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Performance Investigation of Refrigerant Vapor-Injection Technique for Residential Heat Pump Systems

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ABSTRACT

An 11 kW R410A heat pump system with a two-stage vapor-injected scroll compressor has been developed and tested. A conventional heat pump system with a scroll compressor having the same displacement volume to the vapor-injected scroll compressor has been tested under the same conditions to serve as a baseline. The vapor-injected scroll compressor has been tested with the cycle options of both flash tank and internal heat exchanger configurations. The results show that the vapor-injection technique can effectively increase the system performance for both high ambient cooling application and low ambient heating application. A cooling capacity gain of around 14% with 4% COP improvement at ambient 46.1°C, about 30% heating capacity improvement with 20% COP gain at -17.8°C and about 7% HSPF improvement in U.S. Department of Energy's northern Region IV climate have been found for the vapor-injected R410A heat pump system as compared to the conventional system.

1 INTRODUCTION

Although the refrigerant vapor-injection technique has been well justified to improve system performance in refrigeration applications, it has not received much attention for air conditioning applications, particularly with heat pumps for cold climates until recently. The previous experimental studies for the vapor-injection systems were mainly focused on the refrigerant R22 systems. Winandy and Lebrun (2002), Ma et al. (2003) and Wang et al. (2006 and 2007) conducted a series of testing the R22 vapor-injection system in laboratories. Those systems included supplementary heat exchangers serving as subcoolers to subcool main stream refrigerants. He et al. (2006) conducted a field-testing of a R22 vapor-injection heat pump. Heo et al. (2007) tested a twin rotary type compressor with the vapor injection which was applied to a R22 heat pump system. Huang et al. (2007) conducted a field-testing of a R407C vapor-injection heat pump. Nguyen et al. (2007) addressed the control issue of the vapor-injection system. An internal heat exchanger cycle and a flash tank cycle were tested in the heating applications in their study. Nowadays, R410A has been widely recognized as a leading HFC for air-conditioners and heat pumps for replacing R22 (Beeton et al., 2002 and 2003). However, it was reported that approximately 10% performance loss was expected for R410A systems at high ambient temperatures as compared to R22 systems (Chin et al., 1999, Meurer et al., 1999 and Yana-Motta et al., 2000). The performance degradation of R410A conventional residential equipment at very high pressure-ratio operating conditions warrants further investigation of the vapor injection technique, which generally performs better with HFCs than R22. The objective of this study is to determine the performance improvement potential of a R410A heat pump system with a vapor-injected compressor.

2 EXPERIMENTAL SETUP

For this study, a commercially available residential heat pump system was used. The heat pump was originally equipped with a conventional scroll compressor, having a displacement volume of 30.69cc per revolution. Indoor air enthalpy method along with refrigerant enthalpy method was applied to measure the capacity and the COP of the heat pump system based on ASHRAE Standard 116 (1995). The original heat pump was tested to establish a

baseline. The conventional scroll compressor was replaced by a vapor-injected scroll compressor having the same displacement volume after the baseline tests. The system was modified correspondingly to a two-stage vapor-injection system to conduct the vapor-injection tests. The vapor-injection cycle options of both flash tank (FTC) and internal heat exchanger (IHXC) configurations were investigated. All systems were tested under the same ambient conditions. The schematic of the experimental setup for the IHXC is illustrated in Figure 1. The heat pump system was comprised of an indoor unit and an outdoor unit. The indoor unit was mounted to a closed air loop, and the outdoor unit was installed in an environmental chamber. Both the closed loop and the environment chamber were equipped with an air handling unit and a humidifier to condition the air. A plate type heat exchanger, used as an internal heat exchanger, was installed in the environmental chamber. A manually controlled regulating valve was installed in the vapor injection line serving as an expansion valve to control the mass flow rate of the injected vapor. Two three-way valves were installed at the main stream to switch the flow direction between the cooling and heating tests, in order to secure a counter-flow heat transfer between the main stream and the injected stream at the internal heat exchanger. The schematic of the experimental setup for the FTC is illustrated in Figure 2. A manually controlled regulating valve was installed at the outlet of the outdoor heat exchanger to serve as the first stage expansion valve, and to control the mass flow rate of the injected vapor. The refrigerant liquid coming from the outdoor heat exchanger was throttled to two-phase state, and entered a vertically mounted flash tank, a 2.2 liter stainless steel cylinder, in which the two-phase refrigerant is separated to saturated vapor phase and saturated liquid phase due to gravity effect. The saturated vapor refrigerant from the top of the tank is supplied to the intermediate stage of the compressor through the injection line. The saturated liquid from the bottom of the tank is sent to the indoor unit.

