International Journal of Automotive Technology, Vol. 18, No. 1, pp. 41−48 (2017)
DOI 10.1007/s12239−017−0004−x

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pISSN 1229−9138/ eISSN 1976−3832

EXPERIMENTAL STUDY ON PERFORMANCE CHARACTERISTICS OF COLD STORAGE HEAT EXCHANGER FOR ISG VEHICLE

D. W. LEE*

R&D Division, Halla-Visteon Climate Control Corporation, 95 Sinilseo-ro, Daedok-gu, Daejeon 34325, Korea

(Received 21 October 2014; Revised 16 April 2015; Accepted 16 May 2016)

ABSTRACT−The ISG (Idle Stop and Go) system isvery useful in the automobile industry because it increases fuel consumption and reduces green house gas emissions. However, when the engine is on standby, the air-conditioning system does not work due to compressor inactivity, causing thermal discomfort to passengers. This study examines the thermal storage system, which is a cold storage heat exchanger integrated with a current evaporator. The experiments were conducted for an optimum cold storage heat exchanger design with various fin heights and densities, a number of stacking evaporator plates, refrigerant flow circuits inside the evaporator, and PCMs (Phase Change Materials) in the heat exchanger. The effects of coldness-release performance were examined with various ambient temperatures and air flow volume rates to the cold storage heat exchanger. The visualization of PCM's freezing and melting was conducted with the cold storage heat exchanger. From the results, we found that the air discharge temperature of the air-conditioning system that was applied to the optimum cold storage heat exchanger was delayed around 540 seconds compared to the current air-conditioning system to reach 24 °C. Thus we can say that the cold storage heat exchanger integrated with an evaporator is an effective solution for ISG vehicles in maintaining thermal comfort in vehicle cabins during short engine stops.

KEY WORDS : Cold storage heat exchanger, Evaporator, Fuel saving, ISG, Latent heat, PCM, Thermal comfort, Thermal storage system

NOMENCLATURE

D : depth (mm)
H : height (mm)
L : length (mm)
q : cooling capacity (kW)
Q : air flow volume rate (m^3/h)
t : time (s)
T : temperature (°C)
P : pressure (Pa)

1. INTRODUCTION

In light of concerns about environmental protection of the earth and scarcity of fossil fuels, the economization of fuel consumption by vehicles and the reduction in greenhouse gas emissions has become a major global priority. ISG (Idle Stop and Go) system is a vehicle tec-hnology that decreases fuel consumption and exhaust gas by ceasing the operation of an engine during idle mode. In addition, the ISG system increases mileage in the case of waiting frequently at traffic lights during traffic congestion, further protecting the environment from unnecessary pollution. An ISG system, like the one mentioned above, is usually installed in a hybrid electric vehicle due to its known high energy efficiency but nowadays, the system implementation has been widened to include internal combustion engine vehicles. However, vehicles with an ISG system cannot effectively run an air-conditioning system, as the compressor stops that receives driving force via the V-belt, as the engine stops. That is why vehicle air-conditioning systems now face challenges in maintaining thermal comfort of the cabin passengers. To resolve these issues, many possible technologies are being considered, and research has been conducted on thermal storage systems filled with PCMs (Phase Change Materials) in the heat exchanger as a possible solution that provides thermal comfort within a short period of time in the cabin, such as during stops at traffic lights and ad hoc parking (Ao *et al*., 2007).

Konaka and Matsuo (2000) developed a thermal storage system for trucks. For the PCM, around 8 kg of water was used, while simultaneously running the existing air-conditioner to operate the thermal storage system while driving; the cold conditioner operated at 26 °C for 4 hours. As a result of this study, cold storage amounted to approximately 640 Wh, which is necessary after calculation of the time needed for thermal storage during the average driving distance. After reviewing several thermal storage materials under these conditions, water has been identified as the most superior cold storage material.

Various trials involving the use of an existing evaporator as part of a vehicle thermal storage system, have been

**Corresponding author.* e-mail: dlee6@hvccglobal.com

conducted over the past few years. In particular, many efforts to integrate the evaporator and cold storage heat exchanger have been conducted in several countries in order to adapt to the limited space within vehicles (Regine *et al*., 2004); integral type heat exchangers for effective heat conduction between the evaporator and cold storage heat exchanger were suggested as alternatives (Cathy *et al*., 2002; Kitamura *et al*., 2004).

Craig *et al*. (2010) investigated an integrated-type cold storage heat exchanger, which repeats cold storage and thawing using a thermo-siphon effect. Heat conduction through an evaporator tube enables effective cold storage by PCM, and the operating strategy used to control the air temperature and air flow volume rate of air-conditioning systems was also studied by considering thermal comfort and vehicle driving range.

