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# A STUDY ON THE VARIATION OF THE CONVECTIVE HEAT TRANSFER COEFFICIENT ON A COMPRESSOR HOUSING SURFACE WITH OPENINGS OF A REFRIGERATOR MACHINE ROOM

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## ABSTRACT

The openings of refrigerator machine room play an important role in enhancing the heat dissipating performance of the refrigerator compressor. In this study, the compressor housing temperature, the ambient temperature of the compressor and the rotating speed of the fan were measured through experiments. The convective heat transfer coefficient on the compressor housing was then numerically optimized considering various opening shapes. The results show that the convective heat transfer coefficient of the optimized openings increased by about 7 $W/m^{2o}C$ compared to the completely enclosed machine room. Such enhanced convective heat transfer coefficient can contribute to lower superheat of the compressor by efficiently dissipating the internal heat of the compressor.

**Keywords:** refrigerator, machine room, compressor, fan, convective heat transfer coefficient.

## INTRODUCTION

In modern society, a refrigerator is one of household appliances which is essential in home. In order to store food for a long period of time, the refrigerator must continue to operate without interruption. Therefore, the power consumption of the refrigerator is significant compared to other home appliances [1]. In order to reduce the power consumption of the refrigerator, many researchers have been conducting various studies [2].

The efficiency of the refrigerator can be improved by various ways. One way to do this is to increase the thermal performance of the compressor. In 2003, Kim [3] studied internal heat transfer mechanism to improve the thermal performance of a hermetic reciprocating compressor. This has been helpful in predicting the effect of operating parameters on compressor thermal performance. In 2010, Ku [4] improved the thermal performance of compressors using linear compressors instead of reciprocating compressors. Recently, Pilla [5] proposed mixing R-290 refrigerant with R-600a refrigerant to improve refrigeration cycle performance. In addition, there are other ways to improve the thermal efficiency of the refrigerator like improving flow passages. In 2004, Laguerre improved heat transfer and air flow in the refrigerator to enhance the efficiency of the refrigerator [6, 7]. In 2009, Bassiouny [8] found out that the performance of the refrigerator could be reduced in a narrow space where the heat removal of the condenser is not sufficient. In 2013, Afonso [9] improved the performance of refrigerators by reducing heat that infiltrates into the refrigerator by reducing compressor energy consumption. Therefore, air flows around the compressor and in the refrigerator are important factors in improving the thermal performance of the refrigerator.

In the refrigerator machine room, there are openings for dissipating the internal heat of the compressor to the outside with a cooling fan. These openings improve the thermal performance of the compressor and contribute to the total performance of the refrigerator. However, the variation of convective heat transfer coefficient on the compressor housing surface with openings of a machine room has not been known in detail.

Therefore, this study investigated the effect of air flow around the compressor on the convective heat transfer coefficient on the compressor housing surface. To this end, numerical analysis was performed by changing machine room opening. In addition, a heat flux sensor was adopted to verify the convective heat transfer coefficient.

## EXPERIMENTAL AND NUMERICAL METHOD

### Experimental method

The convective heat transfer coefficient on the suction side and the discharge side surfaces of the compressor housing was measured using R600a as a refrigerant. The temperature of the refrigerator was set as -20°C in the freezing room and 2°C in the refrigerating room. The condensing pressure was 5 bars and the evaporating pressure was 0.5 bars. The surface temperature of the compressor dousing, the heat flux, and the ambient temperature were measured using a T-type thermocouple and a heat flux sensor. Figure-1 shows the location of the heat flux sensors and thermocouples attached to the compressor.

The cycle time of the refrigerator was found to be 7 and a half hours by measuring the temperature in the refrigeration room. Therefore, the heat flux and the convective heat transfer coefficient are averaged over the cycle time. They can be obtained from equation (1) using the housing

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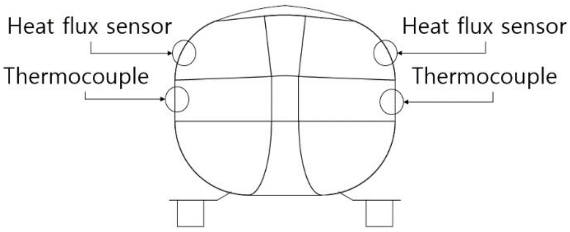


