ICARMA/ARI: GLOBAL REFRIGERANT ENVIRONMENTAL EVALUATION NETWORK PROGRAM

COMPARISON OF HYDROCARBON R-290 AND TWO HFC BLENDS R-404A AND R-410A FOR LOW TEMPERATURE REFRIGERATION APPLICATIONS

Final Report

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ABSTRACT

In order to conclusively establish the inherent relative performance potential of HFC's (R-404A and R-410A) as compared to R-290 for low temperature commercial refrigeration, each refrigerant was tested in a 4 kW capacity refrigeration system at a saturated evaporating temperature of -29^{o}C. Each test was conducted with an adjustable speed scroll compressor sized and operated for equal system capacity. Compressor capacity was controlled by an inverter which regulated the compressor speed. Since the scroll compressor used was initially optimized for R-404A, compressor efficiency was measured and normalized in the data reduction in order to compare inherent refrigerant performance.

The raw test data, without normalization, showed a tested COP for R-410A 9% less than R-290, and a COP for R-404A 10% less than R-290 under the full load test conditions. Under the part load test conditions, the tested COP of R-410A was 3% less than R-290 and the COP of R-404A 5% less than R-290.

COPs were adjusted for equal compressor efficiency to effectively determine the inherent relative performance of the refrigerants. These normalized results show a COP for R-410A that is equal to that of R-290, and a COP for R-404A that is 10% less than that of R-290 under the full load conditions. Under the part load conditions, the normalized COP of the R-410A system is 1% less than that of R-290, and the COP of R-404A is 5% less than that of R-290.

The environmental impact of refrigerants over the entire life cycle of fluid and equipment, including power consumption, is captured in the life cycle climate performance (LCCP) value. The lower the value, the lower the environmental impact. In this report the LCCP of hydrocarbon R-290 and the two HFC blends, R-410A and R-404A, were evaluated for the 4 kW low temperature refrigeration system. Major findings of the LCCP comparison are: The LCCPs of R-404A and R-410A are 10% and 2% higher, respectively, than that of R-290 for tested systems based on the equal compressor efficiency when the annual leakage rate is assumed to be 2%. On an equal compressor efficiency and first cost basis, the LCCPs of R-404A and R-410A are 1% higher and 6% lower, respectively, than that of R-290 for the same 2% annual leakage rate. The underlying assumption is that the first cost of the R-290 system may be, for example, 10% higher due to added safety features, and on an equal first cost basis, the HFC systems would employ the additional cost for brushless DC motors used both for the condensing unit and the unit cooler. Comparisons for other leakage rates are reported also.

EXECUTIVE SUMMARY

There is continued growing environmental awareness at the international level with particular focus on the working fluids of refrigeration systems, heat pumps and air conditioners. Worldwide governmental policy efforts to reduce global warming are directing industry to develop innovative technologies to reduce emissions while also increasing energy efficiency.

Despite the flammability of hydrocarbons, some refrigerator manufacturers especially in European countries and Asian countries have started employing hydrocarbons as refrigerants predominantly in small capacity equipment. These issues have led to call for the careful investigation of currently used refrigerants (HFC's) and potentially applicable HC refrigerants (R-290). To help provide a clear understanding of the relative performance potential of HFC's (R-404A and R-410A) as compared to R-290 for low temperature commercial refrigeration, CEEE conducted an experimental evaluation program under ARI/ICARMA's GREEN Program.

In order to test the performance of three refrigerants for low temperature commercial refrigeration, the experimental facility which was designed and fabricated by CEEE was used for this study. A 4 kW capacity refrigeration system consisting of a unit cooler and a condensing unit, which was originally designed for R-404A, served as the test unit. To match the capacity between refrigerants, compressors having a 19% smaller and 19% larger displacement volume than that for R-404A were selected for R-410A and R-290, respectively from the production compressors. Since the selected displacement volume of the R-410A and R-290 compressor was slightly different from the target displacement, 54 Hz and 57 Hz was used to match the refrigeration capacity by using an inverter drive, respectively for R-410A and R-290 system. In order to know the effects of receiver, the test was conducted with and without a receiver for R-404A and R-410A systems. However, the test for R-290 system was conducted only without the receiver because of the safety reasons to minimize the charge of R-290. The condenser was also modified to integrate a liquid subcooler circuit as a part of the condenser. Based on the optimization of the condenser, a two circuit condenser was used for the testing of R-410A while a three circuit condenser was used for the testing of R-404A and R-290 systems. The air-side configurations and specifications of all condensers were identical.

Charge optimization tests of three refrigerants systems were completed at the full load conditions. Results show that the optimum charge of R-404A was 4.4 kg while the optimum charge of R-410A and R-290 was 89% and 39% of R-404A charge, respectively. Once the refrigerant charge was optimized, each refrigerant was tested both under the full load and part load conditions. Based on equal system capacity test results, the COPs of R-404A and R-410A were 10% and 9% lower, respectively, than that of R-290 under the full load conditions, and they were 5% and 3% lower, respectively, under the part load conditions. COPs were then adjusted for equal compressor efficiency to effectively determine the inherent relative performance of the refrigerants. In and of itself this assumption is not unrealistic as this has been shown to be the case with scroll compressors at other temperature ranges, when appropriately optimized. According to the compressor manufacturer, the R-410A compressor used for the present tests was neither well optimized for the low temperature refrigeration nor representative of future production designs. Based on the same compressor efficiency assumption, the comparison shows that the COPs of R-404A are 10% and 5% lower, respectively, under the full load and part load test conditions as compared to R-290 and the COPs of R-410A are essentially the same with that of R-290 for both test conditions. When the same cost increase of 10% for R-290 to meet safety requirement, is used for the two HFC blends to employ brushless DC (BLDC) motors for the fans to enhance the efficiency, the system simulation results of employing BLDC motors for

show a 8% COP enhancement for both HFC blends under both the full load and part load conditions as compared to the tested system case. In practice, condensing units with HC refrigerants would be used in secondary loop systems. The secondary loop system may require additional cost and energy penalties due to the additional heat exchanger and pumping requirements and the use of heat transfer fluids. When the R-290 employs the secondary loop, the system simulation results shows a 8% to 12% COP enhancement for both HFC blends under both the full load and part load conditions as compared to the tested system case.

Table 1: Comparison of COP (Baseline: R-290 for each case)

Case	Refrigerant	COP Ratio (Full Load)	COP Ratio (Part Load)
Based on the test data	R-404A/R-290	0.90	0.95
	R-410A/R-290	0.91	0.97
Test data corrected based on equal compressor efficiency	R-404A/R-290	0.90	0.95
	R-410A/R-290	1.00	0.99
Simulated results based on the secondary loop R-290 and equal compressor efficiency	R-404A/R-290	0.97	1.05
	R-410A/R-290	1.08	1.11
Simulated results based on equal first cost and compressor efficiency	R-404A/R-290	0.97	1.03
	R-410A/R-290	1.08	1.07

In order to determine the environmental impact of the refrigerants investigated, an LCCP analysis was conducted. To compare the LCCP, it is assumed that the same cost increase of 10% for R-290 to meet safety requirement, is used for the two HFC blends to employ brushless DC motors (BLDC) for the fans to enhance the efficiency. In order to compare the refrigerants at optimum hardware condition, the LCCP of three refrigerants was computed based on the equal compressor efficiency with that measured for R-404A and the same 2% annual leakage rate. Then the LCCP analysis shows that R-404A and R-410A have 10% and 2% higher LCCP, respectively, than that of R-290 when the tested system is considered for all three refrigerants. When the BLDC motors are employed for only HFC systems, the LCCPs of R-404A and R-410A are 1% higher and 6% lower, respectively, than that of R-290. When HFC systems with the BLDC motors are compared with R-290 with the secondary loop, the LCCPs of R-404A and R-410A are 10% and 16% lower, respectively, than that of R-290. Furthermore, it is very clear from these results that the indirect contributions dominate any contributions from refrigerant emissions.

Table 2: Comparison of LCCP (Baseline: R-290)

Unit: CO_2 Ton	Indirect	Direct	Total	Compared to R-290
R-404A with system tested	205.7	7.5	213.2	110%
R-410A with system tested	194.6	3.5	198.1	102%
R-290 with system tested	194.3	0.0	194.3	100%
Unit: CO_2 Ton	Indirect	Direct	Total	Compared to R-290
R-404A with BLDC motors	187.8	7.5	195.3	101%
R-410A with BLDC motors	178.6	3.5	182.1	94%
R-290 with secondary loop	216.1	0.0	216.1	111%
R-290 with safety features	194.3	0.0	194.3	100%

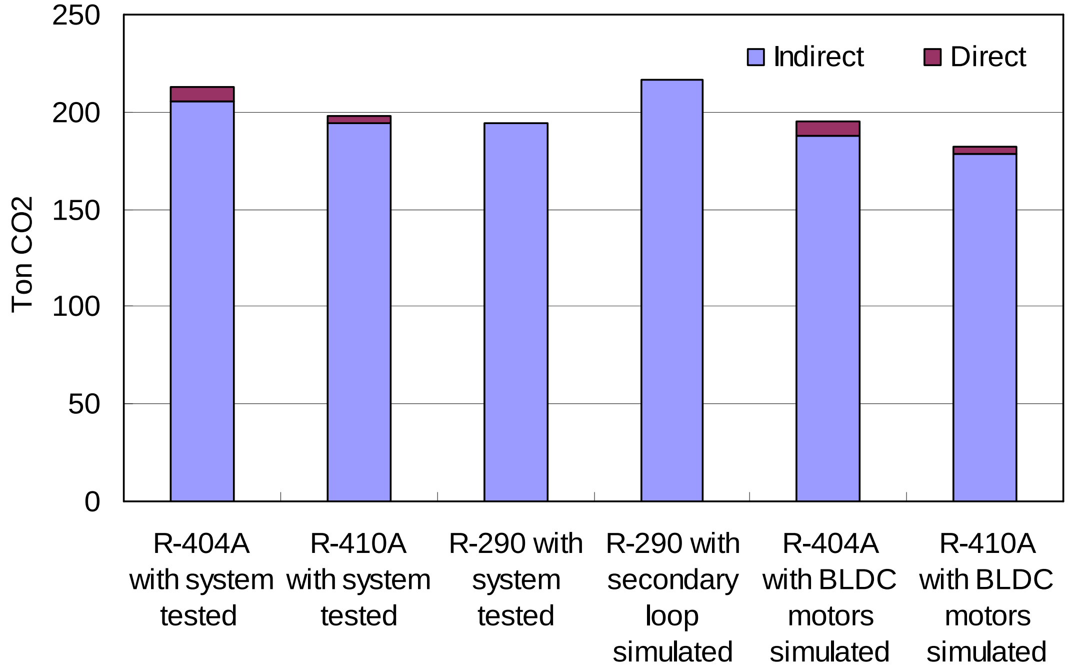

Figure 1: Comparison of LCCP

Working fluid selection should consider many aspects including safety (toxicity and flammability), environmental impact (stratospheric ozone and climate change), cost and performance (capacity and COP). The two most representative commercial refrigeration configurations are the direct expansion and distributed systems, either of which could potentially release the refrigerant into human occupied space. Therefore, the use of either flammable or high toxicity refrigerants is not feasible. To limit these cases, potentially hazardous refrigerants should be limited to unoccupied spaces.

ACKNOWLEDGEMENTS

This work was sponsored by the Air-conditioning and Refrigeration Institute (ARI) under International Council of Air-Conditioning and Refrigeration Manufacturers' Associations (ICARMA)'s GREEN Program. The feedback and technical guidance of the project monitoring subgroup and peer reviewers, including Warren Beeton, Richard Cawley, Piotr Domanski, Kenneth Hickman, Steven Szymurski, Shaobo Jia, Kazumitsu Nishioka, Hung Pham, Ira Richter, John Sheridan and Mark Spatz, is gratefully acknowledged. Furthermore, authors acknowledge material and equipment contributions from Copeland, Heatcraft and Honeywell.

