Advanced Science and Technology Letters
Vol.120 (GST 2015), pp.617-627
http://dx.doi.org/10.14257/astl.2015.120.123

Experimental Study on Cooling Performance Characteristics of Coolant-sourced Heat Pump System with Triple Fluids Heat Exchanger for Fuel Cell Electric Vehicle

Hoseong Lee[1], Jongphil Won[1], Chungwon Cho[1], Taekkyu Lim[1], Hanbyeol Jeon[1], Soobyeong Nam[2], Donggyun Kim[2], Yongchan Kim[3]

[1] Thermal management R&D center, KATECH, 303 Pungse-ro,Cheonan-si, Chungnam, 31214, Korea
[2] HVAC System team, Hanon systems, 1689-1, Sinil-dong, Daejeon, 34325, Korea
[3] Department of Mechanical Engineering, Korea University, Seoul, 136-701, Korea

Abstract. The objective of this study was to investigate the cooling performance characteristics of coolant-sourced heat pump system with triple fluids heat exchanger for a fuel cell electric vehicle. In order to analyze the cooling performance characteristics of the air conditioning system using triple fluids heat exchanger for a fuel cell electric vehicle was developed and tested under various operating conditions according to inlet air conditions of evaporator, coolant conditions and compressor speed. The cooling capacity and coefficient of performance (COP) for cooling of the tested air conditioning system were 7.1 kW and 2.8 under high inlet air temperature with the rated compressor speed, respectively. In addition, developed triple fluids heat exchanger to transfer heat between a refrigerant and two kinds of coolants was analyzed under the same operating conditions. The observed cooling performance of the developed electrical air conditioning system was found to be sufficient for cooling loads under various real driving conditions for a fuel cell electric vehicle.

Keywords: coolant-sourced heat pump, triple fluids heat exchanger, cooling performance, COP, fuel-cell electric vehicle.

1 Introduction

Fossil fuel energy has been used for a long time to power vehicles with internal combustion engines. Internal combustion engines should no longer be used for public transportation and private vehicles because of the world energy crisis and global warming [1]. Because of increasing international usage regulations on fossil fuels and environmental concerns to mitigate global warming and glacier melting, many automotive companies have developed zero-emission vehicles as an alternative to the internal combustion engine (ICE). So, recently, many studies on the development of “green cars” which do not use fossil fuels are being conducted by company engineers

ISSN: 2287-1233 ASTL

and researchers for protection of the environment. Such vehicles have been developed by many automotive makers, although the classifications of green cars have not been officially defined internationally. In this study, green cars would be generally justified by their power source. Electric vehicles, fuel cell electric vehicles and hybrid electric vehicles could be widely classified to deal with environmental regulations [2-4]. Although electric-driven vehicles do not emit air pollutants, their limited driving range is a critical problem for commercialization and popularization. Recently developed electric vehicles generally have maximum driving ranges of 200 km, but this can be reduced by over 40% when the vehicle's heating and cooling systems are operated [5-8]. The efficient heating of the cabin of an electric vehicle is a very important factor in preventing a reduction in driving range. In electric-driven vehicles, heating devices are necessary to heat the cabin air because a high heat source like the engine of conventional vehicles does not exist. Generally, an electric heater of the PTC (positive temperature coefficient) type has been used in electric-driven vehicles. An electric heater system has the advantage of low cost because it is not necessary to modify the established design and add additional devices. However, it could draw heavily from the battery due to the operation conversion characteristics of the electric PTC heater. This can result in a dramatically reduced driving range when the heater is operated. Therefore, effective heating systems for the cabins of electric-driven vehicles are required, to minimize such reductions in range.

The heat pump system has been considered as an alternative to the electric PTC heater for increasing system efficiency [9–13]. Studies on the heat pump system for conventional vehicles were initially made. Antonijevic and Heckt reported that it was superior as an automotive heating unit to other heating solutions with respect to heating performance and fuel consumption [14]. Hosoz and Direk investigated the operating characteristics of an R-134a heat pump system using an air source [15]. The tested heat pump system provided sufficient heating performance in mild weather conditions but its heating capacity dropped rapidly with a decrease in outdoor temperature. The use of an additional heating device was required for the heating load [16].

