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Performance evaluation of propane heat pump system for electric vehicle in cold climate

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ABSTRACT

To solve the problem of driving mileage reduction for electric vehicles during winter resulting from high power consumption for heating and to deal with the global warming issues, propane heat pump is proposed as a novel solution in this paper for the first time. The heating performance characteristics of a mobile propane heat pump system for electric vehicles in cold climate under various operation conditions have been experimentally studied. Different technical solutions for heat pump are also compared in the research. According to the results, propane (R290) system is the strongest competitor at an outdoor ambient temperature above −10 °C among all the strategies while CO_2 system may have small advantages when the ambient temperature is −20 °C. However, due to CO_2's poor cooling COP (Coefficient of performance) at high ambient temperature and its high operating pressure, propane heat pump system will be the most competitive solution for electric vehicles.

Évaluation des performances d'un système de pompe à chaleur au propane pour les véhicules électriques en climat froid

Mots-clés: Propane; Véhicule électrique; Pompe à chaleur; Climat froid; Remplacement du frigorigène

1. Introduction

Nowadays, an increasing attention is drawn to electric vehicles due to the inherent low vehicle emissions. Though replacing gasoline/diesel engine with battery would be an effective method to reduce vehicle emission, it would cause a new problem for vehicles. There will be no waste heat for cabin heating in winter time without engines. Conventionally, PTC (positive temperature coefficient) heater would be a convenient method for heating. However, due to the battery capacity limitation, the power consumption for PTC would significantly affect the driving mileage, resulting in a 50% range reduction (Higuchi et al., 2017). In this case, a heat pump may be a solution for this issue. However, the heating performance of traditional R134a heat pump would significantly deteriorate with a decrease in ambient temperature. According to Kwon et al.'s and Lee's researches (Kwon et al., 2017; Lee, 2015), R134a heat pump cannot supply sufficient heat to the cabin in cold climate (−20 °C). Therefore, multiple researches have raised up different strategies to solve this problem.

Zhang et al. have proposed high efficiency vapor injection system for R134a. According to their research, the R134a vapor injection system can achieve an average improvement of 57.7% in heating capacity under −20 °C ambient temperature and the maximum capacity is 2097 W and COP is 1.17 (Zhang et al., 2016a). However, many countries all over the world have imposed regulations on HFCs (Hydrofluocarbon) application in vehicles. The European Parliament has banned the F-gases with GWP (Global Warming Potential) larger than 150 for all cars from 2017

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Nomenclature

COP	coefficient of performance (-)
GHG	greenhouse Gas
h	enthalpy (kJ·kg-1)
M	motor
P	pressure sensor
T	temperature sensor
W	compressor work (kW)
EXV	electronic expansion valve
GWP	Global Warming Potential
HFC	Hydrofluocarbon
MAC	mobile air conditioner
Q	heating CAPACITY (kW)
V	volume flow rate (m3·s-1)
~HZ	hertz

Greek symbols

ρ	density (kg·m-3)

Subscripts

a	air
aout	air outlet
conout	condenser outlet
ain	air inlet
conin	condenser inlet
ref	refrigerant

(Brown et al., 2010). The U.S. EPA GHG (Greenhouse Gas) regulations (USEPA, 2017) established credits for using low GWP refrigerants for light-duty passenger vehicles and tighter systems and included measures to grant CO_2 credits for MAC (Mobile Air Conditioner) system adopting more efficient components. Japan regulatory prohibitions require the weighted average GWP target for MAC refrigerants are designed to be 150 by 2023 (METI, 2016). China hasn't published any regulations regarding to the refrigerant limitations on MAC system yet. However, China has joined the Kigali Amendment and as the Kigali condition precedent is achieved already, Kigali Amendment will officially come into effect on January 1st, 2019. In the Kigali Amendment (UN Environment, 2016), China has promised to freeze the use of HFCs by 2024. To reach these requirements, CO_2 has been considered as a promising alternative in MAC systems thanks to its environmental friendly properties and excellent heating performance characteristics especially in low temperatures (Bullard et al., 2000). Wang et al., (2018a) have studied the heating performance of CO_2 heat pump system. The results indicated that CO_2 heat pump system can achieve 3.6 kW heating capacity with a COP of 3.1 when both indoor and outdoor temperatures are −20 °C. However, there are some serious concerns by applying CO_2 as a refrigerant in MAC systems. Many researches have reported that the COP of CO_2 air conditioning system is lower at high ambient temperature (above 30 °C) (Kim et al., 2009; Liu et al., 2005; Martin et al., 2005; Yin et al., 1998). Brown et al., have evaluated the performances of CO_2 and R134a systems by semi-theoretical models and the results indicated that the COP of CO_2 was lower than that of R134a by 21% at 32.2 °C and by 34% at 48.9 °C and the COP disparity would be even larger at higher ambient temperatures (Brown et al., 2002). Besides, CO_2 operates at comparably high operation pressures (Baek et al., 2013). It leads to a concern for the system reliability, a high production cost and therefore a large environmental impact during production.

In this article, a new technical solution was proposed by applying propane in mobile heat pump system for electric vehicles. Other than above mentioned technical solutions, propane heat pump system might be a promising method for electric vehicles thanks to the excellent thermo-physical characteristics and environmental friendly properties of propane (Choudhari & Sapali, 2017). It operates at a similar pressure level compared to R134a. No large changes need to be made on current system. Moreover, it has a larger volume cooling capacity compared to R134a leading to it becoming a perfect replacement for R134a in MAC system (Mahmoud, 1999). However, no literature has studied propane mobile heat pump system performances in cold climate before. In this article, heating performance characteristics of proposed propane mobile heat pump system have been studied under various operation conditions. The impacts of ambient air temperature, the recirculated fresh air percentage, the compressor rotational speed and the indoor and outdoor air volume flow rates on the system performance were investigated experimentally. The heating performances of different heat pump strategies including vapor injection technology, CO_2 heat pump system and propane heat pump system have been compared.

