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Performance of R432A to Replace HCFC22 in Residential Air-conditioners and Heat Pumps

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ABSTRACT

In this study, thermodynamic performance of R432A and HCFC22 is measured in a heat pump bench tester under both air-conditioning and heat pumping conditions. R432A has no ozone depletion potential and very low greenhouse warming potential of less than 5. R432A also offers a similar vapor pressure to that of HCFC22 for 'drop-in' replacement. Test results showed that the coefficient of performance and capacity of R432A are 8.5-8.7% and 1.9-6.4% higher than those of HCFC22 for both conditions. The compressor discharge temperature of R432A is 14.1-17.3℃ lower than that of HCFC22 while the amount of charge for R432A is 50% lower than that of HCFC22 due to its low density. Overall, R432A is a good long term 'drop-in' environmentally friendly alternative to replace HCFC22 in residential air-conditioners and heat pumps due to its excellent thermodynamic and environmental properties.

1. INTRODUCTION

For the past few decades, HCFC22 has been predominantly used in residential air-conditioners and heat pumps. The parties to the Montreal protocol, however, decided to phase out HCFC22 eventually since it contains ozone depleting chlorine (UNEP, 1987). Hence, the regulation for HCFC production has begun from 1996 in the developed countries and for new equipment HCFC22 is not currently used in European Union and it can not be used in United States from 2010.

For the past years, various alternatives for HCFC22 have been proposed (Cavallini, 1996; Jung *et al.*, 2000a; Park and Jung, 2007) and tested (Jung *et al.*, 2000a; Park and Jung, 2007) in an effort to comply with the Montreal protocol. At this time, HFC refrigerant mixtures such as R410A and R407C are used in some countries to replace HCFC22 (Cavallini, 1996; Calm and Domaski, 2004). At the same time, many companies expend much effort to develop their own alternatives for HCFC22. Especially, refrigerant mixtures composed of environmentally safe pure refrigerants have gotten a special attention from the industry with the expectation of possible energy efficiency without major changes in the system (Didion and Bivens, 1990).

These days, greenhouse warming has become one of the most important global issues and Kyoto protocol was proposed to resolve this issue, which classified HFCs as greenhouse gases (GECR, 1997). With this global trend in view, HFC134a will be banned in mobile air-conditioners of new vehicles from 2011 according to EU F-Gases regulation and MAC directive which specifically bans the use of refrigerants having global warming potential (GWP) of more than 150 (DEPC, 2006). For reference, the GWP of HFC134a is 1300. At this time, many EU countries seriously consider the ban of the use of even HFCs in residential air-conditioners and heat pumps (Cox, 2004). Even though HFC refrigerants of R410A and R407C are used in some systems, their future is not certain since their GWPs are 1700-2000 which is even higher than that of HFC134a.

One of the possible solutions to avoid HFCs of high GWP is the use of natural refrigerants such as hydrocarbons. For the past few decades, flammable hydrocarbon refrigerants have been prohibited in normal air-conditioning applications due to a safety concern. These days, however, this trend is somewhat relaxed because of the environmental and energy issue. Therefore, some of the flammable refrigerants have been applied to certain

applications (Kruse, 1996; Jung *et al.*, 2000b). Propane (R290) and propylene (R1270) are used for heat pumping applications in Europe (IEHPC, 2002). It is well known that hydrocarbons offer low GWPs of less than 5, low cost, availability, compatibility with the conventional mineral oil, and environmental friendliness (Kruse, 1996; Jung *et al.*, 2000b). Furthermore, dimethylether (DME, RE170) is a good environmentally friendly refrigerant having excellent thermodynamic properties (Jung *et al.*, 1999).

Recently, ASHRAE listed R432A as a possible candidate to replace HCFC22 (ANSI/ASHRAE Addenda, 2007). R432A is a near azeotropic mixture composed of 80% propylene (R1270) and 20% dimethylether (RE170) by mass. It has no ozone depletion potential and very low GWP of less than 5. Its gliding temperature difference during phase change is 1℃ with normal boiling point of -46.6℃.