Previous studies on heat pump systems have focused on conventional vehicles using the internal combustion engine. However, there are few studies on heat pump systems with triple fluids heat exchanger between a refrigerant and two kinds of coolants for fuel cell vehicles. Therefore, the automotive coolant-sourced heat pump system for heating and cooling of the cabin in electric-driven vehicles was considered, with variations in driving conditions, including inlet air temperature of the evaporator, coolant conditions and compressor speeds. The performance characteristics based on cooling capacity and the COP (coefficient of performance) of the heat pump system with developed triple fluids heat exchanger were analyzed.

2 Experimental Setup and Data

2.1 Test setup

Figure 1 shows the schematic diagram of the basic test setup to measure the performance of coolant-sourced heat pump system for the fuel cell electric vehicle using the triple fluids heat exchanger.

The test setup mainly consisted of an electric driven compressor, triple fluids heat exchanger, evaporator, an expansion device (electronic control type), and accumulator. The test setups for triple fluid heat exchanger and evaporator side were installed in a psychrometric calorimeter, which provided pre-controlled ambient temperature. The psychrometric calorimeter, equipped with an air-handling unit including a cooling coil, a heating coil, and a humidifier, was set to 27 °C~42 °C to an accuracy of ±0.2 °C. The psychrometric calorimeter was controlled by using the PID control method. Both an evaporator and a gas cooler with multi-flow type were installed in the psychrometric calorimeters for the purpose of controlling the air-side inlet conditions.

The electric compressor was variable speed type and the current used to drive the compressor was measured by the power meter (WT-210). Compressor work was calculated based on the power input and current. The power input was measured exactly to evaluate the electrical air conditioning system. During the experiments, the major operating parameters were monitored graphically and numerically in real time. In order to calculate and evaluate the performance of the electrical air conditioning system the temperature, pressure, and the mass flow rate were measured. Table 1 shows the specifications of the electrical air conditioning system using two kinds of coolants for a fuel cell electric vehicle. Table 2 shows the test conditions used in this study. During the experiments, the indoor air temperature was set to 27, 35, 42 °C with relative humidity of 50% and the air flow rate was set to 360 m3/hr.

Indoor air temperatures and air flow rates on the evaporator side of the passenger vehicle were varied from 27 °C to 42 °C and 360 m3/hr, respectively. The compressor speed was set variously at 3000 rev/min to 8000 rev/min. The displacement of the compressor was 33.0 cc/rev. The air flow rates at the HVAC module were obtained according to the input voltages of the blower; they were 360 m3/hr at the input voltage of 10.0 V. The above tests were performed to obtain the real air flow rates of the HVAC module for the real fuel cell electric vehicle. The HVAC module consisted of the evaporator assembly and air ventilation parts.

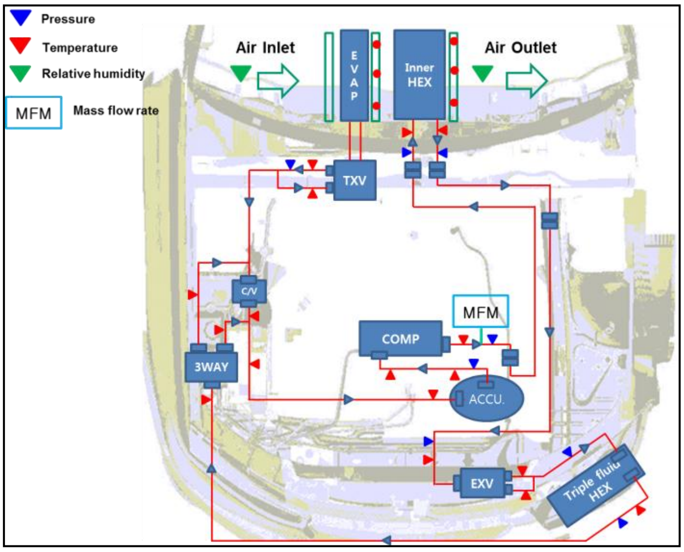


Fig. 1. Schematic diagram of the test setup.

