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ABSTRACT

The heating capacity and coefficient of performance (COP) will significantly decrease when the conventional air source heat pump (ASHP) system is operated under low ambient temperature conditions, with the high discharge temperature. The vapor injection technique with an internal heat exchanger (IHX) has been proposed as an effective way to acquire better performance. In this paper, the performance of a R410A two-stage air source heat pump system with the scroll compressor was studied experimentally. According to the experimental results, the heating capacity increased with the vapor injection at the fixed ambient temperature and water temperature, however the discharge temperature decreased. One interesting founding was that the discharge temperature at the ambient temperature of -6°C and -12°C increased slightly when a little refrigerant was injected into the compressor. The optimal point always occurred when the system COP was maximized with the other constant parameters. Furthermore, the peak points of each COP curve were shifted to the right if the ambient temperature was elevated. With variation of the ambient temperature, the peak points of the heat pump with vapor injection technique were compared with those of the conventional system. The results showed that the vapor injection technique could improve the heating performance and enhance the stability and reliability of the compressor. At the ambient temperature of -20°C, the system could have up to 17.01% improved COP and 13.73% decreased discharge temperature, respectively. In addition, the variation of injection ratio with intermediate pressure and expansion valve operating reliability were also investigated.

1. INTRODUCTION

Due to the low global warming potential (GWP) and high efficiency, the conventional air source heat pumps are widely used in residential houses for space heating and hot water around the world. However, the heating capacity and coefficient of performance (COP, defined as the ratio of heating capacity and total power consumption) will descend dramatically at low ambient temperature in cold regions. Meanwhile, the compressor discharge temperature and compression pressure ratio will rise simultaneously with the decrease of the ambient temperature, which would impair the normal operation reliability of the compressor. Lots of methods have been proposed to overcome this defect. The vapor injection technique, which is marketed for indoor air conditioners since 1979(Umezu and Suma, 1984; Winandy and Lebrun, 2002), has been proved to be a promising method to strengthen the performance of heat pump system, especially at low ambient temperature.

The saturated or super-heated vapor refrigerant can be injected into the compressor injection port with the vapor injection technique. The vapor injection cycles are usually categorized by two elementary configurations that can be regarded as the basis of other more complicated systems. Schematics of the two fundamental vapor injection cycles are displayed in Figure 1 where (a) and (b) show the flash tank vapor injection (FTVI) cycle and the internal heat exchanger vapor injection (IHXVI) cycle respectively. The two-phase refrigerant separates in the flash tank and the saturated vapor is injected into the injection port, while the saturated liquid refrigerant is expanded by the second expansion valve. In the IHXVI cycle, the injection stream refrigerant sub-cools the main stream at the internal heat exchanger, and then turns to super-heated vapor.