Lee *et al*. (2011) conducted an experiment on freezing in the ice tube of the vending machine. The ice tube was made with transparent plastic and mechanically sealed after insertion of a screw-type scrape in order to study the implantation and freezing over time.

Denso Corporation (2012) developed a cold storage evaporator consisting of an integral-type evaporator applied to the Suzuki Wagon R. This cold storage heat exchanger is considered helpful because it increases passenger thermal comfort and vehicle mileage by supplying cold air to the cabin using the latent heat of the PCM during engine stop after activating the cold storage heat exchanger when the air-conditioner is in operation.

Lei *et al*. (2013) modeled the PCM of the thermal storage system using CFD and verified their predictions by their experiment. They were able to increase the cold conditioning system by approximately 6.1 % and decrease power consumption by 61.6 % through an optimized combination of surface roughness and air flow volume rate of the heat exchanger, which influenced the cold conditioning system performance.

In a vehicle with ISG technology, taking into consideration the driving mode, at least 5 % improvement is discernible, according to several research studies. With respect to thermal storage devices, which can minimize energy consumption when an engine stops, Kowsky *et al*. (2014) attempted to resolve this issue with a cold storage heat exchanger using a thermo-siphon effect. By analyzing the freezing and melting temperature of the PCM loaded into the heat exchanger, the cold storage heat exchanger, which is integrated with the evaporator, was developed and researchers were able to prove that the cold storage device increased passenger thermal comfort during testing in a climate wind tunnel.

This study examines a technology that can supply latent heat, stored in a heat exchanger, to a vehicle air-conditioner within a short period. The technology consists of an integral evaporator, which is impractical in an air-conditioning system due to the stopping of engines for vehicles equipped with an ISG device.

The heat exchanger functions as an evaporator when the air-conditioner is operating, by storing ice in the PCM inside the heat exchanger. When the compressor stops, it supplies cold air to the cabin. This cold storage heat exchanger is optimized through experimentation on the geometrical feature change of fin height, density and refrigerant circuits. Also, several kinds of PCMs were investigated to study the influence of thermodynamic characteristic of PCMs on cold storage heat exchangers. The effects of an optimized cold storage heat exchanger on an air handling system were studied to determine applications of cold storage heat exchanger in air-conditioning systems with various air flow volume rates and ambient temperatures.

2. EXPERIMENTAL EQUIPMENT AND METHOD

2.1. Calorimeter of Cold Storage Heat exchanger

Figure 1 illustrates a calorimeter's structure used in a cold storage heat exchanger performance experiment. A calorimeter is thermally isolated to prevent loss of heat to the environment, with an induction guide vane attached to distribute air flow volume equally to the experiment heat exchanger in the test section chamber.

The temperature can be controlled from – 10 °C to 50 °C and the heating device can evenly heat the air. The freezing device controls the temperature of the air-conditioner chamber and consists of a 30-HP compressor, expansion valve and evaporation pressure controller. High airflow volume and low airflow volume are separately measured with a Brandt Corp. B-NZP1000 model nozzle while vapor at 180 °C is provided for humidity, controlled by an enthalpy-calculating humidity sensor.

The temperature is measured by installing the RTD sensor at the entrance and exit of the cold storage heat exchanger. The test section chamber, and the amount of circulating refrigerant was measured by the mass flow meter. In order to minimize the influence of refrigerant oil, two oil separators were installed in series to contain

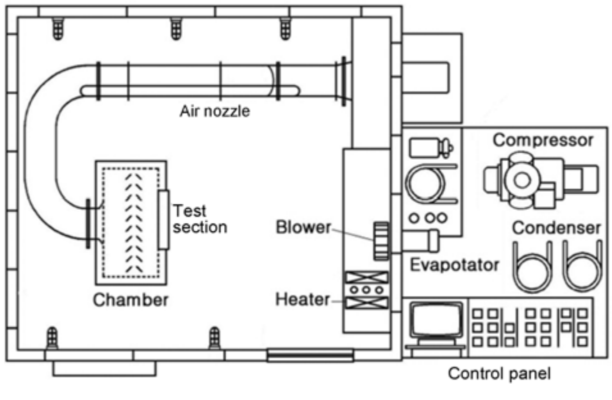
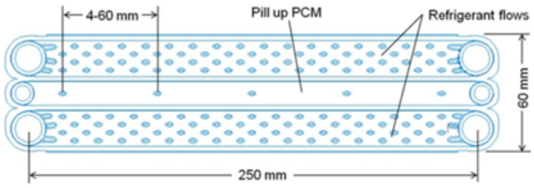
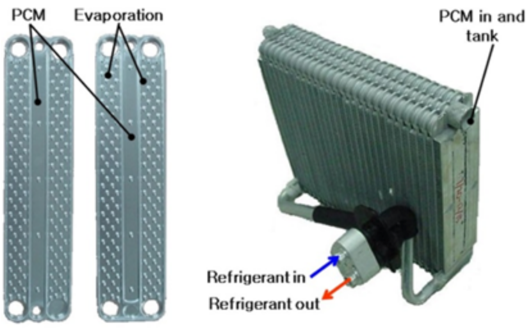