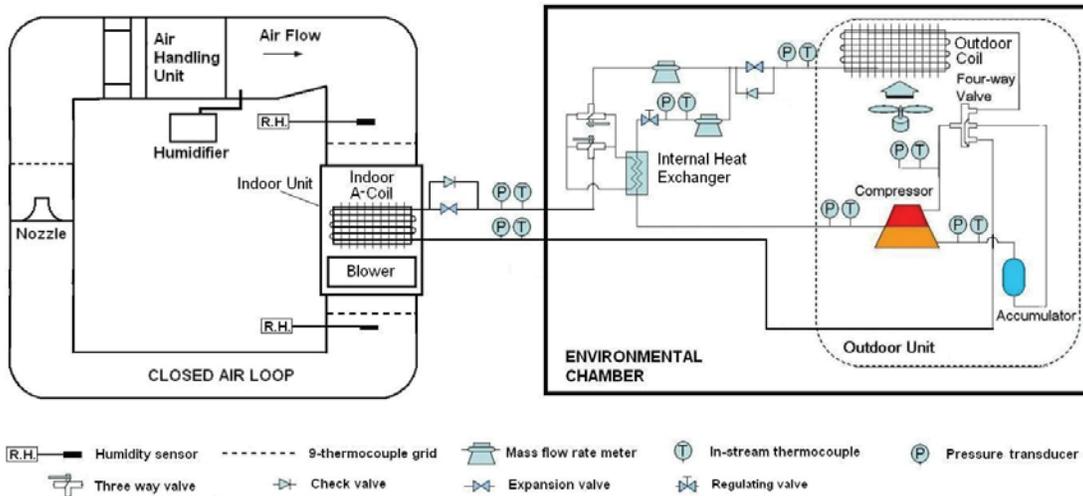


Figure 1: Schematic of the internal heat exchanger vapor-injection system

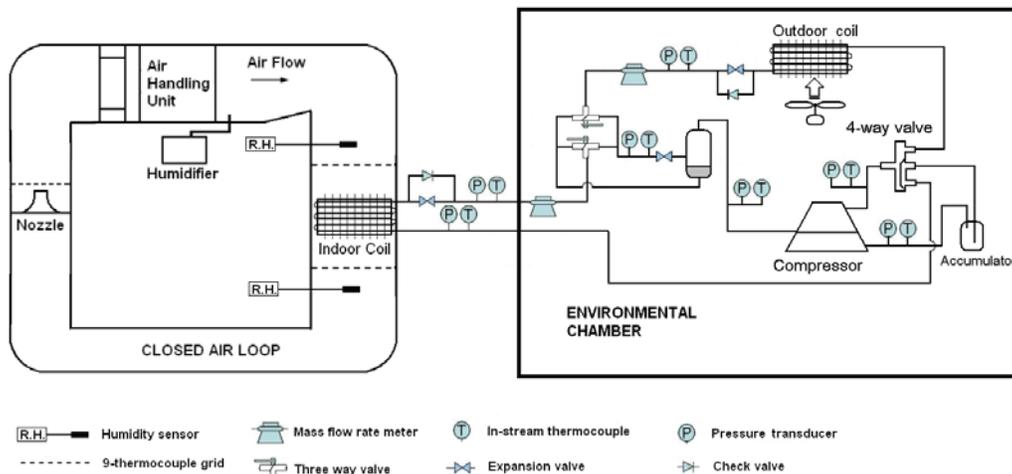


Figure 2: Schematic of the flash tank vapor-injection system

To measure the capacity and the coefficient of performance (COP) of the heat pump system, pressures, temperatures and mass flow rates were measured in both the refrigerant-side and the air-side of the system. Additionally, humidity sensors and differential pressure transducers were applied to the air side to measure the properties and the flow rate of the air circulating in the closed loop. The volume flow rate of the air in the closed loop was 0.57 m³/s. The test conditions were determined based on the steady state test conditions of the ASHRAE Standard 116 (1995). In addition, one high ambient temperature of 46.1°C and one low ambient temperature of -17.8°C were added to the test matrix for the cooling and heating tests, respectively in order to investigate the performance improvement potential of the vapor-injection system at severe weather conditions. The mass flow rate of the injected vapor was varied from 0 g/s to the maximum value, resulting in an injection of saturated vapor.