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NOMENCLATURE

A	Throat area of the orifice, Heat transfer area
ARI:	Air-conditioning and Refrigeration Institute
ASHRAE:	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BLDC:	Brushless DC
CEEE	Center for Environmental Energy Engineering
CFC’s	Chlorofluorocarbons
COP	Coefficient of Performance
Cp_a	Specific heat of air
DP_{cond}	Pressure drop across the condenser
DP_{evap}	Pressure drop across the evaporator
EES	Engineering Equation Solver
F	Function
GREEN	Global Refrigerant Environmental Evaluation Network
GWP	Global Warming Potential
h_{in}	Enthalpy of refrigerant at the indoor unit inlet
h_{out}	Enthalpy of refrigerant at the indoor unit outlet
h_aA_a	Air-side conductance
$h_{dis,isen}$	Enthalpy of refrigerant when the suction gas is isentropically compressed
h_{suc}	Enthalpy of refrigerant at the compressor suction
h_rA_r	Refrigerant-side conductance
HC’s	Hydrocarbons
HCFC’s	Hydrochlorofluorocarbons
HFC’s	Hydrofluorocarbons
HX	Heat exchanger
ICARMA:	International Council of Air-Conditioning and Refrigeration Manufacturers' Associations
LCCP	Life Cycle Climate Performance
$\dot{m}_r$	Refrigerant mass flow rate
n	Number of variables
$P_{cond,avg}$	Average pressure of inlet and outlet of the condenser in absolute pressure
$P_{evap,avg}$	Average pressure of inlet and outlet of the evaporator in absolute pressure
PR	Ratio between the discharge and suction pressure
V	Volumetric air flow rate
Q_{air}	Air-side capacity
Q_{ref}	Refrigerant-side capacity
q_{lci}	Latent air-side capacity
q_{sci}	Sensible air-side capacity
rms:	Root Mean Square
RPM	Revolution Per Minute
T_{ain}	Air temperature entering the indoor unit
T_{aout}	Air temperature leaving the indoor unit
T_{cond}	Condensing temperature
T_{evap}	Evaporating temperature
T_{suc}	Refrigerant temperature at the compressor suction

TEWI	Total Equivalent Warming Impact
TXV	Thermal expansion valve
U	Overall heat transfer coefficient
u_F	Uncertainty of the function
u_n	Uncertainty of the parameter
V_{disp}	Compressor displacement volume
v_n	Parameter of interest (measurement)
v'_n	Specific volume of air at orifice throat
W_n	Humidity ratio of air at orifice throat
W_{iin}	Humidity ratio of air entering the indoor unit
W_{iout}	Humidity ratio of air leaving the indoor unit
W_{comp}	Power consumption of the compressor
W_{total}	Power consumption of the compressor, fans of the unit cooler and condensing unit
ρ	Density of the air
ρ_{suc}	Density of refrigerant at the compressor suction
ΔP	Pressure drop across the orifice
η_{vol}	Volumetric efficiency
η_{comp}	Compressor efficiency

1 INTRODUCTION

Refrigerants should satisfy thermodynamic requirements to efficiently deliver sufficient capacities while being locally safe in equipment and globally safe for environment. Among the three natural refrigerants listed in Table 1, hydrocarbons (HC's) such as propane (R-290), isobutane (R-600a), cyclopropane (R-C270), and their mixtures are already being used especially in some part of the European Union (EU) and Japan, predominantly in domestic refrigerators due to their environmentally benign ozone depletion and low global warming potential characteristics. In 1992, DKK Scharfenstein in Germany developed refrigerators using HC's for both the blowing of insulation foam and the refrigerant (Greenpeace, 1997). Since then, the major household appliance manufacturers in the EU have been marketing HC's based refrigerators. In Japan most of major refrigerator companies have introduced HC's based refrigerators in 2002 (JARN, 2002). The charge of HC's in the refrigerator is very small, about 20 grams in a 130 liter refrigerator, which is almost equivalent to the charge in a cigarette lighter. The use of HC's is growing but their flammability restricts them in other applications where a large quantity of refrigerant is needed such as commercial refrigeration applications. Commercial refrigeration applications include self-contained refrigeration systems similar to domestic refrigerators but also large scale and central refrigeration systems connected to remote evaporators.

Table 1: Environmental Effects of Some Refrigerants (UNEP, 2002)

Refrigerants		ODP	GWP (Time horizons of 100 yrs)
HCFC's	R-22	0.055	1,700
HFC's	R-134a	0	1,300
	R-404A (R125/143a/134a)	0	3,800
	R-410A (R32/125)	0	2,000
Natural Refrigerants	Carbon dioxide (R-744)	0	1
	Ammonia (R-717)	0	<1
	Propane (R-290)	0	20
	Isobutane (R-600a)	0	20
	Cyclopropane (R-C270)	0	n/a

GREEN Program

The International Council of Air-Conditioning and Refrigeration Manufacturers' Associations (ICARMA) established in 1991 initiated the Global Refrigerant Environmental Evaluation Network (GREEN) Program in 2001. Under the GREEN Program, the Center for Environmental Energy Engineering (CEEE) of the University of Maryland jointly with Copeland, HeatCraft, Honeywell, and Air-conditioning and Refrigeration Institute (ARI) started a testing program to develop technically unbiased, credible refrigerant performance information on new and existing refrigerants in a variety of refrigeration, air conditioning, and heat pump applications. The program started its tasks in 2003 to conclusively establish the relative performance potential of HFC's (R-404A, and R-410A) as compared to R-290. In February of 2004, the CEEE completed the experimental evaluation of three refrigerants for the medium temperature commercial refrigeration using an 11 kW capacity system having -20°C to 0°C evaporator saturated refrigerant temperature. In August of 2004, under the GREEN Program, the CEEE started an extended test program for low temperature commercial refrigeration using a 4 kW capacity system having -29°C evaporator saturated refrigerant temperature and updates the results in this report.

2 PROPERTIES OF REFRIGERANTS

Figure 1 shows the saturation pressures of three refrigerants of interest. While R-410A has 33% consistently higher vapor pressure than that of R-404A, the saturation pressure of R-290 is 14% lower at -50°C and 27% lower at 70°C, which indicates a smaller pressure ratio at the same operating temperature levels.

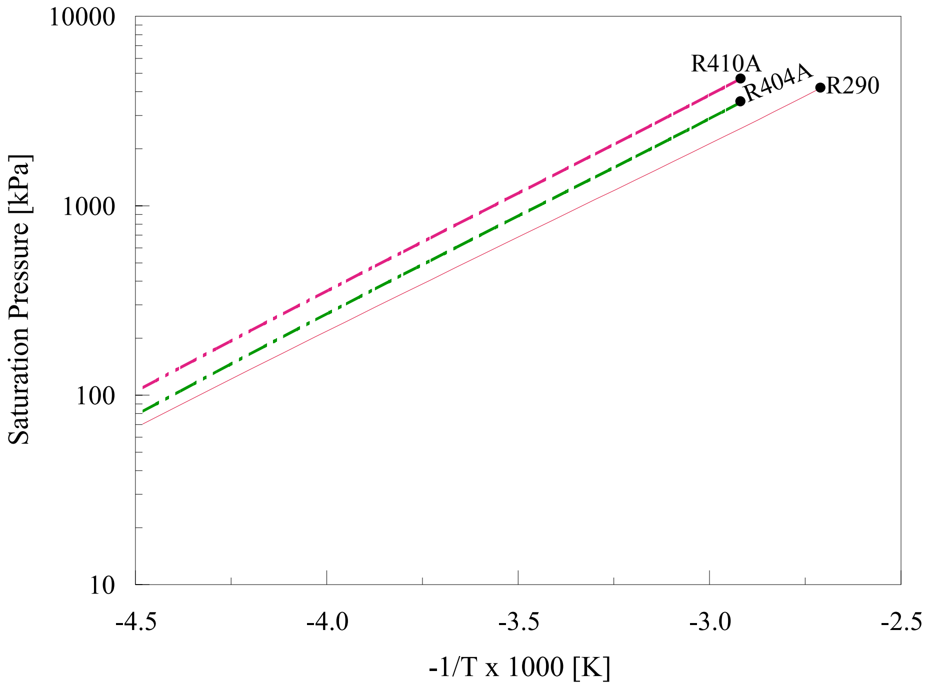
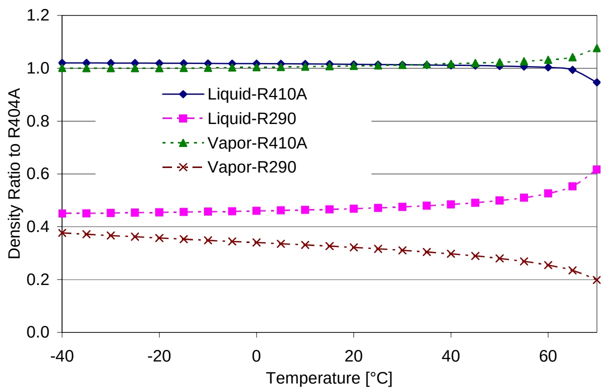


Figure 1: Saturation Pressure of Refrigerants

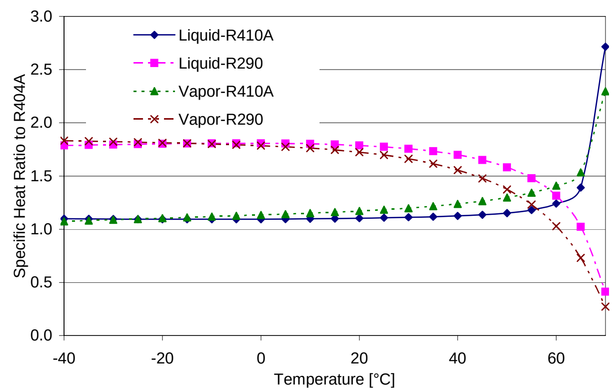
The thermophysical properties of the three refrigerants are compared at typical evaporating and condensing temperatures as shown in Table 2. Both R-410A and R-290 show higher liquid and vapor-specific heat and liquid thermal conductivity than those of R-404A. While R-410A has a 5% lower liquid viscosity and similar vapor viscosity as R-404A, R-290 has a 17 to 30% lower liquid viscosity and about 40% lower vapor viscosity. Figure 2 shows the relative values of these properties of R-410A and R-290 as compared to those of R-404A over the temperature range between -40°C and 70°C. Overall, it is expected that R-290 would have the best transport properties among the three refrigerants and R-410A would have better transport properties than R-404A. While the volumetric capacity of R-410A is 34% higher than that of R-404A, the volumetric capacity of R-290 is 23% lower than that of R-404A, which means a smaller and bigger compressor displacement volume is required for R-410A and R-290, respectively. It should be noted that thermophysical properties change significantly when the temperature exceeds approximately 60°C as could be seen from Figure 2. This is because R-404A and R-410A are approaching the critical point where thermophysical properties change significantly while R-290 does not.

Table 2: Thermophysical Properties of Three Refrigerants (NIST, 2002)

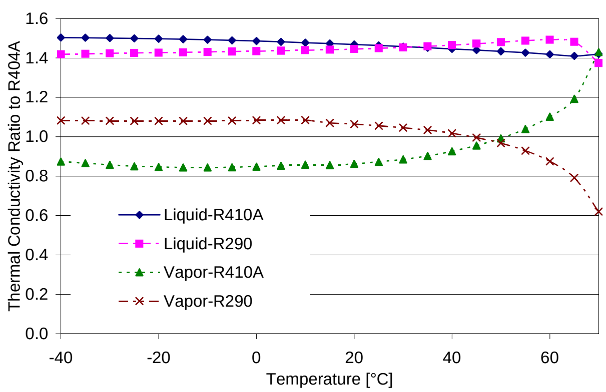
Refrigerant	R-404A		R-410A		R-290	
	-29°C	50°C	-29°C	50°C	-29°C	50°C
Molecular mass [g/mol]	97.6		72.6		44.1	
Normal boiling point [°C]	-46.1		-51.7		-42.1	
Critical temperature [°C]	72.0		71.4		96.7	
Critical pressure [MPa]	3.7		4.9		4.2	
Saturation pressure [kPa]	216	2,304	282	3,067	174	1,713
Sat. liquid density [kg/m^3]	1,252	899	1,270	907	566	449
Sat. vapor density [kg/m^3]	11.3	138	10.8	141	4.0	38.7
Sat. liquid specific heat [kJ/kg-K]	1.29	1.96	1.41	2.26	2.32	3.10
Sat. vapor specific heat [kJ/kg-K]	0.85	1.85	0.93	2.40	1.55	2.54
Sat. liquid viscosity [μ Pa-s]		89		84		74
Sat. vapor viscosity [μ Pa-s]		17.3		17.4		9.4
Sat. liquid thermal conductivity [mW/m-K]		55.7		79.9		82
Sat. vapor thermal conductivity [mW/m-K]		24.3		24.1		23.5
Latent heat [kJ/kg]		104		136		284



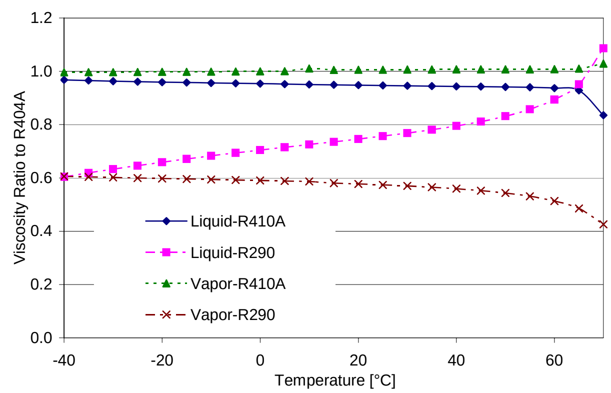
(a) Density



(b) Specific heat



(c) Thermal conductivity



(d) Viscosity

Figure 2: Comparison of Thermophysical Properties

Figure 3 illustrates comparison of the theoretical cycle efficiency of the three refrigerants for various condensing temperatures but a fixed evaporating temperature of the -29°C when the following cycle conditions are used; 5 °C subcooling and superheating, zero pressure drop across heat exchangers, and 100% compressor efficiency. This comparison shows that the three refrigerants have similar performance at low condensing temperatures (within 4% at 14°C condensing temperature) but R-410A and R-290 perform better than R-404A at higher condensing temperatures (11% and 16% better, respectively at 46°C condensing temperature).

