Experimental investigation on an integrated thermal management system with heat pipe heat exchanger for electric vehicle

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Abstract:

An integrated thermal management system combining a heat pipe battery cooling/preheating system with the heat pump air conditioning system is presented to fulfill the comprehensive energy utilization for electric vehicles. A test bench with battery heat pipe heat exchanger (HPHE) and heat pump air conditioning for a regular five-chair electric car is set up to research the performance of this integrated system under different working conditions. The investigation results show that as the system is designed to meet the basic cabinet cooling demand, the additional parallel branch of battery chiller is a good way to solve the battery group cooling problem, which can supply about 20% additional cooling capacity without input power increase. Its coefficient of performance(COP) for cabinet heating is around 1.34 at -20℃ out-car temperature and 20℃ in-car temperature. The specific heat of the battery group is tested about 1.24 kJ/kg·℃. There exists a necessary temperature condition for the HPHE to start action. The heat pipe heat transfer performance is around 0.87 W/℃

on cooling mode and 1.11 W/℃ on preheating mode. The gravity role makes the heat transfer performance of the heat pipe on preheating mode better than that on cooling mode.

Key words: Electric vehicles, Heat pump, Heat pipe, Battery temperature control, Thermal management

Nomenclature:

A_{hp} contact superficial area with coolant of each heat pipe (m^2)

c_b specific heat of battery group(kJ/kg℃)

c_c specific heat of coolant (kJ/kg℃)

m_b mass of battery group(kg)

m_c mass of coolant(kg)

n heat pipe number

Q_{bi} batteries internal heat variation (kW)

Q_c cooling capacity by battery chiller (kW)

Q_{ci} coolant internal heat variation (kW)

Q_g generated heat by batteries (kW)

Q_p preheating heat by PTC (kW)

Q_t transferred heat by HPHE (kW)

q_{hp} heat transfer coefficient of each heat pipe (W/℃)

T_{ba} average temperature of battery group (℃)

T_{bo} coolant outlet temperature (℃)

T_{bi} coolant inlet temperature (℃)

T_{ca} coolant average temperature (℃)

ΔT average temperature difference between the battery group and the coolant (℃)

t time (s)

1. Introduction

Electric vehicle (EV) is an important development orientation to alleviate the traditional automobile exhaust problem. However, thermal management including battery temperature control and cabinet air conditioning is a big challenge for EV, as the traditional engine and oil tank are replaced by electric motor and battery groups.

Lots of heat inside of the battery generated by the electrochemical reaction will raise the battery temperature up sharply, affect its working efficiency badly and even cause safety problem [1, 2]. Sato[3] analyzed the thermal behavior of lithium-ion batteries showing that when the battery temperature was over 50℃, charging efficiency and life cycle would be considerably diminished. Khateeb et al. [4] pointed out that the safety of the Li-ion battery would descend when it operated at the temperature range of 70-100℃. Studies have shown that there is a necessary temperature range for battery to make sure its performance and service life. Pesaran [5] presented that the best range of operating temperature for batteries such as lead-acid, NiMH, and Li-ion are from 25 to 40℃ and suitable temperature distribution from module to module is below 50℃. To control the batteries in the suitable temperature range, there are several methods presented [6-12], such as by air directly, by liquid with plate heat exchanger or by refrigerant phase change with plate or pipe heat exchanger. However, investigations on the thermal behavior of batteries [5,13-14] show that the relationship between the generated heat and discharge rate is nonlinear direct ratio and the higher the discharge rate is, the quicker the increase rate of the generated heat will be. While the discharge rate changes with the working conditions such as acceleration, deceleration, uphill, and downhill. So generated heat of the battery is variable and its instantaneous value is very high. This means the cooling capacity of the battery temperature control system with these normal methods has to be set high enough to avoid the battery on extremely high temperature and lead to an over-size thermal management system. Therefore it is very significant to search for a more efficient battery heat-transfer method to simplify the EV thermal management system.