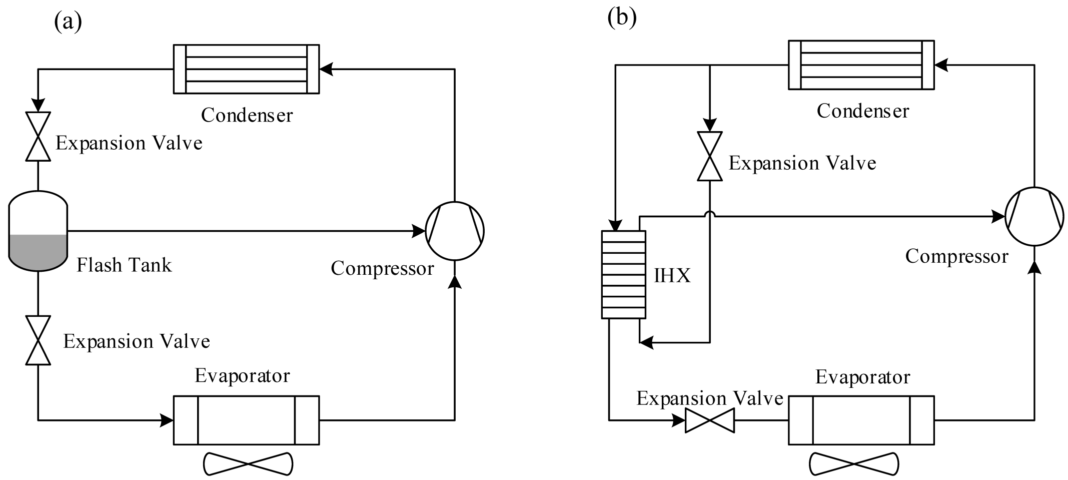

Figure 1: Schematics of the two fundamental vapor injection cycles

Recently, more attention has been focused on the application of the vapor injection technique in the heat pump system. Lifson (2005) theoretically investigated the novel design of the vapor injection ports, and speculated that the heating capacity as well as the efficiency could reach the improvement of 23% and 12%, respectively, at high ambient temperature. In order to optimize the actual operation of two stage cycles, Redón *et al*. (2014) analyzed the influence of design parameters and injection conditions for two different configurations with four refrigerants. They found that the maximum COP could be improved by 30%. (Roh and Kim, 2012) proposed an additional refrigerant injection line into the accumulator, and the experimental results showed that the compressor discharge temperature with the new injection design could drop rapidly at high compressor frequency condition, compared to the classic vapor injection line. Meanwhile, the mass flow rate of injected refrigerant had little effect on that of condenser. Tello-Oquendo *et al*. (2016) experimentally compared the scroll compressor with reciprocating compressor in harsh climates and the results showed that the scroll compressor with vapor injection technique presented better efficiency and COP than that of the reciprocating compressor at low pressure ratios (below 7.5). The experimental analysis which was investigated by (Cao *et al*., 2009), studied the heating performance with mixing refrigerant of R22/R600a, and the optimal intermediate pressure was also proposed. As for the refrigerant type, R410A, a leading HFC to replace R22, was theoretically and experimentally studied by researchers (Beeton and Pham, 2003) in recent years. James *et al*. (2016) developed a semi-empirical model for oil flooding to obtain near-isentropic compression of the R410A scroll compressors with injection technology. The proposed model was validated to be reliable by experiments. Cho *et al*. (2016) compared the system performance in cooling and heating codes with the classic one and the specific improvement value were also presented in the paper.

By reviewing the world-wide academic researches, it could be summarized that the vapor injection technique can improve the overall heating performance of ASHP in low temperature operating conditions. In this paper, the internal heat exchanger vapor injection system was investigated experimentally: the injection ratio (as defined in the section 3.1) at the ambient temperature of -6°C, -12°C and-20°C with respect to the intermediate pressure was acquired. In addition, the effect of vapor injection on the overall heating performance and the comparison with the classic system at the same ambient temperature, such as heating capacity, COP and discharge temperature were also presented in detail. Finally, the effect of the vapor injection on the expansion valve (EV) operating stability was depicted as well.

2. EXPERIMENTAL SETUP AND ERROR ANALYSIS

In order to investigate the effect of the vapor injection technique on the performance promotion of the air source heat pump, a prototype IHXVI cycle heat pump system was developed and experimentally measured in an environmental laboratory.

2.1 Experimental setup

The schematic diagram of IHXVI heat pump system is showed in Figure 2. The test system consists of a scroll compressor, a condenser, an evaporator and an internal heat exchanger, which is also called an economizer in literature. It should be noted that there is a solenoid valve on the injection line, which is designed to switch the operation mode between the IHXVI cycle and the classic ASHP cycle. The details of the IHXVI heat pump system components are summarized in Table 1. The prototype system was investigated in an environmental laboratory where parameters could be controlled and fixed during the experiment process, such as the inlet water temperature, ambient temperature, relative air humidity and water flow rate

