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A Study on the Performance of HCFC22 and Alternative Refrigerants in Heat Pumps

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Key words : HCFC22 alternatives, Hydrocarbons, Refrigerant mixtures, Heat pumps, Industrial chillers

Abstract

This paper is concerned about the performance of HCFC22 alternative refrigerants used in heat pumps and industrial chillers. A water-to-water breadboard heat pump with counter-current heat exchangers and a hermetic compressor was built to carry out the experiments with various refrigerants. For each test, more than 40 temperatures, 4 pressures, power input, mass flow rates of the heat transfer fluids were measured. Refrigerants tested were HCFC22, R290(Propane), an azeotrope of 45%Propane/55%R134a mixture, and a nonazeotropic mixture of Calor 50. All tests were conducted under ARI test A condition. It is found that the COP and capacity of propane were 18% and 2.5% higher than those of HCFC22 while the COP and capacity of 45%Propane/55%R134a mixture were 3.5% and 5.3% higher than those of HCFC22 respectively. Also the COP and capacity of Calor 50 were 17% and 7.8% higher than those of HCFC22. Compressor discharge temperatures of alternative refrigerants were roughly 35℃ lower than that of HCFC22 indicating that these refrigerants are good from the view point of compressor reliability. The charging amounts for the alternative refrigerants were reduced by 40-60% as compared to that of HCFC22. Overall, it can be said that hydrocarbon containing alternative refrigerants are excellent in thermodynamic performance but should be used with considerable care due to their flammability.

1. Introduction

Since 1930s, chlorofluorocarbons(CFCs) and hydrochlorofluorocarbons(HCFCs) have been used extensively in the field of refrigeration and air conditioning due to their favorable characteristics. In particular, for industrial chillers and commercial and residential air conditioning units, HCFC22 has been used extensively due to its excellent thermodynamic and chemical characteristics. In 1974, however, Molina and Rowland suggested that CFCs might be responsible for the destruction of the stratospheric ozone layer. Since then, thorough scientific research efforts were initiated to see if CFCs and HCFCs indeed caused such an environmental hazard. Since the investigation indicated a strong relationship between the CFCs and stratospheric ozone layer depletion, many nations signed the Montreal Protocol in 1987 to regulate the production and trade of ozone depleting substances(UNEP 1987). As a result of this international agreement, CFCs are completely phased out as of January 1996 and HCFCs are regulated to the consumption level of 1989 in developed countries starting from 1996.

For the past decade, many countries proposed and tested various alternative refrigerants in an effort to comply with the Montreal Protocol. Especially, "R22 Alternative Refrigerant Evaluation Program"(R22 AREP) organized by the US Air-conditioning and Refrigeration Institute(ARI 1992-1996) incorporated efforts from the related industries and various research institutions to maximize all the available resources. Initially, R22 AREP suggested a number of possible alternatives including pure HFC134a and propane(R290) as possible candidates to replace HCFC22. As time progressed, however, prospective candidates were narrowed down to the refrigerant mixtures composed of HFCs only.

Nonazeotropic refrigerant mixtures(NARMs) have been proposed for the past few decades as possible candidates to replace the existing refrigerants. Especially, the NARMs composed of environmentally safe pure refrigerants got special attentions from the related industries since the public awareness for the clean environment increased rapidly. Also some NARMs have been shown to improve the energy efficiency along with a capacity control feature under proper conditions, which is not available with pure refrigerants(Kruse, 1981, Vakil, 1981, Moser and Schnitzer, 1984, Berntsson and Schnitzer, 1985, Mulroy et al., 1988, and Jung and Radermacher, 1991).

Figure 1 shows the temperature-concentration diagram of a typical NARM of R22/R114. For NARMs, the saturation temperature changes during evaporation at constant pressure, which is termed a gliding temperature effect. When the secondary heat transfer fluid(HTF) exchanges heat with a NARM in a counter current mode, thermodynamic irreversibility could be reduced by matching the temperature glide of the NARM against that of the HTF, resulting in an increase in the coefficient of performance(COP).