Heat pipe, as a high efficient heat-transfer device combining the principles of both thermal conductivity and phase transition, is a novel idea to apply on the temperature control of EV battery [15]. Actually, because of its highly effective thermal conductivity, heat pipe has been applied successfully in many fields such as electron cooling, solar heater and energy recovering [16, 17]. As for the above mentioned EV battery thermal characteristics, heat pipes between the batteries can help transfer the heat out to the coolant so that the batteries can be maintained in the best operating temperature range under variable working conditions and the temperature difference between batteries can be eliminated [18]. Moreover, because the coolant system has enough thermal capacity, the cooling load can be much lower than that of the instant cooling method. It just needs to meet the average heat dissipation demand instead of the peak generated heat during high discharge rate conditions. Therefore, heat pipe is a promising development orientation for batteries thermal management of EV. Authors' initial investigations have shown that the heat pipe cooling is an effective method [19]. However, the previous study results were mainly concentrating on the basic thermal performance of a single heat pipe unit with a simple experimental apparatus. The thermal performance of the heat pipe heat exchanger (HPHE) for the practical EV battery group still has not been researched, which might be different from that of the single heat pipe because of cluster effect.

On the other hand, since EV has no engine to drive compressor for cooling and no waste engine heat for heating, heat pump system with motor-driven compressor is an important development trend. The investigation on the performance of heat pump system has become a major topic of EV air-conditioning. Suzuki and Katsuya [20] proposed a heat pump system for electric vehicle with functions of cooling, heating, demisting and dehumidifying and their experimental results showed the feasibility of heat pump. However, heat pump system has a shortcoming that its heating capacity drops sharply with the decreasing outdoor temperature. Hosoz and Direk [21, 22] indicated this feature by investigating the performance of R134a heat pump system transformed for the original automobile air conditioning system. In recent years many advances on heat pump system for EV have been presented [23]. Authors [24, 25] have also engaged in the heat pump performance improvement with injection technology and got notable achievement in system heating capacity and COP under extremely cold condition. However when it comes to practical performance of the heat pump system combining with battery temperature control system, there is few literature either.

In this paper, an integrated thermal management system combining a HPHE for battery cooling/preheating with a heat pump air conditioning is presented to investigate its performance characteristics on different working conditions. Cooling and heating performances of the system, as well as the thermal performance of HPHE, are investigated by bench test, hoping to present a significative reference for the EV thermal management.

2. System description and experimental bench setup

2.1 System description

Fig.1 shows the diagram of the heat pump coupling with battery cooling /preheating system based on regular five-chair electric cars and takes R134a as refrigerant. Its working temperature ranges from -20℃to 45℃. The heat pump system mainly consists of a variable-frequency scroll compressor, an outside heat exchanger with a fan, a liquid vapour separator, four refrigerant valves (RV), a condenser followed by an expansion valve (EXV1) for cabinet heating, an refrigerant-air evaporator following with EXV2 for cabinet cooling and a refrigerant-water evaporator for battery cooling called battery chiller. The refrigerant-air evaporator and condenser are installed in the ventilation duct. The system is switched by the RVs for cooling or heating. The battery cooling/preheating system also applies a water-air heat exchanger in front of the car to utilize the natural cooling source and a Positive Temperature Coefficient (PTC) heater to preheat the battery in cold season. A HPHE is installed among the battery group, called battery heat exchanger box here. Please refer the reference [20] for more details of the HPHE.

