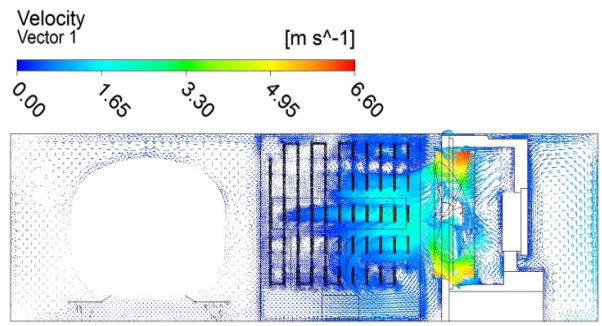
Table-2. Heat flux and convective heat transfer coefficient of the compressor.

Cases	Heat flux	Convective heat transfer coefficient
Natural convection	54.6W/m^2	3.6W/m^2 K
Forced convection	99.6W/m^2	6.6W/m^2 K

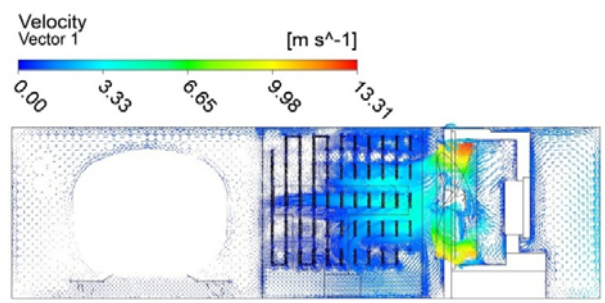
Table-2 shows the heat flux and convective heat transfer coefficient of the compressor surface for natural convection and forced convection. The heat flux for the forced convection is about 1.8 times higher than that for the natural convection. The convective heat transfer coefficient is also about 1.8 times higher for forced convection. This means that the compressor with the cooling fan can release about 1.8 times more heat compared to the compressor with no cooling fan.

Flow distribution inside the machine room with fan speed

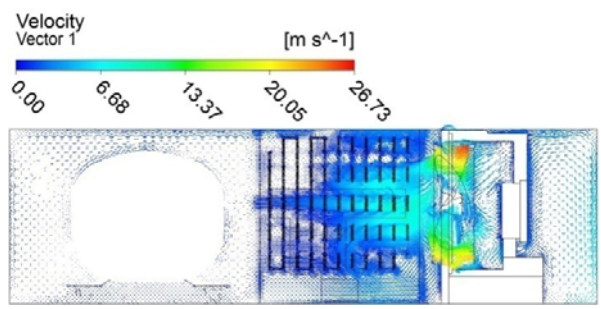
Figure-8 shows the flow distribution inside the machine room according to the fan speed. The fan is mainly used for cooling of the condenser, and the flow rate of the condenser increases proportionally as the fan speed increases. Also, the maximum velocity of the flow field increases to about 6.5 m/s, 13.3 m/s, and 26.7 m/s with



(a) Fan speed = 670 rpm



(b) Fan speed = 1340 rpm



(c) Fan speed = 2680rpm

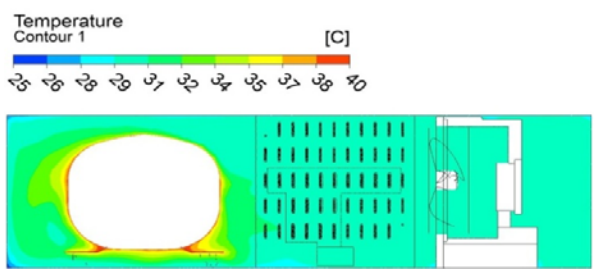
Figure-8. Velocity vectors in the machine room.

Table-3. Velocity of the rotational speed of the fan.