2. Experimental set-up

2.1. Test apparatus

The schematic diagram of the experimental set-up is shown in Fig. 1. The system is similar constructed to a normal heat pump system but charged with propane. The experimental set-up is composed with an electrical compressor, an inner condenser, an outdoor evaporator, an EXV (electronic expansion valve), a flow meter and an accumulator set in a psychometric calorimeter laboratory. The psychometric calorimeter laboratory is composed by an evaporator chamber and a condenser chamber and each chamber with a wind tunnel inside. The heat exchangers are placed at the inlets of the wind tunnels. The wind tunnels are used to regulate air flow rates and measure the inlet and outlet air dry bulb and wet bulb temperatures. The ambient conditions are controlled by the environmental control system consisting of air conditioning units, electrical heaters and humidity control equipment. The electrical compressor is a scroll type with 33cc displacement. The evaporator and the condenser used in this test rig are aluminum micro-channel heat exchangers. The size of the evaporator is 195W*150H*30D (mm) and the size for the condenser is 584W*390H*19D (mm). The electronic expansion valve (EXV) is driven by a stepping motor controlled by a pulse signal generator and the opening of EXV can be regulated from 20 pulse to 550 pulse. An accumulator of 230 ml is installed after the evaporator to serve as a system refrigerant buffer and to ensure only gas enters the compressor. The pressure transducers and thermocouples are installed between different components as shown in Fig. 1.

The uncertainties of measured parameters can be found in Table 1. The heating capacity (Q) of the tested R290 mobile air conditioner was determined by the air side heat transfer rate using Eq. (1). Besides, the heating capacity of the refrigerant side can also be calculated as Eq. (2). Analysis of the experimental results indicated that the error between heat transfer rate for the air side and refrigerant side was within ± 5%. The overall system COP was determined by Eq. (3). Referring to SAE standard J2765 (SAE International, 2008), the system work only includes the work input to the compressor, excluding the work input in the blowers. The properties of R290 refrigerant were calculated according to NIST REFPROP 8.0 (Lemmon et al., 2002).

$$Q = \dot{V}_a \times \rho_{ain} \times (h_{aout} - h_{ain}) \tag{1}$$

$$Q_{ref} = \dot{m}_{ref} \times (h_{conout} - h_{conin}) \tag{2}$$

$$COP = \frac{Q}{W} \tag{3}$$

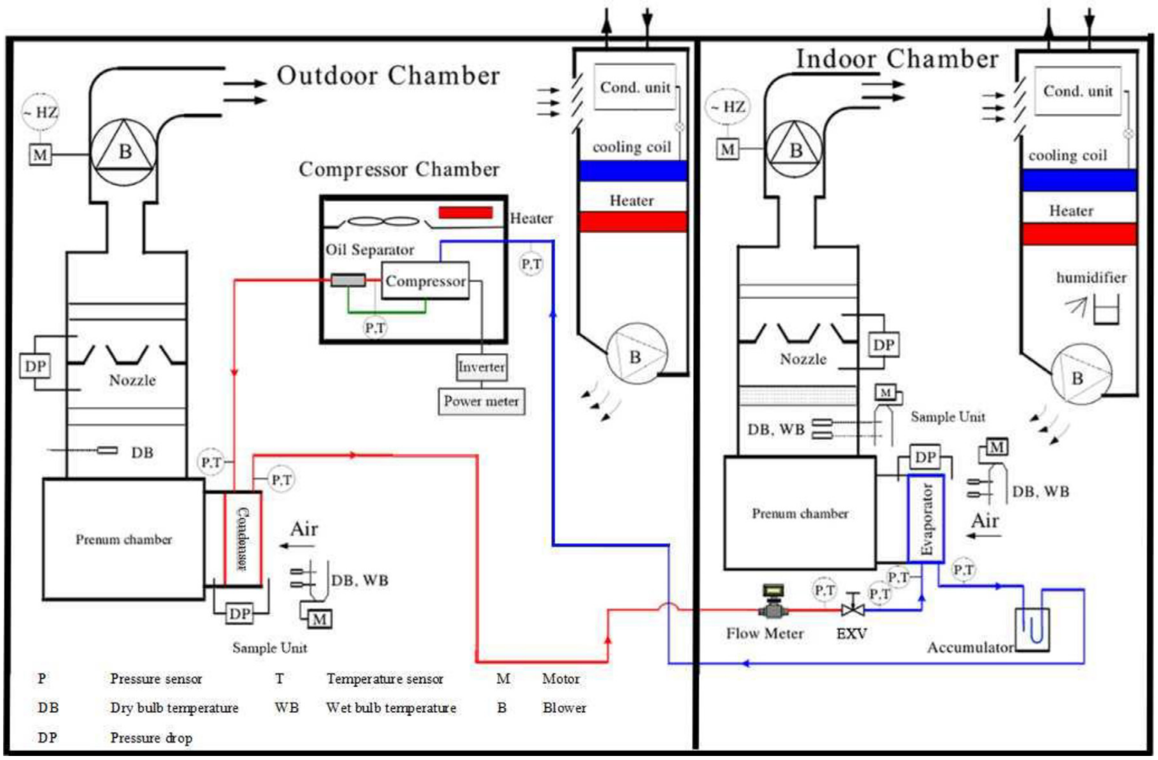

Fig. 1. The schematic diagram of the experimental set-up.

Table 1
Precisions of the measured parameters.

Parameter	Uncertainties
Temperature (°C)	± 0.1K
Pressure transducer (kPa)	±0.25%
Mass flow meter (kg/h)	±0.17%
Power meter (W)	±0.01%
Data logger (34927A, Agilent)	0.004% dcV accuracy
Air side pressure difference (Pa)	±0.2%

Where $\dot{V}_a$ represents the air volume rate flowing into the condenser wind tunnel and it is controlled by the wind tunnel, ρ_{ain} represents the air density flowing in the condenser, h_{ain} and h_{aout} represent the air enthalpy flowing in and out of the condenser, $\dot{m}_{ref}$ represents the refrigerant flow rate, h_{conout} and h_{conin} represent the refrigerant enthalpy flowing in and out of the condenser, W represents the system work input to the compressor. All refrigerants' properties are obtained from NIST REFPROP 8 (Lemmon et al., 2002).