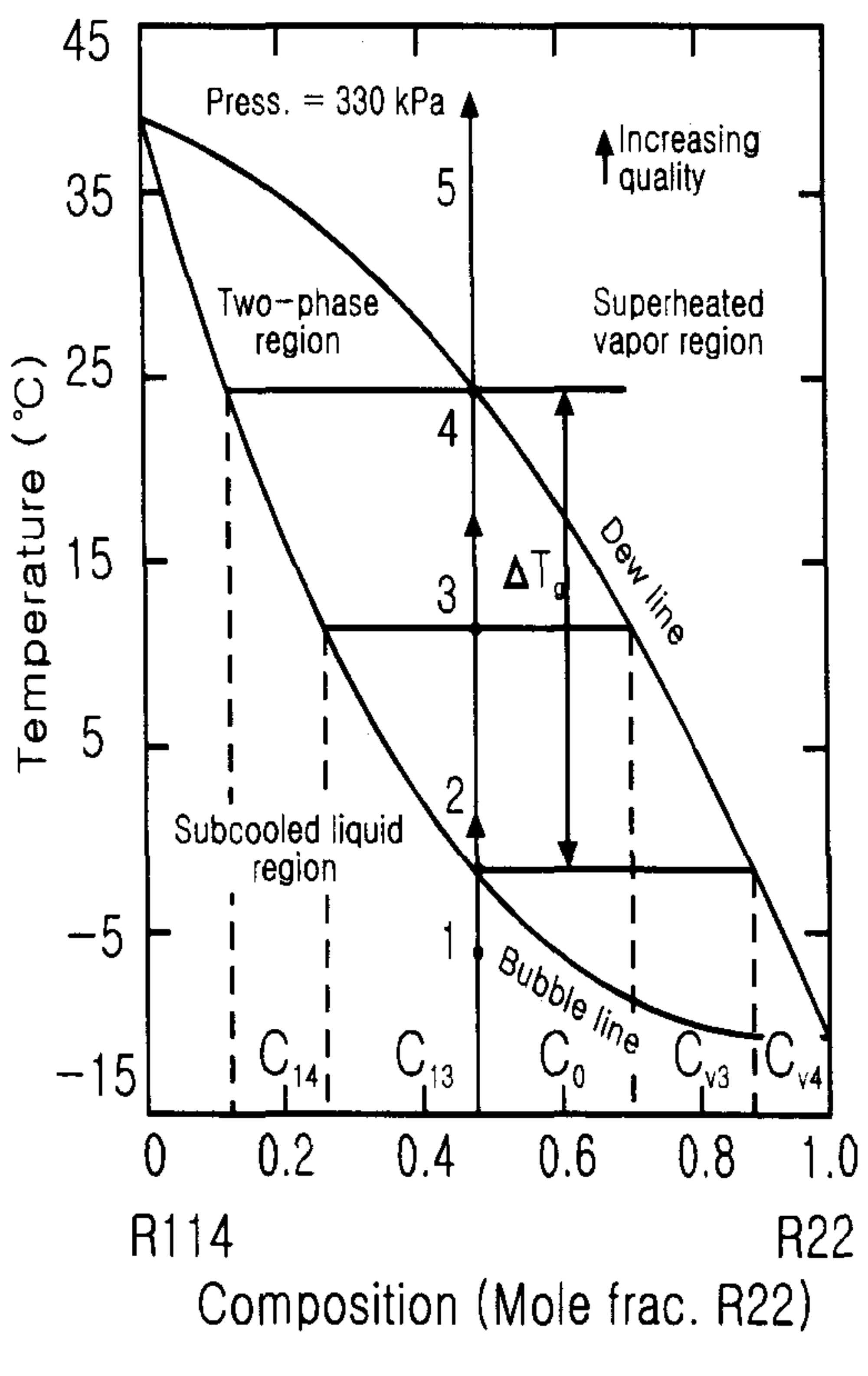

Fig. 1 Temperature-concentration diagram of R22/R114 mixture

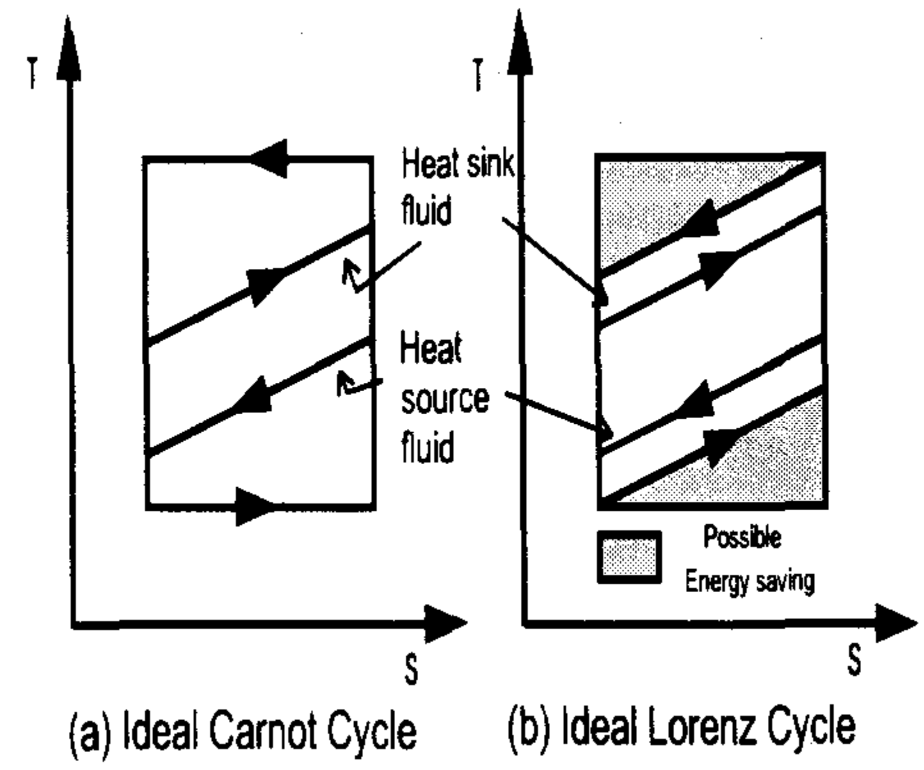

Fig. 2 Ideal Carnot cycle using pure refrigerants and Ideal Lorenz cycle using nonazeotropic mixtures

Fig. 2 shows the ideal Carnot cycle using pure refrigerants and the ideal Lorenz cycle using NARMs. One can see that through the temperature matching, energy efficiency of the Lorenz cycle can be increased significantly as compared to that of the Carnot cycle. Consequently, properly selected NARMs might yield an environmental solution with an increase in energy efficiency without ozone depletion.

Besides the refrigerant mixtures, these days much attention is paid to the so called 'natural fluids' that are claimed to be more environmentally friendly than synthetic fluids. Especially, for domestic refrigerators hydrocarbons and their mixtures were proposed both as a refrigerant and as a foam blowing agent(Liu et al., 1994). Hydrocarbons offer such advantages as low cost, wide availability, compatibility with the conventional mineral oil etc. Their use, however, has been hindered in US and Japan except for some European countries due mainly to their high flammability.

As the concern for the greenhouse warming increases rapidly for the past few years, the use of hydrocarbons seems to be more advocated at the expense of possible risks. Actual test results showed that the charging amounts of hydrocarbons were roughly half that of CFC12 and energy efficiency also increased(Liu et al., 1994). So far, no accidents or damages were reported from the use of the hydrocarbons(GECR 1994). Reflecting this trend, some companies and research institutes have initiated their own evaluation programs to use natural fluids for HCFC22 replacement.

