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The Performance Evaluation of R407C and R410B in a Residential Window Air-Conditioner

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Key Words : Alternative refrigerants, Parallel-cross flow, Counter-cross flow, Evaporator, Condenser, LSHX(liquid-suction heat exchanger)

Abstract

This study presents test results of a residential window air-conditioner using R22 and two potential alternative refrigerants, R407C and R410B. A series of performance tests has been carried out for the basic and liquid-suction heat exchange cycles in a psychrometric calorimeter test facility. For R407C, the same rotary compressor was used as in the R22 system. However, compressor for the R410B system was modified to provide the similar cooling capacity. The evaporator circuit was changed to get a counter-cross flow heat exchanger to take advantage of zeotropic mixture's temperature glide, and liquid-suction heat exchange cycle was also considered to improve the system performance. Test results were compared with those for the basic R22 system. The modified system with a liquid-suction heat exchanger increased cooling capacity and energy efficiency by up to 5%.

1. Introduction

HCFC refrigerant, R22 has been predominantly used as a working fluid in air-conditioning and heat pump applications. Because of increasing concerns about environmental effects of chlorine-containing refrigerants on the earth, the Montreal Protocol established a schedule for regulation of CFC and HCFC substances, including R22. The production and use of CFC was prohibited and HCFC was also started to be regulated in the developed countries from the beginning of 1996. The elimination schedule of HCFC was accelerated in the developed countries and some countries, particularly in Europe, domestic regulations were made to prohibit the use of HCFC in the new system from the year 2000.[1] The refrigerant manufacturers concentrate their effort to find suitable alternative refrigerants, and

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refrigeration and air-conditioner manufacturing industries put spurs to develop system design technologies applying new refrigerants.

The most promising alternatives to R22 seem to be R407C (R32/125/134a : 23/25/52 wt. %) as a short term replacement and R410A (R32/125 : 50/50 wt. %) or R410B (R32/125 : 45/55 wt. %) as a mid or long term replacement. R407C has similar thermo-physical properties with those of R22 and does not require significant system modification. However, R410A or R-410B is a higher pressure refrigerant whose discharge pressure is 50~60% higher than that of R-22, and hence significant design changes including the compressor are needed.

There are several studies on the system performance evaluation of R22 alternative refrigerants in air-conditioning and heat pump applications.[2~15] Domanski and Didion[2] performed theoretical evaluation of R22 and R502 alternative refrigerants. Air-conditioning and Refrigeration Institute (ARI) in the United States constituted R22 Alternative Refrigerant Evaluation Programs (AREP) and evaluated characteristics of R22 alternative refrigerants such as thermodynamic properties, heat transfer, compressor and system performance.[3,4] Torikoshi et al.[5] analyzed the characteristics of heat exchanger using R407C, and Murphy et al.[6] investigated the system performance and cycle characteristics of R407C and R410A in a residential air conditioner. Hwang et al.[7] measured system performance of R407C, R410A and R410B in a heat pump with variation of the refrigerant charge amount and compared the results with those of R22. Nonaka et al.[8] performed a series of tests with variation of heat exchanger circuits and refrigerant charge amount using R407C in a residential air-conditioner, and compared the results with that of R22. Sano et al.[9] analyzed characteristics of refrigeration cycle using R410B systematically. Bivens and Dension[10] performed a study on the possibility of performance improvement of air-conditioning system using R407C and R410A. Kim[11,12] performed experimental study using R22 alternative refrigerants R134a, R407C, R410A and R410B in a heat pump test rig and a household air-conditioner to analyze the characteristics of system and heat exchanger. Burns et al.[13] reported the results of case study related with manufacturing R410A split air-conditioner and Vakil[14] studied the system characteristics with liquid-suction heat exchanger using pure refrigerants and refrigerant mixtures. Domanski et al.[15] analyzed theoretical characteristics using liquid-suction heat exchanger (LSHX) in vapor compression system. Based on the literature reviews, we may conclude that it is necessary to improve the system performance in case of R407C and redesign the system for high pressure refrigerants, R410A and R410B.

