PERFORMANCE OF HCFC22 ALTERNATIVE NATURAL REFRIGERANTS FOR AIR-CONDITIONING AND HEAT-PUMPING APPLICATIONS

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ABSTRACT

In this study, performance of 2 pure hydrocarbons and 7 mixtures was measured in an attempt to substitute HCFC22. The mixtures were composed of propylene (R1270), propane (R290), HFC152a, and dimethylether (RE170, DME). The pure and mixed refrigerants tested have GWPs of 3-58 as compared to that of CO_2 and the mixtures are all near-azeotropic showing the gliding temperature difference (GTD) of less than 0.6℃. Thermodynamic cycle analysis was carried out to determine the optimum compositions and actual tests were performed in a breadboard type laboratory heat pump/air-conditioner at the evaporation and condensation temperatures of 7 and 45℃ respectively. Test results show that the coefficient of performance (COP) of these mixtures is up to 5.7% higher than that of HCFC22. While propane showed 11.5% reduction in capacity, most of the fluids had the similar capacity to that of HCFC22. Compressor discharge temperatures were reduced by 11-17℃ with these fluids. There was no problem with mineral oil since the mixtures were mainly composed of hydrocarbons. The amount of charge was reduced up to 55% as compared to HCFC22. Overall, these fluids provide good performance with reasonable energy savings without any environmental problem and thus can be used as long term alternatives for residential air-conditioning and heat pumping applications.

Key Words: *natural refrigerants, hydrocarbons, residential air conditioners, heat pumps*

1 INTRODUCTION

HCFC22 has been predominantly used in residential air-conditioners and heat pumps for the past few decades and its sales volume has been the largest among various refrigerants. Even though the ozone depleting potential of HCFC22 is not as high as other CFCs, it still contains ozone depleting chlorine and hence the parties to the Montreal protocol decided to phase out HCFC22 eventually and the regulation for the HCFC production has begun from 1996 in the developed countries (UNEP 1987).

For the past few years, various alternative refrigerants have been proposed (Radermacher and Jung 1993, Cavallini 1996) and tested (Jung et al. 2000) in an effort to comply with the Montreal protocol. At this time, HFC refrigerant mixtures such as such as R410A and R407C are being used in some nations (Calm and Domanski 2004). R410A is a near azeotropic mixture with a gliding temperature difference (GTD) of less than 0.2℃. Its vapor pressure is roughly 50% higher than that of HCFC22 and hence the capacity increases significantly with R410A. Due to high pressure, compressors needs to be redesigned completely and also the heat exchangers needs to be optimized to accommodate lower volumetric flow rates associated with the use of R410A. Even though a simple thermodynamic cycle analysis shows that the cycle efficiency of R410A is somewhat lower than that of HCFC22, the actual energy efficiency of R410A is similar to that of HCFC22 due to the improved compressor efficiency and reduced energy losses in some components of the refrigeration system.

On the other hand, R407C is a nonazeotropic refrigerant mixture (NARM) whose GTD is roughly 6℃. Its vapor pressure is similar to that of HCFC22 and hence it is expected that R407C may be used in existing equipment without major changes. Since it is a NARM, however, fractionation may occur in the case of the leak in the system (Didion and Bivens 1990). And also the heat transfer degradation associated with NARMs might cause performance degradation of heat exchangers when R407C is adopted. At present, the trend is such that R410A will be adopted in the new systems while R407C will be used in the existing systems.

At this time, many countries expend much effort to develop their own alternative refrigerants for HCFC22. Especially, refrigerant mixtures composed of environmentally safe pure refrigerants got a special attention from the industry with the expectation of possible energy efficiency increase with these fluids.

These days, greenhouse warming has become one of the global issues and Kyoto protocol was proposed which classified HFCs as one of the greenhouse warming gases (GECR 1997). Hence many EU countries consider the ban of the use of even HFCs in air-conditioners and heat pumps (Cox 2004). For instance, Denmark began taxing HFCs from 2001 and also proposed a regulation that no HFCs should be used in new equipment from 2007 and made very severe regulations on dealing with HFCs and ester oil (Cox 2004). So far, Scandinavian countries led this kind of regulations in EU and this trend will be spread out not only in that region but also in other parts of the world.