Table 1. Components specifications of electrical air conditioning system using triple fluid heat exchanger.

Components	Specifications	
Inner Heat exchanger	Capacity (kW)	5.5 at 6 m³/min and 5,000 rpm
	Type, core size (mm^3)	Multi-flow type, W 222 x H 144 x D 54
Evaporator (Interior heat exchanger)	Capacity (kW)	5.0 at 6 m³/min and 5000 rpm
	Type, core size (mm^3)	Multi-flow type, W 273 x H 230 x D 45
Triple fluids heat exchanger	Capacity (kW)	6.5 at stack coolant 60 °C, electric device coolant 45°C and 5000 rpm
	Type, core size (mm^3)	Count flow type, W 190 x H 225 x D 80 Stack side : 35%, Electric device side : 65%
Compressor	Type	Electric driven compressor

	Displacement (cc/rev)	33
Expansion valve	Type	Electronic control (Pulse)
	Flow rate (kg/h)	50~250
Accumulator	Volume (cc)	950

Table 2. Test Conditions.

Components	Conditions
Compressor speed (RPM)	3000, 4000, 5000, 6000, 7000, 8000
$T_{\text{evaporator, in}}$ (°C)	27.0, 35.0, 42.0
$Q_{\text{evaporator, in}}$ (m^3/hr)	360
$m_{\text{stack coolant, in}}$ (liter/min)	10
$T_{\text{stack coolant, in}}$ (°C)	60.0
$m_{\text{electric device coolant, in}}$ (liter/min)	10
$T_{\text{electric device coolant, in}}$ (°C)	45.0

Table 3. Test equipment and uncertainty of the experimental parameters.

Items	Accuracy
Thermocouples (T-type)	±0.1 ℃
Pressure gage (Sensors, PI3H)	±0.1% (Max 250 bar)
Mass flow meter (Coriolis type)	±0.15%, Max 680 kg/h
Data logger (Gantner)	E. Gate IP (V3) (2.93W @ 12.06 V)
Cooling capacity	4.5%
Cooling COP	5.8%

Table 3 shows the uncertainties of the parameter measurements. Refrigerant and air temperatures were measured with thermocouples. The thermocouples were calibrated to an accuracy of ±0.1 °C. The refrigerant flow rate was measured by a Coriolis type flow meter with an uncertainty of ±0.15% and an upper limit of 680 kg/h. This flow meter was installed between the outlet tube of the internal heat exchanger and the inlet of the expansion valve to minimize the measurement errors. Pressure sensors, which can measure absolute pressure up to 30 bar with an uncertainty of ±0.1%, were installed at the inlet and outlet of each component. In order to verify the measured data of the cooling capacity and the cooling COP, an uncertainty analysis was performed in accordance with the 95% confidence level set by the standards of ANSI/ASME (1985) and Moffat [10,11]. The precision limits and bias limits of all the parameters associated with heating capacity and heating COP were estimated. The average uncertainties of the experimental data on cooling capacity and cooling COP were 4.5% and 5.8%, respectively.