In summary, it can be said that unless pure refrigerants that can replace HCFC22 successfully without causing any penalty in energy efficiency, safety, and cost are found in the near future, hydrocarbons and/or refrigerant mixtures composed of HFCs and hydrocarbons will be the best candidates to replace HCFC22.

In this study, thermodynamic performance of one pure hydrocarbon, one hydrocarbon mixture, and one mixture of a HFC and a hydrocarbon is experimentally measured and compared against that of HCFC22. The results may be applied to any equipment designed for HCFC22 such as room air conditioners, heat pumps, and industrial chillers with and without solar applications.

2. Heat pump cycle description

The concern for the energy conservation has never been greater. From this view point, a certain means has to be considered to improve the energy efficiency of a refrigeration system. One way of increasing the energy efficiency in a refrigeration system is to install a suction line heat exchanger(SLHX), which is a simple device and usually is made of a tube-in-tube type.

Figure 3 shows a schematic diagram of a heat pump with SLHX and the refrigerant flow in this system can be described as follows: Due to the heat exchange with the air stream, evaporation occurs and usually superheated vapor leaves the evaporator at state 1. During evaporation, the refrigerant temperature rises for NARMs(gliding temperature effect) while the temperature remains constant for pure refrigerants if pressure drop is not considered. The refrigerant vapor is heated further via heat exchange through the SLHX to state 2 and compressed to the condenser pressure by a compressor. The vapor at high pressure and temperature at state 3 is desuperheated and condensed in the condenser and usually subcooled liquid leaves the condenser at state 6. This subcooled liquid is further cooled via heat exchange with the suction vapor through the SLHX to state 9. Finally, expansion occurs through an expansion device to complete the cycle and two-phase refrigerant enters the evaporator at state 7.