In this study, thermodynamic performance of R432A was measured in an attempt to examine the possibility of 'drop-in' substitution of HCFC22 for air-conditioning and heat pumping applications

2. EXPERIMENTS

2.1 Experimental apparatus

To achieve the goal of this paper, a breadboard type heat pump/air-conditioner was designed and built in our laboratory. Figure 1 shows the schematic diagram of the experimental heat pump. The nominal capacity of the heat pump is roughly 1 ton of refrigeration (3.5 kW).

The evaporator and condenser were manufactured by connecting 8 pieces of pre-manufactured double tube commercial pipes (E-stick) in series. Each pipe stick is 740mm long and inner and outer diameter are 19.0 mm and 25.4 mm respectively. Figure 2 shows the detailed connection of the pipe sticks. The total length and heat transfer area based on the inner diameter of the evaporator and condenser are 5.92 m and 0.3536 m^2 respectively. Both evaporator and condenser were designed to be counter-current and the secondary heat transfer fluid passed through the inner tube while the refrigerant flowed through the annulus. Throughout the tests, water was used as the secondary fluid for both evaporator and condenser and precision water chiller and heating bath of 0.1℃ accuracy were used to control the temperatures of the water entering into the condenser and evaporator respectively.

A rotary compressor designed originally for HCFC22 was used to lift the pressure. A fine metering needle valve was used as an expansion device to control the refrigerant mass flow rate. Even though a suction line heat exchanger (SLHX) was installed initially to examine the effect of SLHX, it has not been used during this study.

A liquid eye was installed at the exit of the condenser to see the state of the refrigerant coming out of the condenser. A filter drier was installed before the expansion valve to remove contaminants. Charging ports were made at the inlet of the evaporator for liquid and at the inlet of the compressor for vapor. Finally, to reduce the heat transfer to and from the surroundings condenser and evaporator were heavily insulated with polyurethane foams and fiberglass insulation.

2.2 Measurements

More than 40 copper-constantan thermocouples were installed along the evaporator and condenser to measure the refrigerant and water temperatures. Also the compressor dome and discharge pipe temperatures were measured to compare them against those of HCFC22. All thermocouples were calibrated before their use against a precise RTD thermometer of 0.01℃ accuracy. Pressures were measured at the inlets and outlets of the evaporator and condenser using calibrated pressure transducers. Finally, mass flow rates of the secondary heat transfer fluid (HTF) and refrigerant were measured by precision mass flow meters.

Refrigeration capacity was determined by measuring the mass flow rate and temperature difference of water in the evaporator side. This temperature difference of water was measured by a 6-point thermopile whose performance was calibrated by a set of RTDs of 0.01℃ accuracy. On the other hand, the power input to the compressor was measured by a precision torque meter. All data were taken by a computerized data logging system.