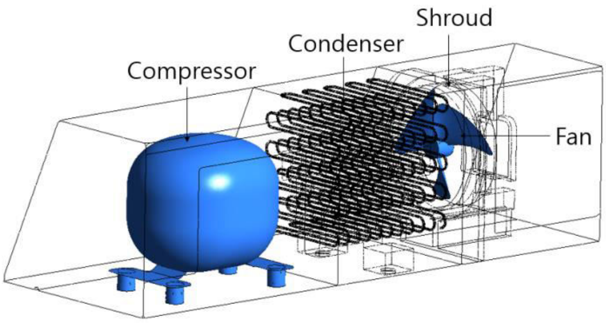

**Figure-1.** Schematic of a compressor.

surface temperature of the compressor.

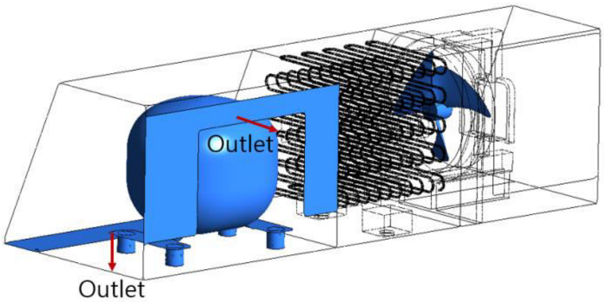
Heat flux= Voltage (mV) x 1000/ST (1)
where, $ST=18.47+(T_{surface}-22.5\ ^{o}C) \times 0.0229$

The experiment was carried out in two cases depending on whether or not fan flow reaches the compressor. When no-fan flow exists, only natural convection around the compressor occurs. This could be done by inserting cardboard between the compressor and the condenser to prevent the flowfrom the fan.

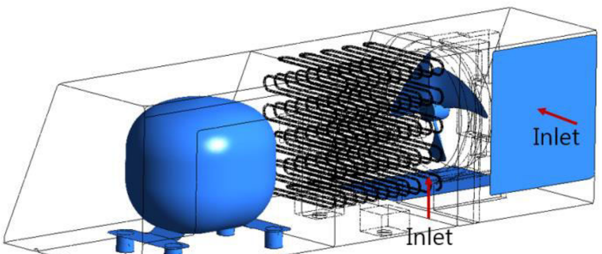
**Numerical method**



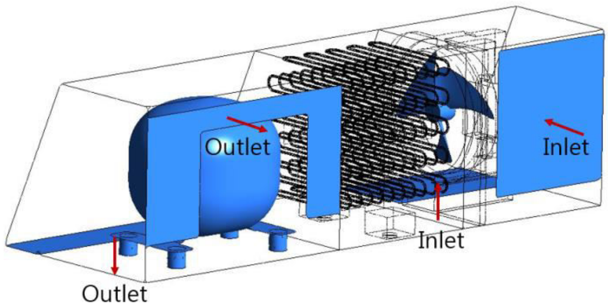

(a) Model 1




(b) Model 2




(c) Model 3




(d) Model 4

**Figure-2.** Machine rooms with varying openings.

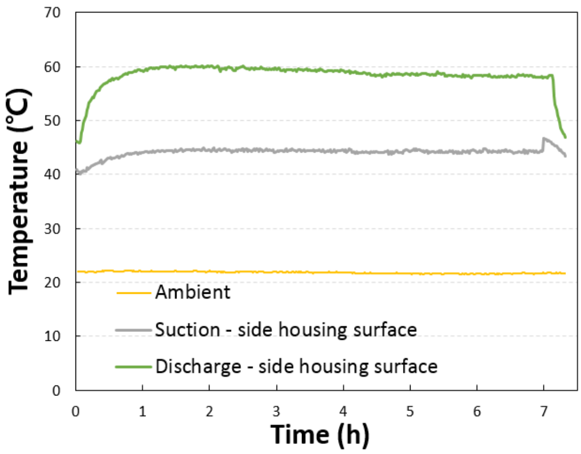
Compressors, condensers, shrouds, fans, etc. in the refrigerator machine room were modeled through the Catia V5 [10]. This study considers four opening shapes as shown in Figure-2. Model 1 is when the machine room is fully closed, model 2 when the compressor-side panel is open, model 3 when the shroud-side panel is open, and model 4 is when both the compressor-side and the shroud-side panels are open.