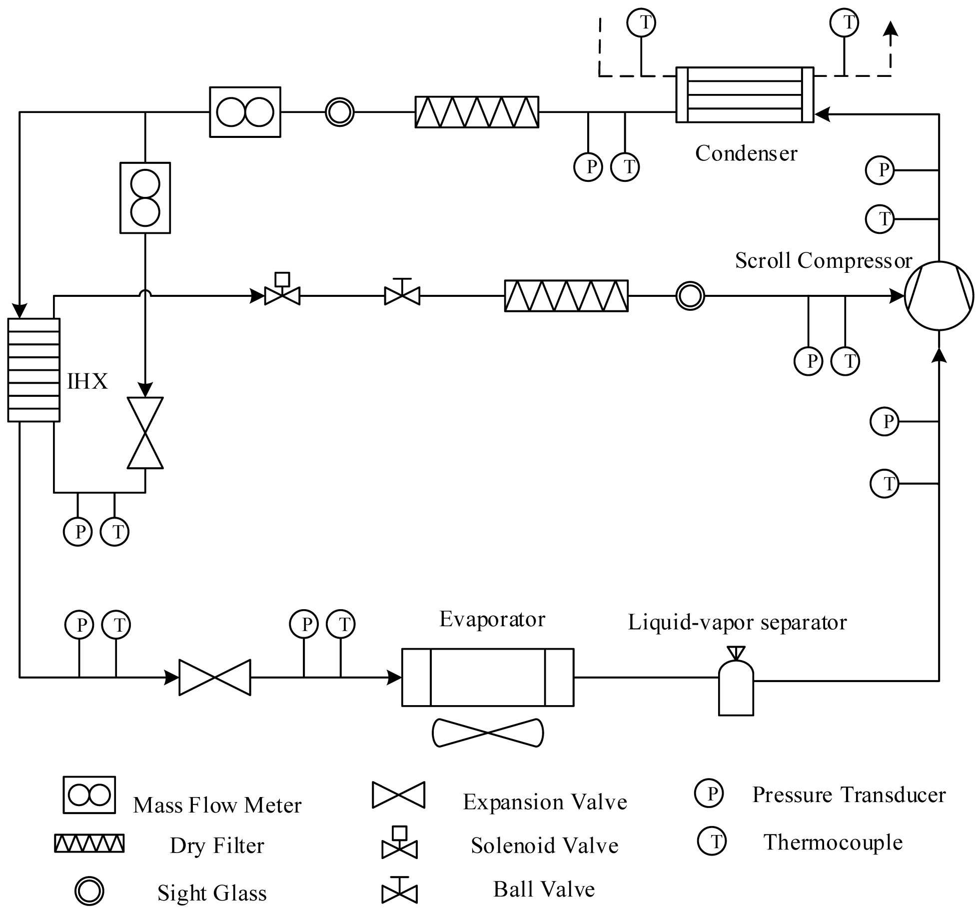

Figure 2: Schematic diagram of IHXVI heat pump system

Table 1: Details of the main components of the IHXVI system

Main components	Type	Characteristics
Compressor	Scroll compressor	Displacement : 24.9 $m^3 h^{-1}$
Condenser	Plate heat exchanger	Number of plates: 40 Plate length: 519mm Plate width: 191mm
Evaporator	Fin and tube	Tube diameter: $\Phi 10 \times 0.7$mm Tube length: 1675mm Number of rows: 2 Number of tube per row: 40
IHX	Plate heat exchanger	Number of plates: 20 Plate length: 280mm Plate width: 71mm

During the test, the total power consumption was measured by an electric power analyzer (WT-500) with a full scale of 0-40kW and an accuracy of ±0.1%. The temperatures of the measurement points were measured using PT100 for the water temperature with an accuracy of ±0.2°C and type K thermocouple for the refrigerant temperature with an accuracy of ±0.5°C. The water flow rate was controlled by a variable frequency water pump and measured by an electromagnetic flow meter with a full scale of 0-6$m^3 \cdot h^{-1}$(±0.01$m^3 \cdot h^{-1}$). The pressure transducers was installed with a full scale of 0-6MPa for the low pressure side and 0-16MPa for the high pressure side respectively, with an accuracy of ±0.25%. The refrigerant mass flow rate (both the injection and main stream) was measured by a mass flow meter (F050) with a full scale of 100$Kg \cdot min^{-1}$(±0.01 $Kg \cdot min^{-1}$).

2.2 Performance calculation

The measured data in the experiment, including refrigerant temperature and pressure, were recorded by portable paperless recorder (MV-2000) to monitor the performance of the system with real-time detection. The ambient temperature, relative humidity, inlet and outlet water temperature and water flow rate were recorded by a data acquisition instrument (34970A).

We can use the following equation (1) to calculate the heating capacity of the system with the measured data such as the water flow rate, water inlet and outlet temperature at the condenser and water specific heat

$$Q_h = c_p m_w (T_{out} - T_{in}) \tag{1}$$

where, c_p represents the specific heat of water, $kJ \cdot kg^{-1} \cdot K^{-1}$; m_w is the water flow rate, $kg \cdot s^{-1}$; T_{in} ,T_{out} are the water inlet and outlet temperatures, °C; Q_h is the IHXVI system heating capacity, kW.