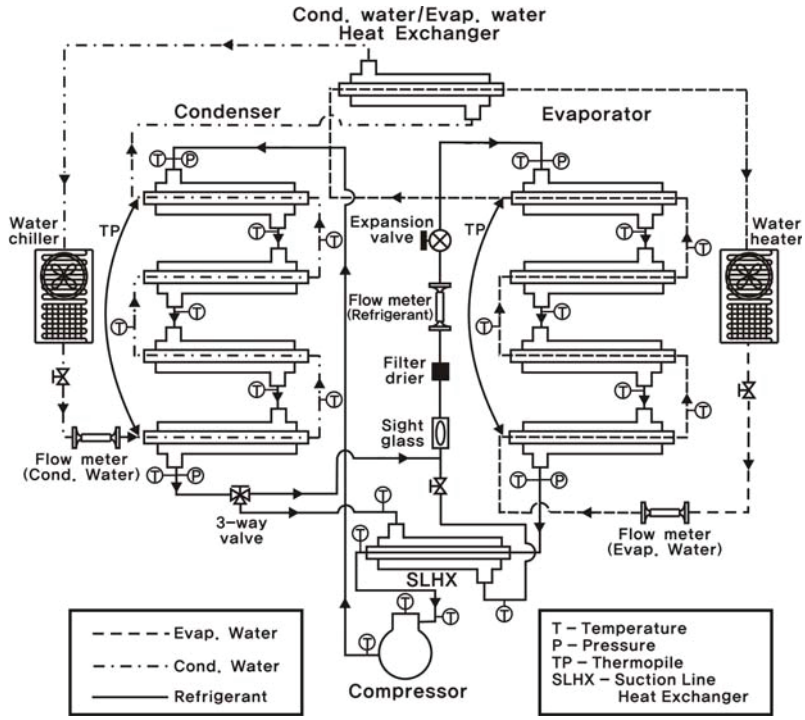

Figure 1: Schematic of a heat pump bench tester

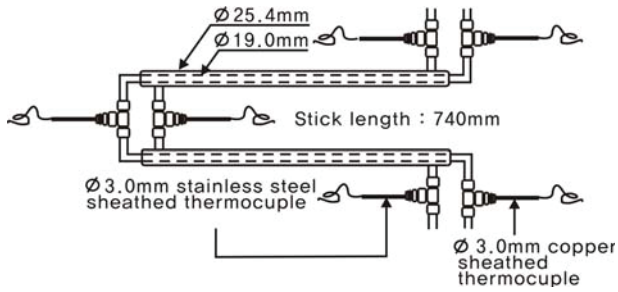

Figure 2: Details of the evaporator and condenser connection

2.3 Test Condition

To compare the performance of refrigerants correctly, a fair test condition should be employed. For this purpose, all tests were conducted with the heat transfer fluid (HTF) temperatures fixed in this study. For air-conditioning simulation, HTF (water/ethylene glycol mixture in the evaporator and water in the condenser) temperatures at the inlet and outlet of the evaporator were set to roughly 26.0℃ and 11.0℃ respectively while those at the inlet and outlet of the condenser were set to roughly 30.0℃ and 42.0℃ respectively. Under this condition, the saturation temperatures of HCFC22 and R432A in the evaporator and condenser were roughly 7℃ and 45℃ respectively.

For heat pumping simulation, HTF temperatures at the inlet and outlet of the evaporator were set to roughly 10.0℃ and 1.0℃ respectively while those at the inlet and outlet of the condenser were set to roughly 30.0℃ and 39.0℃ respectively. Under this condition, the saturation temperatures of HCFC22 and R432A and in the evaporator and condenser were roughly -7℃ and 41℃ respectively. In fact, for both conditions the HTF temperatures at the outlets of the evaporator and condenser varied a little bit due to the difference in capacity between two refrigerants tested.

For all tests, the super heat at the evaporator exit and subcooling at the condenser exit were kept to be 5℃. The amount of charge was adjusted to maintain the same superheat and subcooling under the same external conditional. Finally, as for the lubricant, a conventional mineral oil was used for both R432A and HCFC22.

2.4 Test Procedures

Test procedure is as follows:

(1) The system was evacuated for 2-3 hours before charging.

(2) The temperatures in the chiller and heating bath were set and the secondary HTF was pumped into the evaporator and condenser, and the system was charged with a specific refrigerant. For pure fluids, the system was charged with a vapor refrigerant at the compressor inlet. For a premixed mixture, however, the system was charged with a liquid refrigerant at the evaporator inlet. For new mixtures, the system was charged with a lower vapor pressure refrigerant at the compressor inlet, which was followed by a higher vapor pressure fluid. A digital scale of 0.1 g accuracy was employed to measure the amount of charge.

(3) The expansion valve was controlled, and simultaneously the amount of charge was adjusted to maintain the constant superheat and subcooling, usually 5℃ each, at the exits of evaporator and condenser.

(4) When the system reached steady state for more than 1 hour, data were taken every 30 seconds for more than 30 minutes.