2.2 Data reduction

Figure 2 shows the heat balance between the air side and the refrigerant side used in this study.The heat transfer rate of the refrigerant side was calculated by the refrigerant enthalpy method (ANSI/AMCA 210, 1985 and ASHRAE Standard 116, 1983) [12]. Equation (1) was used to calculate the refrigerant side heat transfer rate. The heat transfer rate of the air side was determined by utilizing both the air flow rate and enthalpy difference, which were calculated by Equation (2), which was used to calculate the air side heat transfer rate:

$$\dot{Q}_{ref} = \dot{m}_{ref} \Delta h_{ref} \tag{1}$$

$$\dot{Q}_{a} = \dot{m}_{a} C_{p,a} (T_{a,in} - T_{a,out}) \tag{2}$$

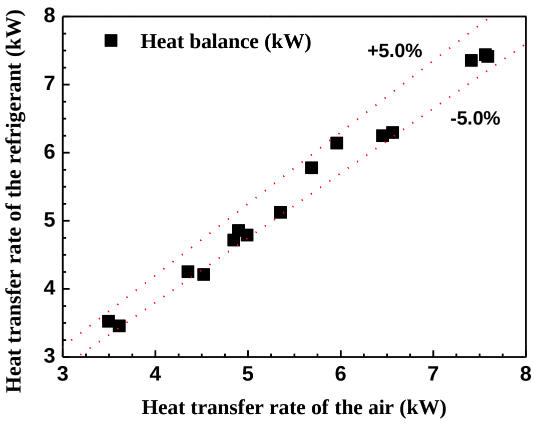


Fig. 2. Heat balance between the air side and the refrigerant side

The heat transfer rate of the air side was consistent with the heat transfer rate of the refrigerant side within ±5%, so the present experimental setup was found to be appropriate. The cooling COP (coefficient of performance) of the electrical air conditioning system was calculated by Equation (3).

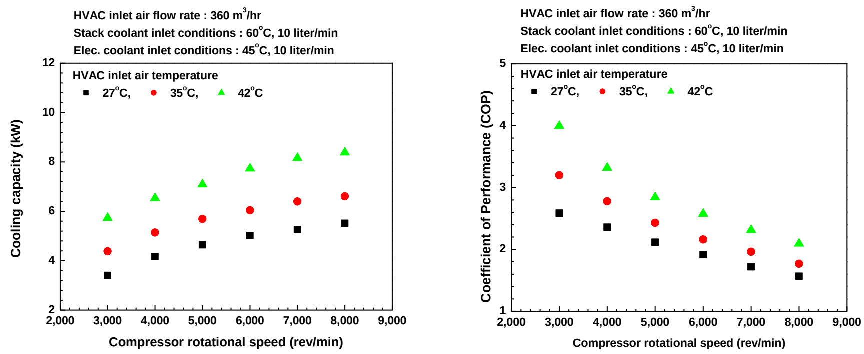
$$COP = \frac{\dot{Q}_a}{\dot{W}_{comp}} \tag{3}$$

3 Results and Discussion

Fig. 3 shows the COP and cooling capacity of coolant-sourced heat pump system with developed tripe fluids heat exchanger in this study. The COP and cooling capacity for the evaporator side were tested with the variation of the inlet air temperature of the evaporator and compressor speed. In this study, the COP and cooling capacity varied from 2.0 to 4.1 and from 6.0kW to 8.1kW at air inlet temperature of 42 oC with the variation of compressor speed, respectively. On the other hand, those varied from 1.5 to 2.7 and from 3.0 kW to 4.4kW at evaporator air inlet temperature of 27 oC along with the same compressor operating conditions.

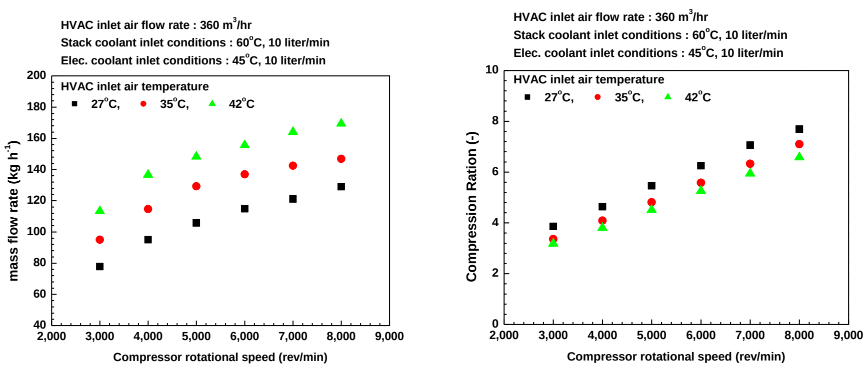
Fig. 4 shows the refrigerant mass flow rate and compression ratio. Refrigerant mass flow rate increased with the increase of the evaporator inlet temperature and compressor speed. However, compression ratio decreased with the increase of the evaporator inlet temperature. In the contrary, compression ratio increased with the compressor speed from 3.0 to 6.0 at air inlet temperature of 42 oC.