The system COP is calculated as listed in equation (2)

$$COP = \frac{Q_h}{W} \tag{2}$$

where, W is the total system consumption, kW.

2.3 Error analysis

Due to the error of the measuring instrument, the systematic errors of the heating capacity and COP could be calculated from error propagation according to the Kline and McClintock method (Bevington and Robinson, 2002) as given in equation (3)

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_{x_1} \right)^2 + \left(\frac{\partial R}{\partial x_2} w_{x_2} \right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_{x_n} \right)^2 \right]^{1/2} \tag{3}$$

where $x_1, x_2, \ldots x_n$ are the independent variables, $w_{x1}, w_{x2} \ldots w_{xn}$ are the uncertainties of the independent variables, R is the given function, w_R is the resultant uncertainty. With the above given equation, the maximum uncertainties of the heating capacity and COP are 3.49% and 3.91%.

3. RESULTS AND DISCUSSIONS

The effect of the vapor injection on the heat pump system performance was experimentally investigated under different test conditions, as arranged in Table 2. During the experimental process, the mass flow rate of injected refrigerant was controlled by the ball valve on the injection line. Meanwhile, the effect of intermediate pressure on injection ratio, the heating performance of IHXVI heat pump system at optimal point and its comparison with the conventional one, and the effect of the vapor injection on expansion valve operating reliability as well as stability were discussed specifically in the following sections.

Table 2 Test conditions

Parameter	Value
Ambient temperature	-6°C, -12ºC, -20ºC
Relative air humidity	75%
Water inlet temperature	36ºC
Water outlet temperature	41ºC

3.1 Effect of the intermediate pressure on injection ratio

The injection ratio α could be calculated by the following equation (4)

$$\alpha = \frac{\mathrm{m}_{inj}}{m_{suc}} \tag{4}$$

where m_{inj} is the injected refrigerant mass flow rate, $\mathrm{kg \cdot s^{-1}}$; m_{suc} is the mass flow rate of the suction (main stream) refrigerant, $\mathrm{kg \cdot s^{-1}}$.

Figure 3 shows the variation of the injection ratio with the intermediate pressure. When the IHXVI heat pump system is operated at various ambient temperatures, the injection ratio increases significantly with the intermediate pressure. Due to the constant ambient temperature of each curve, the corresponding evaporating pressure is fixed. By manually controlling the ball valve, the injected refrigerant flow rate is gradually increased. Due to the coupling relationship between the injected refrigerant flow rate and intermediate pressure, the pressure difference between the intermediate and evaporating pressure also increases, which leads to the increment of the vapor injection, resulting in the elevation of the system injection ratio. Under the condition of same intermediate pressure, the injection ratio increases with the decrease of ambient temperature.

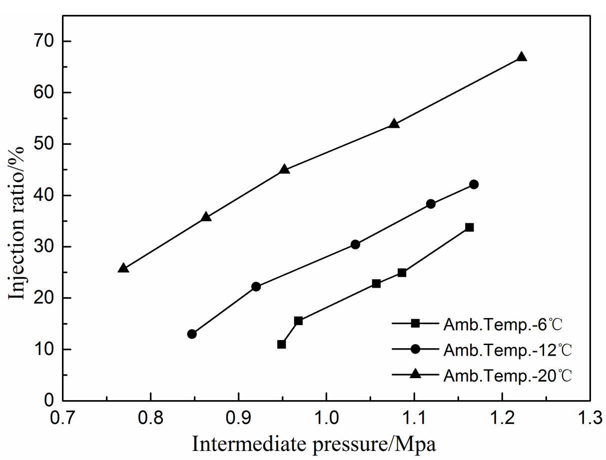

Figure 3: The injection ratio versus intermediate pressure at various ambient temperatures