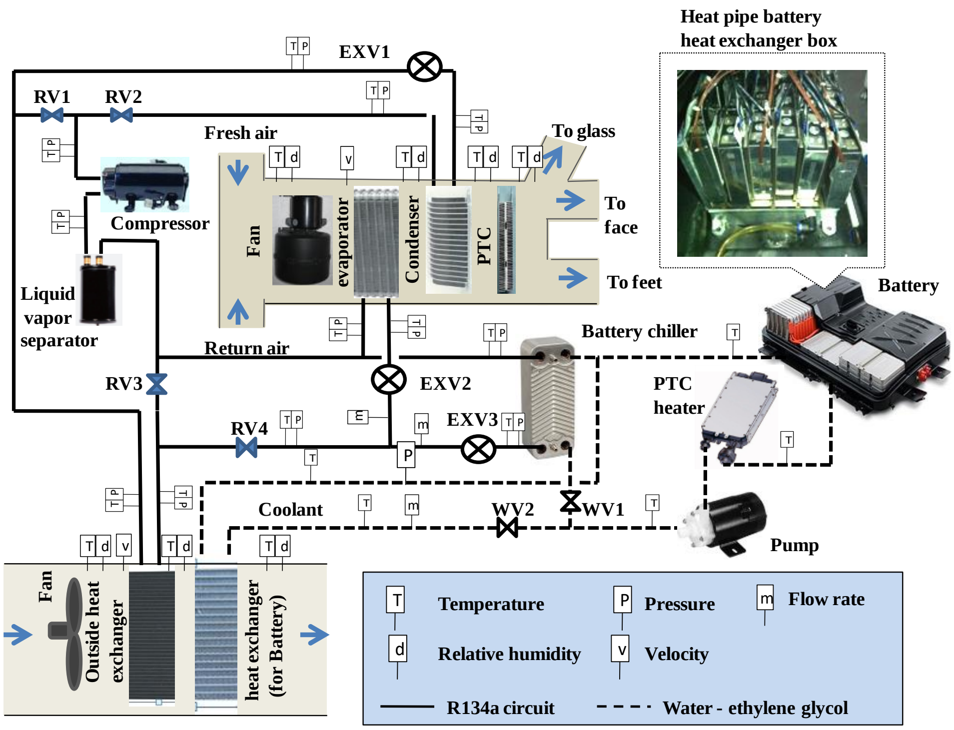


Fig.1.Diagram of the heat pump coupling with the battery cooling/preheating system

2.2 Experimental bench

Correspondingly a test bench is set up inside of a psychrometer testing room to investigate the performance of this system. The experiments are carried out on cooling and heating mode respectively under different working conditions. On cooling mode, the refrigerant valve RV1 and RV4 are open while RV2 and RV3 are closed. The opening of expansion valves EXV2 and EXV3 are changed repeatedly to get the optimum branch refrigerant flow rate of the cabin evaporator and battery chiller. On heating mode, the refrigerant valve RV1 and RV4 are closed while RV2 and RV3 are open. The battery is preheated by the PTC heater. The coolant pipe system and battery heat exchanger box are isolated to prevent unmeasured heat loss. There are 30 real battery modules in the bench for electric cars, but the generated heat during discharging process is simulated by electric films for the sake of safety. The electric films are pasted on the two wide sides of each battery module and thermocouples are pasted on the other two narrow sides to measure the temperature response of the batteries. Each side has three thermocouples. The measurement devices of the bench are shown in Fig.1 and their parameters are shown in Table 1. The relative parameters of the bench are shown in Table 2.

Table 1 Measurement devices

Parameter	Type	Range	Error
Temperature	Thermocouple	-30 to 220℃	±0.5℃
Pressure	Diaphragm	0 to 30bar	±0.5%
Air speed	Hot bulb	0 to 40m/s	±3%
Mass flow rate of Ref.	Coriolis	<370kg/h	±0.1%

Table 2 Parameter of the bench

Item	Symbol	Value(unit)

Mass of battery group	m_b	16.03 kg
Mass of coolant	m_c	3.96 kg
Specific heat of ethylene glycol coolant	c_c	3.334 kJ/kg℃@-20℃ 3.518 kJ/kg℃@35℃ 3.552 kJ/kg℃@45℃
Contact superficial area with coolant of each heat pipe	A_{hp}	0.00188 m^2
Heat pipe number	n	25

2.3 Calculation methodology

To investigate the heat transfer performance of HPHE of battery group, the experiment is carried out to simulate different working modes. The heat composition of the battery temperature control system is shown as Fig.2. The battery internal heat variation and coolant internal heat variation can be expressed as equation (1) and (2) respectively. Here heat transfer coefficient of each heat pipe q_{hp} is applied to indicate the heat transfer performance of the HPHE, which can be expressed as equation(3).

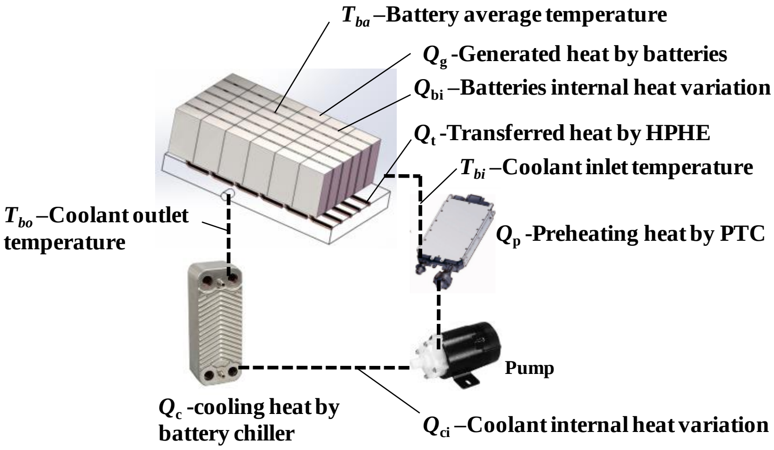


Fig.2 Heat composition of the battery group temperature control system

$$Q_{bi} = c_b m_b \frac{dT_{ba}}{dt} = Q_g + Q_t \qquad (1)$$

$$Q_{ci} = c_c m_c \frac{dT_{ca}}{dt} = Q_t + Q_p + Q_c \tag{2}$$

$$q_{hp} = \frac{Q_t}{n\Delta T} \tag{3}$$

Because the system heat-transfer process goes from dynamic to steady state gradually and the energy during the initial dynamic process is not balanced, it is important to note that this model is only suitable for the final steady state.