In this study, R407C and R410B were selected as R22 alternatives and charge optimization tests for each refrigerant were performed using a residential window system air-conditioner. The circuit optimization for the evaporator and liquid-suction heat exchange cycle were considered to solve the possible degradation of system performance when using alternative refrigerants. Soft optimization test results were compared with those of the R22 basic system. The refrigerant thermodynamic properties necessary in analyzing experimental results are calculated using NIST REFPROP.[16]

2. Experiment

2.1 Experimental unit

The basic unit in the study is a residential window air conditioner which has a rated cooling capacity of 4.13 kW. The unit is composed of the basic elements of a vapor compression system such as a rotary compressor, an expansion device, an evaporator, a condenser, and other components such as an accumulator and fans. The basic specifications of the experimental unit are described in Table 1, and Figure 1 shows a schematic of the instrumented experimental unit.

The compressor of the basic unit is a hermetic vane rotary compressor. For R22 and R407C system, the basic model compressor is used without modification. Specific volume of high pressure refrigerant R410B is smaller by 30% and latent heat is higher by 6% than R22, and its volumetric capacity becomes 30% larger than R22. Consequently, when using R22 compressor in the R410B system, it is

Table 1 Basic specifications of the experimental unit

Items	Specifications
Rated cooling capacity	4.13kW
Refrigerant/oil	R22/Mineral oil(VG56)
Compressor	Hermetic rotary vane compressor(1.5HP, 19.35cc)
Evaporator	2-row 14 tubes, 2 circuits (370×375×25.4mm)
Condenser	2-row 16 tubes, 1 circuit (540×400×43.3mm)
Expansion device	Capillary tube
Indoor fan	Sirocco fan
Outdoor fan	Propeller fan

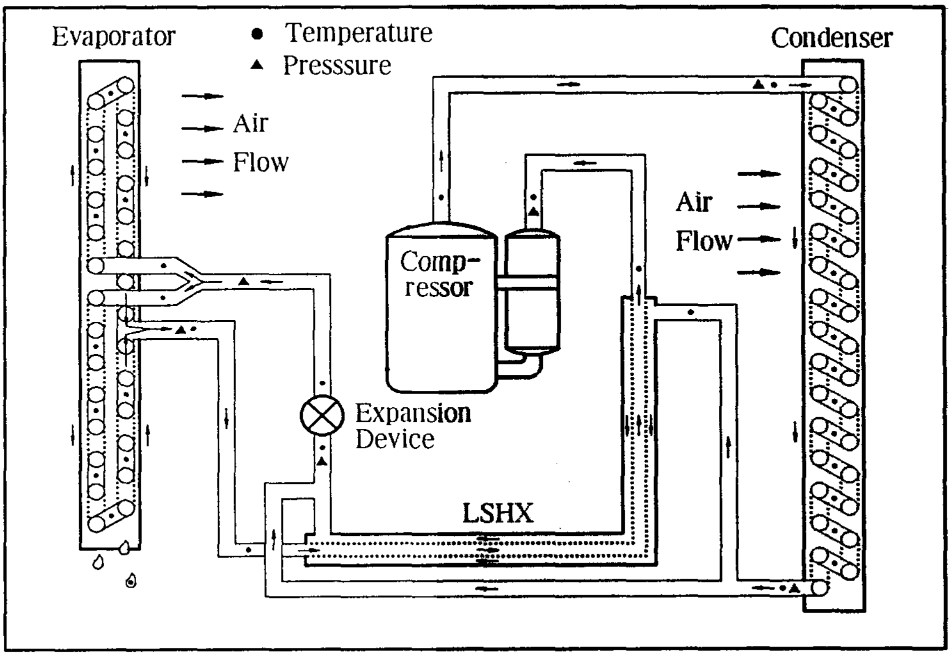

Fig. 1 Schematic diagram for the experimental unit

impossible to compare system characteristics at the same capacity with basic model. For the R410B system, therefore, the compressor is replaced with a 28% smaller displacement compressor.

Heat exchangers are finned-tube type used in a conventional residential air-conditioner. The evaporator is 2-row 14 tubes with two circuits and the condenser is 2-row 16 tubes with one circuit. The parallel-cross flow and counter-cross flow evaporators were considered to investigate the effect of temperature glide for a zeotropic refrigerant mixture on the system performance. The expansion device was used by connecting a capillary tube and needle valve in series to control superheating and subcooling with refrigerant charge amount for each refrigerant. Indoor and outdoor fans are sirocco and propeller fans, respectively and operated by one motor.