The uncertainties of heating capacity and COP can be computed using Moffat's equations as shown in Eqs. (4) and (5), where Xi represents each independent measurement (Moffat, 1988). The maximum relative uncertainties of heating capacity and COP are calculated as ± 2.4% and ± 2.2% respectively.

$$R = R(X_1, X_2, X_3, \ldots, X_N) \tag{4}$$

$$\frac{\delta R}{R} = \left(\sum_{i=1}^{N} \left(\frac{\delta R}{\delta X_i} \delta X_i \right)^2 \right)^{1/2} \tag{5}$$

2.2. Test methods

In this paper, different parameters' effect on the system performance has been studied experimentally including the compressor speed, ambient temperature, indoor air re-circulated percentage, outdoor air velocity and indoor air volume flow rate. The detailed test conditions can be found in Table 2. In each condition, the EXV opening was regulated to a position to ensure the highest system COP.

The first step was to decide the optimal charge mass for the system. The starting point of R290 charge mass was set to 160 g and it was charged into the system in a 20 g increment until the maximum COP value was obtained for the system. The optimal charge mass for the system was 280 g and all the experiments results discussed in this article were carried out based on the optimal charge amount.

Tests 1–16 are designed to investigate the indoor air recirculated percentage and outdoor air temperature influences on the system performance. The indoor air inlet temperature is decided by the indoor air recirculated percentage. The indoor air inlet is comprised of the indoor air recirculated (in our case, it is determined as 20 °C) and outdoor fresh air. The indoor air temperatures are different for different outdoor air temperatures, therefore for each outdoor air temperature condition, 4 tests have been carried out in four different indoor air temperatures. In tests 13–16, the compressor speed is controlled at 2200 rpm because the compressor outlet pressure is too high for the rubber hose (2.5 Mpa in our case). And indoor outlet air temperature is high enough for the cabin heating purpose when the system operates at 2200 rpm when the outdoor ambient temperature is 10 °C. Tests 17–24 are designed to investigate the indoor air volume flow rate and outdoor air velocity influences on the system performance. Tests 25–36 are designed to investigate the compressor speed influences on the system performance. As for vehicles, the indoor outlet air temperature is a designed parameter and usually it will not be designed too high considering the cabin comfort. Thus the compressor is operated in different speed range for different outdoor ambient temperature. For different outdoor ambient temperature, different compressor speeds are tested. The test 37–39 are designed to study

Table 2
Test conditions.

Test no.	Compressor speed (rpm)	Indoor air recirculated percentage (%)	Outdoor air temperature (°C)	Indoor air volume flow rate (m^3/h)	Outdoor air velocity (m/s)	Indoor outlet air temperature (°C)
1–4	3000	100%, 70%, 30%, 0%	−20	350	2	–
5–8	3000	100%, 70%, 30%, 0%	−10	350	2	–
9–12	3000	100%, 70%, 30%, 0%	0	350	2	–
13–16	2200	100%, 70%, 30%, 0%	10	350	2	–
17–20	3000	100%	−20	350	1.5, 2, 2.5, 3	–
21–24	3000	100%	−20	250,300,350,400	2	–
25–27	3000,4000,5000	100%	−20	350	2	–
28–30	1800,3000,4000	100%	−10	350	2	–
31–33	1800,2500,3000	100%	0	350	2	–
34–36	1000,1800,2200	100%	10	350	2	–
37–39	3000,6000,8000	0%	−20	350	2	–
40–43	–	100%	−20,−10,0,10	350	2	40

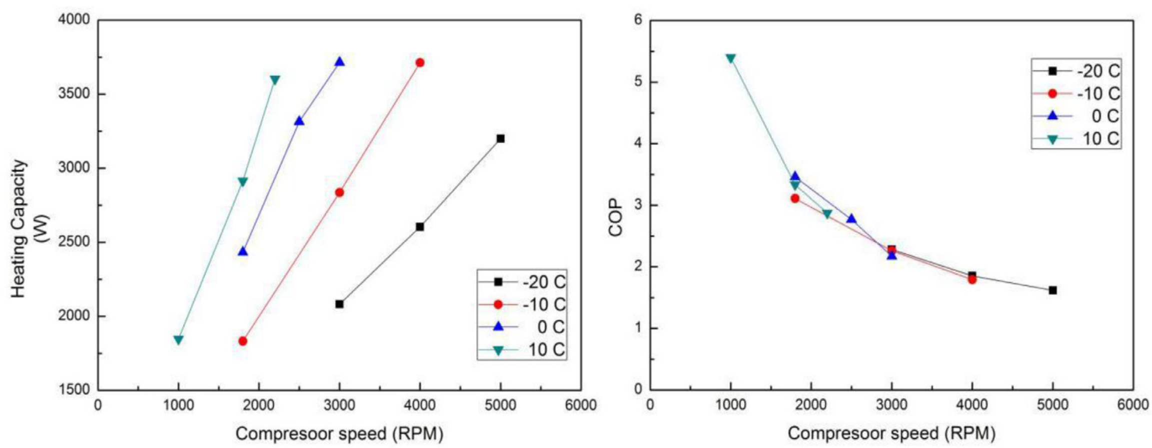

Fig. 2. a) Heating capacity & b) system COP versus compressor speed under different ambient temperature.

the maximum heating capacity of the system at an extreme temperature −20 °C with 0% indoor air recirculated in order to have a general concept of the capability of the mobile heat pump utilizing propane. The tests 40–43 are designed to compare the outdoor ambient temperature influences on the system efficiency with the same indoor outlet air temperature.