3. Experimental result and discussion

3.1 Heat pump system performance

Table 3 shows the system cooling and heating performance under different working conditions.

Table 3 Cooling/heating performance of the heat pump system

Experiment No.	1	2	3	4	5	6
Out-car temperature (℃)	35	35	45	45	-20	-20
In-car temperature (℃)	27	27	45	45	-20	20
EXV1 opening (%)	0	0	0	0	100	100
EXV2 opening (%)	84	84	63	63	0	0
EXV3 opening (%)	0	40	0	90	0	0
Evaporator evaporating temperature (℃)	-1.49	-0.48	9.96	11.7	-23.13	-21.58
Super-heating temperature (℃)	0.93	0.76	1.28	2.59	0.01	1.37
Battery chiller evaporating temperature (℃)	--	-4.30	--	7.49	--	--
Super-heating temperature (℃)	--	18.35	--	0.36	--	--
Condensing temperature (℃)	41.18	41.58	55.4	54.31	20.78	58
Sub-cooling temperature (℃)	0.3	0	7.36	1.98	21.62	9.45
Cabinet refrigerant flow rate (kg/h)	135.16	134.0	192.01	180.4	43.2	47.56
Cabinet cooling/heating capacity (kW)	5.24	5.19	7.22	6.61	2.96	2.75
Battery chiller refrigerant flow rate (kg/h)	--	25.36	--	63.96	--	--

Battery chiller cooling capacity (kW)	--	1.09	--	2.31	--	--
Theoretical compression power (kW)	1.04	1.24	1.43	1.75	0.45	0.86
Actual input power (kW)	2.44	2.46	3.13	3.19	1.48	2.04
Compression efficiency (%)	42.51	50.34	45.53	55.0	30.27	42.05
COP	2.15	2.55	2.31	2.80	2.0	1.34

On cooling mode under out-car 35℃ and in-car 27℃ condition，the opening of EXV1 is kept on 84% while that of EXV3 is changed from 0 to 40%. The evaporator cooling capacity deceases lightly and the battery chiller cooling capacity increases quickly. The total cooling capacity increases about 19.84% and the compressor input power almost keeps the same, so the system COP increases about 18.60%. Under out-car 45℃ and in-car 45℃ condition, the optimum opening of EXV2 is 63% and that of EXV3 is 90%. And also the system total cooling capacity and COP under this conditions are both increases. Compare to the out-car 35℃ and in-car 27℃ condition, the cooling capacity for both cabin and battery as well as system COP are much higher because of the lower compression ratio under this condition. The experimental results of cooling performance show that the additional parallel branch of battery chiller is a good way to solve the battery cooling problem, which can supply about 20% additional cooling capacity without input power increase.

On heating mode under out-car -20℃ and in-car -20℃ condition，system condensing temperature is about 20℃. As the in-car temperature increases to 20℃, system condensing temperature increases to 58℃, heating capacity decreases from

2.96 kW to 2.75kW, the compressor input power increases from 1.48 kW to 2.04 kW, and the heating COP decreases from 2.0 to 1.34. The experimental results show that although the heating COP under -20℃ in-car temperature is higher than that under 20℃ because of the lower compression ratio, the compression efficiency of the scroll compressor is much lower. This is because the motor efficiency of scroll compressor drops rapidly under lower load conditions. The heating capacity is insufficient for the cabinet heating. Therefore PTC heater is suggested to be an auxiliary heat source under extremely cold weather.