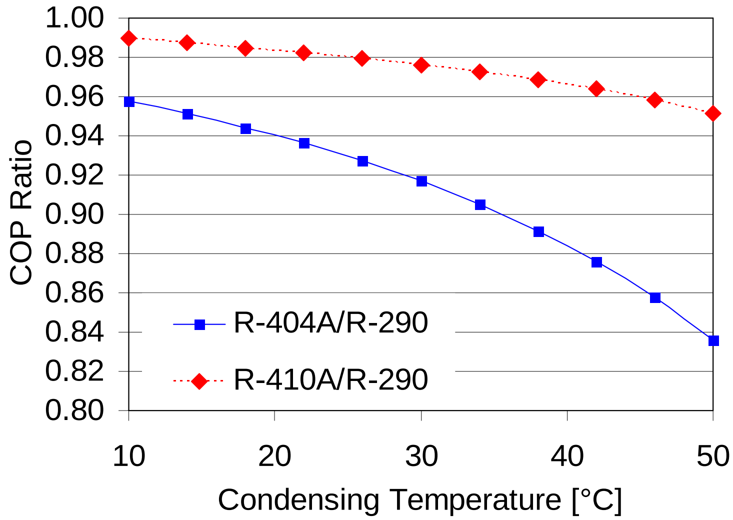
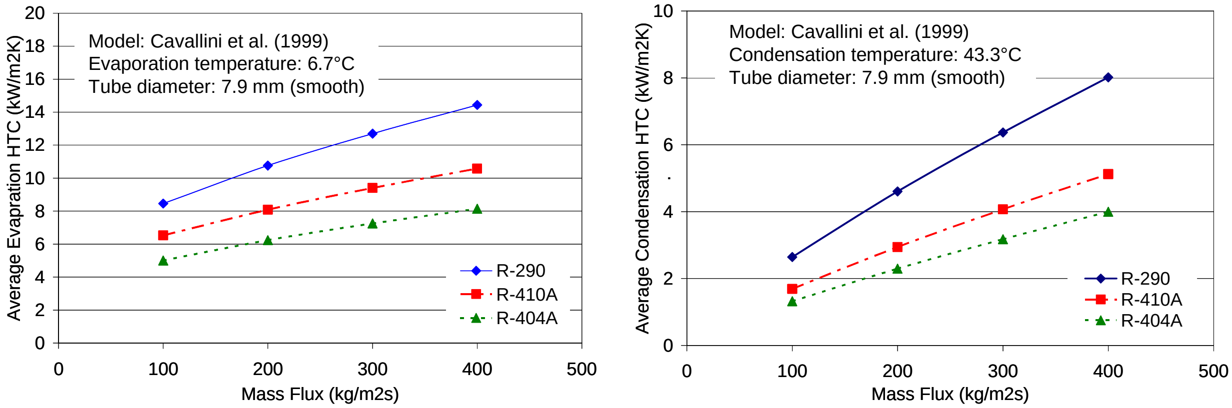


Figure 3: Comparison of Theoretical Cycle Efficiency

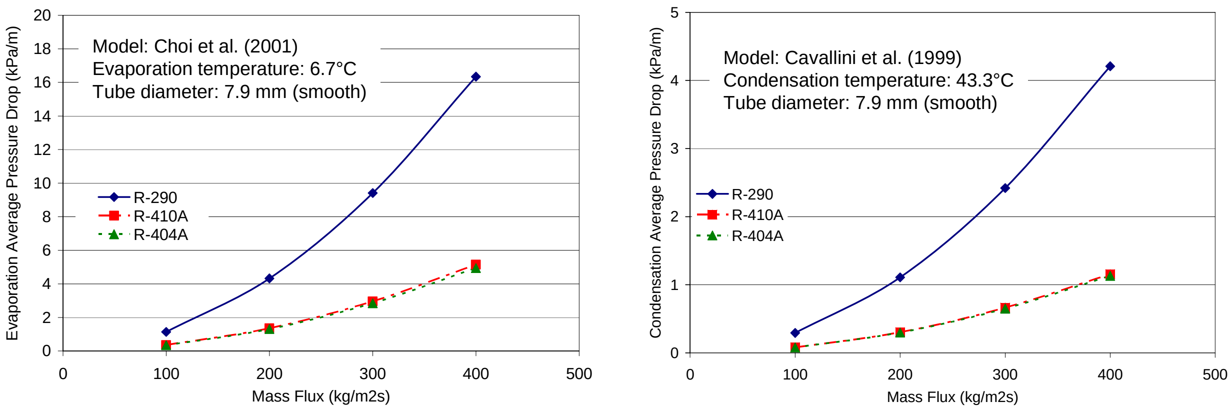
3 HEAT TRANSFER AND PRESSURE DROP

For better comparison of the effect of transport properties, information on the heat transfer and pressure drop characteristics of these refrigerants are required. Figures 4 and 5 show predictions of average heat transfer coefficients and pressure drop for all refrigerants used in this study (Cavallini et al., 10999; Choi et al., 2001). At the same mass flux, R-290 has the best heat transfer among all these refrigerants, but it also suffers the highest pressure drop penalty. These results were expected since the vapor density of R-290 is the lowest, and this property has a large impact in pressure drop predictions. R-410A has superior heat transfer than R-22 and R-404A. Pressure drop plots also show that R-410A suffers the lowest penalty among all the alternatives, which allows further optimization of the heat exchangers design. However, actual comparisons must be conducted at the actual mass fluxes in the system circuits since their mass flow fluxes are different when the system is designed for each refrigerant to produce the same capacity.



(a) Evaporation Heat Transfer (b) Condensation Heat Transfer

Figure 4: Heat Transfer Characteristics of Refrigerants



(a) Evaporation Pressure Drop (b) Condensation Pressure Drop

Figure 5: Pressure Drop Characteristics of Refrigerants

4 SYSTEM PERFORMANCE MODELING

In order to evaluate the system with different refrigerants and make changes to the coil circuit that better suits a particular refrigerant, a detailed system model was used. The model employed for the simulations, Honeywell's Genesym^{TM}, represents a vapor compression refrigeration cycle operating at steady-state conditions. The overall model is composed of sub-models for each component of the system. The major component models include:

- Compressor: Map based and analytical models;
- Evaporator and Condenser: Detailed tube-by-tube modeling;
- Expansion Devices: Analytical models (capillary tubes) and empirical correlations (short tubes, expansion valves).

For an independent verification of the modeling effort, the test and modeling results were also reproduced with "Coil Designer" and "Vapcyc" of CEEE, which are simulation tools for heat exchangers and vapor compression refrigeration cycles.

To model each component, the energy, momentum and mass balance equations are applied together with heat transfer laws, when necessary. This model incorporates some of the most relevant features of existing models, including quasi-local heat transfer analysis of heat exchangers (Domanski, 1989) and simulation of thermostatic expansion devices. Properties are calculated using REFPROP 7 (NIST, 2002), therefore any pure fluid or mixture present in this database can be used in the model. The models for air-side heat transfer coefficient employed were developed by Wang et al. (2000, 1999a, 2001, 1999b) for flat, wavy, lanced, and louvered fins, respectively. Condensation and evaporation refrigerant-side heat transfer coefficients were developed by Cavallini et al. (1999). The two-phase pressure drop models are from Choi et al. (1999).

Figure 6 shows five condenser circuits investigated in this study. The results of this optimization are shown in Table 3. There is not a significant impact of the circuitry changes on system efficiency (around 1% or less). The results shown are for the optimum subcooling found from the charge optimization test. For each refrigerant, the circuit corresponding to the COP value in bold was the one used for testing. It should be noted that the 2:1 circuit with 18 tube subcooling circuit was chosen for R-410A testing because it is resulted in the highest COP or close to the highest COP when the degree of subcooling is either at the optimum or lower than the optimum.

Table 3: Simulated COP of the Investigated Condenser Circuits

Circuit	COP		
	R-404A	R-410A	R-290
3:3 circuits without subcooling circuit	0.817	n/a	n/a
3:1 circuit with 12 tube subcooling circuit	**0.827**	0.899	**0.930**
2:1 circuit with 12 tube subcooling circuit	n/a	0.897	0.925
2:1 circuit with 18 tube subcooling circuit	n/a	**0.903**	0.928
2:1 circuit with 24 tube subcooling circuit	n/a	0.902	0.927

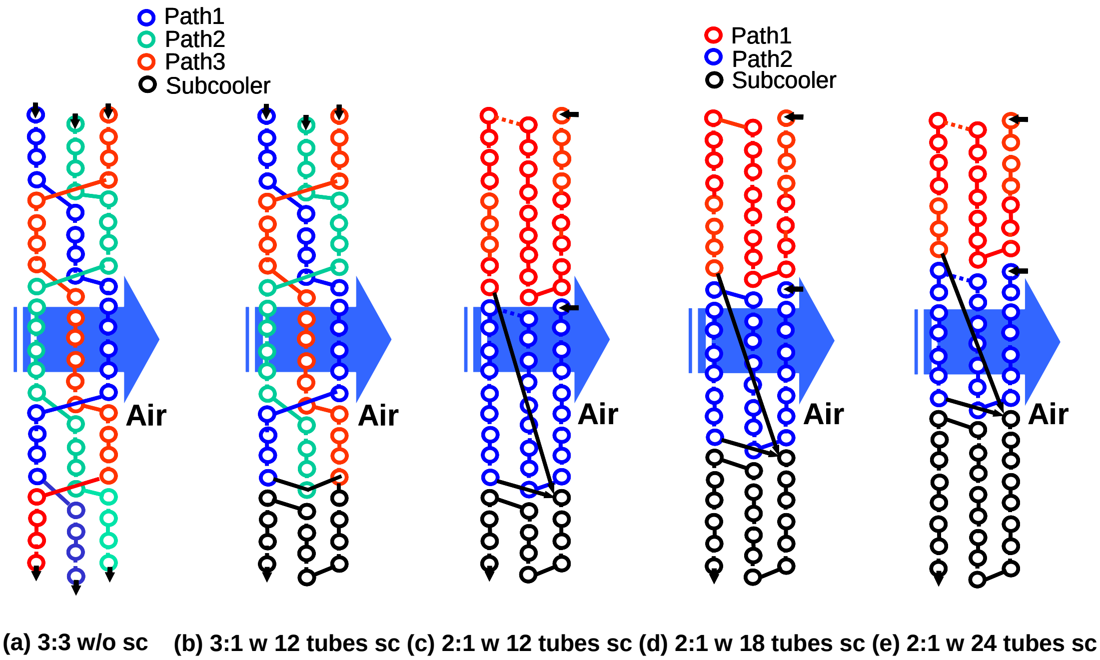


Figure 6: Condenser Circuits

5 EXPERIMENTAL PERFORMANCE EVALUATION

Since the reduction of global warming impact is a major focus for the comparison of refrigerants, the coefficient of performance (COP) of each refrigerant is of concern. However, the performance of HFC's as well as HC's varies much depending upon the test conditions and the degree of system modifications. To contribute to a clear understanding of the relative performance potential of each refrigerant, the hardware was optimized for each refrigerant in the current study.

5.1 Test Facility

The performance of the test unit was measured through the use of a psychrometric test facility constructed at CEEE's heat pump laboratory. This system is comprised of an air-duct and two environmental chambers, which house the indoor and outdoor heat exchangers and the compressor, to measure the capacity based on ANSI/ARI Standard 420 for unit coolers for refrigeration (ARI, 2000) and ANSI/ARI Standard 520 for positive displacement condensing units (ARI, 1997). The arrangement of the test facility is shown in Figures 7 and 8. The unit cooler and the condensing unit were separated from each other through the use of two environmental chambers capable of achieving temperatures ranging from -27 to 43ºC, allowing for the independent control of the inlet air stream conditions. As shown in Figure 7, the indoor duct is equipped with dew point meters to measure the dry bulb and dew temperatures of the air, and an orifice plate to measure the air flow rate. The desired air flow rate was adjusted by an inverter that controlled the speed of a fan that was installed in addition to the original evaporator fans and located in the outlet of the air duct to overcome the additional pressure drop caused by mixing devices, orifice plate, and duct. The duct outlet is open to the chamber to recondition the air stream, after which the air returns to the test unit. This air duct is sealed by a duct sealant to prevent air leakage, and wrapped with insulation to prevent heat losses. The duct size was determined according to ASHRAE Standard 40 (1980).