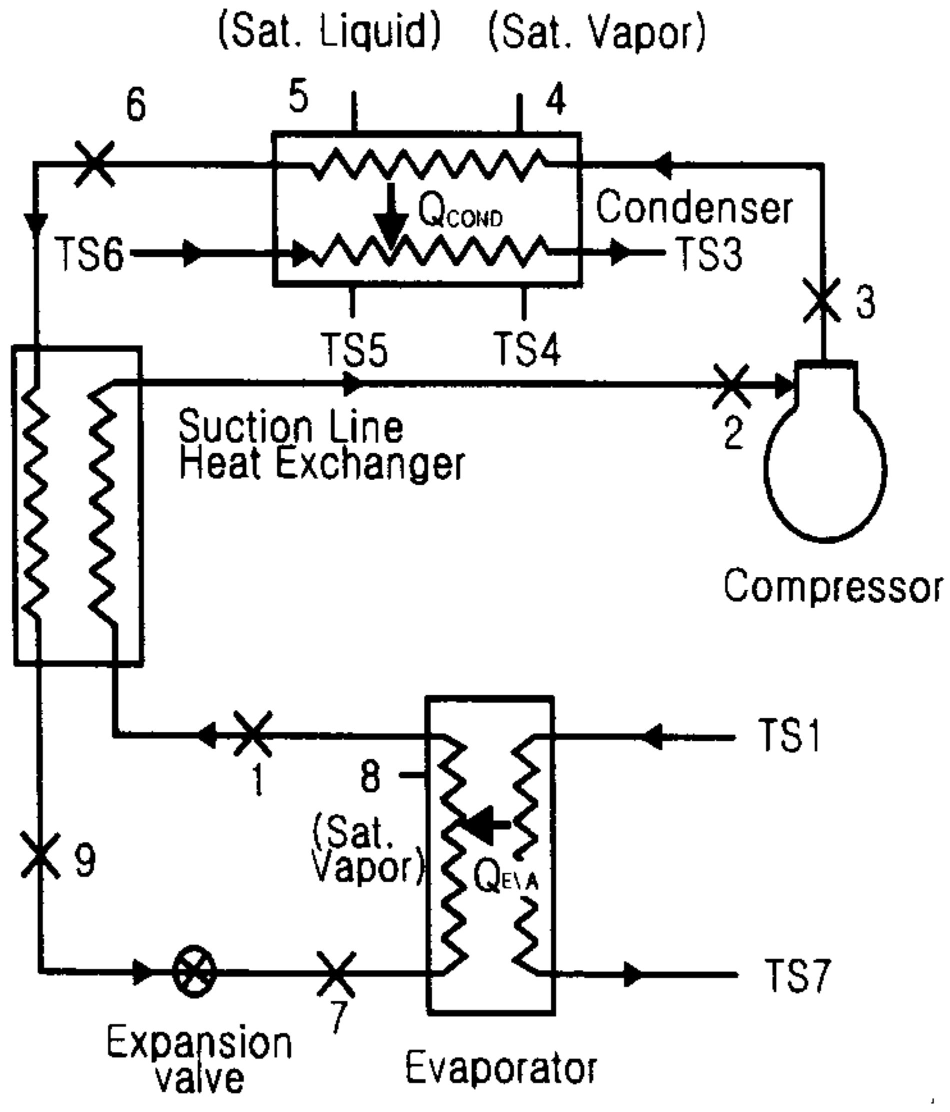

Fig. 3 Schematic diagram of a heat pump with SLHX

3. Experiments

3.1 Breadboard heat pump

To achieve the goal of this paper, a breadboard type heat pump was designed and built in our laboratory. Figure 4 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-made 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 5 shows the details of the connection of the pipes. The total length and heat transfer area based on the inner diameter of the evaporator and condenser are 5.92m and 0.3536m2 respectively. Secondary heat transfer fluid passed through the inner tube while the refrigerant flowed through the annulus. Both evaporator and condenser were designed to be counter-current.

A hermetic compressor designed for HCFC22 was chosen as the system compressor while a fine metering needle valve was used as an expansion device to control the refrigerant mass flow rate. 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. To see the effect of SLHX, another pipe stick of 530mm long, 12.8mm ID, 22.8mm OD was installed. Vapor from the evaporator passed through the inner tube while the liquid from the condenser passed through the annulus in SLHX. A 3-way ball valve and another valve were installed before and after SLHX and by controlling them the refrigerant path to and from the SLHX was controlled and hence the effect of the SLHX could be studied. Charging ports were made at the inlet of the evaporator for liquid charging and at the inlet of the compressor for vapor charging.

Water was used as the secondary heat transfer fluid for both evaporator and condenser and precision water chiller and heating bath were used to control the temperatures of the water entering into the evaporator and condenser. To reduce the heat transfer to and from the surroundings, the evaporator and condenser were insulated with polyurethane foams and fiber glass insulation.