One of the possible solutions to avoid HFCs is the use of natural refrigerants such as hydrocarbons. For the past few decades, flammable hydrocarbon refrigerants have been prohibited in normal refrigeration and air-conditioning applications due to a safety concern. These days, however, this trend is somewhat relaxed because of an environmental mandate. Therefore, some of the flammable refrigerants have been applied to certain applications (Kruse 1996, Jung et al. 2000). Isobutane (R600a) has dominated the European refrigerator/freezer sector for the past decade and is being used in Japan and Korea while propane (R290) and propylene (R1270) are used for heat pumping applications in Europe (IEA Heat Pump Center 2002). It is well known that hydrocarbons offer low cost, availability, compatibility with the conventional mineral oil, and environmental friendliness (Kruse 1996, Jung et al. 2000). Furthermore, dimethylether (DME, RE170) is a good environmentally friendly refrigerant showing excellent thermodynamic properties (Jung et al. 1999).

In this study, performance of 2 pure hydrocarbons and 7 mixed refrigerants containing hydrocarbons, HFC152a, and DME is measured in an attempt to examine the possibility of substituting HCFC22 used in residential air conditioners and heat pumps. These fluids all have no ozone depletion potential and also offer relatively low GWPs of less than 60 and hence can be used as long term candidates. Also, in this study an emphasis is given to those fluids that provide a similar capacity to that of HCFC22 and hence they do not require a compressor change in the existing equipment.

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 (3.5kW).

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 19.0mm and 25.4mm 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 5.92m and 0.3536m^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.

An open type compressor designed for HCFC22 was chosen which was driven by an electric motor. 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.
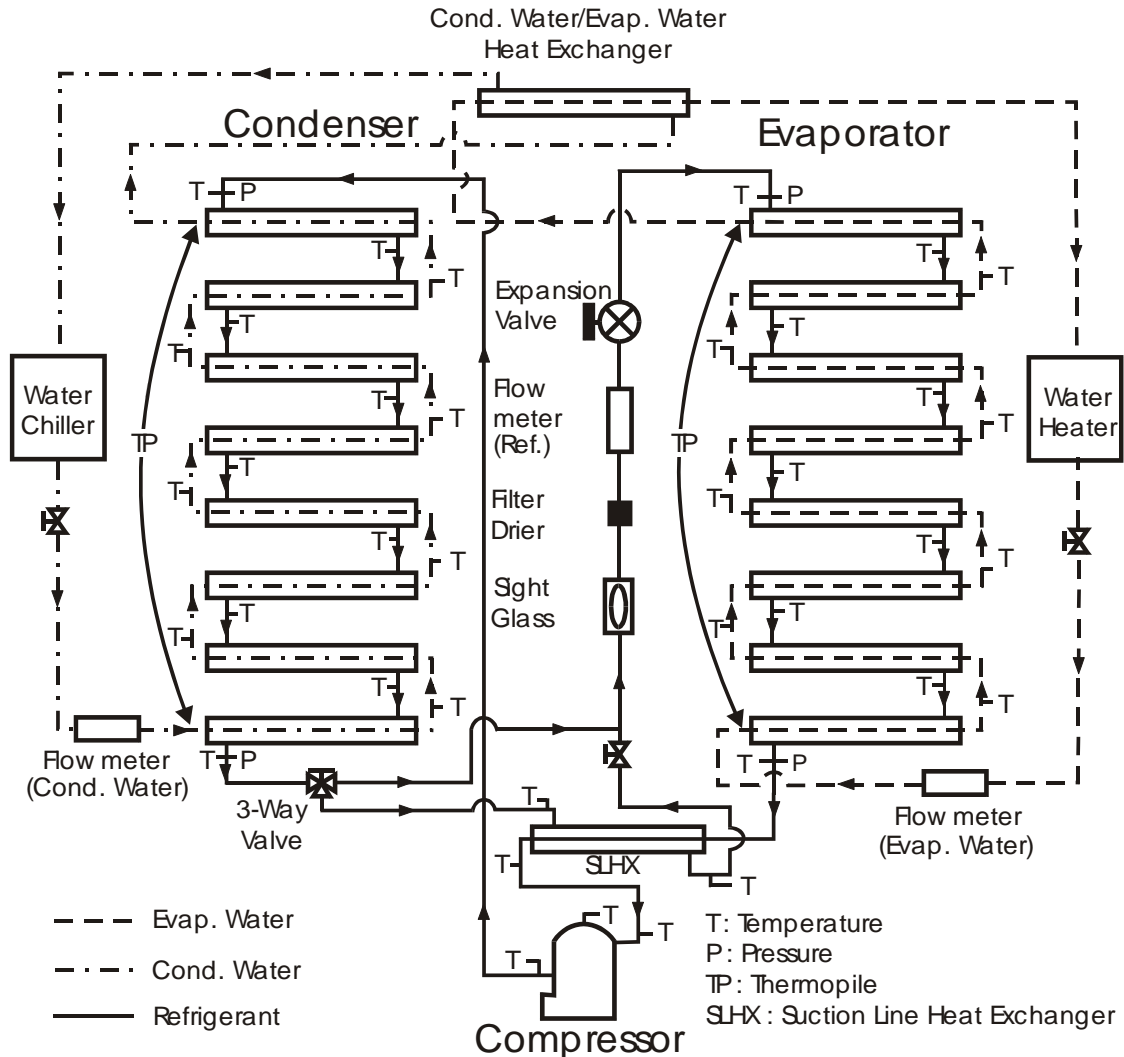

