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Performance of R290 and R1270 for R22 applications with evaporator and condenser temperature variation

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Abstract

In this study, therma l performance of two hydrocarbon refrigerants of R290 and R1270 was measured in an attempt to substitute R22. They wer tested in a heat pump bench tester of 1 ton capacity with a hermetic rotary compressor. Water and water/glycol mixture were employed as the secondary heat transfer fluids in the test bench. All tests were conducted under the same external conditions simulating three different air-conditioning and heat pumping conditions. Test results show that the coefficient of performance of these hydrocarbon refrigerants is up to 11.5% higher than that of R22 under all conditions. Refrigeration capacity of R290 is up to 8.2% lower than that of R22 under normal air-conditioning and heat pumping conditions. Under extremely cold temperature conditions, however, the capacity of R290 is 5% higher than that of R22. On the other hand, the capacity of R1270 is similar to that of R22 under all conditions. Compressor discharge temperatures of these hydrocarbons are reduced by 14-31℃ as compared to R22. The amount of charge is reduced up to 58% as compared to R22. Overall, these hydrocarbons provide good performance with reasonable energy savings without any environmental problems and thus can be used as long-term alternatives for residential air-conditioning and heat pumping applications.

Keywords: Alternative refrigerant; COP; Evaporator; Condenser; Heat pump; Air-conditioning; Discharge temperature

1. Introduction

R22 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 R22 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 R22 eventually; the regulation for the HCFC production began in 1996 in the developed countries [1].

For the past few years, various alternative refrigerants have been proposed [2, 3] and tested [4] in an effort to comply with the Montreal protocol. At this time, HFC refrigerant mixtures such as R410A and R407C are being used in some nations [5]. 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 R22 and hence the capacity increases significantly with R410A. Due to high pressure, compressors need to be redesigned completely and also the heat exchangers need 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 R22, the actual energy efficiency of R410A is similar to that of R22 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

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6℃. Its vapor pressure is similar to that of R22 and hence it is expected that R407C may be used in existing equipment without major changes. Since it is an NARM, however, fractionation may occur in the case of a leak in the system [6]. 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 R22. Especially, refrigerant mixtures composed of environmentally safe pure refrigerants have received special attention from the industry with the expectation of possible energy savings with these fluids.

These days, greenhouse warming has become one a global issue and the Kyoto protocol was proposed to alleviate the warming, which classified HFCs as one of the greenhouse warming gases [7]. Hence many EU countries have considered the ban of even HFCs in air-conditioners and heat pumps [8]. For instance, Denmark began taxing HFCs from 2001 and not only proposed a regulation that no HFCs should be used in new equipment from 2007 but also made very severe regulations on dealing with HFCs and ester oil [8]. So far, Scandinavian countries have led this kind of regulation in the 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 HCFCs and 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. Nowadays, however, this trend is somewhat relaxed because of an environmental mandate. Therefore, some of the flammable refrigerants have been applied to certain applications [9, 10]. 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 [11]. It is well known that hydrocarbons offer low cost, availability, compatibility with the conventional mineral oil, and environmental friendliness [9, 10].

In this study, the thermal performance of two pure hydrocarbons of R290 and R1290 was measured under three different temperature conditions simulating summer and winter conditions in an attempt to examine the possibility of substituting R22 used in residential air conditioners and heat pumps. These fluids have no ozone depletion potential and also offer very low GWPs of less than 3 and hence can be used as long-term alternatives.