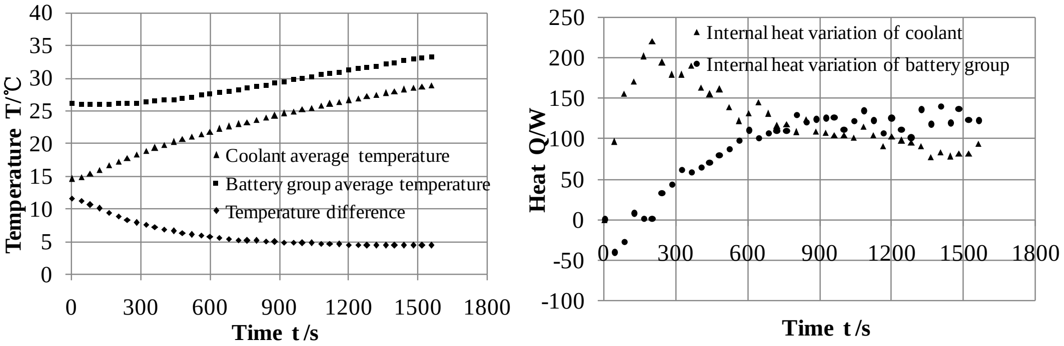
3.2 Heat pipe heat exchanger performance

3.2.1 Testing mode without heating or cooling

Because the battery group is a composition of electrolyte, metal and heat pipe etc., and the thermal capacity characteristics of different material are different, the specific heat of battery group is uncertain. A testing experiment is carried out firstly to test the specific heat of battery group. On this mode, the coolant is circulated by the pump with neither PTC heating nor battery chiller cooling and the generated heat of battery group is dissipated to the coolant by the HPHE. The equation (1) and (2) can be transferred into equation (4).

$$c_b m_b \frac{dT_{ba}}{dt} = Q_g - c_c m_c \frac{dT_{ca}}{dt} \tag{4}$$

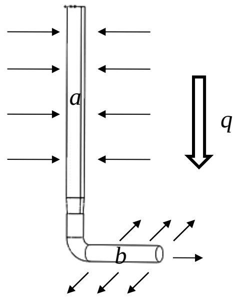
Fig.3(a) shows the average temperature response tendency of the battery group and the coolant from start to steady state. The temperature difference between them decreases from 11.6℃ to a constant 4.4℃. Fig.3(b) shows the internal heat variation response tendency of the battery group and the coolant. As mentioned in the above calculation methodology, the model applied for the steady state is quasi-steady and it is not suitable for the dynamic process. So the initial calculated heat shown in Fig.3(b), which goes up to more than 200W, does not present the actual heat value. The average internal heat variation of the battery group and the coolant at the steady state is around 119.4W and 83.6W respectively. Therefore according to equation (4), the specific heat of battery group can be gained, which about 1.24 kJ/kg℃. This result has also been verified by the following experiment based on the equation (1) and (2).



(a) Temperature response tendency (b) Internal heat variation response tendency

Fig.3 Experimental response tendency

Heat transfer way of the heat pipe under this condition is shown as Fig 4.The top part *a* of the heat pipe absorbs heat from battery and the fluid inside takes the heat to the bottom part *b* to dissipate to the coolant. The fluid inside of the heat pipe evaporates at the top part, goes down to the bottom part by pressure, and goes up by capillary action after condensation. The heat transfer performance of each heat pipe (Fig.5) can be obtained by equation (3) and it shows that as the experiment goes to steady state the heat transfer performance of each heat pipe gets to a relative stable value, which is around 0.86W/℃.



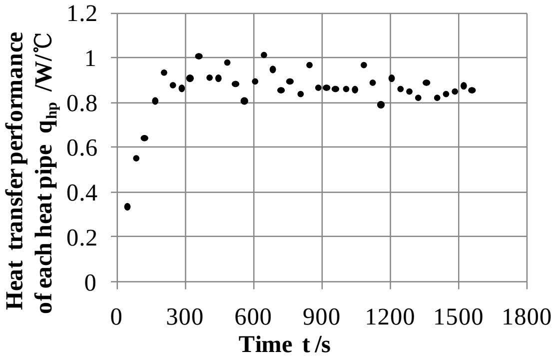
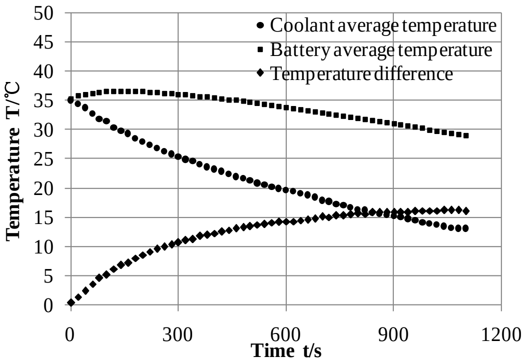