3. RESULTS AND DISCUSSION

Tables 1 and 2 list various measured system parameters such as the coefficient of performance (COP), capacity, compressor work, pressure ratio, discharge temperature, and amount of charge for HCFC22 and R432A obtained under air-conditioning and heat pumping conditions respectively. For each refrigerant, tests were performed at least 3 times and test results agreed within 1% repeatability.

3.1 Coefficient of performance

In refrigeration and air-conditioning, COP is a measure of energy efficiency for a given device charged with a specific refrigerant. Hence, it is important to examine, first of all, the COP of R432A against HCFC22. Figure 3 shows the COPs of HCFC22 and R432A for air-conditioning and heat pumping conditions. As seen in this figure, the COP of R432A is 8.5-8.7% higher than that of HCFC22 under both conditions. One of the reasons for the improved efficiency is the decrease in pressure ratio (PR) across the compressor. As listed in Tables 1 and 2, the PR of R432A decreased 9.7% as compared to that of HCFC22. The decreased pressure ratio in turn results in a decrease in compressor work, which was also seen in Tables 1 and 2. Test results demonstrate that R432A is a good alternative to replace HCFC22 in air-conditioners and heat pumps from the standpoint of energy efficiency.

Table 1: Test results for HCFC22 and R432A under air-conditioning condition

Refrigerants	COP	Diff. in COP (%)	Q_e (W)	Diff. in Q_e (%)	Work (W)	Diff. in Work (%)	PR	T_{dis} (℃)	Charge (g)
HCFC22	3.41		3734		1096		2.99	84.7	1300
R432A	3.70	8.5	3806	1.9	1028	-6.2	2.70	70.6	650

Table 2: Test results for HCFC22 and R432A under heat pumping condition

Refrigerants	COP	Diff. in COP (%)	Q_c (W)	Diff. in Q_c (%)	Work (W)	Diff. in Work (%)	PR	T_{dis} (℃)	Charge (g)
HCFC22	3.68		3472		943		4.27	94.1	1300
R432A	4.00	8.7	3693	6.4	923	-2.1	3.86	76.8	650

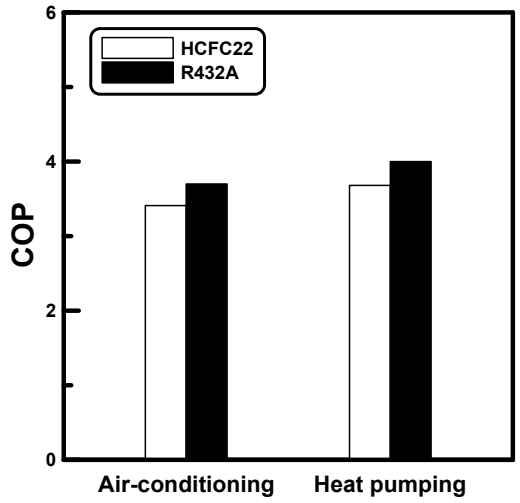

Figure 3: COP of HCFC22 and R432A for two conditions

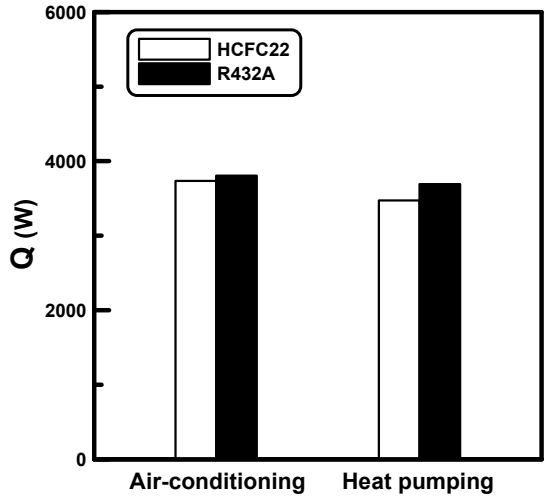

Figure 4: Capacity of HCFC22 and R432A for two conditions

3.2 Capacity

The capacity is as important as COP in refrigeration. Figure 4 and Tables 1 and 2 show the evaporator cooling capacity, Q_e for air-conditioning and the condenser heating capacity, Q_c for heat pumping as well as changes in capacity (Diff. in Q_e and Diff. in Q_c) of R432A as compared to HCFC22 for a given compressor. R432A showed 1.9-6.4% higher capacity under both conditions. Especially, the capacity difference between R432A and HCFC22 was larger under heat pumping condition. This is a good feature considering the inherent problem of heat pumps that the heating capacity decreases as the outdoor temperature decreases.