3.2 Effect of the vapor injection on heating capacity

The variation of heating capacity with the injected refrigerant mass flow rate is illustrated in Figure 4. It is found that the heating capacity increases significantly under various ambient temperatures. With the augment of intermediate pressure, the mass flow rate of injected refrigerant also elevates, which means that the total refrigerant flow rate rises sharply. Although the refrigerant enthalpy at the discharge port of scroll compressor is reduced slightly, which will lower the per unit heating capacity, the superiority of the increment of refrigerant flow rate is more obvious. Meanwhile, due to the function of internal heat exchanger, the expanded refrigerant flowed into the evaporator is sub-cooled, which increases the ability of the evaporator to absorb heat from the surrounding environment, then the heating capacity of the condenser is increased as well. This could be explained in Figure 4, as the environment temperature decreases from -6°C to -20°C, the heating capacity is found to increase by 12.71% and

19.61%, respectively. It means that when the IHXVI heat pump system is operated at a lower ambient temperature, the effect of vapor injection on the heating performance is more obvious.

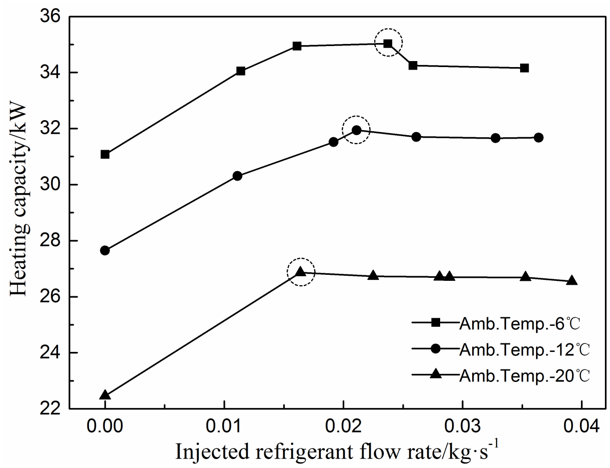

Figure 4: The heating capacity versus injected refrigerant flow rate at various ambient temperatures

3.3 Effect of the vapor injection on COP

The system evaporating temperature is kept within a small range of variation if the ambient temperature is controlled at a constant value varied from -6°C to -20°C. Figure 5 illustrates the effect of the vapor injection on the system COP under the varied working conditions. When the ambient temperature is modified from -6°C to -20°C, the system COP, in each case, also increases. The system COP curves depend on the relative variation between the heating capacity and total power consumption. However, there exists a certain mass flow rate of injected refrigerant for the optimal COP at the constant ambient temperature, which is closely linked to the evaporating temperature. Comparing with each system COP curve, the peak points of each curve are shifted to the left, to be specific, from 2.88 to 2.27, when the ambient temperature drops gradually. It means that the optimal injected refrigerant flow rate corresponding to the maximum COP diminishes when the evaporating temperature declines.

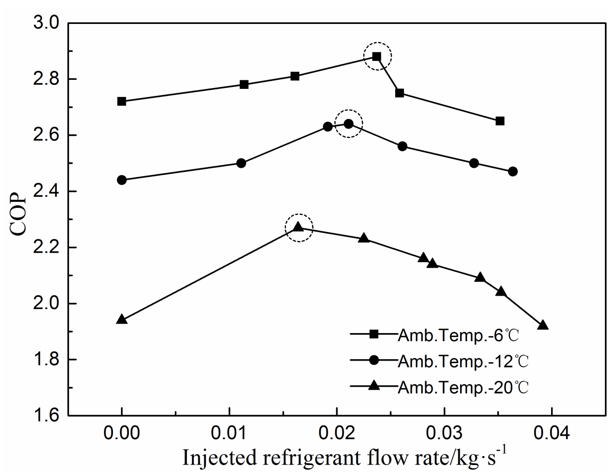

Figure 5: The system COP versus injected refrigerant flow rate at various ambient temperatures

3.4 Effect of the vapor injection on discharge temperature

In order to investigate the relationship between the compressor discharge temperature and vapor injection at different ambient temperatures, by controlling the ball valve manually, the experimental results are depicted in Figure 6. It is found that with the augment of vapor injection, the discharge temperature decreases obviously under condition of low ambient temperature. One interesting phenomenon in Figure 6 is that the compressor discharge temperature at the ambient temperature of -6°C and -12°C increases slightly when a little refrigerant is injected into the compressor. This could be clearly explained that the vapor injection refrigerant is superheated excessively by the liquid refrigerant from the condenser in the IHX, which leads to an increment of the discharge temperature at the same working conditions. On the operating condition of -20°C, the discharge temperature drops by about 20°C during the experiment, which reaches the maximum decreasing amplitude of 13.73%. The compressor discharge temperature corresponding to the peak point of COP curve as illustrated in Figure 6 is about 128°C, which could ensure a safe and stable operation of the compressor under -20°C ambient temperature.