The meshes were created using ICEM CFD [11] and reduced to 3.8 million cells using the polyhedral mesh system of Fluent [12, 13].

Based on the experimental data, the compressor ambient temperature is 22 $^{o}$C, the convective heat transfer coefficient on machine room panels is 5 W/m$^{2o}$C, the compressor housing temperature is 47 $^{o}$C and the condenser temperature is 31 $^{o}$C. The rotational speed of the fan is 1155 rpm at which the fan speed [15] was measured using a tachometer [14] with a reflective tape attached to the hub of the fan.

**RESULTS AND DISCUSSIONS**

**Measurement of convective heat transfer coefficient on the compressor housing surface**




(a) Temperature vs. time

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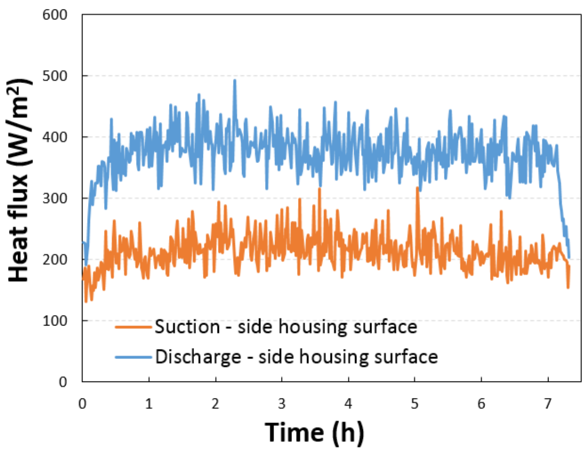
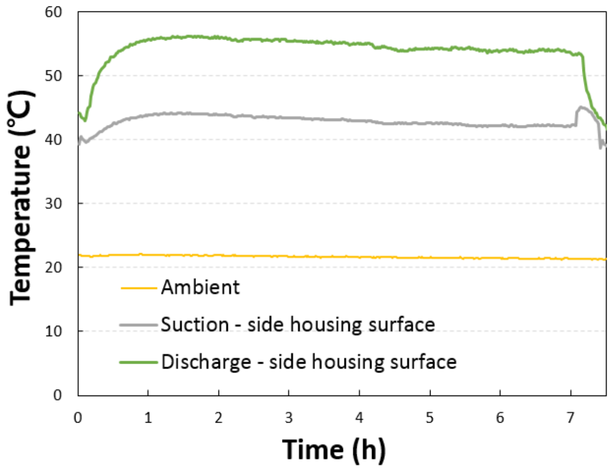
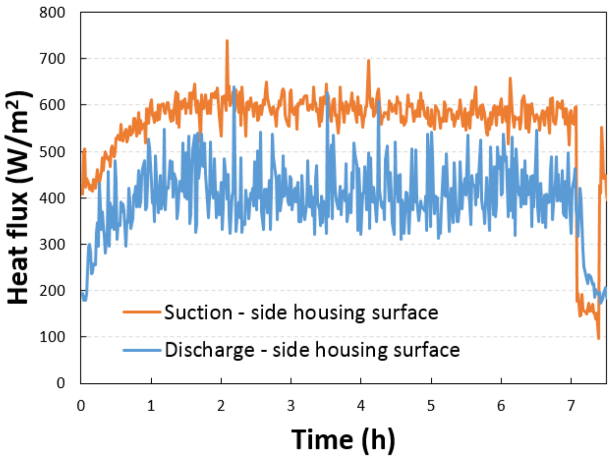



(b) Heat flux vs. time

**Figure-3.**Variation of temperature and heat flux with time for natural convection.




(a) Temperature vs. time




(b) Heat flux vs. time

**Figure-4.**Variation of temperature and heat flux with time for forced convection.

Figure-3 shows the experimental results when the air flow in the machine room is natural convection. Figure-3 (a) represents temperature change with time. The average ambient temperature around the compressor was about 22°C. The average suction-side surface temperature on the compressor housing was about 44°C, and the average discharge temperature was about 14°C higher than the suction temperature. Figure-3 (b) indicates change of heat flux measured on the compressor housing with time. The heat flux on the discharge- side surface of the compressor was higher than that on the suction-side surface. The mean heat flux on the suction-side surface was about 213W/m$^2$ and the mean heat flux on the discharge side was about 372 W/m$^2$. As a result, we have confirmed that the heat flux on the housing surface of the compressor mainly depends on the surface temperature.