The liquid-suction heat exchanger (LSHX) is used to improve the system performance by exchanging heat between cold suction vapor at the evaporator outlet and high temperature liquid refrigerant leaving the condenser, and to prevent liquid refrigerant from entering a compressor. The LSHX has a pure counter-flow configuration where the cold suction vapor is piped through the inner tube of the heat exchanger in counter flow to the hot liquid refrigerant leaving the condenser. Inner tube of LSHX is plain tube and inner diameter is selected to have the same cross sectional area with that of suction line tube of basic model. Outer surface of inner tube is equipped with fins to improve heat transfer with hot refrigerant of liquid-line. The length of LSHX is determined to be 400mm, based on the experimental result for the LSHX capacity. The by-pass valves are installed at the LSHX inlet and outlet. By controlling the valves, the hot liquid refrigerant flows in the annular space of the LSHX or by-passes the LSHX. Heat loss from the LSHX is minimized by insulating it from the surroundings.

2.2 Experimental apparatus

The performance tests are carried out in a psychrometric calorimeter according to Korea Standards (KS B-6369). The calorimeter consists of two test chambers : one is an indoor condition test chamber and the other an outdoor condition test chamber. The desired test conditions in the chambers can be maintained within the tolerances of ±0.1℃ (Korea Standards; dry-bulb ±1.0℃, wet-bulb ±0.5℃) for wet-bulb and dry-bulb temperatures of air entering the test unit. The cooling capacity is calculated by the air enthalpy method in which wet-bulb and dry-bulb temperatures of air entering and leaving the unit, and the associated air flow rate are measured. The refrigerant temperatures and pressures are measured using forty two T-type thermocouples mounted on the surface of the tubes insulated from the ambient and six pressure transducers connected to the static pressure taps. Experimental equipment has reliability within accuracy of 3% because an accuracy of air flow is ±1% and uncertainty on thermodynamic properties of air is within 0.05%.

Experimental data are collected using hybrid recorder for the temperatures and pressures, and power meter for the power consumption, and they are recorded in a personal computer through GPIB.

2.3 Experimental conditions and method

The experiments are performed according to the standard cooling test conditions of the Korea Standards, KS B-6369. The dry-bulb/ wet-bulb temperatures for the indoor and outdoor are 27/19.5 and 35/24 °C, respectively. The baseline tests are first performed using R22 and mineral lubricant (VG56) without any system modifications. After the baseline tests, the expansion device (capillary tube) is changed to an expansion valve and a capillary tube connected in series. The expansion valve is capable of metering the refrigerant flow with a vernier handle and allows accurate control of superheat at the evaporator outlet for each refrigerant. The evaporator coil is also changed to a counter-cross-flow arrangement and the LSHX is installed in the unit. Then, the performance tests for R22 and R407C with a polyol ester lubricant (VG68) are carried out with and without LSHX at the refrigerant charge amount of 800, 900 and 1,000g. For each refrigerant charge, the expansion valve is adjusted for the superheating and subcooling to be 5±1℃ and 10±2℃, respectively. A part of experiments are performed with refrigerant charge over 1000g, but the system efficiency decreases and evaporator wall temperature for R407C decreases below regulation temperature. Therefore, experimental data are collected under refrigerant charge of 1,000g. Finally, the same tests are made for R410B system with a smaller displacement compressor.

3. Results and discussion

The optimization test results for R407C and R410B in a residential window air-conditioner are depicted in Figs. 2~11, compared to those for R22. Figure 2 shows the cooling capacity and energy efficiency ratio of R22 and R407C system with counter-cross flow and parallel-cross flow evaporator. The performance of R22 system with counter-cross flow evaporator is similar to that with parallel-cross flow evaporator. However, the cooling capacity and energy efficiency ratio of R407C system with counter-cross flow evaporator increase by 2.3% and 4.1%, respectively, compared to those with parallel-cross flow evaporator. This is basically caused by the refrigerant characteristics. R407C is a zeotropic refrigerant mixture and has temperature glide of 4~5°C during evaporation. Therefore, heat transfer performance can be improved when the evaporator circuit is designed for refrigerant to be counter-cross flow with air.