Figure 1. Schematic diagram of cold storage heat exchanger calorimeter.



(a) Dimension of cold storage heat exchanger plate



(b) Cold storage heat exchanger plates and assembly

Figure 2. Photo of cold storage heat exchanger.

refrigerant oil of less than 3 % in the calorimeter apparatus.

In Figure 2, the integral-type cold storage heat exchanger was developed in this study and its features were illustrated in detail. The cold storage heat exchanger was formed with brazing thin aluminum plates stacked atop one another as the side ends facilitated evaporation where refrigerant flowed around the center portion of where the PCM was inserted.

Figure 2 (a) shows the main dimension of the cold storage heat exchanger plate, and five small dimples in the center portion, 60 mm apart in order to prevent supercooling during thermal storage of PCM. Figure 2 (b) shows how the cold storage heat exchanger was manufactured by stacking the corresponding cold storage heat exchanger plates.

Table 1 shows various heat exchangers produced for performance experimentation of cold storage heat exchangers. The Base has a size of 274 W (mm) × 250 H (mm) × 45 D (mm) by stacking 27 number of plates on the existing evaporator without a thermal storage part, but the cold storage heat exchanger's size is increased by 15 mm to 274 W (mm) × 250 H (mm) × 60 D (mm), by containing the cold storage part.

In Table 1, cold storage heat exchangers #1 ~ #4 have approximately 185 cm^3 of PCM charged in the cold storage part, and #4 and #5 cold storage heat exchanger have 260 cm^3 of PCM in the center portion which is cold storage part. In this case, the cold storage heat exchanger was experimented on with various fin heights, fin densities, and refrigerant flow circuits inside of the heat exchanger to have superior cold storage performance while maintaining the same cooling performance as the existing evaporator (Base).

Table 1. Experimental components of thermal storage heat exchanger.

No	Number of plate (ea)	Ref. flow path	Fin height (mm)	Fin density (FPDM)
Base	27	7/20 → 12/15	7	68
#1	27	7/20 → 12/15	7	54
#2	27	7/20 → 16/11	7	62
#3	27	7/20 → 19/8	7	68
#4	27	13/14 → 14/13	7	62
#5	38	11/27 → 20/18	4	56
#6	38	11/27 → 22/16	4	56

Table 2. Experimental conditions of cooling performance in calorimeter.

Air inlet temperature	27 °C, 50 % RH
Air volume low rate	450 m^3/h
Refrigerant inlet pressure	15.7 kg/cm^2
Refrigerant inlet temperature	54 °C
Refrigerant outlet pressure	5.5 °C

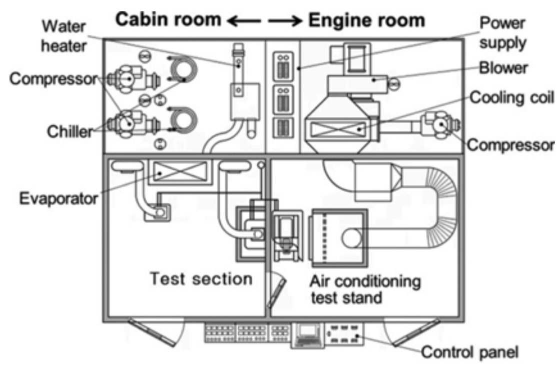


Figure 3. Schematic diagram of cold storage air-conditioning system test apparatus.

Table 3. Cold storage air-conditioning system test components and specifications.

Condenser	Type	PF type, 34 tubes
	Size	677 W × 341 H × 16 D
Evaporator	Base	274 W × 250 H × 45 D, 68 FPDM
	CS HEX #2	274 W × 250 H × 60 D, 62 FPDM
Compressor		VS-18 Swash plate type
Thermal expansion valve		1.5 K at 0 °C, 1.05 Slope

Table 2 consists of experimental conditions for the heat exchanger performance at the calorimeter chamber.

The cooling performance of the heat exchanger used stabilized experimental data below ± 2 % taking into consideration both refrigerant and air side cooling performances which were respectively calculated by a calorimeter.