2. Experiments

2.1 Experimental apparatus

To achieve the goal of this paper, a breadboard type heat pump bench tester was designed and built in our laboratory. Fig. 1 shows the schematic diagram of the experimental heat pump whose nominal capacity is roughly 1 ton of refrigeration (3.5 kW). The evaporator and condenser of the heat pump were manufactured by connecting eight pieces of pre-manufactured double tube commercial pipes (E-stick) in series. Each pipe stick is 740 mm long and inner and outer diameters are 19.0 mm and 25.4 mm, respectively. Fig. 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 countercurrent 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/ethylene glycol chiller and heating bath of 0.1℃ accuracy were used to control the temperatures of the water/ethylene glycol entering

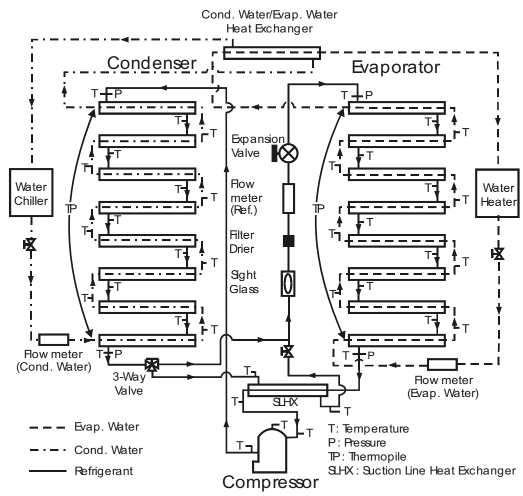

Fig. 1. Schematic diagram of a breadboard heat pump/air-conditioner.

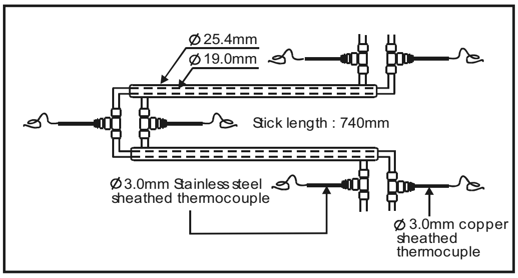

Fig. 2. Details of the evaporator and condenser connection.

into the condenser and evaporator, respectively.

The bench tester was equipped with a hermetic rotary compressor developed for R22. 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 was not 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, the 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 for comparison. 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 by using calibrated pressure transducers. Power input to the compressor was measured by a digital power meter of 0.2% accuracy. Finally, mass flow rates of the secondary heat transfer fluid (HTF) 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. All data were taken by a computerized data logging system.

2.3 Test condition

To properly compare the performance of various refrigerants, a fair test condition should be employed. For this purpose, all tests were conducted with the external HTF temperatures fixed. Tests were performed under three sets of different evaporator/condenser saturation temperatures for R22: 7℃/45℃, -7℃/41℃, and -21℃/28℃. The first condition reflects normal air-conditioning conditions during summer. On the other hand, the second and third ones reflect normal and extreme heat pumping conditions during winter. For a given condition, first of all, tests were carried out for R22 with the adjusted external HTF temperatures to provide the required saturation temperatures in the evaporator and condenser. And then subsequent tests were performed under the same external conditions for two hydrocarbon refrigerants. For a given external condition, actual saturation temperatures of the hydrocarbons in the evaporator and condenser varied a little due to the difference in heat transfer characteristics of these fluids.

2.4 Test procedures

Test procedure for a given condition was as follows:

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

(2) Temperatures in the chiller and heating bath were set, the secondary HTF was pumped into the evaporator and condenser, and the system was charged with a specific refrigerant. For all pure fluids tested in this study, the system was charged with a vapor refrigerant at the compressor inlet. 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.

2.5 Refrigerants and lubricants

In this study, R22 was used a standard reference refrigerant and R290 and R1270 were tested as prospective alternative refrigerants. As for the lubri-cant, a conventional mineral oil was used for all refrigerant tested.

3. Results and discussion

In this study, the thermal performance of two hydrocarbon refrigerants of R290 and R1270 was measured in a breadboard type heat pump tester under three sets of air-conditioning and heat pumping temperature conditions. For each refrigerant, tests were performed at least 2-3 times and test results usually agreed within 1% repeatability. Table 1 lists various measured system parameters such as COP, capacity, discharge temperature, and charge for all fluids tested under three conditions.