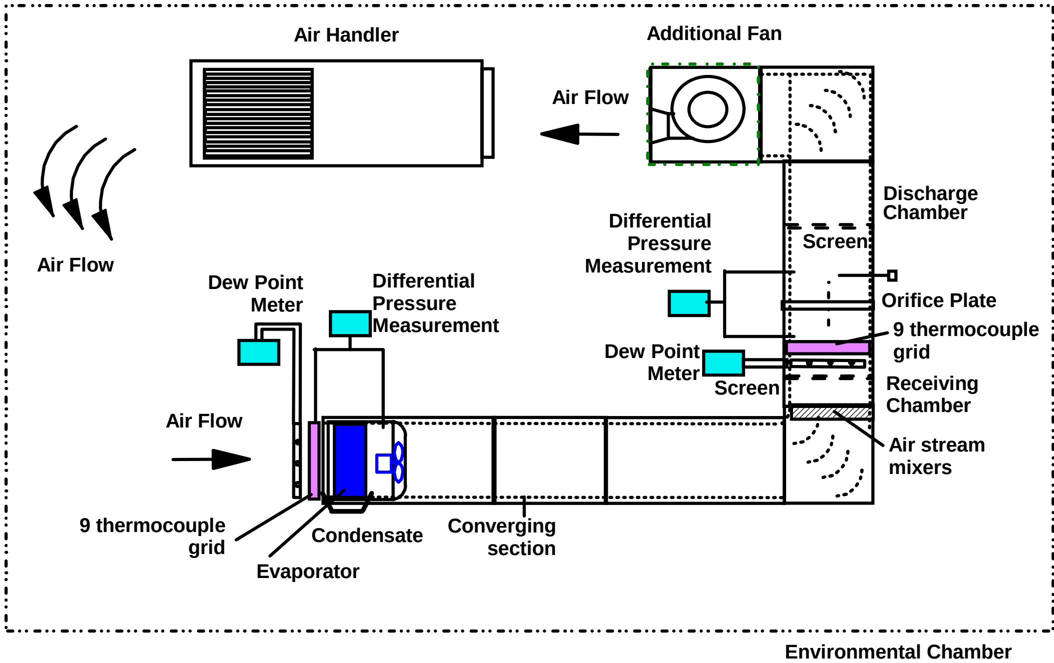


Figure 7: Test Facility for Unit Cooler

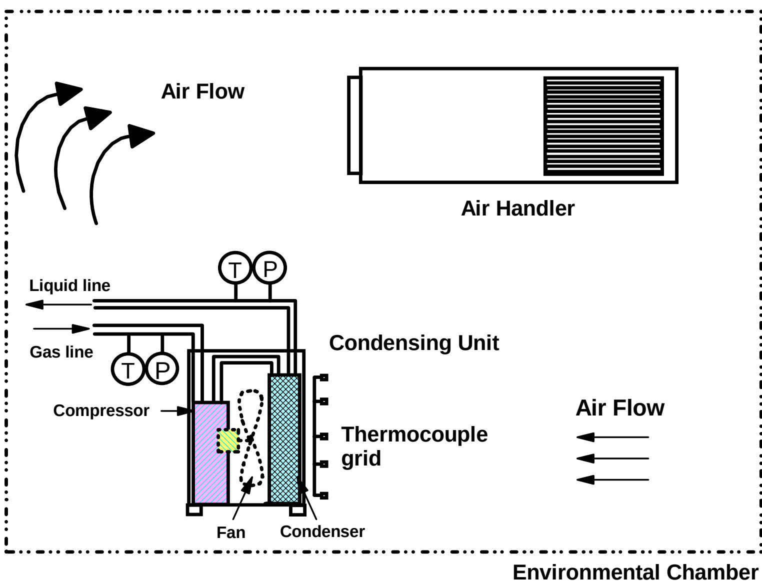


Figure 8: Test Facility for Condensing Unit

5.2 Instrumentation and Measurement

Along with the test facility, instrumentation to measure the performance of the test unit was implemented. The instrumentation was designed to determine the properties of air and refrigerant. There are basically four types of measurements necessary to obtain the required data to calculate and evaluate the performance of the test unit. These are temperatures, pressures, mass flow rate, and power.

Temperature Measurement

To measure the temperature of the air and the refrigerant, T type copper-constantan thermocouples with an accuracy of ± 0.2 ºC were employed. To measure the inlet and outlet air stream temperatures of an evaporator, two thermocouple grids, which have nine thermocouples each, were installed at the inlet and outlet of the unit cooler after the air mixer. Temperature difference between these two thermocouple grids was calibrated such that it is zero when there is no heat transfer in the duct section between the thermocouple grids. For the outdoor unit, six thermocouples were installed at both the air inlet and outlet. To measure the bulk temperature of the refrigerant, in-stream thermocouples were installed at all inlets and outlets of all components. The upstream and downstream air side dew points in the test duct were measured using microprocessor based instruments with an accuracy of ± 0.1 ºC of the coupon temperature. These units were calibrated by the company.

Pressure Measurement

For the pressure measurement of the air and refrigerant, piezoelectric pressure transducers were installed. The static pressures for the air duct were measured with differential pressure transducers with a range of 0 to 623 Pa and an accuracy of ± 1% full scale. Absolute pressure transducers having accuracies of 0.11% full scale were used to measure the refrigerant pressures. These absolute measurements were also made in conjunction with differential pressure transducers used to more accurately measure the pressure drop across the evaporator. The transducers were directly connected to the piping system with tees. The transducers were calibrated by utilizing a pressure calibrator (Omega, PCL5000) after installation into the system. The correlation obtained from the calibration was used in the data acquisition program to convert voltage output into pressure values.

Refrigerant Mass Flow Rate Measurement

Refrigerant mass flow was measured with a Coriolis type mass flow meter with an accuracy of ± 0.4%, which was placed downstream of the condenser outlet. The output signal of 4-20 mA was adjusted to correspond to a range of 0-100 g/s for R-404A, R-410A, and R-290 by using a transmitter calibrator.

Air Volume Flow Rate Measurement

As shown in Figure 7, differential pressure transducers were used for measurement of the air side pressure drop across the flow measurement device. This pressure differential measurement across the flow measurement device was used to determine the volumetric air flow rate (V) in the duct by equation (1) (ASHRAE Handbook, 2001).

$$V = K * A * \sqrt{\frac{2 * \Delta P}{\rho}} \tag{1}$$

where K is a constant determined by combining C, a friction loss coefficient factor, with $1/(1-\beta^4)^{0.5}$ which is an approach factor. The coefficient A refers to the area of the orifice, ρ is the density of the air, and ΔP is a pressure drop across the orifice. The size of the orifice was determined to be 51 cm to keep the pressure drop close to 360 Pa. A constant K was calibrated by using a bank of finned strip heaters having a 5 kW capacity, which were placed between the unit cooler and the orifice.

Compressor RPM Measurement

A piezoelectric accelerometer was used to measure compressor speed.

Power Measurement

The input power to the unit cooler fans and the outdoor fan was measured with watt transducers having an accuracy of $\pm$ 0.2% full scale. For the compressors, the compressor power was measured with a power transducer with a range of 0-12 kW and a digital power meter (Yokogawa, WT-1600). The accuracy of the power transducer is ± 0.2% full scale for a three phase 60 Hz signal. Since an inverter was used to exactly match the cooling capacity of R-290 and R-410A to that of R-404A, input power and line voltage of the R-290 and R-410A compressors were measured before and after the inverter with the digital power meter having an accuracy of 0.3%.

Measurement of Refrigerant Charges

An electronic scale having an accuracy of 1 g is used for charging the refrigerant.

Data Acquisition

Signals from all instruments were fed to a LabView data acquisition software package through the use of National Instruments' FieldPoint DAQ modules. These modules allow for flexibility in instrumentation, as additional channels may be added or removed easily if required later. These modules may also be placed close to the individual parts of the experiment (rather than the computer), eliminating both excessive cable lengths, and problems arising from incorrect wiring. A total of 96 channels of data were collected (64 thermocouples and 32 analog inputs) and sent to the computer for collection and instantaneous on-screen visualization of system parameters (e.g. pressures, temperatures, air flow rates, etc.). The tested sampling rate of this system was 5 seconds. A GUI was written for this experiment, allowing the user quick access to data from the system as it was in operation. Numeric outputs monitored include air side temperatures, air flow rates, dew points, performance (including COP, compressor work, and both latent and sensible cooling loads), refrigerant pressures, mass flow rate, and in-stream temperatures. The graphical portion of the program monitored the history of many of these same measurements. When all measured data reached steady state within 1% variation (temperature variation less than 0.1°C) for more than 30 minutes, the data collection was started for 30 minutes at 5 seconds interval.

5.3 Performance Evaluation

The performance of the test unit was evaluated in terms of its capacity, COP, and compressor efficiencies as described below. To evaluate the capacity experimentally, the air-side capacity and refrigerant-side capacity were calculated from the measured data.

Air-Side Capacity

The sensible air-side capacity (q_{si}) was calculated by equation (2) (ASHRAE Standard 37, 1988).

$$q_{si} = \frac{V}{v'_n(1+W_n)} Cp_a (T_{ain} - T_{aout}) \qquad (2)$$

where Cp_a: Specific heat of air
T_{ain}: Air temperature entering the indoor unit
T_{aout}: Air temperature leaving the indoor unit
v'_n: Specific volume of air at orifice throat
W_n: Humidity ratio of air at orifice throat

The latent air-side capacity (q_{lci}) was calculated from the humidity ratio difference between inlet and outlet by equation (3).

$$q_{lci} = \frac{Q}{v'_n(1+W_n)} (W_{iin} - W_{iout}) \qquad (3)$$

where W_{iin} : Humidity ratio of air entering the indoor unit
W_{iout} : Humidity ratio of air leaving the indoor unit

Then the air-side capacity (Q_{air}) was calculated by summing up the sensible air-side capacity (q_{sci}) and the latent air-side capacity (q_{lci}).

Refrigerant-Side Capacity

The refrigerant-side capacity (Q_{ref}) was calculated using the mass flow rate of refrigerant and enthalpy difference between inlet and outlet of the evaporator. The evaporator inlet enthalpy was obtained from the expansion valve inlet enthalpy by assuming an isenthalpic expansion process. These enthalpies were calculated based on the measured pressures and temperatures by using thermodynamic property routines, REFPROP V7 (NIST, 2002). Then the refrigerant-side capacity (Q_{ref}) was calculated using equations (4).

$$Q_{ref} = \dot{m}_r (h_{out} - h_{in}) \qquad (4)$$

where $\dot{m}_r$: refrigerant mass flow rate
h_{in}: enthalpy of refrigerant at the indoor unit inlet
h_{out}: enthalpy of refrigerant at the indoor unit outlet

To confirm that the data are reliable, the capacity determined using these two methods should agree within 6% of each other as required by ASHRAE Standard 116 (1995). The reported capacity and COP values were based on refrigerant-side values. The air-side values were used only to check the total energy balance.

COPs

COPs were calculated for both the air-side and the refrigerant-side based on the capacity and total system power consumption (W_{total}) including the condenser and unit cooler fan motor power consumption in addition to the compressor power consumption by using equation (5).

$$COP_{air} = Q_{air} / W_{total}$$
$$COP_{ref} = Q_{ref} / W_{total} \qquad (5)$$

Compressor Efficiencies

For the compressor performance evaluation, volumetric (η_{vol}) and compressor (η_{comp}) efficiencies were calculated as defined by equations (6) and (7) (ASHRAE, 2000; ANSI/ARI Standard 550, 1997):

$$\eta_{vol} = \frac{\dot{m}_r}{\rho_{suc} \times V_{disp} \times RPM} \qquad (6)$$

$$\eta_{comp} = \frac{(h_{dis,isen} - h_{suc}) * \dot{m}_r}{W_{comp}} \qquad (7)$$

where ρ_{suc}: refrigerant density at the compressor suction
V_{disp}: compressor displacement volume
RPM: compressor revolution speed
$h_{dis,isen}$: refrigerant enthalpy when the suction gas is isentropically compressed
h_{suc}: refrigerant enthalpy at the compressor suction
W_{comp}: compressor power consumption

5.4 Error Analysis

During experimentation, the bias (or systematic) error and the precision (or random) error are two important parameters to be mindful of (Beckwith et al., 1993). Detailed error analysis to determine the magnitude of these values is described as follows.