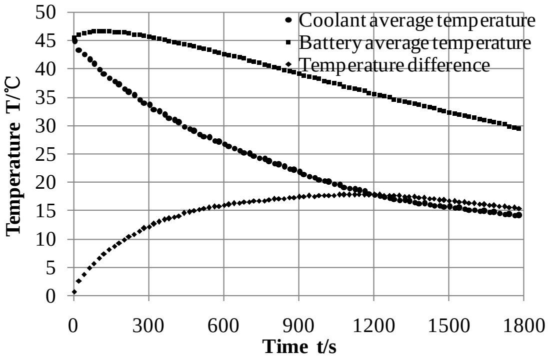
Fig.4 Heat transfer way of the heat pipe Fig.5 Heat transfer performance of each heat pipe

3.2.2 Cooling mode

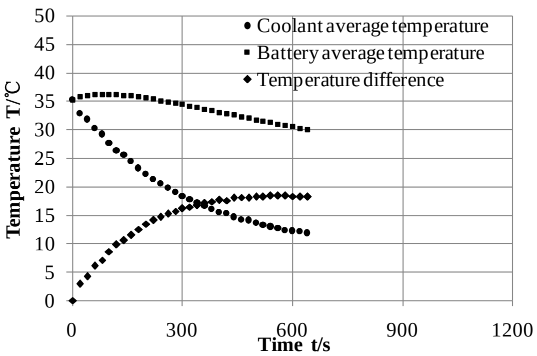
On this mode the coolant is circulated by the pump with battery chiller cooling and the battery group generates heat. The cooling capacity of the battery chiller is adjusted by changing the opening degree of EXV3.

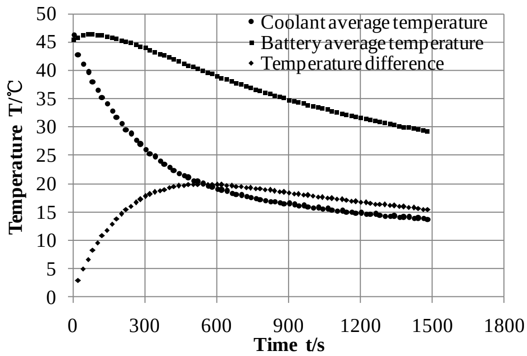
Fig.6 shows the temperature response of the battery cooling process. At the beginning, the battery group and the coolant are on the same temperature condition. As the cooling mode starts, the coolant average temperature inside of the battery exchanger box decreases quickly. While the battery average temperature increases to a higher value firstly and then begins to decrease with the coolant. It is worthy of mention that as the battery average temperature increases to the highest point, the temperature differences between the coolant and the battery group are all in the range of 7~8℃under every experimental condition. This result shows that at the beginning of this experimental mode, the battery begins to generate heat as it supplies power while the HPHE does not start until the temperature difference goes up to 7~8℃. Therefore the battery group temperature will go up before the HPHE starts to transfer heat from the battery to the coolant. And finally the battery temperature decreases to its suitable range. The battery cooling rate depends on the EXV3 opening degree. Under the conditions of 33% EXV3 opening degree, it takes about 1040s to be cooled from 35℃ to 30℃, 1720 s from 45℃ to 30℃. Under the conditions of 60% EXV3 opening, it takes 600s from 35℃ to 30℃, 1180s from 45℃ to 30℃.