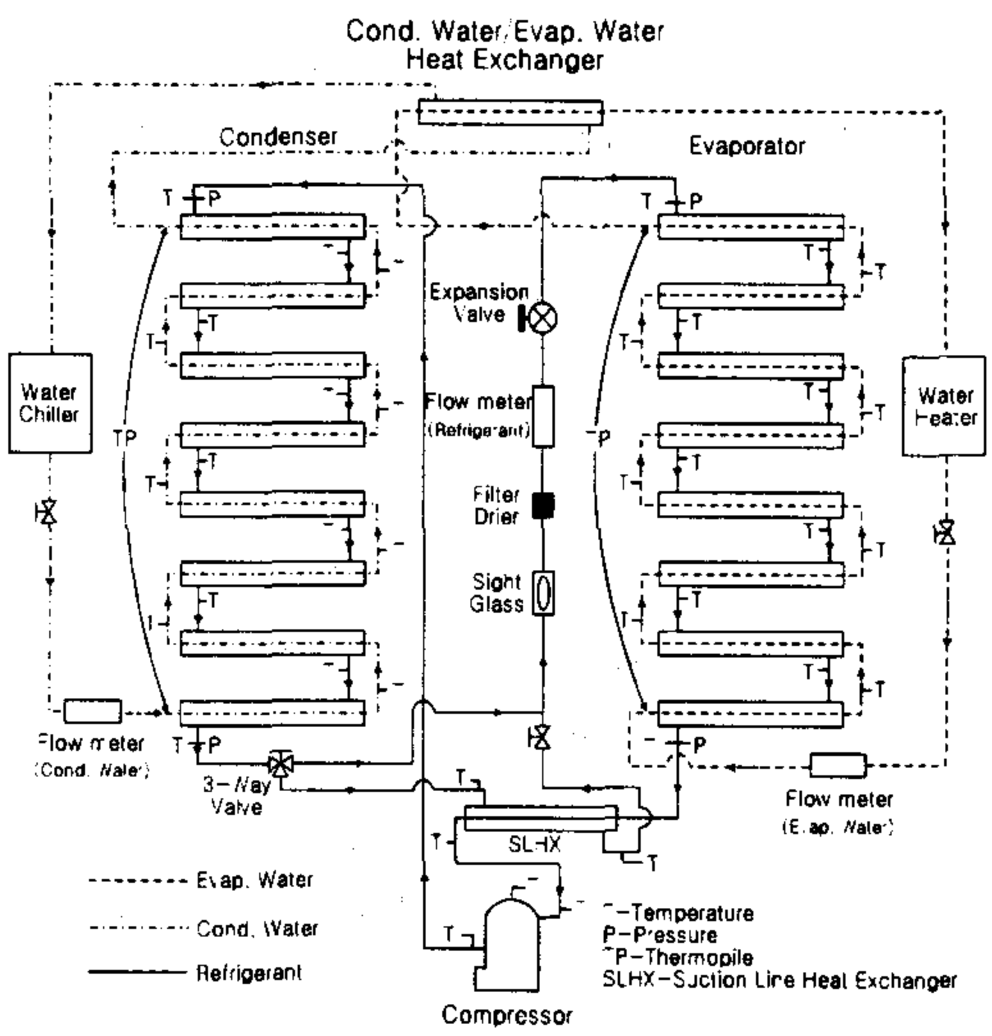

Fig. 4 Breadboard heat pump diagram

3.2 Measurements

More than 40 copper-constantan thermocouples(TCs) were installed along the evaporator and condenser to measure the refrigerant and water temperatures. These TCs were calibrated before their use against a precise RTD thermometer of 0.01C accuracy. 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 0.01C accuracy. Temperatures on the compressor dome and at the compressor discharge pipe were measured also to determine the reliability of the system with alternative refrigerants. Pressures were measured at the inlets and outlets of the evaporator and condenser using precision pressure transducers of 0.1% FS accuracy. On the other hand, power input to the compressor was measured by a digital power meter of 0.1% accuracy. Finally, mass flow rates of the secondary fluid and refrigerant were measured by precision mass flow meters of 0.2% accuracy utilizing coriolis force effect. All data were taken by HP 3852 data logging system which was connected to a personal computer and stored in hard disk for later use.