3.1 Energy efficiency

To alleviate greenhouse warming, the energy efficiency of energy conversion devices should be improved. In air-conditioning and heat pumping, the 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 tested hydrocarbon refrigerants against the reference fluid in selecting alternative fluids.

Figs. 3 and 4 show the coefficient of performance (COP) of three fluids tested and the change in COP as compared to R22 under three conditions. As seen in these figures, the COP of R290 is 6.1-11.5% higher than that of R22 for all three conditions. In fact, R290 showed the lowest pressure ratio across the compressor among three fluids tested, and hence showed the lowest compressor power. This, in turn, resulted in the highest COP. On the other hand, the COP of R1270 is similar to that of R22 for all three conditions. Even though the refrigeration capacity of R1270 was larger than that of R22, the compressor power of R1270 also increased accordingly.

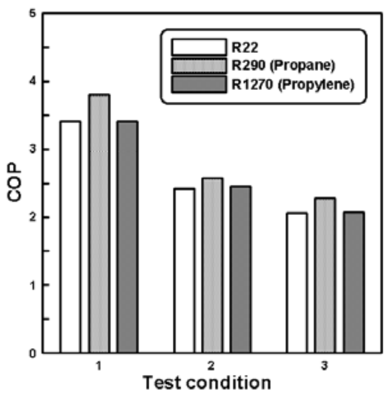

Fig. 3. COP of various refrigerants for three conditions.

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.

Figs. 5 and 6 show the refrigeration capacity (Q_e) of three fluids tested and the change in Qe as compared

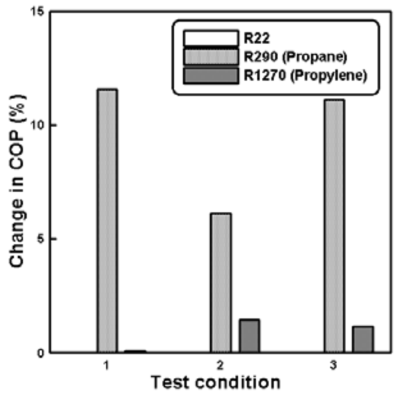

Fig. 4. Change in COP of various refrigerants as compared to R22.

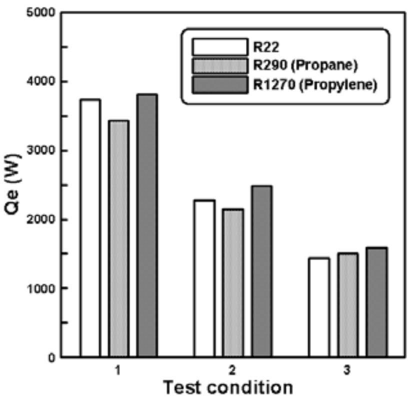

Fig. 5. Refrigerating capacity of various refrigerants for three conditions.

Table 1. Summary of test results for various refrigerants.

Test condition No.	Refrigerants	COP	Compressor power (W)	Pressure ratio	Q_e (W)	$T_{dis.}$ (℃)	Charge (g)
1	R22 R290 (propane) R1270 (propylene)	3.41 3.80 3.41	1096 901 1117	2.74 2.37 2.53	3734 3427 3811	84.8 62.3 70.7	1300 550 580
2	R22 R290 (propane) R1270 (propylene)	2.42 2.57 2.45	943 837 1014	3.96 3.39 3.63	2282 2149 2489	94.1 63.2 75.4	1350 580 600
3	R22 R290 (propane) R1270 (propylene)	2.05 2.28 2.08	698 661 766	4.78 4.16 4.62	1432 1508 1591	93.8 65.8 75.2	1350 620 650

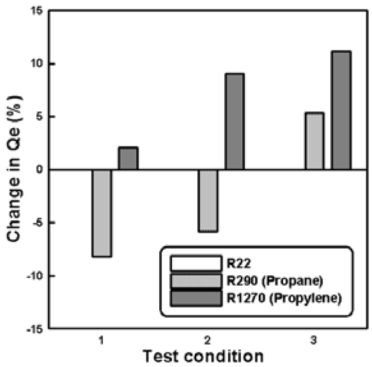

Fig. 6. Change in refrigerating capacity of various refrigerants as compared to R22.