Bias Error

The bias error is an uncertainty that occurs in the same way each time a measurement is made. The total uncertainty of a measurement due to the uncertainty of individual parameters is referred to as the propagation of uncertainty (Beckwith et al., 1993). Also referred to as bias, the total uncertainty of any function may be calculated using the Pythagorean summation of uncertainties which is defined by equation (8) (Kline and McClintock, 1953):

$$u_F = \sqrt{\left(\frac{\partial F}{\partial v_1} * u_1\right)^2 + \left(\frac{\partial F}{\partial v_2} * u_2\right)^2 + \left(\frac{\partial F}{\partial v_3} * u_3\right)^2 + \ldots\ldots + \left(\frac{\partial F}{\partial v_n} * u_n\right)^2} \qquad (8)$$

where:
u_F = uncertainty of the function
u_n = uncertainty of the parameter
F = function
v_n = parameter of interest (measurement)
n = number of variables

The partial derivatives of each independent measurement for the relevant calculated parameters were determined using the uncertainty propagation function in the Engineering Equation Solver (EES), and applied within the program to the root mean square (rms) outcome. The results of this effort are shown in Table 4.

Precision Error

The precision error is different for each successive measurement but have an average value of zero. This minimum/maximum error in the measurements of importance was calculated with a spreadsheet based upon the rated deviation of the system's instrumentation. The precision error was calculated to have a confidence level of 99.7%.

Total Error

After evaluating the bias and precision errors, the total errors are calculated by summing up these two errors. Table 4 shows the results of the total error calculation applied to those quantities important in this study. From this, it was determined that the air side calculations for capacity and COP generated the most uncertainty, primarily due to the accuracy of the instruments involved in the measurement (thermocouple grids, air side pressure transducers and dew point meters), and this is the reason for reporting the refrigerant-side performance as the primary method.

Table 4: Measurement Errors

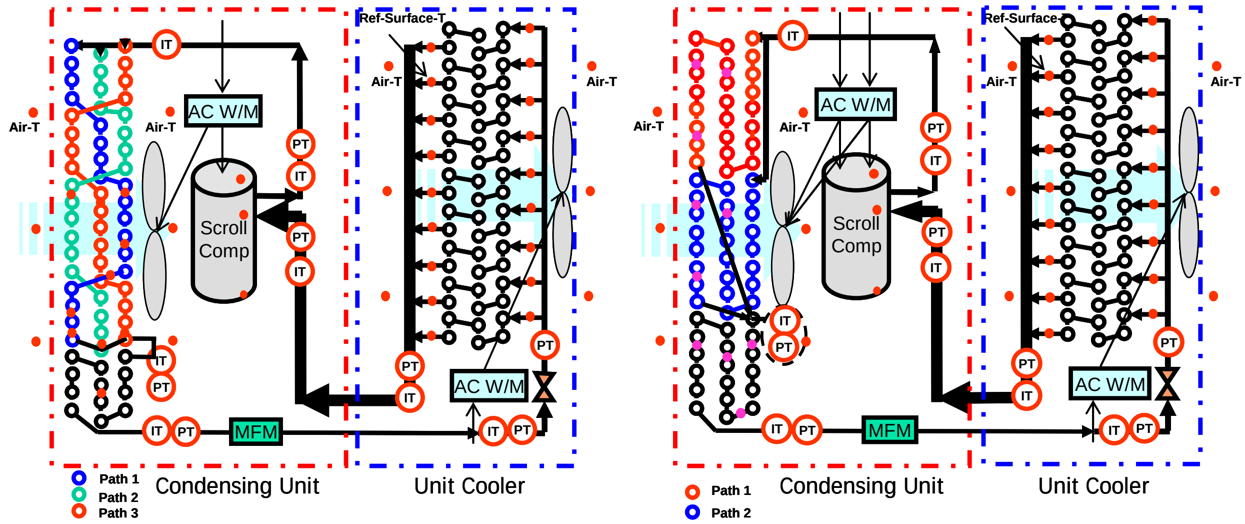
Parameter	Air-side capacity	Air-side COP	Refrigerant-side capacity	Refrigerant-side COP
Bias error [%]	± 0.8	± 0.8	± 1.1	± 1.8
Precision error [%]	± 1.6	± 1.6	± 0.4	± 0.5
Total error [%]	± 2.4	± 2.4	± 1.5	± 2.3

5.5 Test Unit

The test unit consists of a unit cooler and a condensing unit having a 4 kW refrigeration capacity for low temperature refrigeration and both designed for R-404A. A unit cooler incorporates two fans that produce a flow rate of 2.22m^3/s with a capacity of 4 kW at -29°C evaporator saturated refrigerant temperature. The condensing unit has a single axial fan delivering an air flow rate 1.6 m^3/s. The refrigeration cycle of the test unit is shown in Figure 9.

Heat Exchangers

The evaporator has 9 circuits and each circuit consists of 6 tubes in three rows. Each circuit is distributed along the vertical direction. The overall flow direction of the refrigerant is against the air stream. To maintain consistency in the testing with the simulation, which assumed the system was not equipped with a receiver, the condenser circuit was redesigned to have a subcooler. Based on the simulation described earlier, the same condenser, but two different circuits, was used in testing. A three circuit condenser (Figure 9 (a)) was used for testing of R-404A and R-290, and a two circuit condenser (Figure 9 (b)) was used for testing of R-410A. The three-circuit condenser consists of 18 tubes in each circuit and the three circuits join just before the last 12 tubes to form the subcooler. The two-circuit condenser consists of 24 tubes in each circuit and they join just before the last 18 tubes to form the subcooler.



(a) R-404A and R-290 **(b) R-410A**

Figure 9: Heat Exchanger Circuits and Instrumentation

Details of both heat exchangers are listed in Table 5. The measured surface temperatures of the inlet and outlet of each condenser circuit for all three refrigerants were comparable within 0.3°C and 1.3°C, respectively, which indicates a fair distribution also. The symmetrical distributor was placed in a vertically downward direction in order to reduce the possibility of flow maldistribution in the evaporator. To check the uniformity of the refrigerant distribution, the surface temperature of each evaporator circuit was measured with thermocouples as illustrated in Figure 9. The measured surface temperatures of the inlet and outlet of each evaporator circuit and the intermediate path of the condenser circuits for all three refrigerants were comparable within 0.5°C and 1.0°C, respectively, which indicates an acceptable distribution.

Table 5: Specifications of Heat Exchangers

Specification		Evaporator	Condenser
Dimension	W x H x D [cm]	142 x 57 x 8	99 x 70 x 8
	Frontal area [m^2]	0.81	0.69
Air flow	Air flow rate [m^3/s]	2.22	1.6
	Frontal air velocity [m/s]	2.73	2.35
Fin	Shape	Wavy	Wavy
	Fin pitch [mm]	4.23	2.1
	Thickness [mm]	0.19	0.13
Tube	No. of row	3	3
	No. of tubes per each row	18	22
	Tube diameter [mm]	9.52	9.52
	No. of circuit	9	3 (R-404A, R-290), 2 (R-410A)
	Tube shape	Inner grooved	Inner grooved

Compressor

The test unit employs a scroll compressor from Copeland Corp. Three scroll compressors having different displacement volume as shown in Table 6 were used. All compressors were sized to produce as closely as possible the same cooling capacity for each respective refrigerant and use the motors having the closest possible motor efficiency. Since the selected displacement volume available for R-410A and R-290 is larger than the target displacement, an inverter drive was used to match the refrigeration capacity by adjusting the inverter. Two different lubricants were utilized in testing, POE for R-404A and R-410A, and mineral oil for R-290. This requires system flushing in addition to compressor changes. Especially, when the test moved from R-410A system to R-290 system, system flushing was done using another compressor with same oil as that of R-290 system until the index of refraction of the oil becomes that of the new oil to be used for the R-290 system. Then the appropriate compressor, precharged with the correct lubricant, was installed.

Table 6: Specifications of Compressors

Refrigerant	R-404A	R-410A	R-290
Oil	POE	POE	Mineral oil
Displacement [cc]	82.6	67.1	98.0
Displacement Volume Ratio	1	0.81	1.19
Motor	3 phase, 208-230 V AC, 60 Hz		

Expansion Device

A single hand adjusted needle valve was used as the expansion device between the condenser and evaporator in the system in order to maintain an equal evaporator superheating for all three refrigerants.

Vapor Line

It should be noted that to minimize the effect of the pressure drop across the vapor suction line on the system performance, the pressure drop across the suction line was always maintained less than 1°C saturation temperature drop by using 28.7 mm tube diameter and 5 m length for the suction line.

Receiver

A typical commercial refrigeration system has a receiver for refrigerant management. Because of that, in order to know the effect of a receiver, the test was done with and without a receiver for the R-404A and R-410A system. However, for R-290 system, the test was done only without receiver to minimize the refrigerant charge for safety reasons. For this test, a vertical type receiver having 127 mm diameter and 241 mm height was used.

Refrigerants

Two HFC's (R-404A and R-410A) were supplied from Honeywell and propane was purchased from a local chemical supplier. Since the refrigerant purity of two HFC's is higher than 99.5% (ANSI/ARI Standard 700, 1999), the effects of impurities are negligible. There are three grades of propane potentially used as the refrigerant (Table 7). Among the three major impurities in Table 7 (isobutane, butane and ethane), R-600a has the highest composition and can potentially affect the property of propane. Since the boiling temperature of R-600a is higher than

R-290, more R-600a means a higher saturation temperature. The saturated liquid temperatures of instrument grade and chemically pure grade are very similar to that of pure R-290 within a 0.2°C deviation. The saturated vapor temperatures of these two grades are 0.2 – 0.3°C and 0.5°C higher than that of pure R-290, respectively. However, these differences are almost same as the thermocouple measurement error. Moreover, the saturation enthalpies and densities of these three grades are almost same within 0.1% variation. The difference in the refrigerant-side capacity calculated by assuming pure R-290 and the other two grades is less than 0.5%. Therefore, three grades shown in Table 7 can be technically treated as pure R-290. In the R-290 testing, the instrument grade was used.

Table 7: Impurities of Propane (Airgas, 2003)

Grade	Composition [wt.%]			
	Propane (C_3H_8)	Isobutane (C_4H_{10})	Butane (C_4H_{10})	Ethane (C_2H_6)
Research grade	99.99	< 0.01	< 0.01	< 0.01
Instrument grade	99.53	0.40	0.07	0.01
Chemically pure grade	98.98	0.77	0.20	0.03

5.6 Test Procedure

Test conditions for full load and part load are summarized in Table 8. Here, the part load means a reduced temperature lift from the evaporator to the condenser. During the part load operation the cooling capacity of the system was not fixed and was determined by the system's response to the reduced ambient conditions. First the refrigerant charge was optimized to maximize the system COP by running a series of tests at full load conditions (ambient temperature for condenser side at 35°C). During the charge optimization tests, the degree of superheating was kept constant to be 5°C by adjusting the opening of the metering valve to simulate the control of a TXV. The optimum refrigerant charge was decided when the COP became the maximum. Then the part load test (ambient temperature for condenser side at 18.3°C) was conducted at the optimum charge that was determined from the full load tests. It should be noted that the part load condition was referred from the 50% part load condition for the water chillers with an air cooled condenser (ANSI/ARI Standard 550/590, 1997). For both tests, the evaporator inlet air was kept at -23.3°C dry-bulb temperature and dry condition. Air flow rates through the evaporator and condenser were fixed at 2.22 m^3/s and 1.6 m^3/s, respectively. Moreover all test data was acquired after the system reached steady state.

Table 8: Test Conditions

Test	Heat Exchanger	Inlet air dry-bulb/wet-bulb temperature [°C]	Air flow rate [m^3/s]	Superheating [°C]
Full load	Evaporator	-23.3/-23.8	2.22	5
	Condenser	35.0/24.0	1.6	
Part load	Evaporator	-23.3/-23.8	2.22	5
	Condenser	18.3/11.0	1.6	

5.7 Full Load Test Results

Frequency Adjustment and Effects of Inverter

To compare the performance of each refrigerant under fair conditions, it was decided to match the cooling capacity of R-410A and R-290 to that of R-404A as close as possible. Since the compressors were selected from the commercially available platform as shown in Table 6, the frequency of the inverter was varied in R-410A and R-290 testing. As a result of the frequency variation, 54 Hz and 57 Hz were selected for the inverter setting of R-410A and R-290, respectively. Based on the test results from the medium temperature application test, R-410A system was evaluated both using the inverter and without the inverter to investigate the effects of inverter. By using the inverter the input line voltage dropped by 5 V but the current and power consumption of the compressor were only changed within 0.2% between two cases. Furthermore, the difference of the performance between with and without the inverter is less than 1% in the performance. Similar results were found for R-290 as well.