2.2. Experiment of Coldness Release Performance

Figure 3 is a schematic of the cold storage and release system experiment apparatus with the cold storage heat exchanger in the air-conditioner.

The experimental device was built according to the vehicle cabin and engine room. Various controls are available such as temperature, humidity and air flow volume rate etc. In the cabin room, which corresponded to the interior of the vehicle air-handing system, the cold storage heat exchanger was installed in the test section. In the engine room, a compressor and condenser were installed, and each component was connected with plumbing. The expansion valve was installed at the end of the cold storage heat exchanger to control the amount of refrigerant flow against the heat load of the cabin.

The experiment was conducted with a cold storage air-conditioning system using a coldness release function after stopping the compressor with an optimized cold storage heat exchanger based on the results in the calorimeter. Table 3 shows detailed specifications of the air-conditioning system's compartments used in the thermal storage air-conditioning system performance.

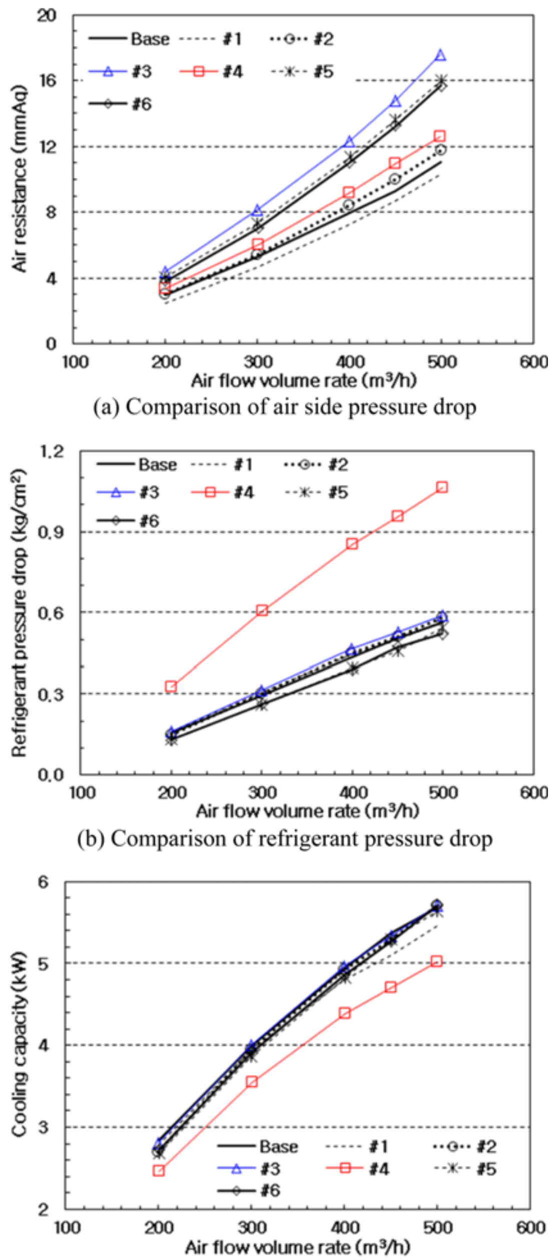
Table 4. Thermodynamic property of PCMs.

No	Melting point (°C)	Congealing point (°C)	Specific heat (kJ/kg·K)	Heat of fusion (kJ/kg)
A	6	2	1.8	214
B	9	6	1.8	174
C	9	7	1.9	190
D	9	8	2.0	210
E	10	9.9	2.0	207

Table 5. Experimental conditions of cold storage air-conditioning system.

Cabin room	
Evaporator inlet temperature	24 °C, 35 % RH
Air volume flow rate	450 m³/h
Engine room	
Condenser inlet air temperature	45 °C
Air volume flow rate	2,745 m³/h
Compressor rotational speed	2,580 rpm

In order to study the effects of PCM on the cold storage air-conditioning system, an experiment was conducted with various PCMs, which have different thermodynamic characteristics. Generally, PCMs must not cause corrosion of the heat exchanger because it is made of aluminum, does not have a change of thermal property during freezing and melting over a long period of time, and does not have significant volume change during the freezing and melting process. The PCM used was made of paraffin, an organic chemical in this experiment that influences coldness release function by inserting different PCMs with different



(a) Comparison of air side pressure drop

(b) Comparison of refrigerant pressure drop

(c) Comparison of cooling capacity

Figure 4. Experimental results of calorimeter with various cold storage heat exchangers.

thermal properties. Refer to Table 4 regarding the detailed thermodynamic properties of the PCMs.