Figure 3 shows the evaporator temperature profiles for different evaporator circuits. Temperature glide for R407C decreases by pre-

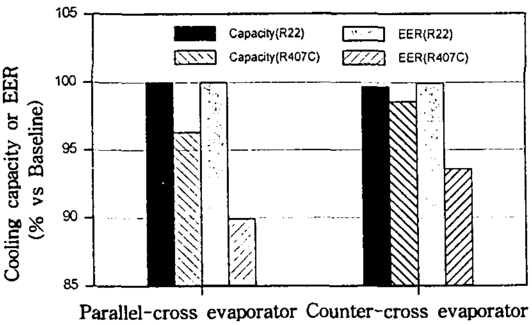

Fig. 2 System performance for different evaporator circuits(Baseline : R22 with parallel-cross evaporator)

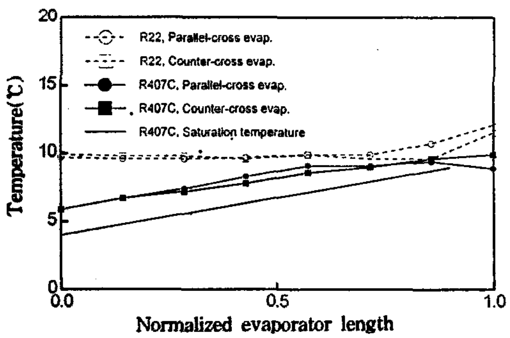

Fig. 3 Evaporator wall temperature profiles for different evaporator circuits

ssure drop, compared with theoretical value due to composition change during evaporation. Therefore average evaporating temperature of R407C is lower than that of R22. It may be one factor for increasing the cooling capacity and decreasing energy efficiency ratio of the R407C system. Decreasing evaporator wall temperature increases temperature difference between evaporator inlet air and evaporator surface temperatures and consequently cooling capacity increases. Temperature inequality for each pass at the evaporator outlets is due to the difference of pressure drop in each pass and partially caused by irregularity of air flow through the evaporator.

Figure 4 shows condenser wall temperature distributions. Temperature glides for R22 and R407C in two phase region are 3°C and 5°C, respectively. In contrary with the case of evaporator, temperature glide for R407C increases by refrigerant-side pressure drop, compared with that during condensation under constant pressure.

Figures 5 and 6 present the cooling capacity and energy efficiency ratio, respectively, for the system with and without LSHX. The system capacity and energy efficiency ratio increase with the refrigerant charge. All the three refrigerants show the highest energy efficiency at refrigerant charge of 1,000g. With the refrigerant charge over 1,000g, the energy efficiency decreases even though the cooling capacity increases, and evaporator wall temperature, particularly for R407C, decreases below 6°C locally. In general, household air-conditioning system is designed so that evaporating temperature is over pre-determined value (about 6°C) for the standard cooling test condition. Refrigerant charge over 1,000g might lower evaporating temperature below