3. Experimental results and discussion

3.1. The effect of compressor speed

Fig. 2 show the compressor speed effect on the system performance under various outdoor ambient air temperatures. The EXV is regulated to a position to ensure the highest system COP. At 5000 rpm, −20 °C outdoor temperature and 20 °C indoor temperature (100% indoor recirculated percentage), the system heating capacity reaches 3200 W, the air outlet temperature reaches 46 °C while the system COP is 1.62. Therefore the propane heat pump system can provide enough heat under extremely cold weather without PTC. As observed from Fig. 2(a) and (b), with the increase of compressor speed, the system heating capacity increased rapidly while the system COP decreased. At 3000 rpm, −20 °C outdoor temperature and 20 °C indoor temperature (100% indoor recirculated percentage), the system heating capacity is 2082 W and the system COP is 2.28. Comparing to the condition of 5000 rpm, −20 °C outdoor temperature and 20 °C indoor temperature (100% indoor recirculated percentage), the system heating capacity is decreased by 35% while the system COP increased by 40.7% at 3000 rpm. Considering the compressor speed effects on system heating capacity and power input, it is interesting to notice when the compressor speed increases from 4000 rpm to 5000 rpm, the system heating capacity increased by 595.45 W while the power input increased by 570 W which is almost equal to the heating capacity increase. And also, the system noise will increase with the compressor speed increase which will largely affect the vehicle comfort. In this case, an option of 4000 rpm compressor speed + PTC will be better than the option of 5000 rpm compressor speed.

3.1.1. The effects of outdoor ambient temperature

Moreover, as indicated from the figures, the outdoor ambient temperature's effect on the system heating capacity is also tremendous. With the decrease of outdoor ambient temperature, the system heating capacity largely reduced while the system COP experiences a small variation. This can be explained from the p-h diagram as shown in Fig. 3. Fig. 3 shows the p-h diagrams of the system working states of test condition 1, 5, 9. The p-h diagrams are drawn based on the real test data. The enthalpy value for each point is calculated from the P-T measurement at each component inlet using NIST REFPROP 8.0. The only assumption made during plotting p-h diagram is the expansion process is an isenthalpic expansion process and the same for the following p-h diagrams. These conditions provide the outdoor ambient temperature effects on the system performance under same compressor speed and indoor air recirculated percentage (3000 rpm, 100%). As shown in Fig. 3, with the increase in outdoor ambient temperature, the evaporation pressure increases resulting in the increase in the refrigerant density at the compressor suction. As the mass flow rate dominates the variation in system heat capacity, it experiences a large increment with the increase in outdoor ambient temperature. The system COP is a combined effect of both the enthalpy differences between the condenser and the compressor and the compressor

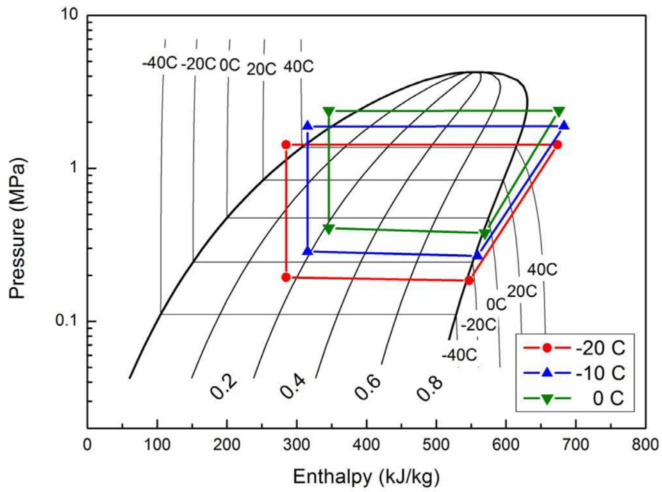

Fig. 3. Logp-h diagrams of system states maintaining same compressor speed under different outdoor ambient temperature.

isentropic efficiency. With the increase in outdoor ambient temperature, both enthalpy differences decrease, and the compressor isentropic efficiency doesn't vary much (maintaining at the level from 73.4% to 80.3%), the system COP is almost the same at the same compressor speed with varying outdoor ambient temperature.

3.1.2. Same indoor air outlet temperature

For vehicle heating, one important parameter evaluating the system performance is to the indoor air outlet temperature. In order to ensure the cabin comfort, the indoor air outlet temperature shall be raised to 40 °C. The impact of the outdoor ambient temperature on the system COP has been studied experimentally when 100% re-circulated air is heated up to 40 °C. When the outdoor temperature increases, the system COP increases almost linearly with the outdoor ambient temperature. As shown in Fig. 4(a), as the outdoor ambient temperature increases from −20 °C to 10 °C, the COP increases 223% and this is because same indoor outlet temperature means almost the same heating capacity (around 2450 kW) and as the compressor speed decreases, the compressor work decreases resulting in the increase in system COP. Fig. 4(b) shows the p-h diagrams of the system working states of test conditions 40–43. These conditions provide the outdoor ambient temperature effects on the system performance under the same indoor air outlet temperature (40 °C). As shown in Fig. 4(b), to maintain the same indoor air outlet temperature, the condensation pressure and the enthalpy difference between the inlet and outlet of the condenser are kept at the same level regardless of the outdoor ambient temperature variation. As the outdoor ambient temperature decreases, the evaporation temperature decreases causing the decrease in refrigerant density at the compressor inlet. Therefore, to compensate the mass flow rate loss causing by the decrease in refrigerant density at the compressor inlet, the compressor speed has to be increased.

3.2. The effect of indoor re-circulated air percentage

To ensure the vehicle cabin comfort, including the indoor humidity, CO_2 ppm (parts per million), etc., a mix of the indoor air with outdoor fresh air is required. Thus it is necessary to study the system performance variations with different indoor re-circulated air percentage.