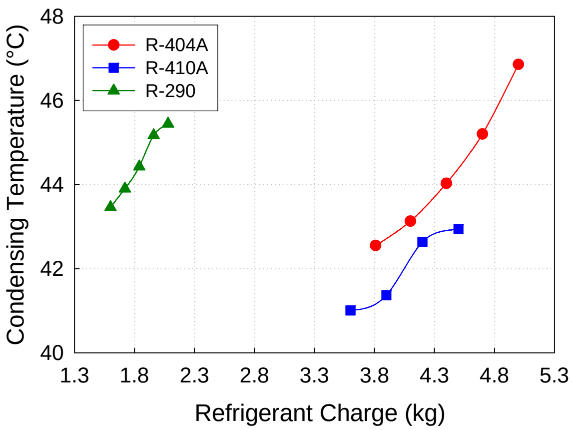
Charge Optimization

The refrigerant charge optimization tests were performed under full load test conditions. By varying refrigerant charge, the optimum charge resulting in the highest COP was experimentally obtained. It should be noted that the power consumption of the unit cooler fans and the condensing unit fan was constant at 0.66 and 0.36 kW, respectively throughout all tests. As described earlier, the compressor line frequency was adjusted during the R-410A and R-290 charge optimizations to match the cooling capacity to that of R-404A. When refrigerant charge was increased with the fixed degree of superheating, the condensing temperature and the degree of subcooling increased as shown in Figure 10 (a) and (b). This increase of the subcooling contributed to the higher latent heat of evaporation. At the same time, the higher condensing temperature yielded a higher pressure ratio, which contributed to the higher compressor work as shown in Figure 10 (c) and (d).

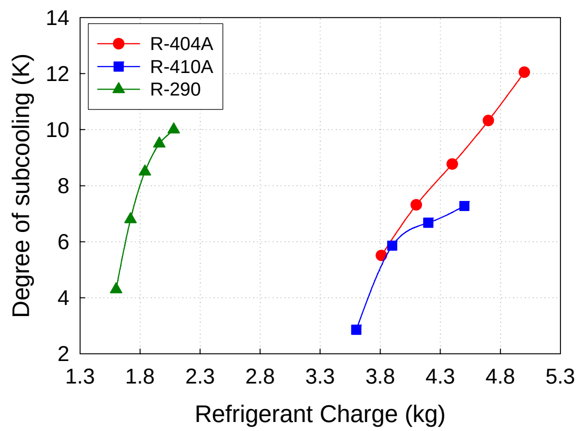
However, the effective increase of the available latent heat of evaporation diminishes as the condensing temperature increases as can be seen from the pressure-enthalpy diagram of each refrigerant while the compressor work keeps increasing. Therefore, the refrigeration capacity and COP increase until they reach their maximum and then decrease as illustrated in Figure 10 (e) and (f). The system performance of the three refrigerants as well as their cycle parameters at three different charges is summarized in Table 9. The optimum charge of R-404A was 4.4 kg with a capacity of 3.7 kW and the COP was 0.784. The optimum charge of R-410A and R-290 was 89% and 39% of that of R-404A. The COPs of R-410A and R-290 were 2% and 12% higher than that of R-404A. These results reflect the thermodynamic characteristics of R-290, but does not reflect the characteristics of R-410A because the compressor efficiency for R-410A is about 10% lower than that for R-404A as can be seen Table 12.

The calculated performances of these three refrigerants for the cycle condition (5°C subcooling and superheating, zero pressure drop across the heat exchangers, and 100% compressor efficiency) are compared in Table 10. It shows 11% and 16% higher COP as compared to R-404A for R-410A and R-290 respectively under the first condition in Table 10. Under the second condition in Table 10, it also shows 10% and 14% higher COP for R-410A and R-290, respectively compared to that of R-404A. The condensing temperatures for the three refrigerants of the tested conditions are between these two simulated condensing temperatures. In addition, Table 10 shows that the pressure ratio (PR) of R-410A is 2% higher but the PR of R-

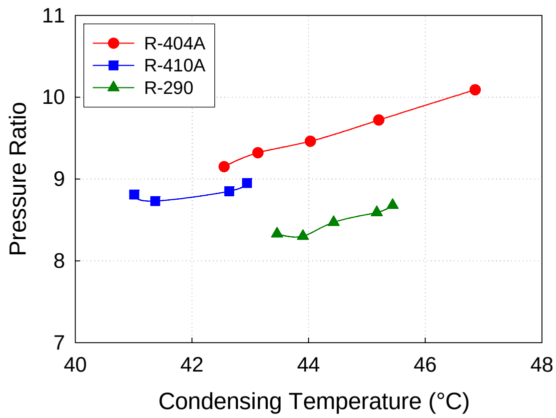
290 is 7% lower compared to R-404A, which results in a thermodynamically more favorable compressor operating condition for R-290. Even though this comparison explains the inherent thermodynamic difference, further analysis, which can account for the effects of heat transfer, pressure drop, and subcooling is required because the actual system operating conditions are different from the simulated conditions used in this comparison.



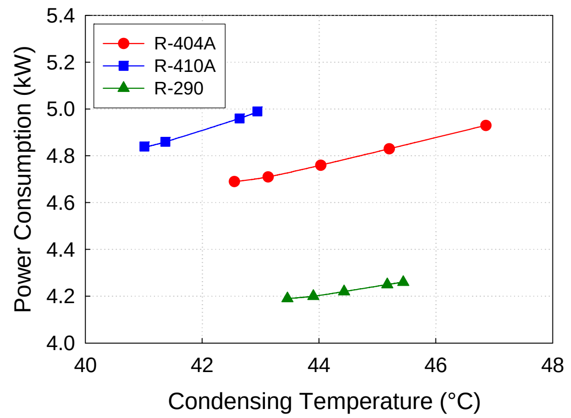
(a) Condensing Temperature vs. Charge



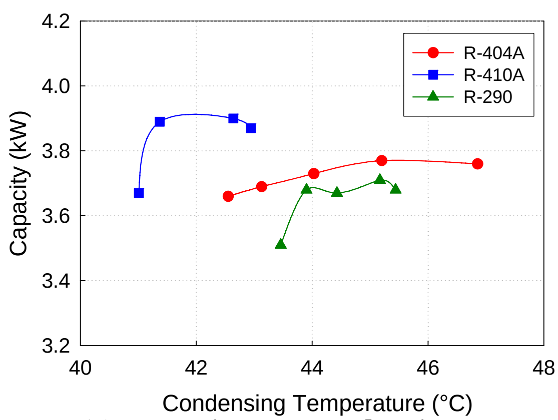
(b) Subcooling vs. Charge



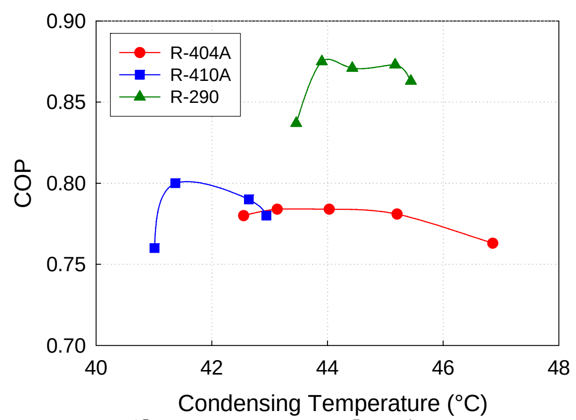
(c) Pressure Ratio vs. Condensing Temp.



(d) Power Consumption vs. Condensing Temp.



(e) Capacity vs. Condensing Temp.



(f) COP vs. Condensing Temp.

Figure 10: Charge Optimization Results

Table 9: Full Load Test Results (Optimum charge in bold)

Refrigerant	Parameter	Data		
R-404A (60 Hz)	Charge [kg]	4.1	**4.4**	4.7
	Capacity [kW]	3.7	**3.7**	3.8
	COP	0.784	**0.784**	0.781
	DP_{evap} [kPa]	21.0	**20.8**	20.8
	DP_{cond} [kPa]	11.7	**9.9**	8.3
	$P_{evap,avg}$ [kPa, abs]	226	**227**	228
	$P_{cond,avg}$ [kPa, abs]	1,971	**2,013**	2,069
	Subcooling [°C]	7.3	**8.8**	10.3
	Superheating [°C]	4.8	**4.6**	4.9
	Pressure Ratio (P_{dis}/P_{suc})	9.32	**9.46**	9.72
	T_{evap}/T_{cond} at P_{suc}/P_{dis} [°C]	-29.3/43.7	**-29.1/44.6**	-29.1/45.7
	Mass flow rate [g/s]	43	**43**	43
	η_{vol}	0.85	**0.84**	0.84
	η_{comp}	0.56	**0.56**	0.56
R-410A (54 Hz)	Charge [kg]	3.6	**3.9**	4.2
	Capacity [kW]	3.7	**3.9**	3.9
	COP	0.76	**0.80**	0.79
	DP_{evap} [kPa]	8.1	**8.1**	8.2
	DP_{cond} [kPa]	38.4	**34.1**	27.2
	$P_{evap,avg}$ [kPa, abs]	290	**295**	299
	$P_{cond,avg}$ [kPa, abs]	2,486	**2,507**	2,585
	Subcooling [°C]	2.9	**5.9**	6.7
	Superheating [°C]	5.2	**4.8**	4.6
	Pressure Ratio (P_{dis}/P_{suc})	8.81	**8.73**	8.85
	T_{evap}/T_{cond} at P_{suc}/P_{dis} [°C]	-28.7/41.4	**-28.3/41.8**	-27.9/43.0
	Mass flow rate [g/s]	28	**28**	28
	η_{vol}	0.79	**0.79**	0.78
	η_{comp}	0.50	**0.50**	0.49
R-290 (57 Hz)	Charge [kg]	1.6	**1.72**	1.84
	Capacity [kW]	3.5	**3.7**	3.7
	COP	0.84	**0.88**	0.87
	DP_{evap} [kPa]	9.3	**9.3**	9.1
	DP_{cond} [kPa]	5.3	**2.4**	2.3
	$P_{evap,avg}$ [kPa, abs]	183	**185**	183
	$P_{cond,avg}$ [kPa, abs]	1,482	**1,497**	1,515
	Subcooling [°C]	4.3	**6.8**	8.5
	Superheating [°C]	5.1	**4.6**	4.9
	Pressure Ratio (P_{dis}/P_{suc})	8.33	**8.30**	8.47
	T_{evap}/T_{cond} at P_{suc}/P_{dis} [°C]	-28.4/43.6	**-28.1/44.0**	-28.3/44.6
	Mass flow rate [g/s]	17	**17**	17
	η_{vol}	0.82	**0.82**	0.82
	η_{comp}	0.56	**0.56**	0.56

Table 10: Thermodynamic Comparison of Three Refrigerants

Condition	Refrigerant	COP Ratio	PR Ratio
-29.0/46.0/5.0 (T_{evap}/T_{cond}/Subcool)	R-410A/R-404A	1.11	1.02
	R-290/R-404A	1.16	0.92
-29.0/42.0/5.0 (T_{evap}/T_{cond}/Subcool)	R-410A/R-404A	1.10	1.02
	R-290/R-404A	1.14	0.93

(5°C subcooling, zero pressure drop across the heat exchangers, and 100% compressor efficiency)

Condensation Heat Transfer

Since the same air-side conditions were used for these three refrigerants with equal capacity, the refrigerant-side thermal resistance is responsible for the difference in overall condenser thermal resistance and refrigerant-side pressure drop, and thus for the pressure ratio of each refrigerant. Table 11 shows the ratio of the measured refrigerant mass flux and UA values (Overall heat transfer coefficient multiplied by heat transfer area) for the refrigerants. When UA values were calculated, the measured refrigerant-side condenser capacity and log mean temperature difference were used. If it is assumed that the air-side heat transfer coefficient is same for the refrigerants due to the same air velocity and air inlet temperature, the UA values indicate the difference of refrigerant-side heat transfer coefficient. These results indicate that R-410A has the best condensation heat transfer coefficient among three refrigerants, and R-290 has almost same values as those for R-404A. The latter results are due to R-290's better transport properties even though it has the smaller mass flux due to the lower density. These results show the same trend as that in Figure 4. In addition, the condensing temperature for R-410A system is about 2°C lower than R-290 and R-404A system as can be seen in Table 16 due to the better condensation heat transfer coefficient.

Table 11: Contribution of Condensation Heat Transfer

Refrigerant	Mass flux (kg/m^{2}s)		Calculated UA value (W/°C) (based on measured refrigerant capacity)	
	Full load	Full load	Full load	Part load
R-404A	211 (100 %)	214 (100 %)	1070 (100 %)	1135 (100 %)
R-410A	209 (99 %)	223 (104 %)	1599 (150 %)	1269 (118 %)
R-290	83 (39 %)	88 (41 %)	1075 (100 %)	1073 (95%)

Compressor Efficiency

Figure 11 shows the measured compressor efficiencies of the three refrigerants as defined by equation (7). As shown in Figure 11, the compressor efficiency varies as a function of the pressure ratio. Since higher compressor efficiency would mean smaller compressor power consumption, lower pressure ratio is desirable and can be achieved with better heat exchanger design. As shown in the Figure, the compressor efficiency of the R-410A system is about 10% lower than those for other two refrigerants system. As can be seen from Table 10, the theoretical pressure ratio of R-290 is 7~8% lower than that of R-404A, improving the compressor efficiency. However, it should be noted that compressor efficiency is also affected by the compressor design in addition to the above factors.