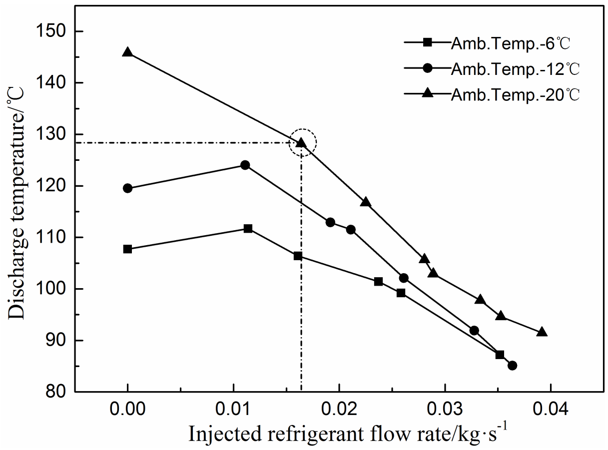

Figure 6: The discharge temperature versus injected refrigerant flow rate at various ambient temperatures

3.5 Effect of the vapor injection on EV operating reliability

During the experiments, the condensing temperature of the IHXVI system is closely related to the water inlet temperature. Therefore, the condensing temperature remains almost unchanged when the water inlet temperature at the condenser is fixed at 36°C. The favorable benefit of the IHXVI system with vapor injection technique is the refrigerant temperature differences between the condenser outlet and second EV inlet. In order to validate the operating reliability of the expansion valve, the experiments were conducted in the different working conditions. Figure 7 contains the information of the refrigerant temperature at the condenser outlet, refrigerant temperature at second EV inlet and refrigerant temperature difference between them with respect to the ambient temperature. It indicates that the maximum refrigerant temperature difference can reach to 29°C in a -20°C environment, and the IHX provides over 20°C sub-cooling of the main stream refrigerant, which has quite a larger effect on the safe temperature range of the EV.

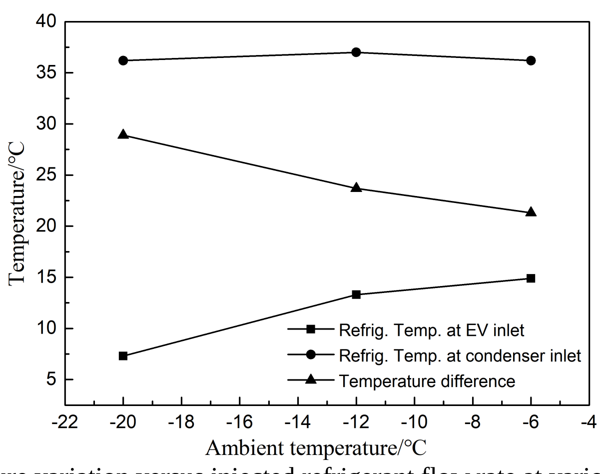

Figure 7: The temperature variation versus injected refrigerant flow rate at various ambient temperatures

3.6 Comparison of the conventional and IHXVI system

In order to analyze the effect of vapor injection on the improvement of the system performance, the variation of heating capacity, COP and discharge temperature with respect to the ambient temperature between the conventional and IHXVI system at the optimal value are showed in the following Figure 8 and Figure 9. As illustrated in Figure 8, the heating capacity of the IHXVI system is much larger than that of the classic ASHP system, with the ambient temperature modified during the experiment. This phenomenon demonstrates the superiority of the vapor injection technique in improving the heating capacity.

As showed in Figure 9, the COP of the conventional cycle decreases significantly with the ambient temperature due to the lower evaporating temperature and higher compression ratio. This defect almost exists for the single stage air-source heat pump system. The COP of the IHXVI system is larger than that of the conventional cycle at the same ambient temperature, especially for the extremely low temperature of -20°C. The COP increases from 1.94 to 2.27 at the working condition of -20°C, which indicates that the lower the operating ambient temperature, the more significant the effect of vapor injection on the system COP. As for the discharge temperature, the value of the IHXVI system is obviously lower than that of the conventional system. As mentioned previously, a higher discharge temperature will degrade the volume efficiency and increase the power consumption, and reduce the viscosity of lubricating oil simultaneously. The discharge temperature of the IHXVI system could decrease from 145°C to 128°C, enhancing the reliability of the compressor.