Fig. 5 show the indoor re-circulated air percentage effect on the system performance under various outdoor ambient air temperatures. As illustrated before, the performance curve of 10 °C outdoor temperature was tested at 2200 rpm compressor speed considering the rubber hose bearing pressure while other curves shown in Fig. 5 were tested at 3000 rpm compressor speed. As shown in Fig. 5, with the increase of indoor air recirculated percentage, the system COP decreases rapidly while the system heating capacity experiences small variation. At 3000 rpm, −20 °C outdoor temperature and −20 °C indoor heat exchanger air inlet temperature (0% indoor recirculated percentage), the system heat capacity is 2380 W while its COP is 5.22. Compared to results of the condition at 3000 rpm, −20 °C outdoor temperature and 20 °C indoor heat exchanger air inlet temperature (100% indoor recirculated percentage), the system COP decreased by 44% while the heating capacity decreased only 12.5% at 0% indoor recirculated percentage. This can be explained from the p-h diagrams as shown in Fig. 6. Fig. 6 shows the p-h diagrams of the system working states of test condition 1–4. With the increasing indoor recirculated air percentage, the evaporation pressure slowly increases and the condensation pressure significantly increases. However, as the subcooling degree remains almost the same, the enthalpy differences between the condenser inlet and outlet doesn't vary much. As the evaporation pressure also varies very small, the heating capacity remains at the same level regardless of the indoor air recirculated percentage variation. While with the increase in the indoor air recirculated percentage, the condensation pressure increases

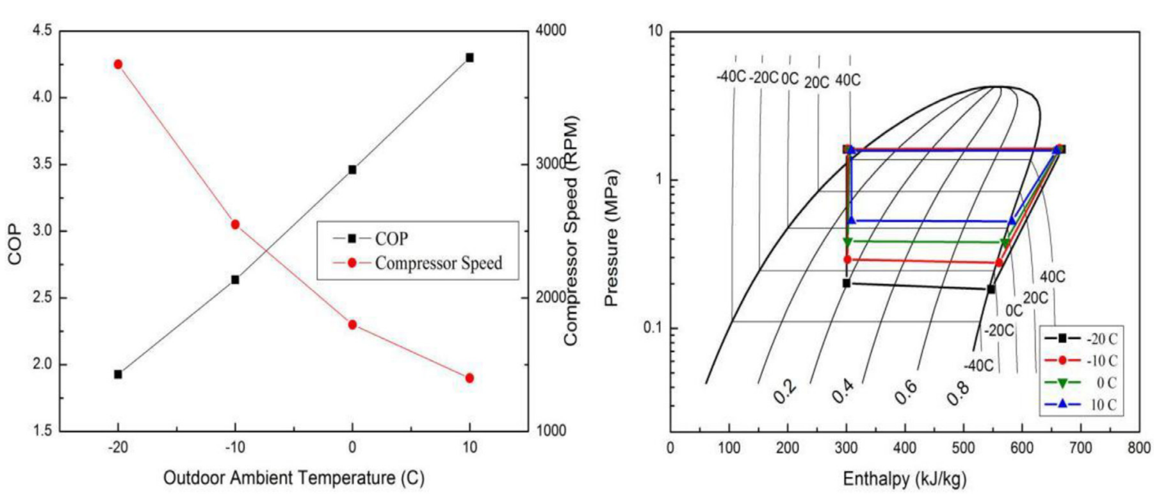

Fig. 4. Effects of outdoor ambient temperature with same indoor air outlet temperature on a) system performance and b) logp-h diagram.

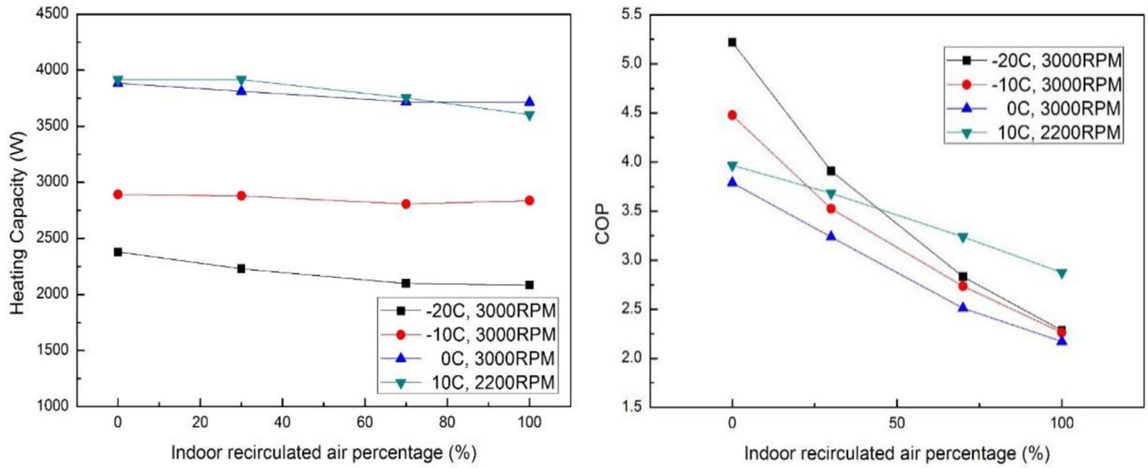

Fig. 5. a) Heating capacity and b) system COP versus indoor recirculated air percentage.

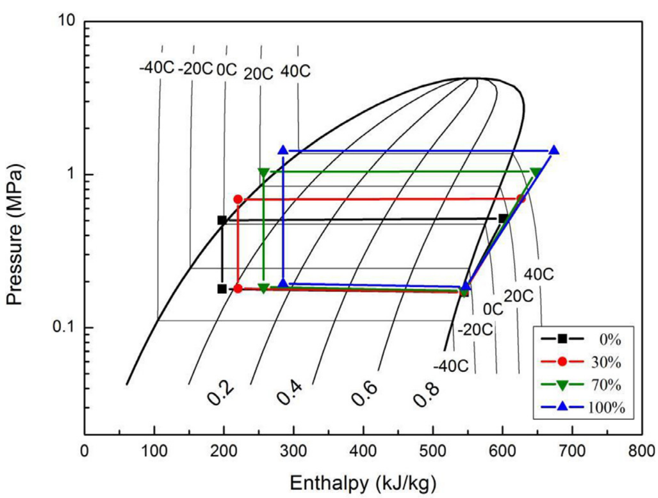

Fig. 6. Logp-h diagrams of system states maintaining compressor speed under different indoor air recirculated percentage.

resulting in the increase in the compressor work thus decrease in system COP.