Functional testing of the thermal storage air-conditioning system was performed with a cold storage heat exchanger. Experimental devices controlled and measured data for: revolution per minute, torque, work of compressor, suction and discharge refrigerant pressure and temperature at compressor manifold, supplying air flow volume, air temperature to condenser, condenser inlet and outlet pressure and temperature, supplying air flow volume, air temperature, and humidity to the evaporator.

The air supplied to the condenser was set to 45 °C, taking into consideration the hot weather conditions of summer. Air flow volume was set to 2,745 m^3/h, which was entered the air flow volume at the front bumper when a car drove at a speed of 50 km/h. At this driving speed, the compressor ran at 2,580 rpm. The air temperature supplied to the cabin was set to 24 °C, and the air flow volume was set to 450 m^3/h. Specific experimental conditions are presented in Table 5.

An experiment was conducted on the freezing and melting process of the PCM into the cold storage heat exchanger. The PCM was frozen by thermal conduction from the evaporator, which became cold when the air-conditioner was in operation. Then the iced, phase changed material released coolness into the cabin while it melted. A visualization experiment was conducted via transparent cold storage heat exchanger, which composed the end plate of the transparent plastic, rubber seal and bolt between the heat exchanger and end plate.

3. RESULTS AND DISCUSSION

3.1. Experimental Results for Calorimeter

Figure 4 shows the experimental results of the cold storage heat exchanger made as a prototype from the heat exchanger calorimeter. The cold storage heat exchanger consisted of one body with an evaporator, so it has to have equal cooling capacity as the current evaporator to satisfy the target cooling performance. The experiment was conducted with a various number of stacked heat exchanger plates, refrigerant circuit change of the inside heat exchanger, fin height, and density of the cold storage heat exchanger.

Figure 4 (a) shows the results of the air side pressure drop experiment. Cold storage heat exchanger #1 had a lower air resistance in all air flow volume range experiments against the Base. Cold storage heat exchanger #2 had a similar air side pressure drop to the Base, except that over air flow volume of 400 m^3/h was shown slightly higher than the Base. Meanwhile, cold storage heat exchangers #5 and #6 had a high pressure drop of air side in all ranges of air flow volume due to the 4 mm height of fin and the stack of 38 number of plates against the Base.

Figure 4 (b) shows the comparison results of the refrigerant pressure drop in the cold storage heat exchanger.

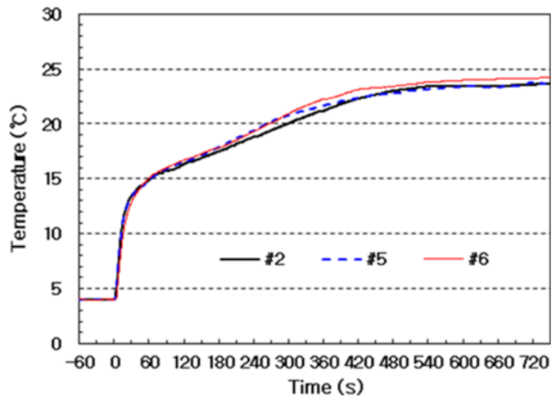


Figure 5. Experimental results of coldness release using various PCMs.

All experiment cold storage heat exchangers were relatively good except for cold storage heat exchan exchanger calorimeter experiment, cold storage heat excger #4. The cold storage heat exchangers #1, #2 and #3 showed a similar refrigerant pressure drop with the Base, and #5 and #6 had a lower refrigerant pressure drop compared to the Base.

Figure 4 (c) shows the results of a cooling capacity experiment. All cold storage heat exchangers except for #4 had a similar cooling capacity to the Base. However, the cooling capacity of cold storage heat exchanger #1 slightly dropped at a air flow volume of greater than 400 m^3/h.

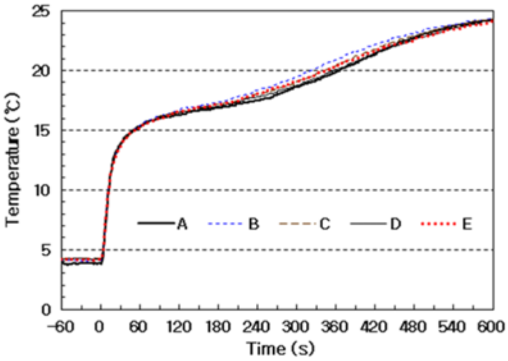
In the results from the cold storage heathanger #2, with 27 stacked plates, was rated as the most outstanding cold storage heat exchanger whereas cold storage heat exchanger #5 and #6, with 38 plates stack, were disadvantageous to air side pressure drop, but advantageous in refrigerant pressure drop, and coldness release performance because the internal volume of the heat exchanger linked-charge capacity of PCM increased by more than 40 % than cold storage heat exchanger #2.