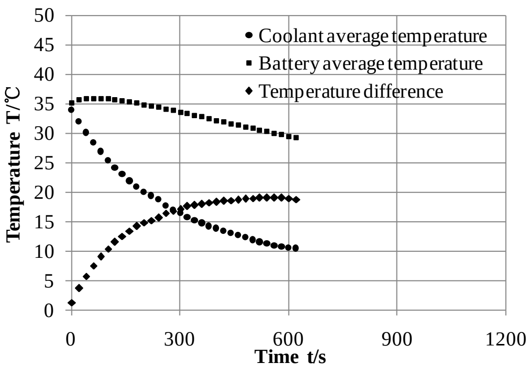


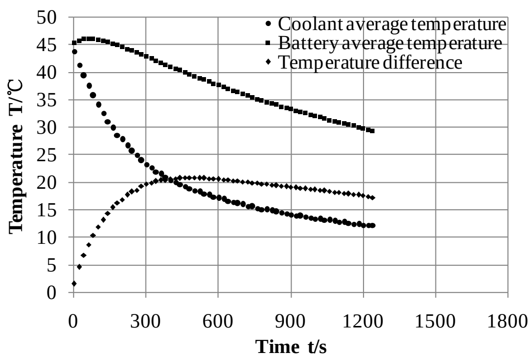
(a)35℃/33% (b)45℃/33%





(c)35℃/60% (d)45℃/60%





(e)35℃/100% (f)45℃/100%

Fig.6 Temperature response tendency on cooling mode

Fig.7 shows the heat transfer performance of each heat pipe according to equations (1)~(3). Heat transfer way of the heat pipe under this condition is the same as fig 3.The results show that as the system runs to steady state the values of the heat pipe heat transfer performance on different working conditions go to be coincident, which is around 0.87 W/℃. This result is also fitting very well with the above experimental result. According to this result, the total heat transfer performance of the HPHE is easily to be estimated, so that the temperature difference between coolant and battery group can be determined accordingly in the real application.

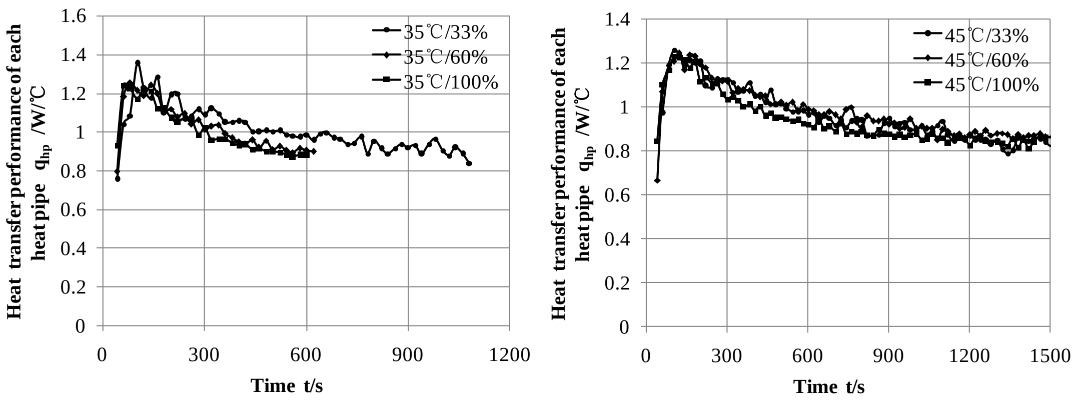
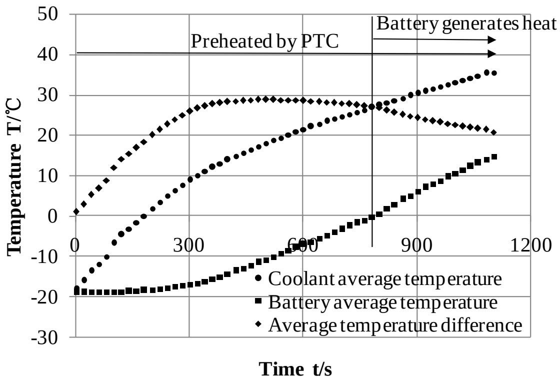


Fig.7 Heat transfer performance of each heat pipe on cooling mode

3.2.3 Preheating mode

On this mode the coolant is circulated by the pump with PTC heating and the battery group begins to generate heat when its temperature gets to be higher than 0°C.

Fig.8 shows the temperature response of the battery preheating process under -20°C out-car temperature. The coolant average temperature increases quickly as the PTC heater is on. But the battery response temperature does not change at the first stage of 200s until the coolant temperature goes up to be 2°C. After then it begins to increase gradually with the increasing of the coolant temperature. This means the heat pipe also has a start condition on heating mode. The start temperature of its bottom evaporating terminal is about 2℃ and the temperature difference between the evaporating terminal and condensing terminal under this condition is about 22℃.The battery begins to generate heat as its temperature gets higher than 0℃, so the increase rate of the battery temperature is getting higher after 0℃.