As a reference, the system has also been tested at compressor speeds of 6000 rpm and 8000 rpm when both indoor and outdoor temperature are −20 °C. As a result, at 6000 rpm, the system capacity can reach 3754 W while the COP is 2.76 and at 8000 rpm, the system heating capacity can reach 4777 W while the COP is 2.28. Therefore, the propane heat pump can provide enough heat during the warm-up period in cold climate with a high efficiency.

3.3. The effect of indoor air volume flow rate

Fig. 7(a) shows the experimental results of test conditions 21–24 presenting the indoor air volume flow rate effects on the system performance. As indicated from Fig. 7(a), with the increase in the indoor air volume flow rate, the system COP and heating capacity increase at the same time. When the indoor air volume flow rate increased from 250m^3/h to 400m^3/h, the system heating capacity increased by 13.7% and the COP increased by 32.4%. Fig. 7(b) shows the p-h diagrams of the system working states of test condition 21–24. As the indoor air volume flow rate increases, the evaporation pressure remains almost constant while the condensation pressure decreases largely resulting an improvement in the system COP. The condensation pressure is not linearly decreases with the increase of indoor air volume flow rate which explains why COP is not linearly increase with the increasing indoor air volume flow rate.

3.4. The effect of outdoor air velocity

Fig. 8(a) shows the experimental results of test conditions 17–20 presenting the outdoor air velocity effects on the system performance. As indicated from Fig. 8(a), with the increase in the outdoor air velocity, the system heating capacity increases very slowly with the increase in the outdoor air velocity thanks to the evaporation pressure increase as shown in Fig. 8(b) while system COP remains almost the same. When the outdoor air velocity increases from 1.5 m/s to 3 m/s, the system heating capacity increased by 8.5% while the system COP is almost constant. This is because the compressor work increases at the same time with increasing heating capacity. And this can be explained as the system frictional work increases with an increasing system mass flow rate resulting in the increase in the compressor work.

4. Different heat pump strategies comparison

To solve the problem of driving mileage for electric vehicles during winter, heat pump is proposed as a solution. A heat pump with higher efficiency can greatly improve the driving mileage for electric vehicles. R134a heat pump system, R1234yf heat pump system, R134a EVI (Economized vapor injection) heat pump system and CO_2 heat pump system have been recognized as the most common technical solutions for heat pump. In order to get a whole picture of how propane heat pump system performs, this chapter gives a comparison among different heat pump solutions. The authors gathered the experimental data for different heat pump solutions of electric vehicles from open literature and compare it with the experimental data for propane heat pump system.

Figs. 9 and 10 show the heating capacity and system COP of different heat pump strategies including R134a heat pump system, R1234yf heat pump system (Zou et al., 2017), R134a EVI (Economized vapor injection) heat pump system (Zhang et al., 2016b), CO_2 heat pump system (Wang et al., 2018b), R290 heat pump system under different operation condition as show in Table 3.

Comparing to normal R134a and R1234yf system, R290 system heating capacity is about 70% larger than both systems at 0 °C while more than 90% larger than both systems at −10 °C. If excluding the compressor displacement effect, R290 system still has around 40% larger heating capacity at 0 °C and more than 55%

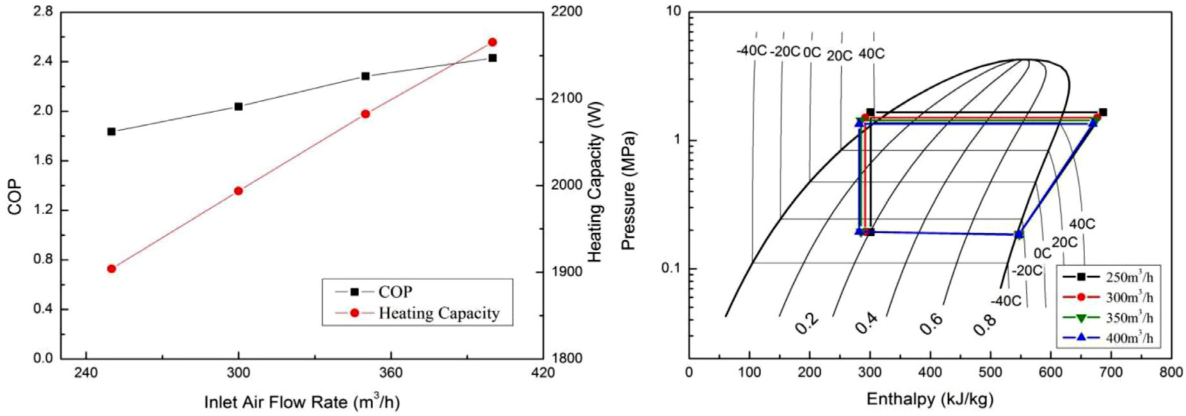

Fig. 7. Effects of indoor air flow rate on a) system capacity and b) logp-h diagram.

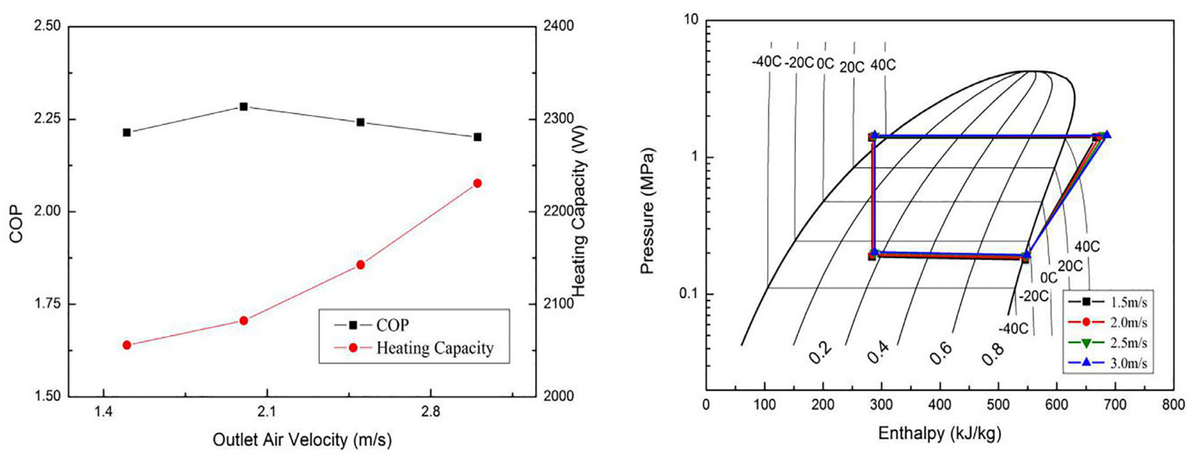

Fig. 8. Effects of outdoor air velocity on a) system capacity and b) logp-h diagram.