3.2. Coldness Release of Cold Storage System Experiment

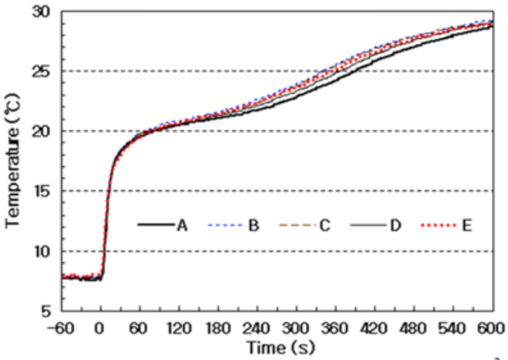
Coldness release performance was conducted in an air-conditioning system with candidates of cold storage heat exchangers.

Figure 5 shows the results of coldness release performance of cold storage heat exchangers #2, #5 and #6, which were relatively excellent performance in the calorimeter experiment.

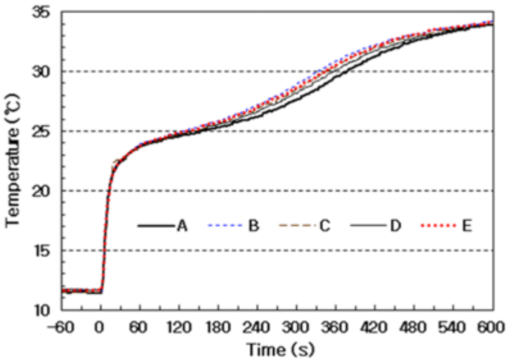
The study examined the temperature change of air discharge from the cold storage heat exchanger after the compressor stops, that PCM melting caused by delied the air discharge temperature increases. After the compressor stopped operating, discharging air temperature rapidly increased, and the cold storage heat exchanger #2 has a slightly lower discharging air temperature than #5 and #6 after 60 seconds. Cold storage heat exchanger #5 and #6 had more charge of PCM than #2, but each had a similar



(a) Discharge air temperature (T_{amb} = 24 °C, Q = 450 m^3/h)



(b) Discharge air temperature (T_{amb} = 29 °C, Q = 450 m^3/h)



(c) Discharge air temperature (T_{amb} = 34 °C, Q = 450 m^3/h)

Figure 6. Experimental results of coldness release performance with various PCMs and ambient temperature.

coldness release performance. This means that the cold storage heat exchanger does not have a significant influence of coldness release performance for 10 minutes after the compressor stops.

According to the results of Figures 4 and 5, it can be seen that cold storage heat exchanger #2 is the best heat exchanger in terms of performance and economy.

Figure 6 shows the result of experiments on coldness releasing performance by utilizing various kinds of PCM in cold storage heat exchanger #2, which was chosen as the optimal cold heat exchanger.

Figure 6 (a) shows the coldness releasing performance when the temperature of air supplied by a cold storage heat

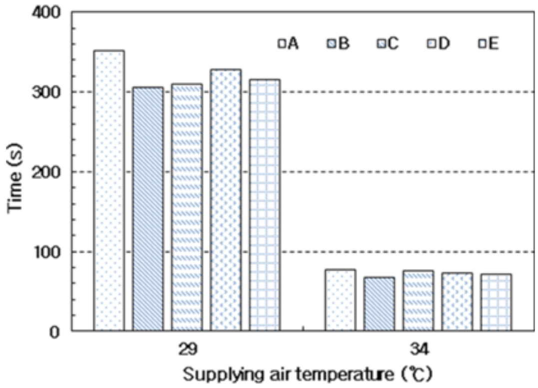


Figure 7. Comparison of 24 °C air discharge temperature under various ambient temperatures (Q = 450 m^3/h).

exchanger is 24 °C. The difference in cold releasing performance due to the characteristics of PCM charged in the cold storage heat exchangers was almost negligible, but it showed a slight difference after approximately 120 seconds.

Figure 6 (b) shows the results of a coldness releasing performance experiment when air at a temperature of around 29 °C was supplied through a cold storage heat exchanger; the general tendency as that, very similar to Figure 6 (a), there as no major difference in coldness releasing performance according to the PCM, and as the temperature of air supplied increased, the temperature of the air discharge increased under all test conditions.

Figure 6 (c) is the result of experimenting on coldness releasing performance at 34 °C, which mimics a tropical environment, is supplied by a cold storage heat exchanger. It has the same profile as Figure 6 (a) and Figure 6 (b), and as the outdoor temperature increased, the temperature of air, discharged from the cold storage heat exchanger, also increased.