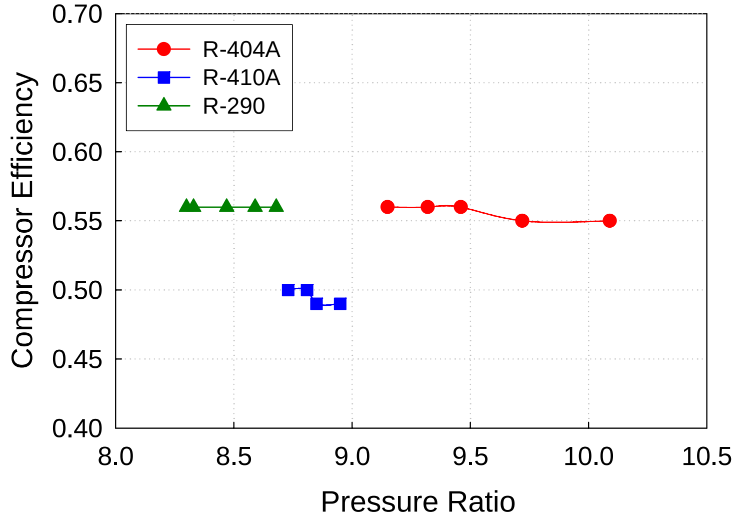


Figure 11: Compressor Efficiency vs. Pressure Ratio

Effects of Compressor Efficiency under Full-Load Conditions

Production compressors were selected to match the capacity requirement of each refrigerant as closely as possible. Since the compressor is optimized based on a pre-determined built-in scroll set volume ratio, it may not be as optimized under a higher or lower pressure ratio that may occur under the system operating condition. According to the compressor manufacturer, the compressor selected for R-410A is designed for the medium temperature refrigeration and the same compressor efficiency can be expected, if each compressor is optimized for each refrigerant at the system operating condition by adjusting the built-in scroll set volume ratio and the motor. Table 12 compares the system performance of these three refrigerants for two cases. The first case is based on the measured compressor efficiency from Table 9. The second case is based on the re-calculation of the compressor power assuming the compressor efficiency of R-410A and R-290 being equal to that of R-404A. This adjustment is intended to compare the performance of each refrigerant while eliminating the effect of compressor efficiency assuming the compressor manufacturer could achieve the same compressor efficiency for each refrigerant. Then the COP of R-410A is equal to that of R-290.

Table 12: Effects of Compressor Efficiency under Full Load Condition

Case	Refrigerant	Compressor Efficiency Ratio	COP Ratio
Based on measured value (from Table 9)	R-404A/R-290	1.00	0.90
	R-410A/R-290	0.89	0.91
Assuming equal compressor efficiency	R-404A/R-290	1.00	0.90
	R-410A/R-290	1.00	1.00

Effects of a Receiver under Full-Load Conditions

Since a typical commercial refrigeration system has a receiver for refrigerant management, the test was done with and without the receiver under the full-load conditions for the R-404A and R-410A system in order to know the effects of receiver. However, for R-290 system, the test was done only without the receiver to minimize refrigerant charge for safety reasons. The receiver was installed at the outlet of the condenser. The effects of the receiver for R-290 system could be assumed based on the test results for the R-404A and R-410A system.

The test was done with various refrigerant charges to know the effects of refrigerant charge variation. Based on the test results, the capacity, COP, evaporating pressure, and condensing pressure including other data are independent of refrigerant charge variation until the receiver is full of refrigerant. After the receiver is filled with refrigerant, the behavior of the system for additional refrigerant charge is same as that for system without the receiver. Table 13 shows the effects of the receiver under full load conditions. The capacity and COP for both refrigerants is reduced to about 6 to 7%. The capacity for the systems with the receiver is decreasing because the enthalpy at the expansion valve inlet is higher than that without the receiver due to the smaller degree of subcooling, therefore the enthalpy difference between the expansion valve inlet and the evaporator outlet is decreasing for the fixed evaporator outlet conditions. The COP is affected mostly by the decreasing capacity. Since the two-phase refrigerant occupies in larger portion and the subcooled liquid refrigerant occupies in lesser portion for the systems with the receiver than those without the receiver, the refrigerant-side pressure drop across the condenser is higher for the system with the receiver. In addition, the pressure ratio is getting smaller because the subcooled liquid goes to the receiver.

Table 13: Effects of Receiver under Full Load Condition

Refrigerant	R-404A		R-410A	
Receiver	Without	With	Without	With
Frequency [Hz]	60		54	
Charge [kg]	4.4	4.9	3.9	4.7
Capacity [kW]	3.7	3.4	3.9	3.6
COP	0.78	0.74	0.80	0.75
DP_{evap} [kPa]	20.8	21.3	8.1	8.4
DP_{cond} [kPa]	9.9	25.4	34.1	35.8
$P_{evap,avg}$ [kPa, abs]	227	227	295	292
$P_{cond,avg}$ [kPa, abs]	2,013	1,918	2,507	2,496
Subcooling [°C]	8.8	2.3	5.9	2.3
Superheating [°C]	4.6	4.6	4.8	4.8
Pressure Ratio (P_{dis}/P_{suc})	9.46	9.06	8.73	8.77
Mass flow rate [g/s]	43	43	28	28
η_{vol}	0.84	0.85	0.79	0.78
η_{comp}	0.56	0.56	0.50	0.49

5.8 Part Load Test Results

After finishing all full load tests, the part load tests were conducted at the optimum charge that was determined from the full load tests. Table 14 shows the comparison of part load test results. The capacity of the R-290 system was 3% lower than that of the R-404A system but the capacity of the R-410A system was same. The measured COPs of the R-410A and R-290 systems were 2% and 6% higher than that of R-404A system. These results illustrate that the COP of R-290 system is reduced by 6% compared to the COP under full load test conditions. As a result, the performance enhancement of the R-410A system is essentially same as that of R-290 system under the part load conditions assuming same compressor efficiency as can be seen in Table 15. These changes in the performance with the part load conditions agree well with the theoretical cycle efficiency comparison as shown in Figure 3.

Table 14: Part Load Test Results (Ambient temperature at 18.3°C)

Refrigerant	R-404A	R-410A	R-290
Frequency [Hz]	60	54	57
Charge [kg]	4.4	3.9	1.72
Capacity [kW]	4.9	4.9	4.7
COP	1.24	1.27	1.31
DP_{evap} [kPa]	18.9	8.5	8.5
DP_{cond} [kPa]	26.5	64.9	13.6
$P_{evap,avg}$ [kPa, abs]	219	285	179
$P_{cond,avg}$ [kPa, abs]	1,327	1,711	1,004
Subcooling [°C]	7.9	4.7	7.2
Superheating [°C]	5.8	5.3	5.0
Pressure Ratio (P_{dis}/P_{suc})	6.70	6.38	6.00
Mass flow rate [g/s]	44	30	18
η_{vol}	0.91	0.88	0.91
η_{comp}	0.61	0.59	0.61

Effects of Compressor Efficiency under Part Load Conditions

Similar to the full load test conditions, table 15 compares the system performance of these three refrigerants for two cases. The first case is based on the measured compressor efficiency from Table 14. The second case is based on the recalculation of the compressor power consumption assuming the compressor efficiency of the R-410A and R-290 systems being equal to that of the R-404A system. Then the COP of R-410A approaches to that of R-290.

Table 15: Effects of Compressor Efficiency under Part Load Conditions

Case	Refrigerant	Compressor Efficiency Ratio	COP Ratio
Based on measured value (from Table 14)	R-404A/R-290	1.00	0.95
	R-410A/R-290	0.97	0.97
Assuming equal compressor efficiency	R-404A/R-290	1.00	0.95
	R-410A/R-290	1.00	0.99

6 LIFE CYCLE CLIMATE PERFORMANCE (LCCP) ANALYSIS

There are two types of global warming effect. The first one is the direct global warming contribution due to the emission of refrigerants itself. The second is the indirect global warming contribution due to the emission of CO_2 by consuming the energy which is obtained by combustion of fossil fuels. In order to determine the effects of the refrigerants investigated and to analyze both the direct and indirect contributions to global warming calculations were conducted by applying the similar approach used by Spatz and Motta (2003).

6.1 Safety Issue and Energy Efficiency

To meet the safety requirement for R-290 system, the first cost of R-290 system would increase up to 30% as estimated by Threadwell (1994) for a typical residential unit. Powell et al. (2000) also reported the cost increase of HC's related with the electrical safety enhancement as much as $240 to $500 for commercial refrigeration and air-conditioning type applications. If a moderate cost increase of 10% of the first cost is used to enhance the efficiency of HFC blends, this will result in a lower life cycle climate performance (LCCP) for HFC blends. To investigate this scenario, it was assumed that a 10% increase in the first cost was used to enhance the efficiency of the HFCs. Then in order to enhance the efficiency, two options of employing better components were investigated: brushless DC motors (BLDC) for the fans and 48% larger condenser. The comparison of the efficiency shows that the BLDC motors ranged in 200 W to 400 W can reduce the power consumption of the fan motors by 20% as compared to the existing permanent split capacitor motors under their rated condition (ADL, 1999). On the other hand the larger condenser has only minimal benefit. The reduced fan motor power consumption results in a higher capacity and COP for HFCs. Comparisons in Table 16 were conducted for three scenarios. The first scenario implies that the test data are reevaluated on an equal compressor efficiency based on the measured values for the R-404A system. The second scenario implies that the secondary loop unit is employed for only R-290 for safety reason in addition to the equal compressor efficiency. In practice, condensing units with HC refrigerants would be used in secondary loop systems. The secondary loop system may require additional cost and energy penalties due to the additional heat exchanger and pumping requirements and the use of heat transfer fluids. By employing the secondary loop, 9% COP degradation, reported by Sand et al. (1997), was assumed. The third scenario implies that the unit first cost is matched for the three refrigerants by assuming that BLDC motors are employed for only HFC blends and additional safety features are employed only for R-290 without employing the secondary loop in addition to the equal compressor efficiency. Again, the underlying assumption is that the first cost of the R-290 system may be, for example, 10% higher due to the added safety features, and on an equal first cost basis, the HFC systems would employ the additional cost for the BLDC motors. In the second and third scenarios, the relative performances of R-404A and R-410A to that of the R-290 improve approximately 8% to 12% for both the full load and part load test conditions. Since the COP enhancement directly affects the results of the LCCP analysis, it was decided to include three scenarios in the LCCP analysis.

Table 16: Comparison of COPs of Three Refrigerants for Three Scenarios

Case	Refrigerant	COP Ratio (Full Load)	COP Ratio (Part Load)
Test data corrected based on equal compressor efficiency	R-404A/R-290	0.90	0.95
	R-410A/R-290	1.00	0.99
Simulated results based on the secondary loop R-290 and equal compressor efficiency	R-404A/R-290	0.97	1.05
	R-410A/R-290	1.08	1.11
Simulated results based on equal first cost and compressor efficiency	R-404A/R-290	0.97	1.03
	R-410A/R-290	1.08	1.07

6.2 LCCP Comparison (Based on the Equal Compressor Efficiency)

The environmental impact of refrigerants over the entire lifecycle of fluid and equipment, including power consumption, is captured in the LCCP value. The lower the value, the lower the environmental impact. In order to determine the power consumption of a typical refrigeration system over the course of a year, a bin analysis was performed using weather data from an ANSI/ARI Standard for chillers (ANSI/ARI Standard 550, 1998). It uses data averaged from 29 cities across the U.S. Table 17 shows the results of this analysis which was used to determine the indirect global warming contribution by extrapolating the test results under the full load and part load conditions. It should be noted that the following assumptions were used in the calculation.

The refrigeration load is distributed linearly from 100% at 36.4°C and to 75% at 8.6°C. The load ratio is defined as the ratio of the refrigeration load at each temperature bin and that at 36.4°C. The cooling capacity and the power consumption of the system are distributed linearly by using the performance measured under two test conditions. The actual operating hours are obtained by multiplying the load factor (the ratio of the refrigeration capacity and the refrigeration load) to the bin hours. By assuming that the fan motors and compressor are turned on and off simultaneously during the cyclic operation, the system power consumption is obtained by multiplying the power consumption at each bin temperature and the actual operating hours. The compressor efficiencies of three refrigerants are equal. The fan speed is fixed. Four assumptions used in the ADL report (2002) were used: a 0.65 kg of CO_2 per kW-hr of electrical production, a 2% annual leakage rate, a 15% end-of-life loss, and a 15-year life. It should be noted that these assumptions were taken from a split unitary air conditioning system since test equipment consisting of the condensing unit and unit cooler for the walk-in cooler application is very similar in design.

To compare the relative performance potential of each refrigerant at optimum hardware, the effects of varying compressor efficiency are excluded. With this consideration the LCCP was calculated based on an equal compressor efficiency using that measured for R-404A.