Figure-4 shows the experimental results when the air flow in the machine room is forced convection. Figure-4(a) represents temperature variation with time. The average ambient temperature around the compressor was 22°C. The average suction-side surface temperature on the compressor housing was about 42 °C, and the average discharge-side surface temperature was about 11 °C. Compared with the natural convection, it can be seen that the surface temperature of the compressor housing is reduced by the fan flow. Figure-3(b) indicates change of heat flux measured on the compressor housing surface with time. The mean heat flux on the suction-side surface was about 558 W/m$^2$ and the average heat flux on the discharge side was about 396 W/m$^2$. As a result, it was confirmed that the heat flux on the housing surface of the compressor can significantly increase by the fan flow.

**Table-1.** Heat flux and heat transfer coefficient for natural convection.

| Location | Heat flux | Heat transfer coefficient |
|---|---|---|
| Suction-side housing surface | 213.0W/m$^2$ | 9.6W/m$^2$ °C |
| Discharge-side housing surface | 372.2W/m$^2$ | 10.3W/m$^2$ °C |

**Table-2.** Heat flux and heat transfer coefficient for forced convection.

| Location | Heat flux | Heat transfer coefficient |
|---|---|---|
| Suction-side housing surface | 558.2W/m$^2$ | 26.8W/m$^2$ °C |
| Discharge-side housing surface | 396.1W/m$^2$ | 12.4 W/m$^2$ °C |

Tables 1 and 2 shows the convective heat transfer coefficients and the heat fluxes for natural convection and forced convection, respectively. For natural convection, the convective heat transfer coefficient is higher at higher surface temperature of the compressor housing. However, for forced convection, the convective heat transfer coefficient is higher at the suction-side surface than at the discharge-side surface. As a result, it was confirmed that the fan air flow has greater effect on convective heat

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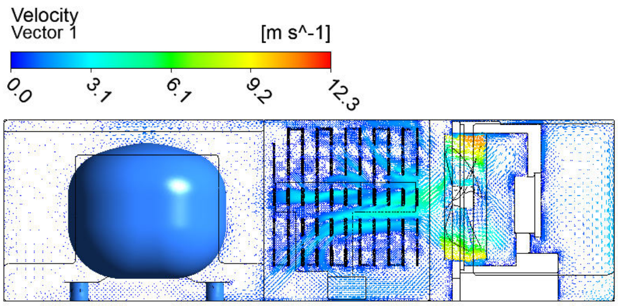
transfer coefficient compared to the surface temperature for forced convection.

**Numerical analysis with opening shape**

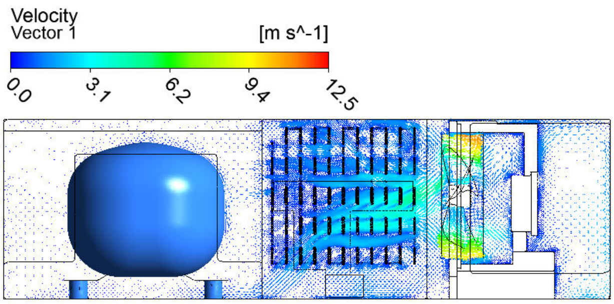
Figure-5 shows the variation of velocity vector with model for forced convection. The velocity vectors of model 1 and model 2 are directed downward. On the other hand, the velocity vectors of model 3 and model 4 are directed toward the compressor. Among four models, model 4 has the most efficient air flows around the compressor to dissipate the heat from the compressor inside.

Figure-6 shows the contour of velocity magnitude near the compressor housing surface. The higher velocity magnitude near the compressor housing indicates the more heat release from the compressor inside to the outside ambient air. Model 1 shows the maximum velocity of 0.65 m/s, while model 4 shows the maximum velocity of 2.17m/s. Therefore, it can be said that the heat dissipation of the compressor to the outside becomes maximum for model 4.