Fig. 5 shows pressure drop characteristics of the developed triple fluids heat exchanger with the variation of operating conditions, such as evaporator air inlet temperature and compressor speed. Triple fluids heat exchanger transferred heat between two kinds of coolants and the refrigerant. Stack coolant portion of triple fluids heat exchanger is about 35.0% and electric device coolant portion of it is about 65.0% due to normal operating temperature, over 60 oC and under 50 oC, respectively. With respect to pressure drop, stack coolant side was higher than electric device coolant side by 28.4%. And refrigerant side pressure drop had first order equation along to the refrigerant flow rate.



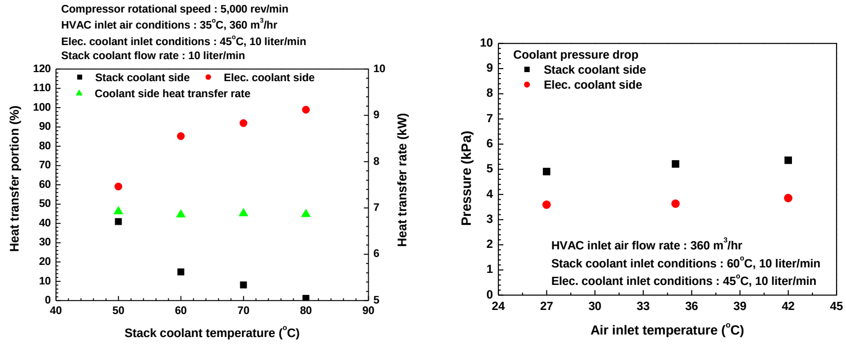
(a) Cooling capacity (b) Coefficient of Performance

Fig. 3. Effects of the inlet air temperature of evaporator on the COP and cooling capacity with the variation of compressor speeds.



(a) Refrigerant mass flow rate (b) Compression ratio

Fig. 4. Effects of the inlet air temperature of evaporator on the refrigerant mass flow rate and compression ratio with the variation of compressor speeds.



(a) Heat transfer rate portion among triple fluids (b) Pressure drop for coolants

Fig. 5. Performance characteristics of the triple fluids heat exchanger with the variation of operating conditions.

4 Conclusions

The cooling performance characteristics of coolant-sourced heat pump system with developed triple fluids heat exchanger were experimentally investigated by varying the inlet air temperatures for evaporator side and compressor speed. Experimental results showed that cooling capacity and coefficient of performance (COP) were sufficient to cover the cooling load of the automobile in the summer season.

Therefore, tested coolant-sourced heat pump system using the triple fluids heat exchanger has sufficient cooling performance to cope with cooling load under various actual driving conditions. In the future, more researches about triple fluids heat exchanger will be performed and analyzed, especially, coolant side conditions variation.

(1) COP and cooling capacity varied from 2.0 to 4.1 and from 6.0kW to 8.1kW at air inlet temperature of 42 oC with the variation of compressor speed, respectively. On the other hand, those varied from 1.5 to 2.7 and from 3.0 kW to 4.4kW at evaporator air inlet temperature of 27 oC along with the same compressor operating conditions.
(2) Refrigerant mass flow rate increased with the increase of the evaporator inlet temperature and compressor speed. However, compression ratio decreased with the increase of the evaporator inlet temperature. In the contrary, compression ratio increased with the compressor speed from 3.0 to 6.0 at air inlet temperature of 42 oC.
(3) With respect to triple fluid heat exchanger which has the role to transfer heat between two kinds of coolants and the refrigerant, stack coolant side was higher than electric device coolant side by 28.4%. And refrigerant side pressure drop had first order equation along to the refrigerant flow rate.