Figure 7 compared the time difference for increase in temperature of air discharge after the compressor was stopped at 24 °C, which is generally accepted as a comfortable indoor temperature during summer season. The PCM is completely melted at 24 °C, discharge air temperature and at 29 °C, outdoor temperature. Otherwise at 34 °C outdoor temperature, the PCM melted faster than at an outdoor temperature of 29°C (Supplyign air temperature).

PCM A was the slowest to reach 24 °C air discharge temperature, followed by D, E, and C when outdoor temperature was 29 °C. Generally it took around 350 seconds to reach 24 °C of discharge air temperature after the compressor stopped, so it is possible for the PCM to completely dissolve while waiting for a traffic signal. However, at 34 °C outdoor temperature, the time to reach 24 °C discharge air temperature decreased. There was a slight difference of 68 ~ 78 seconds to reach 24 °C air discharge temperature.

Taking into considerdation the results of the experiment

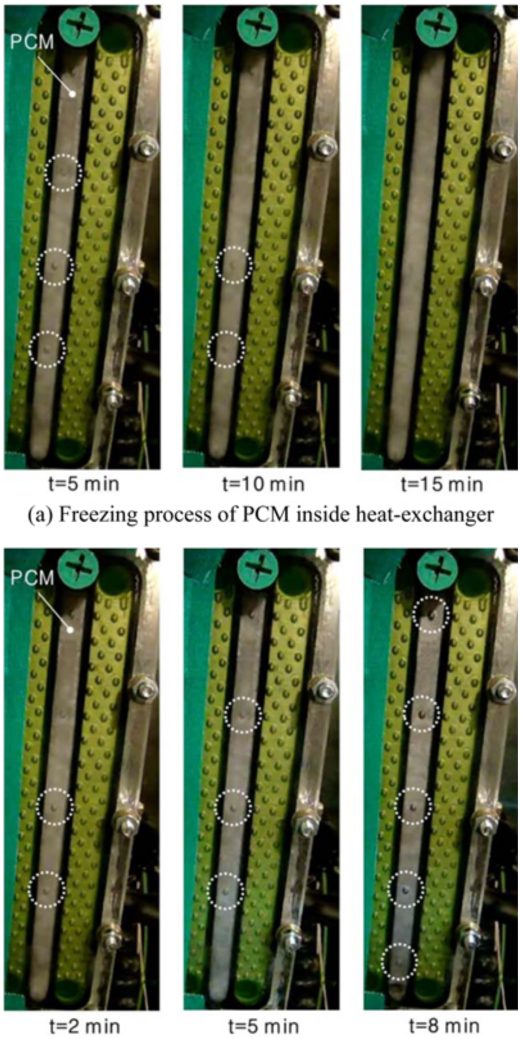


Figure 8. Visualization of freezing and melting process inside cold storage heat exchanger.

with various PCMs in cold storage heat exchanger #2, PCM A was chosen as the best cold storage medium. The thermal storage system was considered to be very efficient when indoor average temperature was below 24 °C in the car with an air-conditioner, due to the cold storage heat exchanger.

Figure 8 shows a visibility experiment on the freezing and melting process of the PCM, charged in the thermal storage unit located at the center of the cold storage heat exchanger. Thermal storage material had to solidify by storing ice for a short period of time. When releasing coldness, it was good to liquefy the whole area of the thermal storage unit in the heat exchanger.

Figure 8 (a) shows a ice storage process using a distinguishing dimple formed as a means to prevent a sub-cooling phenomenon of the PCM around the cold refrigerant inside the cold storage heat exchanger. PCM was completely frozen after approximately 10 minutes of operating the air-conditioning system and the solidification of the PCM could be seen.

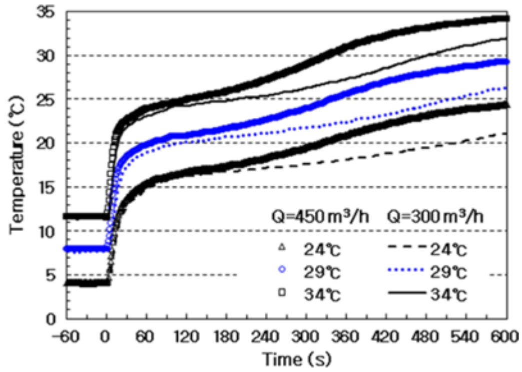


Figure 9. Coldness release performance with various air temperatures and air flow volume rates.