3 EXPERIMENTAL RESULTS AND DISCUSSIONS

In total, 288 tests were conducted. The energy balance error between the air-side and the refrigerant-side capacity was evaluated for all tests. The comparison of the air-side and the refrigerant-side capacities is shown in Figure 3. The energy balance error of all the tests was less than 6%, which is in compliance with the ASHRAE standard 116 (1995).

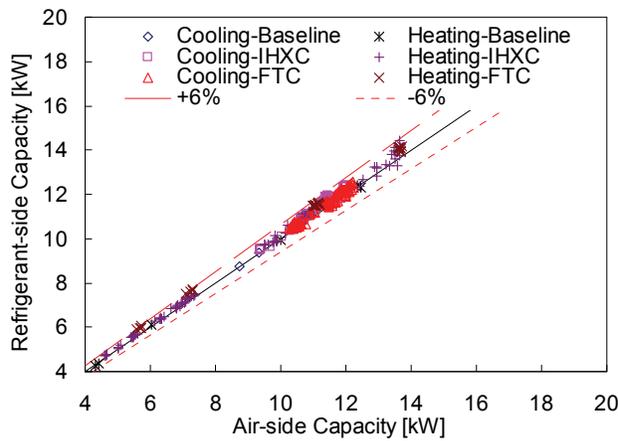


Figure 3: Comparison of air-side and refrigerant-side capacities for all performance tests