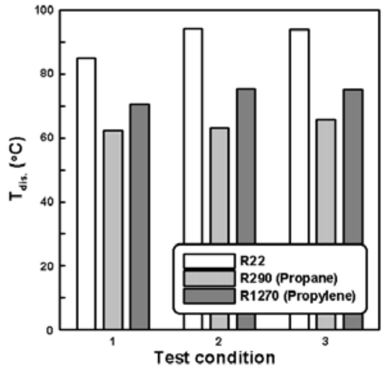

Fig. 7. Discharge temperature of various refrigerants for each test conditions.

to R22 under three conditions. As seen in these figures, the capacity of R290 is 5.8-8.2% lower than that of R22 under test condition 1 and 2. But under test condition 3, the extreme winter condition, the capacity of R290 is 5.3% higher that that of R22. The capacity of R1270 is 2.1-11.1% higher than that of R22 for all three conditions. As the evaporator temperature decreases during winter, the capacity of a heat pump decreases as well. But the rate of decrease in the capacity for hydrocarbons is smaller with a decrease in evaporator temperature as seen in Fig. 6.

Thus, hydrocarbons are good especially for heat pumping applications. Test results also indicate that mixing of R290 and R1270 may produce a mixture whose capacity is similar to that of R22. Also, if one wants to use pure R290 for R22 application, the compressor displacement volume should be increased by 6-10% as compared to that of R22.

3.3 Compressor discharge tmperatures

In applying alternative refrigerants, the lifetime and reliability of the system as well as the stability of the refrigerant and lubricant should be considered. 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 was very small.

Fig. 7 and Table 1 show the compressor discharge temperatures of three fluids tested under three conditions. R290 and R1270 showed a decrease in compressor discharge temperature of 14.1-30.9℃ under three conditions. Especially, R290 showed the lowest discharge temperatures among three fluids tested. For heat pumps used in extremely cold weather, compressors are often destroyed due to the excessive discharge temperature. As seen in Fig. 7, the discharge temperatures of hydrocarbons under extreme conditions (test condition 3) are even lower than that of R22 under normal air-conditioning conditions (test condition 1). This is a good characteristic and will be very beneficial to the manufacturers since it will lead to an improvement in system reliability and lifetime. 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 hydrocarbons have smaller density than most halocarbons and hence the amount of charge decreases significantly with hydrocarbons [12]. As Table 1 illustrates, R290 and R1270 showed a decrease in charge of up to 58% as compared to R22. This will help alleviate further the direct emission of refrigerant which is responsible for the greenhouse warming.

4. Conclusions

In this study, thermal performance of R22 and 2 pure hydrocarbons of R290 and R1270 was measured in a breadboard-type water cooled heat pump tester under three typical air-conditioning and heat pumping conditions. Various performance characteristics of these fluids were measured and following conclusions were drawn.

(1) COPs of R290 and R1270 are up to 12% higher than those of R22 under three conditions. Especially, R290 showed 6-12% increase in COP as compared to R22 under three conditions.

(2) Capacity of R290 under normal air-conditioning and heat pumping conditions is 6-8% lower than that of R22. But under extremely cold conditions, the capacity of R290 is 5% higher than that of R22. On the other hand, the capacity of R1270 is up to 11% higher than that of R22 under three conditions. The rate of decrease in capacity for R290 and R1270 with decreasing evaporator temperature is lower than that of R22. This characteristic is good especially for winter heat pumping applications.

(3) Compressor discharge temperatures of R290 and R1270 are lower than that of R22 by 14.1-30.9℃ under three conditions. 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 58% as compared to R22 due to their lower density.

(5) Finally, more elaborate tests are to be performed in actual residential air-conditioners and heat pumps 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.

Acknowledgment

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