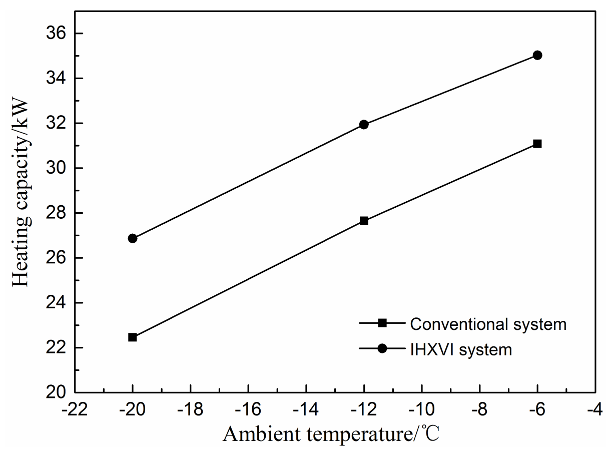

Figure 8: The comparison of heating capacity at various ambient temperatures

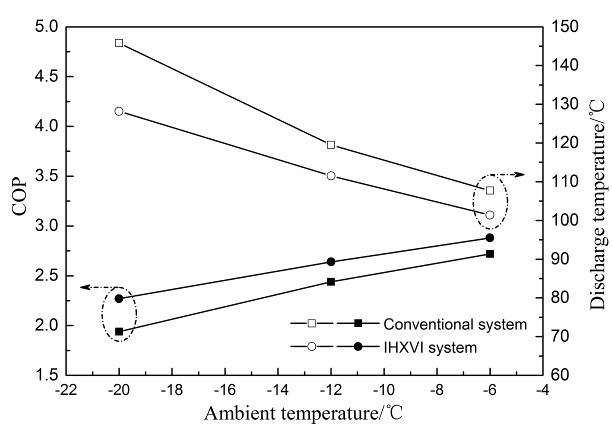

Figure 9: The comparison of COP and discharge temperature at various ambient temperatures

4. CONCLUSIONS

The conventional air-source heat pump with vapor injection under various ambient temperatures is experimentally studied in this paper. The effect of vapor injection on the promotion of heating performance and the comparison with the conventional system are presented in detail. In addition, the effect of the vapor injection on the expansion valve operating reliability is also conducted. The experimental results could be summarized as follows:

(1)Under the working condition of same intermediate pressure, the injection ratio increases with the decrease of ambient temperature.

(2)The heating capacity and COP change with the ambient temperature and vapor injection. There exists an optimal value of injected refrigerant mass flow rate, corresponding to the peak point of each COP curve. Meanwhile, the peak points will shift to the left as the ambient temperature decreases.
(3)According to the experiment, the IHX provides maximum 29°C sub-cooling of the main stream refrigerant under all tested working conditions. Therefore, the effect of the IHX on the EV operating reliability is also important for the stability of the system.
(4)With variation of the ambient temperature, the heating capacity, discharge temperature and COP of the IHXVI system are compared with those of the conventional heat pump system. It could be interpreted that the system COP is maximized at the optimal point, however the discharge temperature degrades significantly at the same point. For the -20°C ambient temperature, the IHXVI system could have up to 17.01% improved COP and 13.73% decreased discharge temperature, respectively.

NOMENCLATURE

ASHP	air source heat pump	
COP	coefficient of performance	
c_p	specific heat capacity	($kJ \cdot kg^{-1} \cdot K^{-1}$)
EV	expansion valve	
FTVI	flash tank vapor injection	
IHXVI	internal heat exchanger vapor injection	
m	mass flow rate	($kg \cdot s^{-1}$)
GWP	global warming potential	
Q	heating capacity	(kW)
T	temperature	(℃)
W	total system consumption	(kW)
α	injection ratio	

Subscript

h	heating
in	inlet
inj	injection
out	outlet
suc	suction
w	water

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