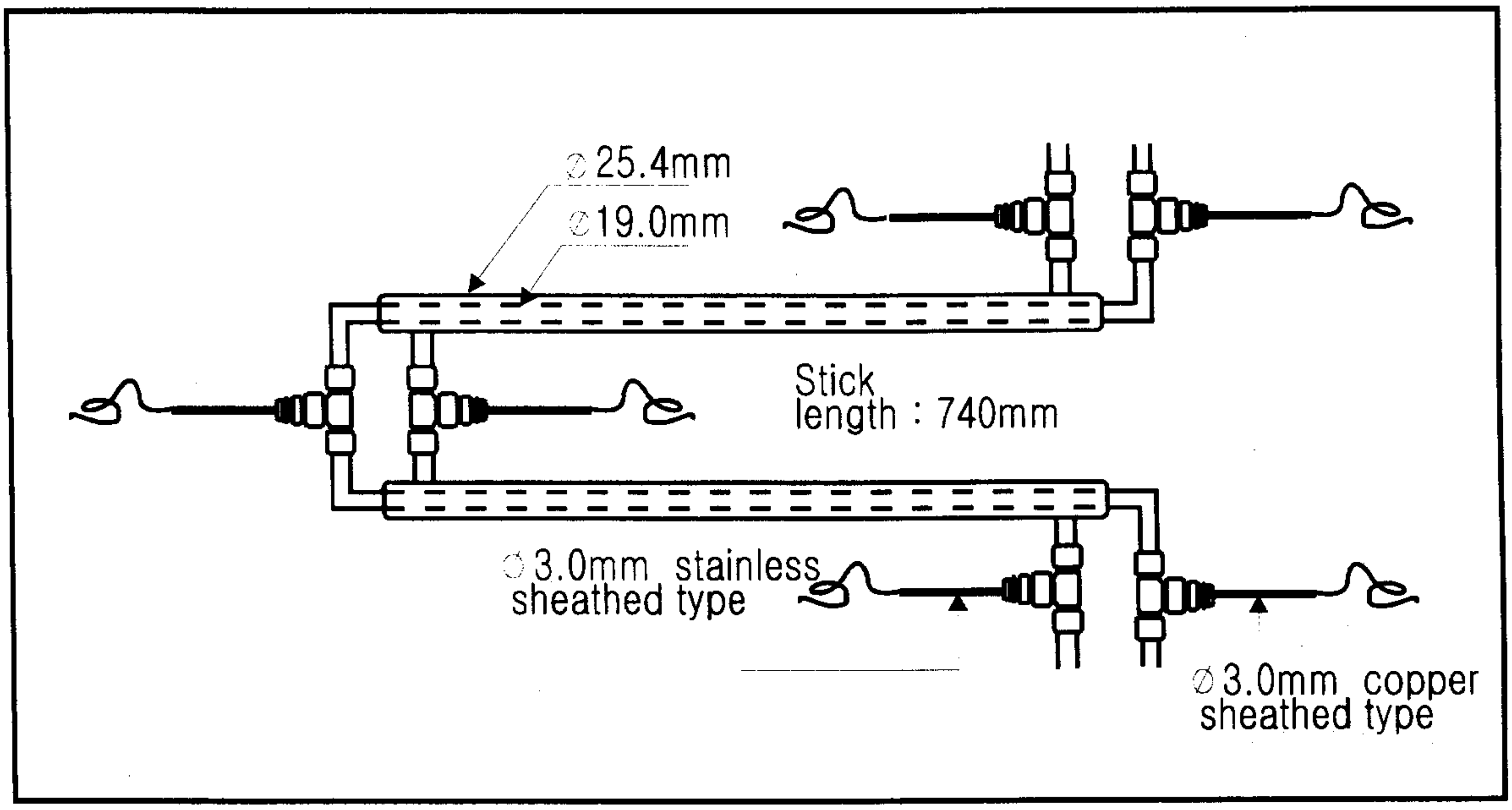

Fig. 5 Details of evaporator and condenser connection

3.3 Test condition and procedure

To compare the performance of various refrigerants, all tests were conducted under ARI test A condition for air conditioners and heat pumps(ARI 1989). Under this condition, water temperatures at the inlet and outlet of the evaporator were fixed to 26.7℃ and 14.4℃ respectively while those at the inlet and outlet of the condenser were fixed to 35.0℃ and 43.2℃ respectively. In fact, the water temperature at the outlet of the condenser varied a little during the tests, but were tried to be maintained at about 43.2℃.

Test procedure is as follows:

(1) Evacuate the system for 2-3 hours before charging the system with a given refrigerant.

(2) Set the temperatures in the chiller and heating bath, pump the secondary fluid into the evaporator, and condenser and charge the system with a specific refrigerant. For pure refrigerants, the system was charged with vapor refrigerant at the compressor inlet. For a premixed mixture, however, the system was charged with liquid refrigerant at the evaporator inlet. For a new mixture, the system was charged with lower vapor pressure refrigerant at the compressor inlet, which was followed by charging the

higher vapor pressure refrigerant. A digital scale of 0.1g accuracy was employed to determine the amount of charge.

(3) Control the expansion valve to maintain the same superheat and subcooling 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.

3.4 Refrigerants and lubricants.

In this study, 4 refrigerants were tested. HCFC22 is the base refrigerant while R290(Propane) is a pure hydrocarbon and Calor 50 is a nonazeotropic mixture of R290 and R170(Ethane) whose composition is not disclosed in public. Finally, 45%R290/55%R134a is an azeotrope. For HCFC22, R290, and Calor 50, traditional mineral oil was used as a lubricant while for 45%R290/55%R134a, polyolester oil of the same viscosity grade was used.

4. Results and discussion

Table 1 shows the COP, refrigeration capacity, discharge temperature, and amount of charge for 4 refrigerants.

Table 1 Test results for 4 refrigerants

Refrigerant	COP	Qe(W)	Dis. Temp. (℃)	Dome Temp. (℃)	Oil	Charge (gr)
HCFC22	2.435	3526	112.2	51.8	Mineral	1200
Calor 50	2.867	3821	77.5	39.0	Mineral	468
R290	2.884	3441	76.9	39.8	Mineral	445
45%R290/55%R134a	2.521	3715	76.0	37.5	Ester	720

Figure 6 shows the COP and change in COP as compared to HCFC22 for the refrigerants tested. As compared to HCFC22, COPs of Calor 50 and R290 are 18-19% higher while the COP of 45%R290/55%R134a is 3% higher. From these results, it can be said that hydrocarbons or hydrocarbon containing mixtures provide better thermodynamic efficiency as compared to HCFC22.