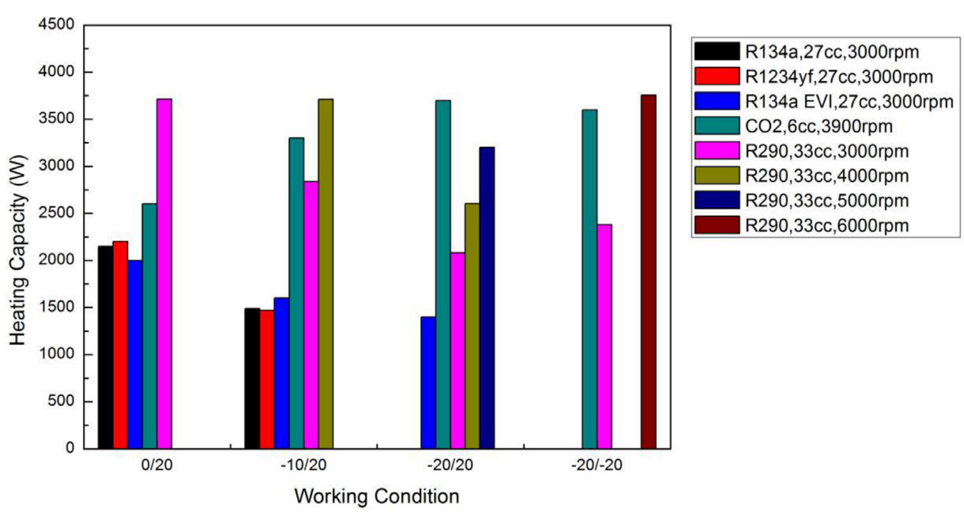

Fig. 9. Heating capacity comparisons among different heat pump strategies.

larger heating capacity at −10 °C than both systems, assuming linear interpolation. While comparing system COP at same compressor rotational speed, R290 system is about 4.29% and 12.3% larger than R134a and R1234yf system at 0 °C, respectively and 25.5% and 7.6% higher than R134a and R1234yf system at −10 °C, respectively. If the COP is compared at the same heating capacity, R290 system will be more advantageous.

Comparing to R134a EVI system, R290 system is still more beneficial. R290 system heating capacity is 85.7%, 77.2% and 48.7% larger than R134a EVI system at 0 °C, −10 °C, −20 °C, respectively. Excluding the compressor displacement influences, the heating capacity of R290 system is still about 51.9%, 45.0% and 21.7% larger than R134a EVI system at 0 °C, −10 °C, −20 °C, respectively. While comparing system COP at same compressor rotational speed, R290

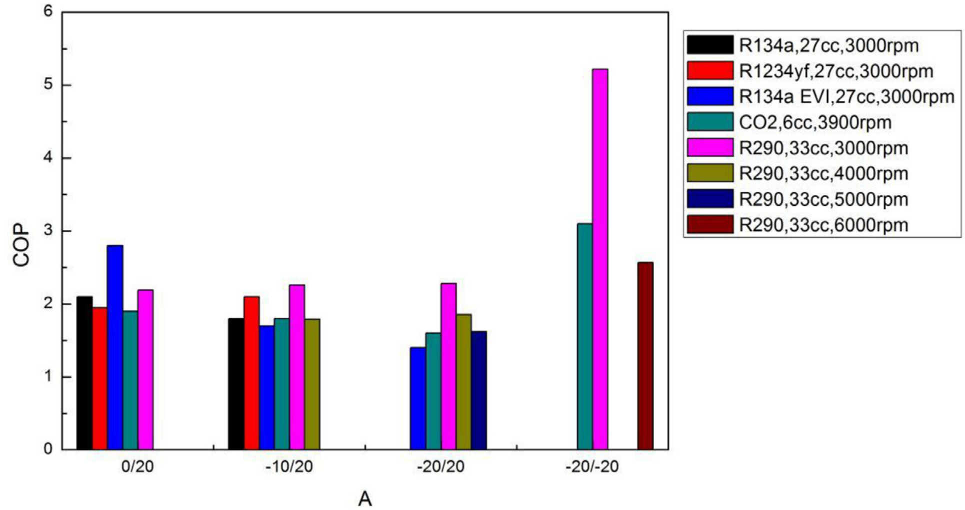

Fig. 10. System COP comparisons among different heat pump strategies.

Table 3
Operation conditions for different heat pump strategies.

Heat pump strategy	Compressor speed (rpm)	Compressor displacement (cc)	Indoor air temperature (°C)	Outdoor air temperature (°C)
R134a without EVI	3000	27	20	0, −10
R1234yf without EVI	3000	27	20	0, −10
R134a with EVI	3000	27	20	0, −10, −20
CO_2	3900	6	20	0, −10, −20
CO_2	3900	6	−20	−20
R290	3000	33	20	0, −10, −20
R290	3000	33	−20	−20
R290	4000	33	20	−10, −20
R290	5000	33	20	−20
R290	6000	33	−20	−20

system is 22% smaller than R134a EVI system at 0 °C and 32.9% and 62.8% larger than R134a EVI system at −10 °C and −20 °C, respectively. Though the COP of R290 system is about 22% smaller than R134a EVI system at 0 °C, R290 system heating capacity is 48.7% larger than the system. Comparing both systems at the same heating capacity, R290 system could be still beneficial. Besides, R290 system has tremendous advantages at low ambient temperatures comparing to R134a EVI system.