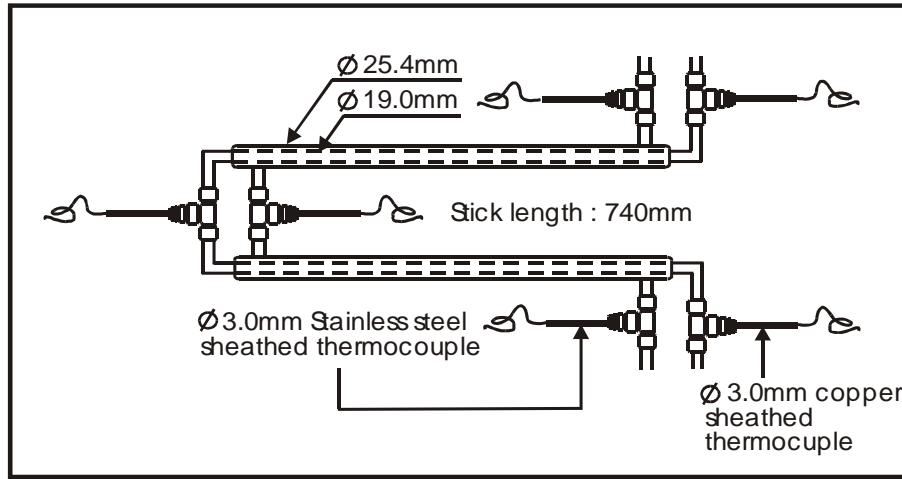**Fig. 1. Schematic diagram of a breadboard heat pump/air-conditioner**

Fig. 2. Details of the evaporator and condenser connection

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.

2.3 Test Condition

To compare the performance of various refrigerants, a fair test condition should be employed. For this purpose, all tests were conducted with the HTF temperatures fixed. Hence, HTF of water temperatures at the inlet and outlet of the evaporator were roughly 23.0℃ and 12.0℃ respectively while those at the inlet and outlet of the condenser were roughly 35.0℃ and 43.0℃ respectively. In fact, the water temperature at the outlet of the condenser varied a little bit due to the difference in capacity among the fluids tested. After setting the external HTF conditions, the evaporation and condensation temperatures of all refrigerants tested were close to 7℃ and 45℃ respectively.

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 0.1g 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.

2.5 Refrigerants and Lubricants

Table 1 lists the refrigerants tested in this program. HCFC22, 2 pure hydrocarbons and 7 mixtures were tested altogether. In fact, a thermodynamic cycle analysis was performed to obtain optimum compositions for the mixtures. As for the lubricant, a conventional mineral oil was used for all refrigerant tested.

Table 1. Refrigerants tested in this study

Ref. number	Refrigerants	Temp. glide (℃)	GWP
1	R22	0	1,700
2	R290(propane)	0	<3
3	R1270(propylene)	0	<3
4	20%R1270/80%R290	0.57	<3
5	50%R1270/50%R290	0.33	<3
6	80%R1270/20%R290	0.03	<3
7	60%R290/40%R152a	0.25	58
8	71%R290/29%R152a	0	43
9	75%R290/25%R152a	0.25	37
10	45%R1270/40%R290/15%DME	0.59	<3

3   RESULTS AND DISCUSSION

In this study, performance of 10 refrigerants was measured in a breadboard type heat pump/air-conditioner under a typical air-conditioning temperature condition. For each refrigerant, tests were performed at least 2-3 times and test results usually agreed within 1% repeatability. Table 2 lists various measured system parameters such as COP, capacity, discharge temperature, and charge for all fluids tested.