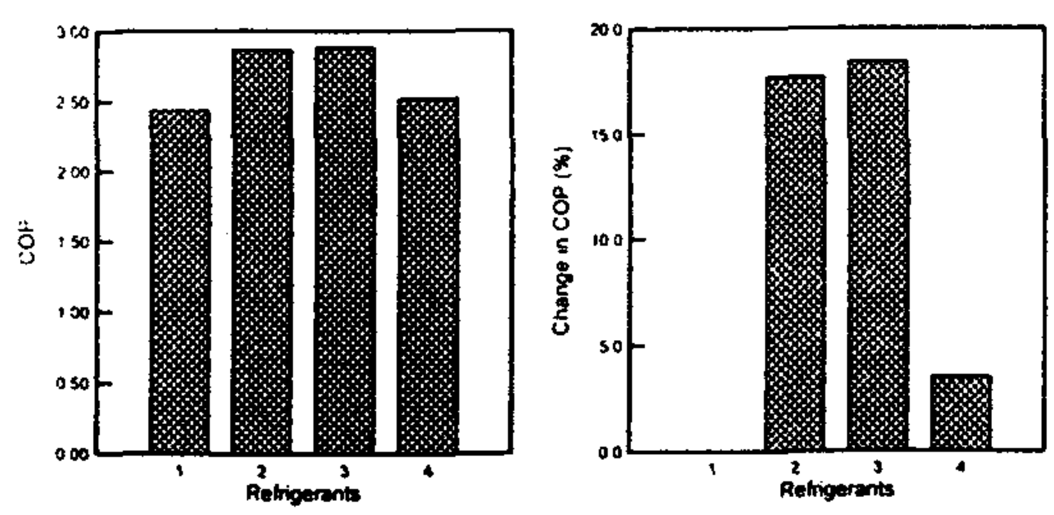

Fig. 6 COP and change in COP as compared to HCFC22

Figure 7 shows the capacity and change in capacity for the refrigerants tested. As compared to HCFC22, the capacities of Calor 50 and 45%R290/55%R134a mixtures are 8.4 and 5.3% higher while that of R290 is 2.4% lower. Overall, it can be said that alternative refrigerants tested in this study provide similar or better capacities than HCFC22.

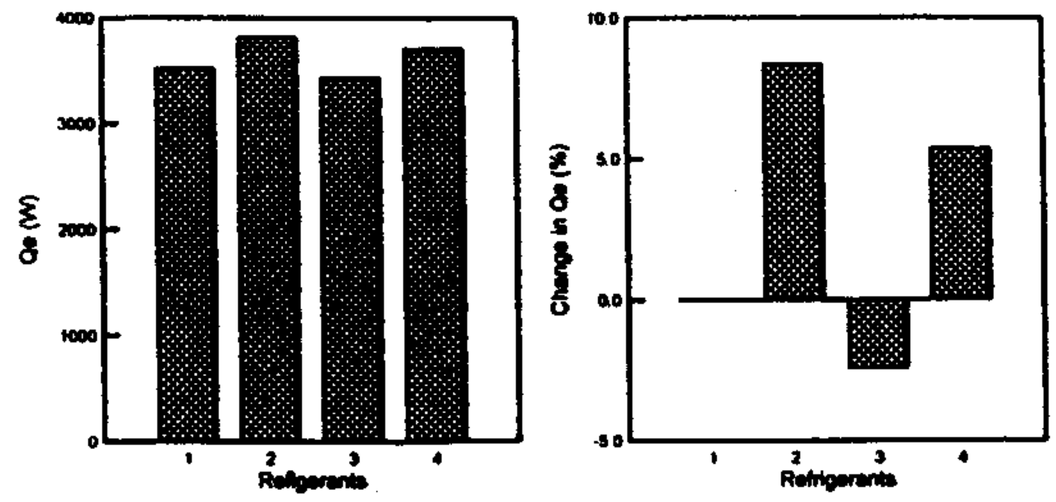

Fig. 7 Capacity and change in capacity as compared to HCFC22

Figure 8 shows the changes in COP and capacity when SLHX was used. The change was based on the performance of each refrigerant when SLHX was not used. Note that no data with SLHX were taken for R290. With the use of SLHX, in general, both COP and capacity were increased for all refrigerants. Especially, 45%R290/55%R134a took the advantage of using SLHX greatly. It, however, can be said that the use of SLHX should be decided carefully for each refrigerant.