Comparing to CO_2 system (Wang et al., 2018a), R290 is competitive at an outdoor ambient temperature above −10 °C. According to the reference, the CO_2 system was tested at a compressor speed at 3900 rpm since our system is tested based on 3000 rpm baseline and only −10 °C and −20 °C ambient temperature, the system performances at 4000 rpm are tested. According to the results, the heating capacity of R290 system at 3000 rpm is 42.8% larger than CO_2 system at 3900 rpm and the COP is 15.3% higher at 0 °C ambient air temperature. The heating capacity of R290 system at 4000 rpm is 12.5% higher than CO_2 system at 3900 rpm and the COP is almost the same at −10 °C ambient air temperature. While the outdoor ambient temperature goes to −20 °C, the heating capacity of R290 system is only 70.4% of that for CO_2 system while the system COP is 15.8% higher than that for CO_2 system. If we compare the results of R290 system at 5000 rpm with CO_2 system at −20 °C, the heating capacity of R290 system is around 14% lower than that for CO_2 system while the system COP is 1.3% higher than that for CO_2 system. Moreover, when both air inlets temperatures of the indoor and outdoor heat exchangers are −20 °C, R290 system at 6000 rpm can reach 104.3% heating capacity and 83.2% system COP of CO_2 system at 3900 rpm which means to achieve almost the same heating capacity, the efficiency of R290 system may be a little lower than CO_2 system in this case.

By comparing different low temperature heat pump strategies, it is easy to conclude that R290 system is the most competitive candidate at an outdoor ambient temperature above −10 °C among all the strategies while CO_2 system may have small advantages when the ambient temperature is −20 °C. This is because when comparing to R134a and R1234yf system with/without of EVI, the latent heat of R290 is about twice large of those for R134a and R1234yf. As the heating capacity is a combined result of the system mass flow rate and latent heat, the latent heat still dominates resulting in a larger heating capacity of R290 system. While for COP, as indicated from Navarro et al. (2013) research, the compressor efficiency and volumetric efficiency of R290 are much higher than those for R134a and R1234yf and this can explain the COP improvement of R290 system. When comparing to CO_2 system, the vapor density of CO_2 drops much slower than that of R290 with decreasing temperature, thus it is reasonable that at −20 °C, CO_2 system is more advantageous while at a temperature above −10 °C, R290 system is more competitive. However, as CO_2 is operated at really high pressure, up to 12 MPa (Aprea and Maiorino, 2008), the system reliability and components cost will be much higher than R290 system. The system cost will also be a very important factor when manufacturers compare different solutions. However, as neither of CO_2 and R290 heat pump system are in mass production right now, it is relatively hard to estimate the system costs. Table S1 in Appendix shows a estimated cost calculation for different heat pump systems the material cost and refrigerant cost. According to the estimation, the cost for R290 heat pump system is almost at the same level as R134a system which is much cheaper than R1234yf system and CO_2 system. However, further investigation on system cost including the technology cost and associated safety operation cost for R290 and CO_2 heat pump systems are still needed. As a conclusion, among all the compared system strategies, R290 heat pump system edges out all the competitors by its competitive system performances and relatively low system cost.

The major drawback for R290 is its flammability. By now, there have been several studies reporting on the safety studies regarding R290 application in split type household air conditioners and small refrigeration systems. Nagaosa (2014) has proposed a new numerical formulation to simulate the R290 spread in a room and examined the leakage rate effect on concentration profile. Colbourne et al. presented quantitative risk analysis on estimating the severity of the R290 leakage consequences for ice cream cabinet (Colbourne and Espersen, 2013), refrigerator and split type household air conditioner (Colbourne and Suen, 2015). Zhang et al., (2016a) have studied the explosion characteristics associated with the indoor and outdoor units and they concluded that the explosion overpressure is sufficiently low so as to not damage the air conditioner. Tang et al. (2017) have proposed a quasi-liquid nitrogen method to investigate the refrigerant distribution in R290 split air conditioner and the leaking rate under various conditions. They suggested by installing a solenoid valve, the leaking rate will be reduced and the safety can be improved. Besides, the standards have been developed to allow R290 safe use after a long experience of using it in refrigeration system (Corberan et al., 2008). However, the researches for the safety application of R290 are mostly focused on residential air conditioner and refrigeration system. As a liquified natural gas, R290's application in mobile AC/HP systems still needs a series of safety precautions in the future.

5. Conclusions

The paper studied the heating performance characteristics of a mobile propane heat pump system for electric vehicle in cold climate. The effects of outdoor ambient temperature, indoor recirculated air percentage, compressor speed, indoor air volume flow rate and outdoor air velocity on the system performances are studied experimentally. The results indicate that outdoor ambient temperature has a great effect on both heating capacity but has little effect on system COP while the indoor ambient temperature has an opposite effect. A decreasing indoor air volume flow rate would lead to a decrease in the system COP due to the increase of system condensation pressure. This is because the compact design the condenser to fit HVAC size inside cabin while the outdoor air velocity has little influence on the system performances.

Different heat pump strategies are also compared in the research. According to the results, R290 system is the most competitive candidate at an outdoor ambient temperature above $-10\,°C$ among all the strategies while CO_2 system may have small advantages when the ambient temperature is $-20\,°C$. However, due to CO_2's poor cooling COP at high ambient temperature and its high operating pressure, propane heat pump system may be the best solution for electric vehicles in terms of performances and cost. As propane operates at a pressure slightly higher than R134a, the system connecting rubber hose may be optimized in order to bear the pressure. Also, due to the flammability of propane, a series of safety precautions in the future will be needed for propane's application in mobile air conditioner and heat pump systems.

Supplementary materials

Supplementary material associated with this article can be found, in the online version, at doi:10.1016/j.ijrefrig.2018.08.020.

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