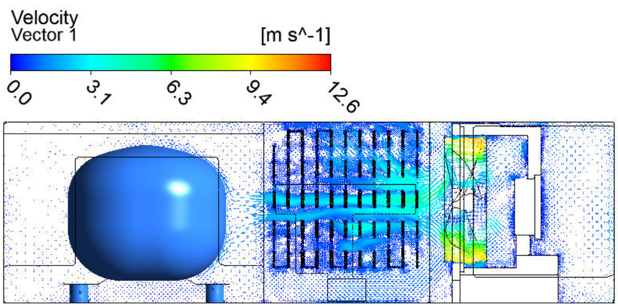
Figure-7 shows the temperature distribution inside the machine room. The mean temperature in the machine room was about 31°C for model 1, about 30°C for model 2, about 27°C for model 3 and about 28°C for model 4. Therefore, the mean temperature inside the machine room was lowest for model 3.



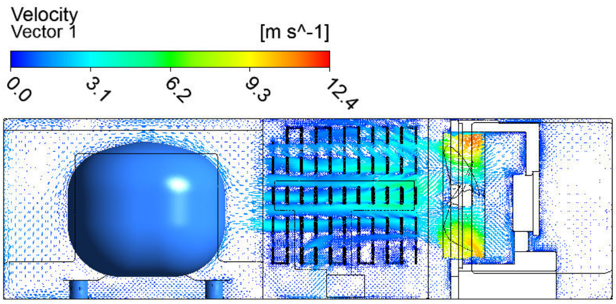

(a) Model 1




(b) Model 2

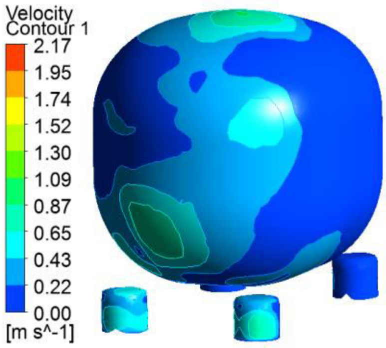



(c) Model 3

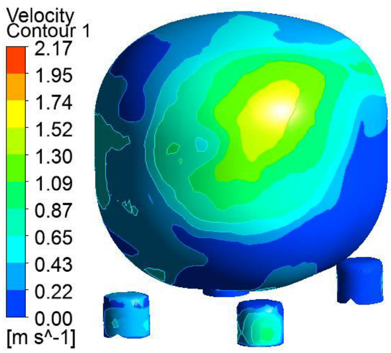



(d)Model 4

**Figure-5.** Velocity vectors for models.




(a) Model 1




(b) Model 2

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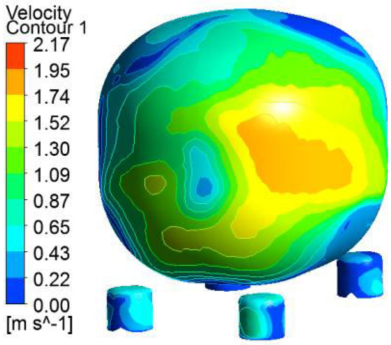
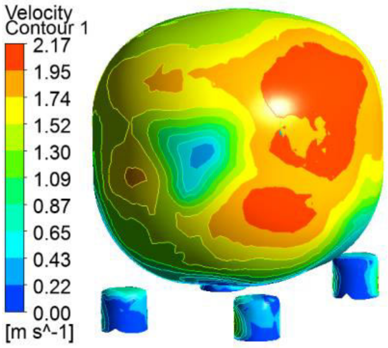