3.1 Energy Efficiency

In order to alleviate greenhouse warming, the energy efficiency of energy conversion devices should be improved. In refrigeration and air-conditioning, coefficient of performance (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, COPs of various refrigerants against the reference fluid in selecting alternative fluids.

Figure 3 and Table 2 show the measured COPs and changes in COP (Diff. COP) as compared to HCFC22 for various refrigerants tested. As shown in Figure 3 and Table 2, the COPs of all alternative refrigerants are up to 5.7% higher than that of HCFC22 except that the COP of R1270 is 0.7% lower than that of HCFC22. These results show that all fluids tested can be alternatives to HCFC22 from the standpoint of energy efficiency.

Table 2. Test results for various refrigerants

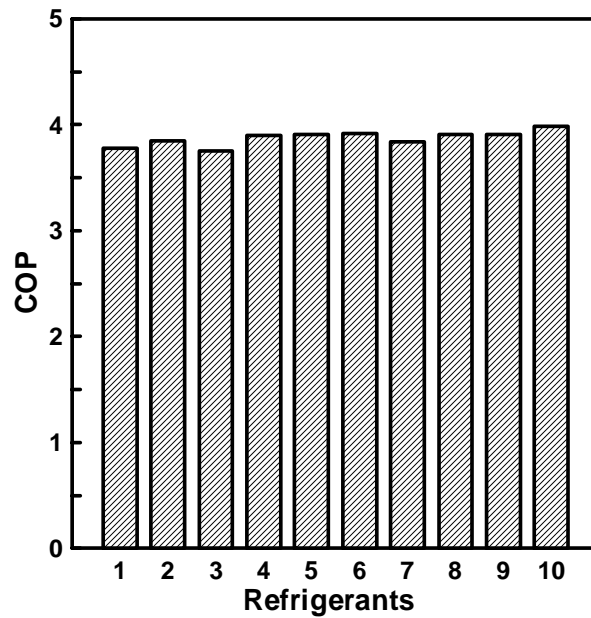| Ref. number	Refrigerants	COP	Diff. COP (%)	Q_e (W)	Diff. Q_e (%)	T_{dis} (℃)	Diff. T_{dis} (℃)	Charge (g)
1	R22	3.78		3600		80.2		1170
2	R290(propane)	3.85	1.9	3187	-11.5	63.0	17.3	520
3	R1270(propylene)	3.75	-0.7	3808	5.8	69.1	11.7	540
4	20%R1270/80%R290	3.90	3.4	3362	-6.6	63.8	16.5	525
5	50%R1270/50%R290	3.91	3.5	3589	-0.3	65.5	14.7	550
6	80%R1270/20%R290	3.92	3.8	3729	3.6	67.4	12.9	530
7	60%R290/40%R152a	3.84	1.8	3572	-0.8	64.9	15.3	630
8	71%R290/29%R152a	3.91	3.6	3533	-1.9	64.4	15.9	600
9	75%R290/25%R152a	3.91	3.6	3527	-2.0	64.6	15.6	600
10	45%R1270/40%R290/15%DME	3.99	5.7	3551	-1.4	67.5	12.7	540

Fig. 3. COP of various refrigerants

3.2 Capacity

Refrigeration capacity is as important as COP in refrigeration. If the capacity of an alternative refrigerant deviates too much from that of the reference fluid, the compressor must be redesigned completely which would be quite costly. Therefore, it would be good for the alternative refrigerants to provide a similar capacity to that of the reference fluid.