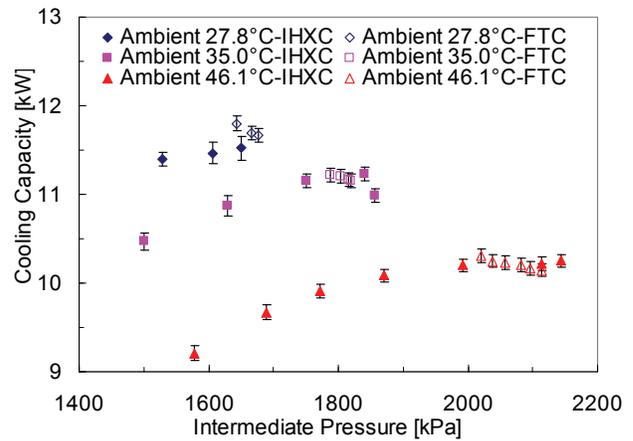


Figure 4: VI cooling capacity vs. intermediate pressure

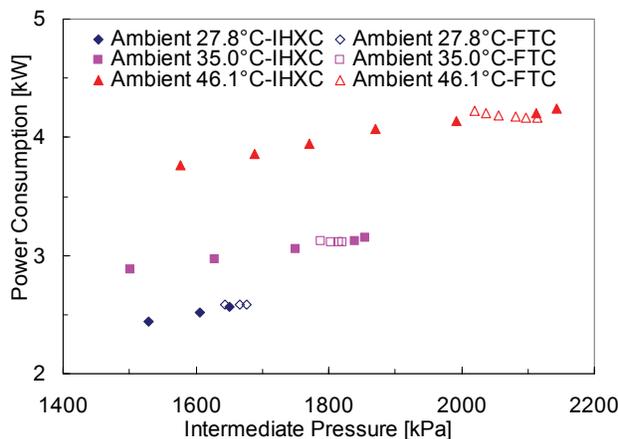


Figure 5: VI cooling power consumption vs. intermediate pressure

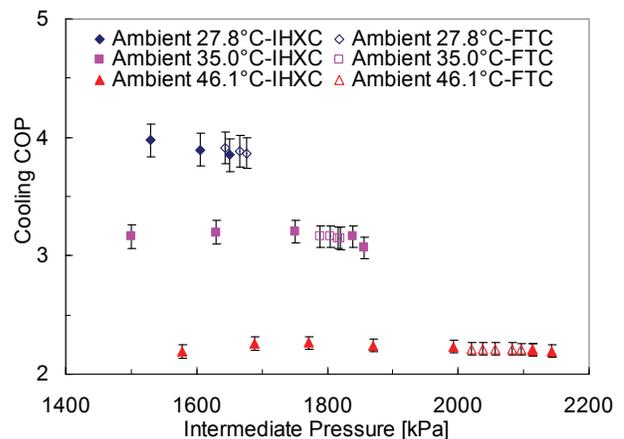


Figure 6: VI cooling COP vs. intermediate pressure

3.1 Steady-State Cooling Performance

The system capacity, the power consumption and the COP at different ambient conditions, are plotted with different intermediate pressures at the injection port, and illustrated in Figure 4 through 6. As shown in Figure 4, the cooling capacity of the IHXC increases with the intermediate pressure, but the increment is saturated at higher intermediate pressure. This is because the temperature difference between the injected vapor and the main stream refrigerant decreases as increasing the intermediate pressure, which affects the subcooling effect at the internal heat exchanger. On the other hand, the cooling capacity of the FTC decreases with increasing the intermediate pressure. This is because the refrigerant enthalpy at the evaporator inlet increases with increasing the intermediate pressure. This reduces the enthalpy span across the evaporator, which has a negative effect on the FTC capacity. As shown in Figure 5, the power consumption of the IHXC increases with the intermediate pressure for all cases. This is because the injection mass flow rate increases with the intermediate pressure. The compressor compresses extra amounts of refrigerant coming from the injection line at its higher stage. Contrary to the IHXC, the power consumption of the FTC has little change with increasing the intermediate pressure. The power consumption even gradually decreases about 1.3% with increasing the intermediate pressure at the ambient condition of 46°C. This effect can be explained by the following reasons. One reason is that the compressor compresses more refrigerant at its higher stage with increasing the intermediate pressure. This contributes to the increased power consumption. On the other hand, increasing the intermediate pressure reduces the liquid refrigerant in the condenser so that the compressor head pressure is reduced. This effect contributes to the reduced power consumption. The combination of the two effects makes the change of the compressor power consumption not obvious. Figure 6 shows that the vapor injection has a negative impact to the COPs of the IHXC and the FTC at the low ambient cooling application (ambient temperature of 27.8°C). The COPs of the IHXC and the FTC decrease about 3% and 1%, respectively, with increasing the intermediate pressure. The COPs of the IHXC at the ambient conditions of 35°C and 46.1°C do not show obvious improvement with increasing the intermediate pressure. The maximum COP change within the test points is 2%. The COP of the FTC at such condition is almost constant. This is because the increase of the power consumption somewhat diminishes the benefit of the capacity improvement. The system performance of the IHXC and the FTC in the cooling mode has been compared to the baseline system. The changes of the system capacity and the COP at different injection ratios are illustrated in Figure 7. Overall, the IHXC and the FTC show a comparable performance improvement. However, the IHXC has a wider operating range of the intermediate pressure than the FTC does. This is because the superheat of the injected vapor can be adjusted in the IHXC, but the injected vapor in the FTC is saturated. Both systems indeed improve the system cooling capacity. The higher the ambient condition is, the more capacity improvement is observed. The maximum capacity gain is 15%, associated with a 2% COP gain, at the ambient condition of 46.1°C. The COP improvement of the vapor-injection system having the same compressor displacement volume to the baseline is not very obvious. Overall, the maximum improvement is around 2~4% depending on the ambient conditions, which means that the vapor injection almost equally affects the capacity and the power consumption. The results show that this technique is more favorable for the high ambient cooling application.