Figure 8 (b) showed the results of experiments on PCM's melting process as it released coldness after the compressor stopped. Liquefaction of PCM started after approximately 2 minutes. It released coldness after about 5 minutes, and completely turned into liquid after 8 minutes.

Figure 9 shows the effects on the air discharge temperature and air flow volume rate supply to the cold storage heat exchanger in the air-conditioning system after the compressor stops. As seen previously, if air that flows into the thermal storage heat exchanger, increases, discharge air temperature from the thermal storage heat exchanger shows to be relatively high, as well. Moreover, the air flow volume supplied to the cold storage heat exchanger was the most significant factor in the temperature of discharging air drops. If supplied air flow volume rate was 300 m^3/h, the temperature of discharge air was lower and the heating-up time was also delayed when compared with the 450 m^3/h air flow volume rate.

Figure 10 shows the results of experimenting with cooling capacity and temperature of discharging air from a air-conditioning system with a cold storage heat-exchanger

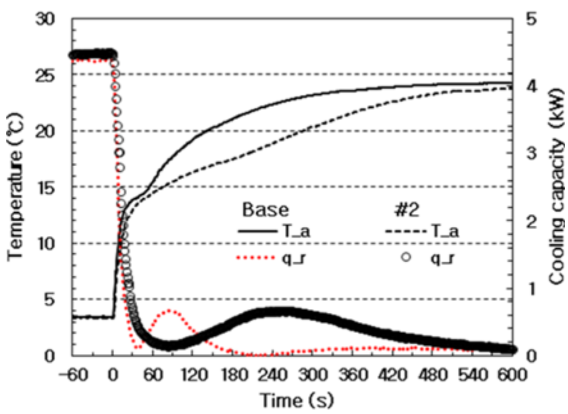


Figure 10. Comparison of coldness release capacity between Base and CSH (T_{amb} = 24 °C, Q = 450 m^3/h).

and Base heat exchanger after the compressor stops. For the Base, the temperature of discharging air increased drastically after the compressor stopped, and after hesitation at 45 seconds, it continued to rise quickly.

The cooling capacity of the evaporator was 4.2 kW, and it suddenly dropped neaboth Base and coldstorage heat exchanger #2 is arounf 4.2 kW, and it suddenly drops near to zero due to the compressor stops. After that, the cooling capacity of the Base is up to 650 W, as evaporation of the refrigerant remained inside the heat exchanger, then dropped again after rapid evaporation in the evaporator. In the case where a cold storage heat exchanger was used, though the temperature of discharging air increased after the compressor stopped, cooling capacity maintained for a long period of time due to latent heat of the PCM in the cold storage heat exchanger #2, and the discharging temperature slowly increased due to ice-stored in the PCM of the cold storage heat exchanger #2.

Temperature of discharge air increase delayed time was 360 seconds for the Base to reach 24 °C, and around 540 seconds for the thermal storage system which was adapted #2 cold storage heat exchanger and PCM A.

Meanwhile, it was found that the thermal storage system provided thermal comfort to the cabin because the Base had a 22.2 °C temperature of discharging air, and the cold storage system had a temperature of 17.4 °C, which was lower by about 4.8 °C since the 180 seconds after compressor stopped.

4. CONCLUSION

When operating in ISG mode in a vehicle, since the compressor of an air-conditioning system stops, thermal comfort in cabin worsens. As an alternative, a thermal storage system with an evaporator-assembled cold storage heat exchanger was studied and the following conclusions were obtained.

Cold storage heat exchanger performance was optimized by changing refrigerant circuit, fin height and fin density.

As a result of experimenting with various cold storage PCMs, the time for the temperature of discharging air to reach 24 °C from the supplying air temperature of 29 °C was around 310 ~ 360 seconds.

It was confirmed by visualization for the thermal storage and thawing of the thermal storage unit that PCM is ice-stored to solid over 10 minutes, and it can last more than around 2 minutes when releasing coldness.

Supplying air temperature and air flow volume rate has a significant influence on the thermal storage system, and it was found that coldness releasing performance dropped when temperature was higher and air flow volume rate was larger.

It took around 540 seconds for the temperature of discharing air to reach 24 °C for the air-conditioner with cold storage heat exchanger #2 and PCM A, and the temperature rise was lower by 4.8 °C at around 180 seconds in contrast to the current Base.

Therefore, the evaporator-assembled cold storage heat exchanger was optimized through the experiment; as a result of analyzing the influence on performance of air-conditioning systems using the applied heat exchanger, the thermal storage system is considered to be very useful in maintaining occupant comfort while also increasing vehicle fuel efficiency during idle stop time or while waiting at a traffic light signal.

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