Figure 4 and Table 2 show the capacities (Q_e) and changes in capacity (Diff. Q_e) as compared to HCFC22 for various refrigerants for a given compressor. Propane showed 11.5% decrease in capacity as compared to HCFC22 while propylene showed 5.8% increase in capacity. All other mixtures showed similar capacity as compared to HCFC22. 50%R1270/50%R290 and 60%R290/40%R152a showed almost the same capacity as that of HCFC22.
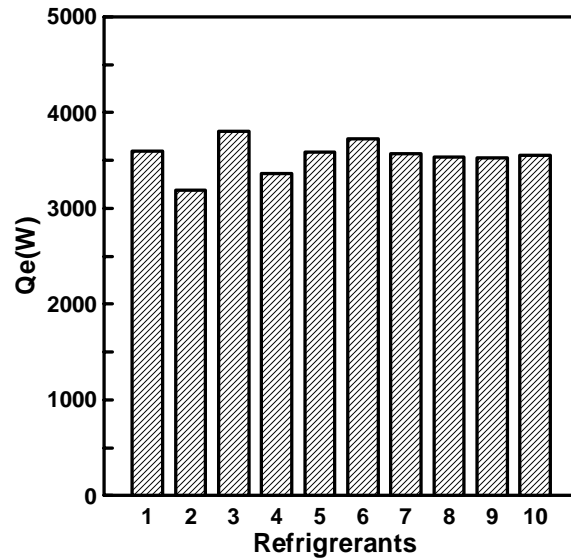

Fig. 4. Refrigeration capacities of various refrigerants

3.3 Compressor Discharge Temperatures

In applying alternative refrigerants, the lifetime and reliability of the system as well as the stability of the refrigerant and lubricant should be considered as well. These characteristics can be examined indirectly by measuring the compressor discharge temperature (T_{dis}). In this study, a thermocouple was attached to the compressor discharge line with 3mm insulation around the sensors and hence the temperature deviation due to the change in surrounding is very small.
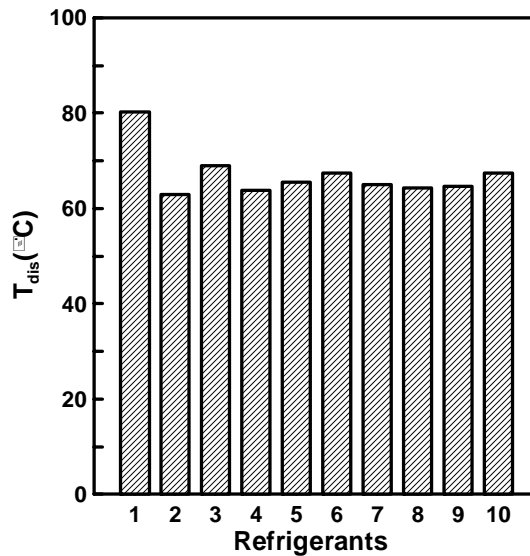

Fig. 5. Compressor discharge temperatures of various refrigerants

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Figure 5 and Table 2 shows the compressor discharge temperatures. All alternative fluids tested in this study showed 11-17℃ decrease in compressor discharge temperature. From this observation, it can be safely concluded that these alternative fluids would be appropriate from the viewpoint of system reliability and fluid stability.

3.4 Refrigerant Charge

Most of the hydrocarbons have smaller density than that of most of the halocarbons and hence the amount of charge decreases significantly with hydrocarbons (Maclaine-cross and Leonardi 1997). As Table 2 illustrates, all refrigerants tested showed a decrease in charge of up to 55% 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, refrigeration performance of HCFC22, 2 pure hydrocarbons and 7 mixed refrigerants composed of hydrocarbons and HFC152a and DME was measured in a breadboard type water cooled heat pump/air-conditioner under a typical air-conditioning condition. Various performance characteristics of these fluids were measured and following conclusions were drawn.

1. COPs of these fluids are similar to or better than that of HCFC22. 45%R1270/40%R290/15%DME mixture showed the highest COP which is 5.7% higher than that of HCFC22.

2. Capacities of propane (R290) and 20%R1270/80%R290 are lower than that of HCFC22 by 11.5 and 6.6% respectively while other fluids showed a similar capacity to that of HCFC22.

3. Compressor discharge temperature of all fluids tested were lower than that of HCFC22 by 11-17℃. This indirectly indicates that these fluids would show long term stability and reliability.

4. The refrigerant charge for all refrigerants tested was reduced up to 55% as compared to HCFC22 due to their lower density.

5. Finally, more elaborate tests are to be performed in actual residential air-conditioners before applying any of these fluids tested in this study since the test heat pump is designed only for the preliminary evaluation of the refrigerants. Actual COPs and capacities might vary due mainly to the difference in heat exchangers used for the tests.

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