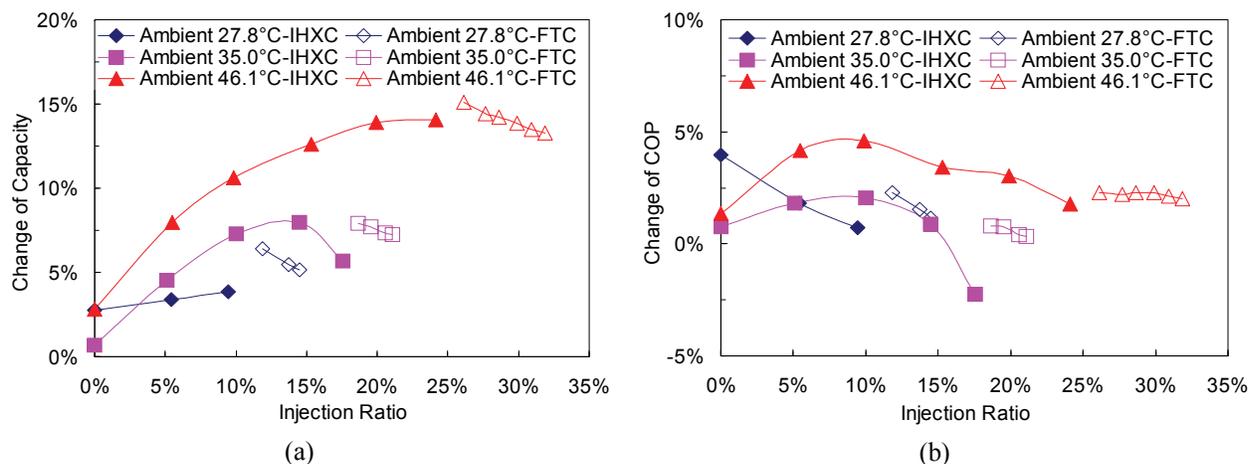


Figure 7: Comparisons of VI cooling performance to the baseline system: (a) Capacity, (b) COP

3.2 Steady-State Heating Performance

The vapor-injection effects on the system performance in the heating application are illustrated in Figure 8 through 10. For the case of the IHXC, the heating capacity and the power consumption, shown in Figure 8 and 9, respectively, increase almost the same trend when the intermediate pressure increases, which overall results in a fairly constant COP for the high ambient heating applications ($T_{amb}=16.7^{\circ}\text{C}$ and $T_{amb}=8.3^{\circ}\text{C}$) as shown in Figure 10. However, for the low ambient temperature heating applications ($T_{amb}=-8.3^{\circ}\text{C}$ and $T_{amb}=-17.8^{\circ}\text{C}$) in the same charts, the increase of the heating capacity has a greater extent than the increase of the power consumption when the intermediate pressure increases, so that the COP increases almost linearly when the intermediate pressure increases. The rises of both the heating capacity and the power consumption come from the increase of the mass flow rate of the injected vapor refrigerant with increasing the intermediate pressure. The compressor needs more power to overcome the increased mass flow rate at its higher stage. Meanwhile, the heating capacity increases with more refrigerant flowing through the condenser.