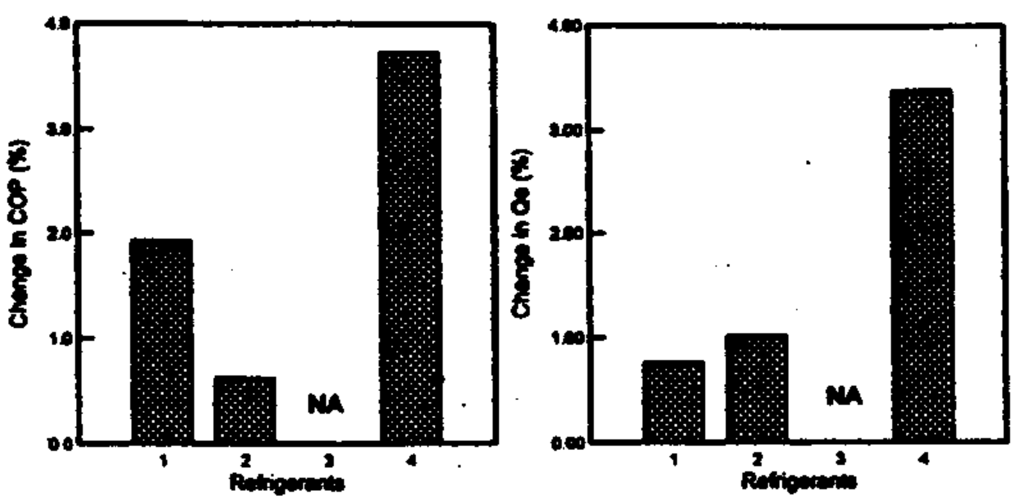

Fig. 8 Effect of SLHX on COP and capacity

Figure 9 shows the compressor discharge and dome temperatures for the refrigerants tested. One can easily see that alternative refrigerants tested in this study show much lower discharge and dome temperatures(35.0 ℃ and 10℃ respectively). From these data, one can say that the system reliability with these alternative refrigerants would be better as compared to HCFC22.

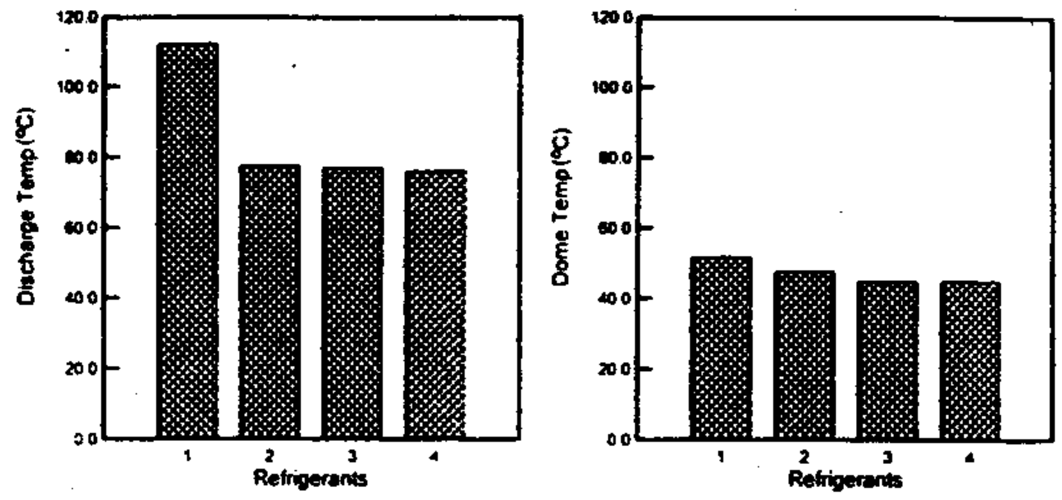

Fig. 9 Compressor discharge and dome temperatures

Finally, as seen in Table 1, the amount of charge decreased significantly as hydrocarbons were used. For example, the amounts of charge for Calor 50 and R290 were almost 40% that of HCFC22 while that of 45%R290/55%R134a was 60% that of HCFC22.

Conclusion

In this study, HCFC22, hydrocarbon R290, Calor 50(hydrocarbon nonazeotropic mixture), and 45%R290/55%R134a (hydrocarb on azeotrope) were tested in a breadboard heat pump under ARI test condition. From the measured data, following conclusions can be drawn.

(1) The COP of the alternative refrigerants tested are up to 18% higher than that of HCFC22.
(2) The capacity of R290 is slightly lower and those of Calor 50 and 45%R290/55% R134a are more than 5% higher than that of HCFC22.
(3) The COP as well as capacity increase up to 3-4% as the suction line heat exchanger is used.
(4) Compressor discharge temperatures of the alternative refrigerants are 35.0C lower than that of HCFC22.
(5) The amounts of charge for hydrocarbons and hydrocarbon containing mixtures are 40-60% lower than that of HCFC22.

Acknowledgment

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