Table 17: System Power Consumption - Weather Bin Analysis

Temp. bin (^oC)	Hrs	Load Ratio	kW-hours					
			R-404A with system tested	R-410A with system tested	R-290 with system tested	R-290 with secondary loop	R-404A with BLDC fan motors	R-410A with BLDC fan motors
36.4	37	1.000	180	158	160	173	173	154
33.6	120	0.975	517	462	467	507	497	448
30.8	303	0.950	1,163	1,054	1,063	1,161	1,117	1,021
28.1	517	0.925	1,779	1,633	1,644	1,804	1,709	1,579
25.3	780	0.900	2,421	2,246	2,257	2,488	2,324	2,167
22.5	929	0.875	2,612	2,447	2,455	2,716	2,507	2,357
19.7	894	0.850	2,286	2,160	2,164	2,401	2,193	2,077
16.9	856	0.825	1,997	1,901	1,902	2,117	1,916	1,825
14.2	777	0.800	1,659	1,589	1,588	1,773	1,590	1,524
11.4	678	0.775	1,327	1,279	1,277	1,429	1,272	1,225
8.6	2,869	0.750	5,158	4,997	4,986	5,592	4,944	4,783
Total	8,760	-	21,100	19,925	19,962	22,161	20,243	19,161

(Calculation was done based on the equal compressor efficiency with R-404A system)

With this information, a LCCP analysis was performed for six cases and results are shown in Figure 12 and Table 18. The LCCP analysis shows that R-404A and R-410A have 10% and 2% higher LCCP, respectively, than that of R-290 when the tested system is considered for all three refrigerants. When the BLDC motors are employed for only HFC systems, the LCCPs of R-404A and R-410A and R-290 are 1% higher and 6% lower, respectively, than that of R-290. When HFC systems with the BLDC motors are compared with R-290 with the secondary loop, the LCCPs of R-404A and R-410A are 10% and 16% lower, respectively, than that of R-290. Furthermore, it is very clear from these results that the indirect contributions dominate any contributions from refrigerant emissions.

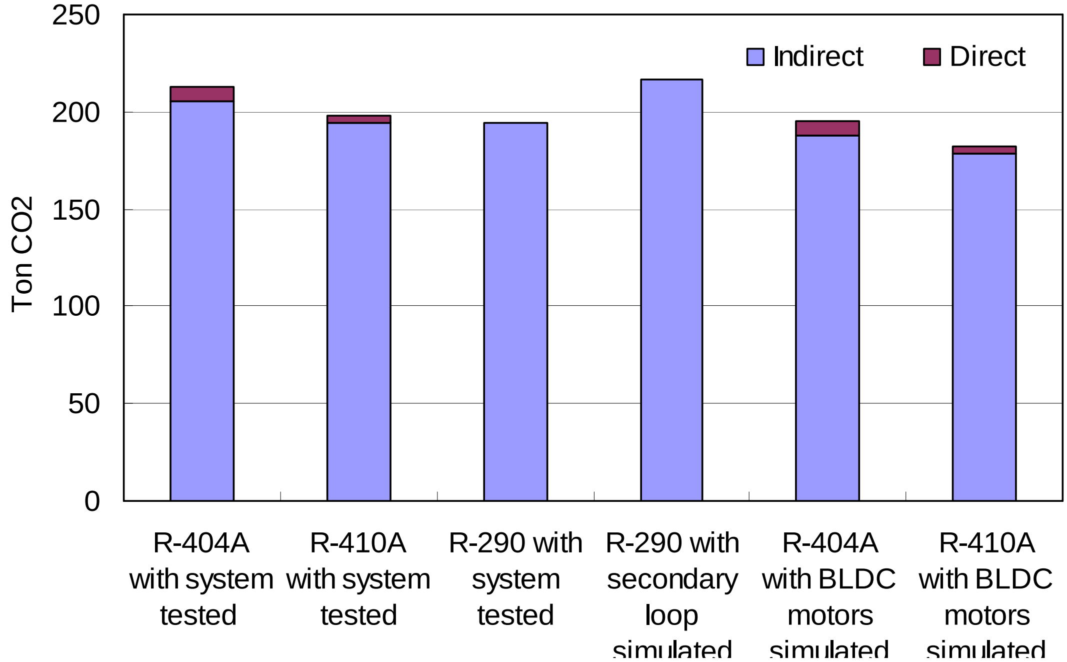


Figure 12: Comparison of LCCP (Based on the equal compressor efficiency)

Table 18: Comparison of LCCP (Based on the equal compressor efficiency)

Unit: CO_2 Ton	Indirect	Direct	Total	Compared to R-290
R-404A with system tested	205.7	7.5	213.2	110%
R-410A with system tested	194.6	3.5	198.1	102%
R-290 with system tested	194.3	0.0	194.3	100%
Unit: CO_2 Ton	Indirect	Direct	Total	Compared to R-290
R-404A with BLDC motors	187.8	7.5	195.3	101%
R-410A with BLDC motors	178.6	3.5	182.1	94%
R-290 with secondary loop	216.1	0.0	216.1	111%
R-290 with safety features	194.3	0.0	194.3	100%

Effects of Annual Emission

ADL report (2002) assumed 15% and 4% leakage rates for the direct expansion system and the distributed system, respectively, whereas it also assumed 2% leakage rate for the unitary equipment. 2% leakage rate was also considered because the hardware configuration of the test unit in the current study is similar to the unitary equipment. To reflect these estimations, five different levels (0%, 2%, 5%, 10%, and 15%) of annual leakage rates were examined. With this information, a LCCP analysis was performed on the basis of equal compressor efficiency and first cost, and the results are shown in Figure 13. If the system is tight (no leakage), then the LCCPs of R-404A and R-410A are 2% and 7% lower, respectively, than that of R-290. This result reflects the fact that the direct emission contribution by the leakage is eliminated and the difference in the annual power consumption dominates the LCCPs of three refrigerants. However, as the annual leakage rate increases from 0% to 5%, the LCCP of R-404A is 4% higher than that of R-290, while the LCCP of R-410A is 4% lower than that of R-290. If the annual leakage rate increases up to 10%, then the LCCPs of R-404A and R-410A are 11% higher and 1% lower than that of R-290. If the annual leakage rate increases up to 15%, then the LCCPs of R-404A and R-410A are 17% and 2% higher than that of R-290. This result indicates that the LCCP of R-404A is always greater than the two other refrigerants as the annual emission is kept greater than 2%. The LCCP of R-410A is lower than that of R-290 as long as the annual emission is kept below 10%.

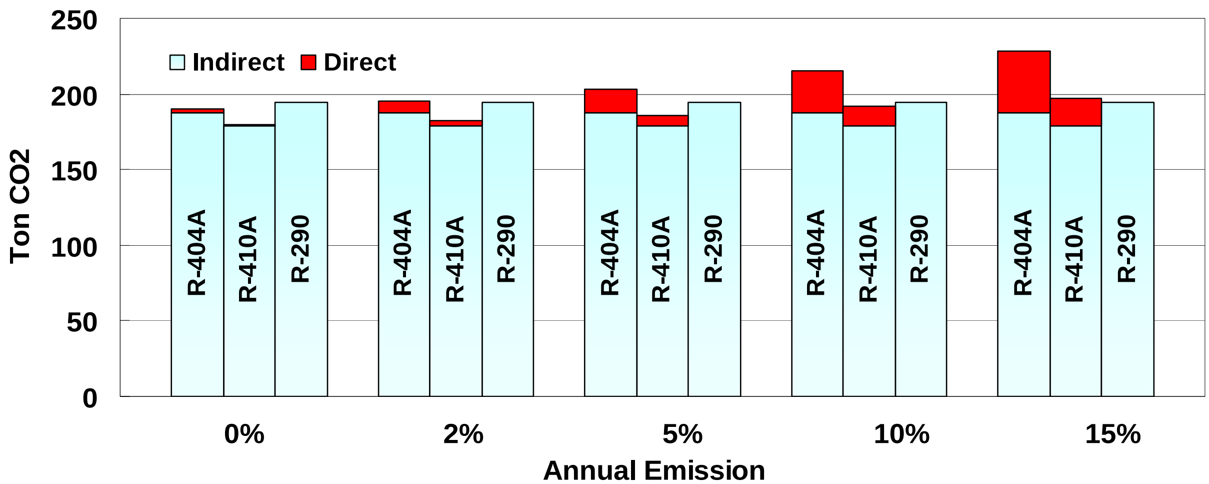


Figure 13: Comparison of LCCP at Various Annual Emissions (Based on the equal compressor efficiency and first cost)

7 CONCLUSIONS

There is continued growing environmental awareness at the international level with particular focus on the working fluids of refrigeration systems, heat pumps and air conditioners. Worldwide governmental policy efforts to reduce global warming are directing industry to develop innovative technologies to reduce emissions while also increasing energy efficiency. Despite the flammability of hydrocarbons, some refrigerator manufacturers especially in European countries and Asian countries have started employing hydrocarbons as refrigerants predominantly in small capacity equipment. These issues have led to call for the careful investigation of currently used refrigerants (HFC's) and potentially applicable HC refrigerants (R-290). To help provide a clear understanding of the relative performance potential of HFC's (R-404A and R-410A) as compared to R-290 for low temperature commercial refrigeration, CEEE conducted an experimental evaluation program under ARI/ICARMA's GREEN Program.

In order to test the performance of three refrigerants for low temperature commercial refrigeration, the experimental facility which was designed and fabricated by CEEE was used for this study. A 4 kW capacity refrigeration system consisting of a unit cooler and a condensing unit, which was originally designed for R-404A, served as the test unit. To match the capacity between refrigerants, compressors having a 19% smaller and 19% larger displacement volume than that for R-404A were selected for R-410A and R-290, respectively from the production compressors. Since the selected displacement volume of the R-410A and R-290 compressor was slightly different from the target displacement, 54 Hz and 57 Hz was used to match the refrigeration capacity by using an inverter drive, respectively for R-410A and R-290 system. In order to know the effects of receiver, the test was conducted with and without a receiver for R-

404A and R-410A systems. However, the test for R-290 system was conducted only without the receiver because of the safety reasons to minimize the charge of R-290. The condenser was also modified to integrate a liquid subcooler circuit as a part of the condenser. Based on the optimization of the condenser, a two circuit condenser was used for the testing of R-410A while a three circuit condenser was used for the testing of R-404A and R-290 systems. The air-side configurations and specifications of all condensers were identical.

Charge optimization tests of three refrigerants systems were completed at the full load conditions. Results show that the optimum charge of R-404A was 4.4 kg while the optimum charge of R-410A and R-290 was 89% and 39% of R-404A charge, respectively. Once the refrigerant charge was optimized, each refrigerant was tested both under the full load and part load conditions. Based on equal system capacity test results, the COPs of R-404A and R-410A were 10% and 9% lower, respectively, than that of R-290 under the full load conditions, and they were 5% and 3% lower, respectively, under the part load conditions.

Since the compressor selected for R-410A is designed for the medium temperature refrigeration and not optimized for the low temperature refrigeration, and the compressor manufacturer states that the compressor can be designed to have the same compressor efficiency for three refrigerants, the other comparison was made based on the equal compressor efficiency. Based on the same compressor efficiency assumption, the comparison shows that the COPs of R-404A are 10% and 5% lower, respectively, under the full load and part load test conditions as compared to R-290 and the COPs of R-410A are essentially the same with that of R-290 for both test conditions.

In order to determine the environmental impact of the refrigerants investigated, an LCCP analysis was conducted. To compare the LCCP, it is assumed that the same cost increase of 10% for R-290 to meet safety requirement, is used for the two HFC blends to employ brushless DC motors (BLDC) used both for the condensing unit and the unit cooler to enhance the efficiency. The system simulation results of employing BLDC motors for R-404A and R-410A show a 8% COP enhancement under both the full load and part load conditions as compared to the tested system case. In order to compare the refrigerants at optimum hardware condition, the LCCP of three refrigerants was computed based on the equal compressor efficiency with that measured for R-404A and the same 2% annual leakage rate. Then the LCCP analysis shows that R-404A and R-410A have 10% and 2% higher LCCP, respectively, than that of R-290 when the tested system is considered for all three refrigerants. When the BLDC motors are employed for only HFC systems the LCCPs of R-404A and R-410A are 1% higher and 6% lower, respectively, than that of R-290. When HFC systems with the BLDC motors are compared with R-290 with the secondary loop the LCCPs of R-404A and R-410A are 10% and 16% lower, respectively, than that of R-290. Furthermore, it is very clear from these results that the indirect contributions dominate any contributions from refrigerant emissions.

Working fluid selection should consider many aspects including safety (toxicity and flammability), environmental impact (stratospheric ozone and climate change), cost and performance (capacity and COP). The two most representative commercial refrigeration configurations are the direct expansion and distributed systems, either of which could potentially release the refrigerant into human occupied space. Therefore, the use of either flammable or high toxicity refrigerants is not feasible. To limit these cases, potentially hazardous refrigerants should be limited to unoccupied spaces.

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