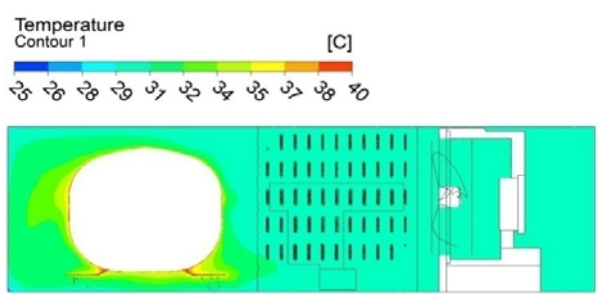
Fan speed	Maximum velocity	Compressor surface average velocity
670 rpm	6.6m/s	0.09m/s
1340 rpm	13.3m/s	0.24m/s
2680rpm	26.7m/s	0.50m/s

increasing fan speed. Therefore, the velocity reaching the compressor surface also increases proportionally. The maximum flow velocity and the flow velocity at the compressor surface are summarized in Table-3. The average flow velocity at the compressor surface increases in proportion to the rotational speed of the fan.

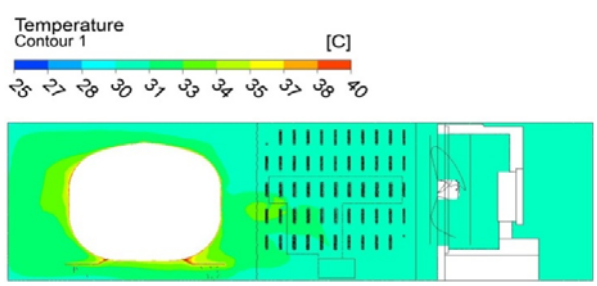
Figure-9 shows the temperature contour inside the machine room according to the fan speed. It can be seen that the temperature distribution around the compressor is relatively high when the fan speed is 670 rpm. Also, the temperature distribution inside the machine room is uneven. On the contrary, when the fan speed is



(a) Fan speed = 670 rpm



(b) Fan speed = 1340 rpm



(c) Fan speed = 2680rpm

Figure-9. Temperature contours in the machine room.

Table-4. Heat flux and convective heat transfer coefficient on the outer surface of the compressor shell.

Fan speed	Heat flux	Convective heat transfer coefficient
670 rpm	62.0 W/m^2	4.13 W/m^2 K
1340 rpm	99.6 W/m^2	6.64 W/m^2 K
2680rpm	143.7 W/m^2	9.58 W/m^2 K

2680 rpm, heat is efficiently released from the compressor surface and the temperature around the compressor is relatively lower. Also, the temperature distribution inside the machine room is uniform. This means that the heat inside the machine room can be released to the outside more effectively.

Table-4 shows the heat flux and the convective heat transfer coefficient on the surface of the compressor according to the fan speed. The heat flux is about 2.3 times higher than that at 670 rpm when the fan speed is 2680 rpm. Likewise, the convective heat transfer coefficient is also 2.3 times higher at 680 rpm than at 2680 rpm. Therefore, if the fan speed is 2680 rpm, the internal heat of the compressor can be released about 2.3 times more compared to the case with 670 rpm.

According to Oliveira [9], when the convective heat transfer coefficient is changed from 5 W/m^{2}K to 10 W/ m^{2}K, the refrigerant temperature at suction port is reduced by about 7 K. Therefore, when the fan speed is changed from 670 to 2680 rpm, the convective heat transfer coefficient changes by about 5.4 W/m^{2}K, so that the refrigerant temperature at suction port is also decreased by 7 K or more. That is, when the superheat of the compressor is decreased, the volume efficiency is increased resulting in increased cooling capacity of the refrigerator.

CONCLUSIONS

In this study, the variation of the convective heat transfer coefficient on the compressor surface with fan speed in the refrigerator machine room was investigated. Through this, the superheat of the compressor and the compressor performance was predicted. The following conclusions can be drawn from this study.

a) When the cooling fan of the condenser operates at the low speed of 670rpm, it can release the internal heat of the compressor about 1.8 times higher compared to the case with no fan.
b) If the fan speed is increased from 670 rpm to 2680 rpm, the convective heat transfer coefficient increases by 5.4 W/m^{2}K, which has the effect of lowering the compressor superheat by about 7 K.

Therefore, the flow distribution around the compressor can affect the performance of the compressor, so this must be taken into consideration when designing

the refrigerator machine room. In the future, experiments for the verification are needed.

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