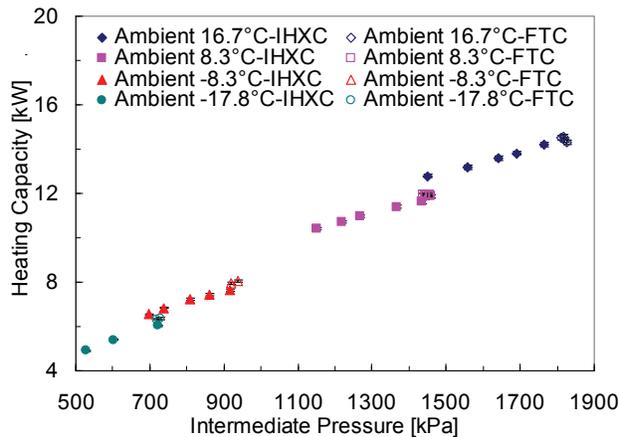


Figure 8: VI heating capacity vs. intermediate pressure

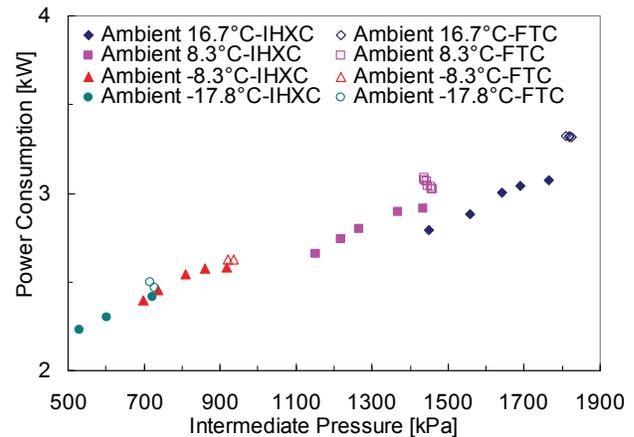


Figure 9: VI heating power consumption vs. intermediate pressure

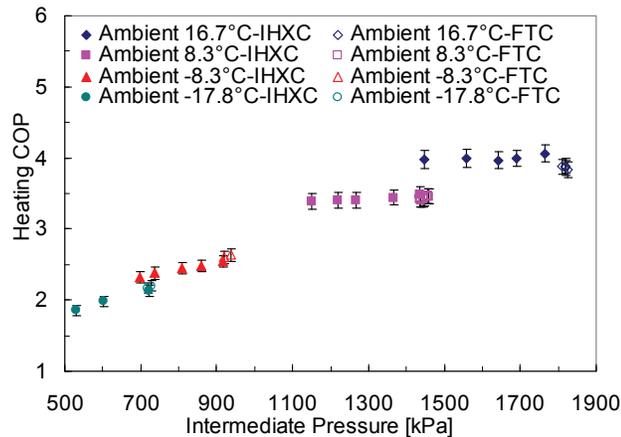


Figure 10: VI heating COP vs. intermediate pressure

The heating performance of the vapor-injection system has been compared to the baseline system. The changes of the system capacity and COP at different injection ratios are illustrated in Figure 11. The results show that the vapor-injection technique indeed improves the heating capacity significantly, and it is more favorable for the low ambient heating applications. The lower the ambient temperature is, the more capacity improvement is found. The maximum heating capacity gain varies from 13% to 33% as the ambient temperature decreases from 16.7°C to -17.8°C . The improvement of the heating COP is more significant at the low ambient conditions than that at the high ambient conditions. The maximum COP improvement is 23% for the FTC at the ambient temperature of -17.8°C . As compared to the IHXC, the FTC has very limited range of the intermediate pressure since the injected vapor is

saturated. Overall, the FTC shows better performance improvement in terms of the capacity and the COP gains at the low ambient heating than the IHXC does. This is because the intermediate pressure of the FTC is slightly higher than the IHXC, so that it injects more refrigerant into the second stage of the compressor than the IHXC does. This effect makes more refrigerant flow through the FTC's condenser, and delivers higher heating capacity than the IHXC.

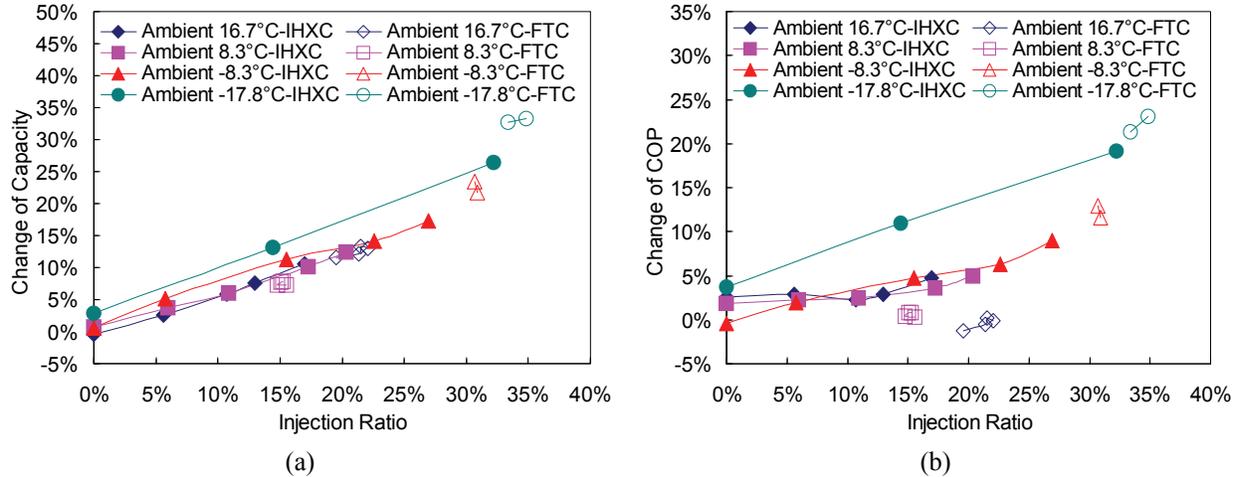


Figure 11: Comparisons of VI heating performance to the baseline system: (a) Capacity, (b) COP

3.3 Cooling and Heating Seasonal Efficiency

In this study, the seasonal energy efficiency ratio (SEER) and the heating seasonal performance factor (HSPF) of the baseline, the IHXC, and the FTC are carried out. The results of the SEER and HSPF are summarized in Table 1 and 2. The inputs for evaluating the SEER are the system cooling capacity and the power consumption at the ambient temperatures of 27.8°C and 35°C. The inputs for calculating the HSPF are the system heating capacity and the power consumption at the ambient conditions of 8.3°C, 1.7°C and -8.3°C. Those capacities and the power consumptions are evaluated from the experimental study; the capacities are converted from SI units to the required units.