Test results indicate that R432A is a good 'drop-in' replacement without requiring major changes in major components including the compressor. In fact, resizing and redesigning of compressors is very costly and the 'drop-in' feature of R432A is very advantageous from the viewpoint of manufacturing cost.

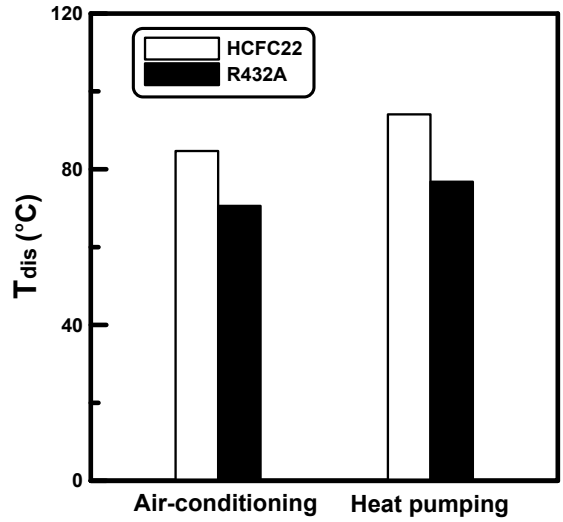

Figure 5: Discharge temperature of HCFC22 and R432A for two conditions

3.3 Compressor discharge temperatures

The lifetime and reliability of the system as well as the stability of the refrigerant and lubricant should be considered when alternative refrigerants are considered. These characteristics can be examined indirectly by measuring the compressor discharge temperature (T_{dis}). Figure 5 shows the compressor discharge temperatures of HCFC22 and R432A for air-conditioning and heat pumping conditions. As listed in Tables 1 and 2, R432A showed 14.1-17.3℃ decrease in discharge temperature. From this observation, it can be safely concluded that R432A would be appropriate from the viewpoint of system reliability and refrigerant stability.

3.4 Refrigerant charge

Most of the hydrocarbons have smaller density than that of the halocarbons and hence the amount of charge decreases significantly with hydrocarbons (Maclaine-cross and Leonardi, 1997). As listed in Tables 1 and 2, R432A showed a decrease in charge of 50.0-51.9% as compared to HCFC22. This will help alleviate further the direct emission of refrigerant which is responsible for the greenhouse warming.

4. CONCLUSIONS

In this study, thermodynamic performance of R432A and HCFC22 was measured in a breadboard type heat pump/air-conditioner under typical air-conditioning and heat pumping conditions. Various performance characteristics were measured and following conclusions were drawn.

- The COP of R432A is 8.5-8.7% higher than that of HCFC22.
- The capacity of R432A is 1.9-6.4% higher than that of HCFC22.
- The compressor discharge temperature of R432A is 14.1-17.3℃ lower than that of HCFC22.
- The amount of charge for R432A is 50% lower than that of HCFC22 due to its low density.
- R432A is a good long term 'drop-in' environmentally friendly alternative refrigerant to replace HCFC22 in residential air-conditioners and heat pumps due to its excellent thermodynamic and environmental properties.

NOMENCLATURE

COP	coefficient of performance
GWP	global warming potential
PR	pressure ratio

Subscripts

c	condenser
dis	discharge

Q	capacity	(W)	e	evaporator
T	temperature	(℃)		

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ACKNOWLEDGEMENT

This study was supported by MK Chemical, Inc. and Inha University.