Acknowledgments. This work was supported by a grant "Development of 3.5kW heat pump system and components for the electric vehicle and high efficient operating technologies under low temperature (20142020104520)" from the Korea Institute of Energy Technology Evaluation and Planning.

References

1. Kim SC, Won JP, Park YS, Lim, TW, Kim, MS. Performance evaluation of a stack cooling system using CO2 air conditioning system in fuel cell vehicle. International Journal of Refrigeration 2009;32:70-77.
2. Takahisa S, Katsuya I. Air conditioning system for electric vehicle. Proceedings of SAE International Congress & Exposition 1996;960688.
3. Al-Alawi BX, Bradley TH. Total cost of ownership, payback, and consumer preference modeling of plug-in hybrid electric vehicles. Applied Energy 2013;103:488–506.
4. Brouwer AS, Kuramochi T, Broek M, Faaij A. Fulfilling the electricity demand of electric vehicles in the long term future: an evaluation of centralized and decentralized power supply systems. Applied Energy 2013;107:33–51.
5. Doucette RT, McCulloch MD. Modeling the prospects of plug-in hybrid electric vehicles to reduce CO2 emissions. Applied Energy 2011;88:2315–2323.
6. Lee MG, Park YK, Jung KK, Yoo JJ. Estimation of fuel consumption using in-vehicle parameters. International Journal of u- and e- Service, Science and Technology 2011;4:37-46.
7. Wang H, Liu Y, Fu H, Li G. Estimation of state of charge of batteries for electric vehicles. International Journal of Control and Automation 2013;6:185-194.
8. Kim DW, Lee MY. Theoretical approach on the heating and cooling system design for an effective operation of Li-ion batteries for electric vehicles. Journal of the Korea Academia-Industrial Cooperation Society 2014;15:2545-2552.
9. Jang JY, Bae HH, Lee SJ, Ha MY. Continuous heating of an air-source heat pump during defrosting and improvement of energy efficiency. Applied Energy 2013;110:9–16.
10. Aikins KA, Lee SH, Choi JM. Technology review of two-stage vapor compression heat pump system. International Journal of Air-Conditioning and Refrigeration 2013;21:1330002.
11. Byrne P, Miriel J, Lent Y. Experimental study of an air-source heat pump forsimultaneous heating and cooling – Part 1: Basic concepts and performanceverification. Applied Energy 2011A;88:1841–1847.
12. Byrne P, Miriel J, Lent Y. Experimental study of an air-source heat pump for simultaneous heating and cooling – Part 2: Dynamic behavior and two-phase thermosiphon defrosting technique. Applied Energy 2011b;88:3072–3078.
13. Lee HS, Cho CW, Won JP, Lee MY. Study on cooling performance characteristics of air conditioning system using R744 for a passenger vehicle. Journal of the Korea Academia-Industrial Cooperation Society 2011;1(12):1-7.
14. Antonijevic D, Heckt R. Heat pump supplemental heating system for motor vehicles. Journal of Automobile Engineering 2004;218(10):1111–1115.
15. Hosoz M, Direk M. Performance evaluation of an integrated automotive air conditioning and heat pump system. Energy Convers Manage 2006;4(47):545–549.
16. ASHRAE Standard 116. Methods of testing for seasonal efficiency of unitary air-conditioners and heat pumps. American Society of Heating, Refrigerating and Air-Conditioning Engineers Atlanta (GA); 1983.

17. Kim SC, Won JP, Kim MS. Effects of operating parameters on the performance of a CO2 air conditioning system for vehicles. Applied Thermal Engineering 2009;29:2408-2416.