Table 1: Summary of SEER results

	Ambient (°C)	Capacity (kW)	Power (kW)	Degradation Coefficient*	SEER (Btu/W-h)	SEER Improvement
Baseline	27.8	10.4	3.310	0.08	12.50	0.0%
	35.0	11.1	2.910			
IHXC ⁺	27.8	11.0	3.583		13.03	4.2%
	35.0	11.4	2.867			
IHXC	27.8	11.0	3.583		12.62	1.0%
	35.0	11.5	2.992			
FTC	27.8	11.2	3.548		12.82	2.6%
	35.0	11.8	3.018			

*A typical OEM value for cooling degradation coefficient is used in this study.
⁺ Injection port is turned on at the ambient of 35°C, and turned off at the 27.8°C.

In Table 1, it is observed that the IHXC and the FTC have little effect on improving the system SEER when the compressor injection port is open. If the baseline SEER is regarded as a base, the maximum SEER improvement is 2.6%, which is achieved by the FTC. The IHXC shows only 1% improvement. This is because the system COP at the low ambient condition (27.8°C) has to be enhanced in order to improve the SEER. However, the vapor-injection cycles have little benefit or even negative effect on the system COP at such an ambient condition. This diminishes the fact that the system performance is indeed improved at 35°C by the vapor injection. To eliminate the negative effect, another operating option is examined in the IHXC, in which the injection port is shut off at the ambient temperature of 27.8°C. In this case, the SEER of the IHXC shows 4.2% improvement as compared to the baseline.

Table 2 shows that the IHXC and the FTC can significantly improve the system HSPF. In average, about 8% improvement in the HSPF can be achieved as compared to the baseline. Overall, the SEER and the HSPF for the IHXC and the FTC are really close to each other. The difference is less than 1.6%, which is within the measurement uncertainties. Therefore, it is hard to conclude which one is particularly better than another.

Table 2: Summary of HSPF results

	Ambient (°C)	Capacity (kW)	Power (kW)	Degradation Coefficient*	HSPF**	HSPF Improvement
Baseline	8.3	10.4	3.110	0.20	8.22	0.0%
	1.7	8.0	2.930			
	-8.3	6.5	2.790			
IHXC	8.3	12.0	3.395			
	1.7	9.3	3.183			
	-8.3	7.8	3.018			
FTC	8.3	12.0	3.453			
	1.7	9.2	3.225			
	-8.3	8.0	3.048			

*A typical OEM value for heating degradation coefficient is used in this study.
 **HSPF is evaluated under the following conditions:
 The climate region is Region IV defined by U.S. Department of Energy; the demand-defrost credit is 1.03; the compressor is turned on/off at -34.4°C/-37.2°C.

4 CONCLUSIONS

In this study, the performance potential of the two-stage heat pump system with a vapor-injected scroll compressor was experimentally investigated. The FTC and the IHXC options of the two-stage vapor-injection system were explored. The experimental results show that the IHXC has a wider operating range of the intermediate pressure than the FTC due to its freedom of setting for a certain amount of superheat at the injection port. Overall, the IHXC and the FTC show a comparable performance improvement as compared to the baseline system. A cooling capacity gain of around 14% with 4% COP improvement at ambient 46.1°C and about 30% heating capacity improvement with 20% COP gain at -17.8°C and about 7% HSPF improvement in U.S. Department of Energy's northern Region IV climate were found for the vapor-injected R410A heat pump system as compared to the conventional system. It is concluded that the vapor-injection system is more favorable in the high ambient temperatures for the cooling mode and the low ambient temperatures for the heating mode.

NOMENCLATURE

COP	Coefficient of Performance
FTC	Flash Tank Cycle
HSPF	Heating Seasonal Performance Factor
IHXC	Internal Heat Exchanger Cycle
SEER	Seasonal Energy Efficiency Ratio

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