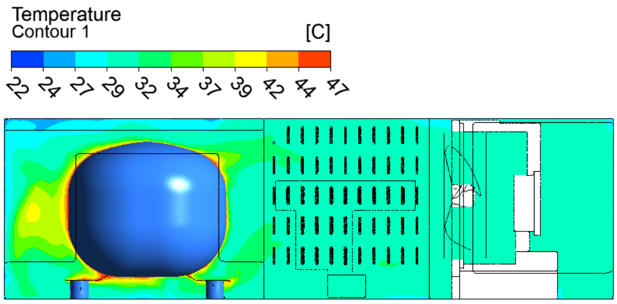

(c) Model 3



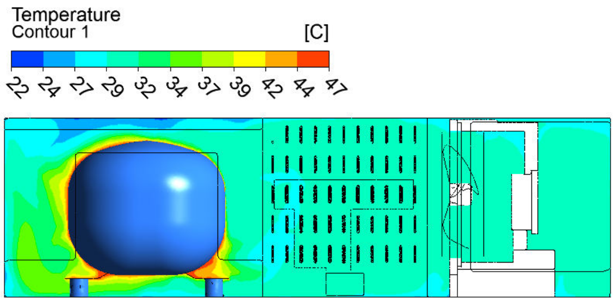

(d) Model 4

**Figure-6.** Contour of velocity magnitude near the compressor housing.

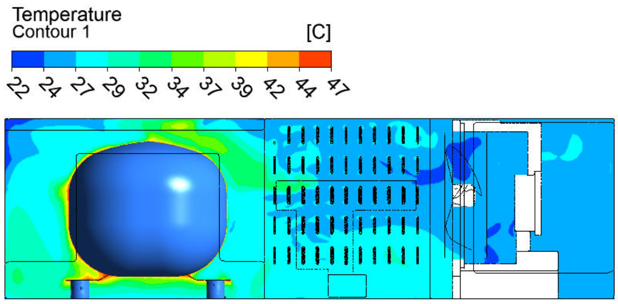
Table-3 shows the mean heat flux and the mean convective heat transfer coefficient on the compressor housing surface. The ambient temperature was 22 °C and the surface temperature was 47 °C. The convective heat transfer coefficient for model 4 is greater by about 7 W/m$^2$ °C, compared to model 1. Therefore, model 4 is an optimized model that allows the internal heat of the compressor to be well dissipated to the outside.



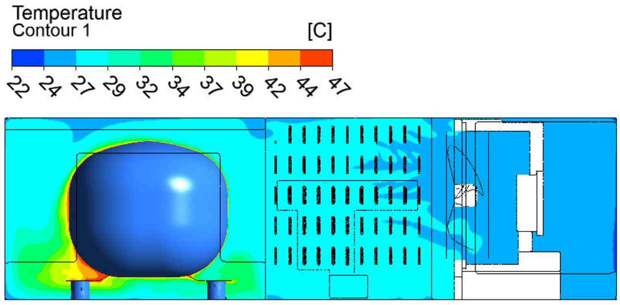

(a) Model 1




(b) Model 2




(c) Model 3




(d)Model 4

**Figure-7.** Temperature contours for models.

**Table-3.**Convective heat transfer coefficient based on ambient temperature and housing surface temperature.

| Cases | Heat flux | Convective heat transfer coefficient |
|---|---|---|
| Model 1 | 132.2 W/m$^2$ | 5.3 W/m$^2$ °C |
| Model 2 | 149.3W/m$^2$ | 6.0W/m$^2$ °C |
| Model 3 | 220.4W/m$^2$ | 8.8W/m$^2$ °C |
| Model 4 | 309.1W/m$^2$ | 12.4W/m$^2$ °C |

## CONCLUSIONS

In this study, convective heat transfer coefficient on the compressor housing surface was experimentally obtained. In addition, the openings of the refrigerator machine room are numerically optimized. The following conclusions can be drawn in this study.

a) Model 4 has the highest convective heat transfer coeff icient and is greater by about 7 W/m$^2$°C compared to model 1.
b) The heat flux on the housing surface is mainly influen ced by the surface temperature for natural convection while it more depends on the air velocity for forced co nvection. Thus, air flow in the machine room needs to be optimized for better thermal efficiency of the com pressor.

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In the future, it is necessary to study the variation of heat dissipation performance of the compressor with flow direction of the fan.


## ACKNOWLEDGEMENTS

This research was supported by Basic Science Research Program through the National Research Foundation of Korea (NRF) funded by the Ministry of Education, Science and Technology (NRF-2016R1D1A1B02009582). This work was supported by the Human Resources Development program (No. 20154030200940) of the Korea Institute of Energy Technology Evaluation and Planning (KETEP) grant funded by the Korea government Ministry of Trade, Industry and Energy.


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