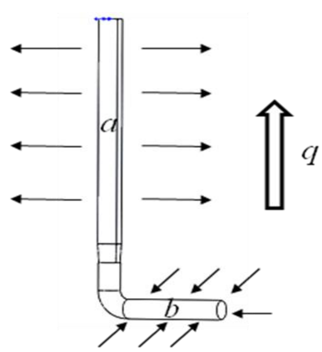


Fig.8 Temperature response on preheating mode Fig.9 Heat transfer way of the heat pipe

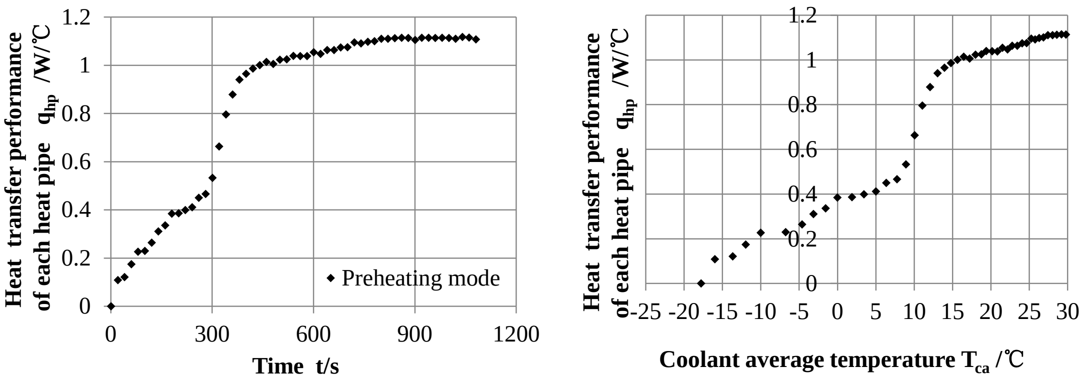


Fig.10 Heat transfer performance on preheating mode

Fig 9 shows the heat transfer way of the heat pipe under preheating mode. The bottom part *b* of the heat pipe absorbs heat from coolant and the top part *a* preheats the battery. The fluid inside of the heat pipe evaporates at the bottom part, goes up to the top part by the central route of the pipe with the heat and gets down by the inside wall of the pipe after condensation. Fig.10 shows the heat transfer performance of each heat pipe on preheating mode. At first the heat transfer performance increases gradually at a relative low level with the increasing of the temperature difference. As the coolant temperature goes up to be higher than 8℃, the heat transfer performance jumps up quickly to a higher value, and then increases slowly to a relative stable value, which is about 1.11W/℃. This result verifies that the coolant temperature has important effect on the heat transfer performance of the HPHE because the evaporating and condensing process of the fluid inside of the heat pipe depends on the temperatures of its two terminals. Meanwhile, compare to the heat transfer performance of the heat pipe on cooling mode, that on heating mode is higher. This is because on heating mode the heat pipes can take advantage of the gravity role to get better heat transfer performance. The HPHE designed for this experimental bench can be on a good heat transfer performance when the coolant average temperature is higher than 15℃.

4. Conclusion

According to the above experimental research on an integrated thermal management system for EV, the cooling and heating performance of heat pump system and heat transfer performance of HPHE are investigated. The research results show that the presented system works well as an effective thermal management method for EV. The main conclusions go as following:

(1) The system cooling performance shows that the additional parallel branch of battery chiller is a good way to solve the battery group cooling problem, which can supply about 20% additional cooling capacity without input power increase. The cooling capacity distribution of each branch under different working conditions can be optimized by adjusting the expansion valve.

(2) The system heating performance under extremely cold condition shows that although the heating COP under -20℃ in-car temperature is higher than that under 20℃, the compression efficiency of the scroll compressor is much lower because the motor efficiency of scroll compressor drops rapidly under lower load conditions. So improving the heating performance under high temperature difference condition is still an important future work for EV.

(3) The specific heat of the battery group is tested about 1.24 kJ/kg℃. On cooling mode, there is a delay for the HPHE to start heat transfer and the temperature difference for the HPHE to start between its two terminals is about 7~8℃. The heat pipe heat transfer performance on different cooling working conditions is around 0.87 W/℃.

(4) On preheating mode, the HPHE also has a necessary start condition. The start temperature of the bottom evaporating terminal is about 2°C and the temperature difference between the two terminals is about 22°C. The coolant temperature has important effect on the heat transfer performance of HPHE. As the coolant temperature goes up to be higher than 8°C, the heat transfer performance jumps up

quickly to a higher value, and then increases slowly to a relative stable value, which is about 1.11W/℃. The heat transfer performance of the heat pipe on preheating mode, is higher than that on heating mode because it can take advantage of the gravity role.

(5) The research results show that the heat transfer performance of HPHE can meet the demand of battery temperature control on different working conditions. According to the heat transfer performance of the HPHE and specific heat of the battery group, the design parameters of the coolant system can be determined based on the calculation methodology in the real applications.

5. Acknowledgements

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