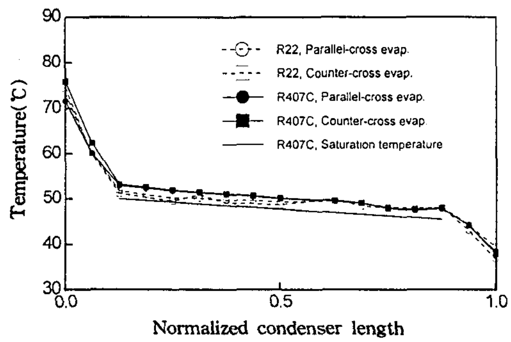

Fig. 4 Condenser wall temperature profiles for different evaporator circuits

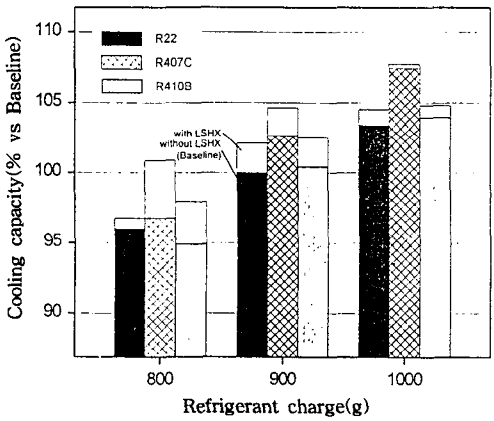

Fig. 5 Cooling capacity with and without LSHX

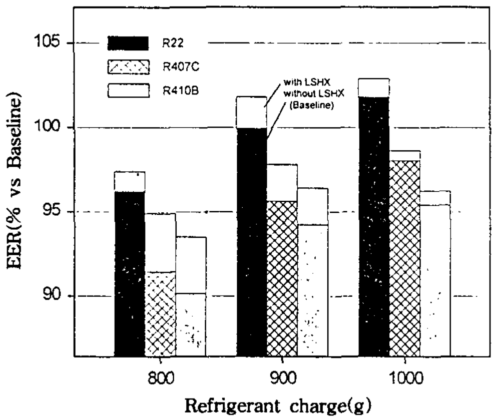

Fig. 6 Energy efficiency ratio with and without LSHX

0°C and frost could be accumulates on the evaporator surface at the low temperature cooling test condition, and hence amount of refrigerant charge should be smaller than 1,000g in the system. In the case of basic cycle, the cooling capacity and energy efficiency for the R407C system are 1~4% higher and 5% lower than those for the R22 system, respectively. It is originated from that the average evaporating temperature for R407C is low and the compression ratio is high, compared with those for R22. For R410B, the cooling capacity is equal to that of R22 and the energy efficiency ratio is 7% lower than R22. When using LSHX, the system performance is improved for all the selected refrigerants; the cooling capacities are increased 1~2% for R22, 0~5% for R407C, and 1~3% for R410B, and the improvements of EER are 2% for R22 and 1~3% for R407C and R410B. These results are different from the theoretical interpretation by Domanski et al.[14], and it is partially because the actual system used in this study is different from the inverse Rankine cycle used by Domanski et al. and system performance depends on the system and operating condition.

Figures 7 and 8 show the suction and discharge pressures for the basic cycle with variation of refrigerant charge amount. The suction pressure for R407C is similar to that for R-22 and discharge pressure is 10~20 % higher than that for R22. However, the suction and discharge pressures for R410B are 155~160% of those for R22. These results are similar with the estimation using NIST REFPROP.[16] For high pressure refrigerant R410B, significant design change should be

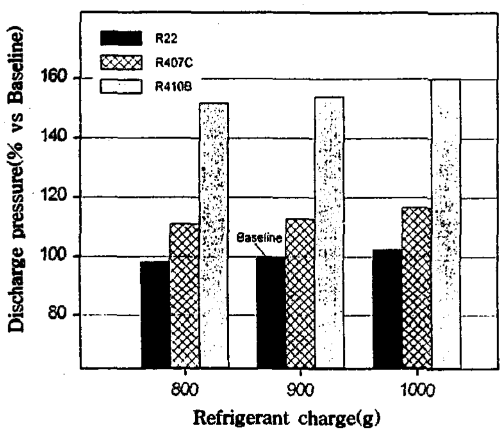

Fig. 8 Discharge pressure for different refrigerants

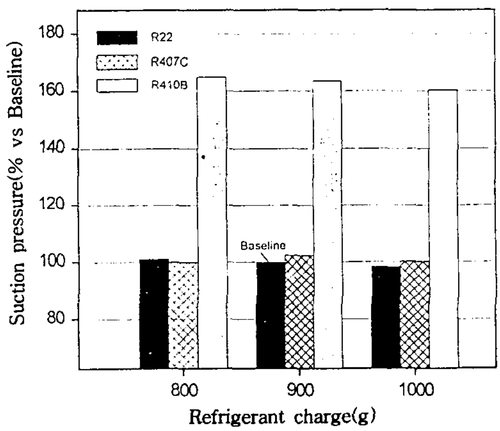

Fig. 7 Suction pressure for different refrigerants

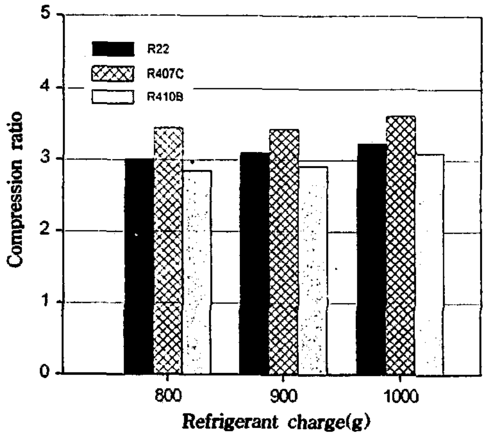

Fig. 9 Compression ratio for different refrigerants

required for the high side components including the compressor. However, because the standard for the burst pressure is mitigated about 40% and the fatigue test is added, systematic investigations for the reliability should be needed.

Figure 9 depicts compression ratio for all the test refrigerants with variation of refrigerant charge. It is observed that compression ratio increases with refrigerant charge amount. In a vapor compression cycle, refrigerant mass flow increases with refrigerant charge amount and refrigerant mass flow discharged from the compressor becomes larger than that passed through the expansion device, and consequently, subcooling at the outlet of condenser and discharge pressure increase. The compression ratios for R407C and R410B are 11% higher and 6% lower than that of R22, respectively and this tendency is good agreement with the theoretical results calculated using refrigerant thermodynamic properties at the same conditions.[17] For R407C, compression ratio to maintain the same condensation and evaporation temperatures is higher than R22, because operating pressures are similar to those of R22 but the slope of vapor pressure curve is larger than that of R22. In the case of R410B, operation pressures are much higher than those of R22 but compression ratio corresponding to condensation and evaporation temperatures is lower than that of R22. This result indicated that the optimization of the compressor design should be needed, appropriate to the refrigerant characteristics, because it affects the reliability and power consumption of the compressor and the system efficiency directly.

Figure 10 and 11 show tube wall temperature distributions in the evaporator and condenser with and without LSHX for the refrigerant charge of 900g. Evaporating temperature profiles for R22 and R410B systems show very similar characteristics and the average refrigerant temperature is maintained as about 9°C constantly. R407C has temperature glide of about 4°C as in the results for evaporator circuit design change from parallel-cross flow to counter-cross flow, and it has general tendency for the average temperatures of the evaporator to decrease a little by using LSHX. For the condenser, all the three refrigerants have the similar temperature pro-

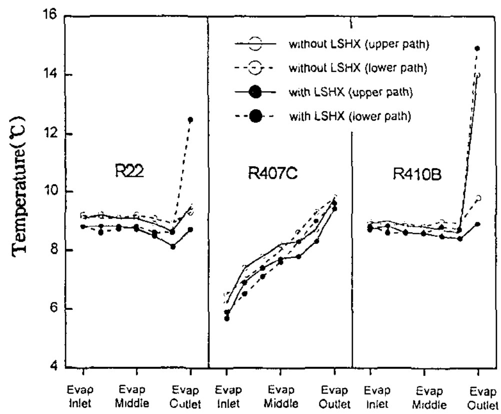

Fig. 10 Evaporator wall temperature profiles with and without LSHX

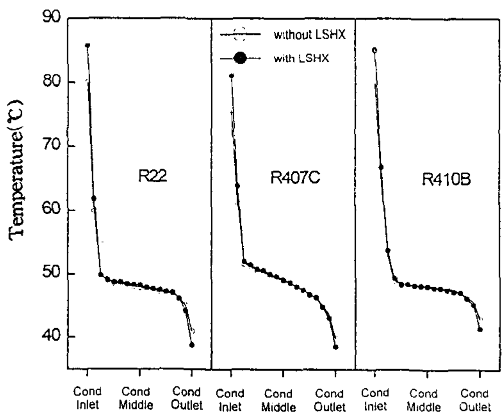

Fig. 11 Condenser wall temperature profiles with and without LSHX

files and temperature glide appears in case of R407C due to the characteristics of zeotropic refrigerant mixture. When using LSHX, it is observed that the condenser inlet temperature gets higher because the compressor discharge temperature increases by the increase of suction temperature at the compressor inlet, but other parts have very similar temperature profiles.

4. Concluding remarks

The performance evaluation of R407C and R410B as replacements for R22 were investigated in a 4.13 kW household window air conditioning unit. The modification of evaporator coil to the cross-counter flow configuration increased the cooling capacity and energy efficiency ratio of R407C by 2.3 and 4.1%, respectively. When using LSHX, the cooling capacities were increased 1~2% for R22, 0~5% for R407C, and 1~3% for R410B, and the improvements of energy efficiency ratio were 2% for R22 and 1~3% for R407C and R410B. The performance was improved for all the three refrigerants and R407C indicated the highest performance improvement by applying LSHX. The performance improvements by using LSHX depended on the refrigerant characteristics and system operating conditions. Therefore, the further study on the selection of LSHX capacity and appropriate operating condition is needed. After the soft optimization with R407C, the cooling capacity and energy efficiency ratio were 105% and 98% of the R22 baseline, respectively and the optimal charge amount of refrigerant was 1,000g which was increased by 100g compared with 900g of basic model. The important operating conditions of R407C system, such as the evaporating and condensing pressures and the compressor discharge temperature, did not significantly deviate from those of R22 system. The near-azeotrope R410B, however, had a 57% higher discharge pressure than that of R22. The cooling capacity and energy efficiency ratio for R410B system with the 28% smaller compressor were 103% and 96% of the R22 baseline, respectively.

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