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Performance of R170/R1270 mixture under air-conditioning and heat pumping conditions †

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Abstract

In this study, the performance of R170/R1270 mixture was measured on a heat pump bench tester to substitute for R22. The tester was equipped with a hermetic compressor providing a nominal capacity of 3.5 kW. All tests were conducted under typical summer and winter conditions of 7/45ºC and -7/41ºC in the evaporator and condenser, respectively. During the tests, the composition of R170 varied from 0 to 10% with an interval of 2% for the R170/R1270 mixture. Test results showed that the capacities of the R1270 and R170/R1270 mixtures were 3.2-10.0% and 4.2-20.4% higher than those of R22, respectively. The coefficient of performance (COP) of R1270 was 2.7-3.6% higher than that of R22. On the other hand, the COP of R170/R1270 mixture decreased at a constant rate as R170 was added to R1270. The COP of R170/R1270 mixture was similar to that of R22 at 2% R170 and then became lower than those of R22 as R170 increased. For the mixture, the compressor discharge temperature was 8-20ºC lower than that of R22. The amount of charge was reduced up to 55% compared with R22. Overall, pure propylene and R170/R1270 mixtures are good long-term candidates from the viewpoint of energy efficiency and greenhouse warming to replace R22 in residential air-conditioners and heat pumps.

Keywords: Alternative refrigerant; COP; Capacity; R1270 (Propylene); R170 (Ethane); Refrigerant mixture; Heat pump

1. Introduction

In the past decades, chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) have been used extensively in various refrigeration and air-conditioning applications due to their excellent thermodynamic properties and chemical stability. However, due to their effects on stratospheric ozone layer depletion, CFCs and HCFCs are now controlled substances under the Montreal protocol [1]. Due to international regulation, non-ozone depleting hydrofluorocarbons (HFCs) have been used in most of the refrigeration and air-conditioning applications for the past two decades.

Greenhouse warming has currently become one of the most important global issues. The Kyoto protocol, which was proposed to resolve this issue, classified HFCs as one of the greenhouse warming gases [2]. Consequently, refrigerants with low greenhouse warming potential (GWP) and zero ozone depletion potential (ODP) are required to be used in refrigeration and air-conditioning applications. At the same time, the performance of refrigeration and air-conditioning equipment has to be improved to reduce indirect greenhouse warming caused by the use of electricity generated mainly by the combustion of fossil fuels. In fact, for most refrigeration and air-conditioning equipment, the indirect warming effect is more than 80% of the total warming.

R22 has been used predominantly in residential air-conditioners and heat pumps and has the largest sales volume among all refrigerants. R22, however, is an HCFC that contains the ozone-depleting chlorine atom and should be phased out eventually. As part of the environmental protection effort, R22 can no longer be used in newly manufactured air-conditioners in the US beginning 2010. Most developed countries are also expending research and development efforts to replace ozone-depleting R22 with environment-friendly refrigerants.

One of the best ways to solve energy and environmental problems in the refrigeration industry is the use of natural refrigerants, such as hydrocarbons. Hydrocarbons have zero ODP and very low GWP. In general, hydrocarbons offer 10-15% increase in energy efficiency in various refrigeration and air-conditioning applications. Despite these advantages, for the past few decades, hydrocarbon refrigerants have been prohibited in normal refrigeration and air-conditioning applications due to safety concerns. These days, however, this prohi-

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bition has somewhat relaxed because of an environmental mandate. Therefore, some of the flammable refrigerants have been applied to certain applications.

Purkayastha and Bansal [3] measured the performance of propane and R22 in a heat pump with 15 kW capacity and found that the coefficient of performance (COP) of propane is 18% higher than that of R22 with a decrease in heating capacity of 15%. The refrigerant mass flow rate of R290 is half that of R22, and the compressor discharge temperature of R290 is much lower than that of R22. Granryd [4] also performed a thermodynamic cycle and heat transfer analysis for propane and R22 and arrived at a similar conclusion. Chang et al. [5] measured the performance of four pure hydrocarbons of isobutane, butane, propane, and propylene, as well as two binary mixtures of propane/isobutane and propane/butane, and discovered that both propane and propylene have better performance than R22. Fernaldo et al. [6] used propane in a heat pump with 5 kW capacity and determined the optimum amount of charge for the use of mini-channel aluminum heat exchangers. Recently, Hwang et al. [7] carried out a study to determine energy consumption for HFC mixtures of R404A and R410A, as well as propane in walk-in refrigeration systems, and observed that the COP of propane is up to 10% higher than that of R404A and R410A.

Reflecting the recent interest in the use of hydrocarbons, ASHRAE listed many refrigerant mixtures such as R429A, R430A, R431A, R432A, and R433A, which contain hydrocarbons and flammable refrigerants for better energy efficiency and environmental protection [8]. Due to these continuing efforts, certain hydrocarbons are being applied to refrigeration and air-conditioning applications [9-11]. At this time, propane is being used in small scale air-conditioners, heat pumps, and vending machines because of its good material, lubricant compatibility, and low cost [11].

According to the literature survey, although the energy efficiency of propane is higher than that of R22, its capacity is up to 15% lower than that of R22. Therefore, propane is not a good choice particularly for heat pumping during the winter season. The survey shows that although heat pumps are widely used for energy saving during winter time, little information is available for the performance of hydrocarbons under heat pumping conditions. During winter time, building heating load increases due to the increase in the temperature difference between the indoors and outdoors; however, the heating capacity of heat pumps decreases as the outdoor temperature decreases, which is a severe drawback of heat pumps. To overcome this difficulty, refrigerants should be provided with more capacity during winter time. From this viewpoint, propylene (R1270) and the mixture of ethane (R170) and propylene are good choices for vapor compression heat pumps.

Zeotropic refrigerant mixtures can increase the energy efficiency of certain refrigeration equipment under optimized conditions. For zeotropic refrigerant mixtures, a good temperature matching between the refrigerant and secondary heat transfer fluid (HTF) can be achieved in heat exchangers due to their temperature gliding effect during phase change [12, 13]. To alleviate the effects of greenhouse warming in the future, applying zeotropic refrigerant mixtures to air-conditioning and refrigeration equipment needs to be considered. In the literature, however, few studies are found dealing with zeotropic refrigerant mixtures composed of hydrocarbons applied to heat pumps and air-conditioners.

In this study, the cooling and heating performances of a hydrocarbon mixture composed of ethane (R170) and propylene (R1270) were measured under typical summer and winter conditions in a heat pump bench tester, and the results were compared with those of R22.

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 schematics of the experimental heat pump whose nominal capacity is roughly one 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 a series. Each pipe stick was 740 mm long with inner and outer diameters of 19.0 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 were 5.92 m and 0.35 m^2, respectively. Both evaporator and condenser were designed to be counter-current. 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. Precision water/ethylene glycol chiller and a heating bath of 0.1°C accuracy were used to control the temperatures of the water/ethylene glycol entering the condenser and evaporator, respectively.

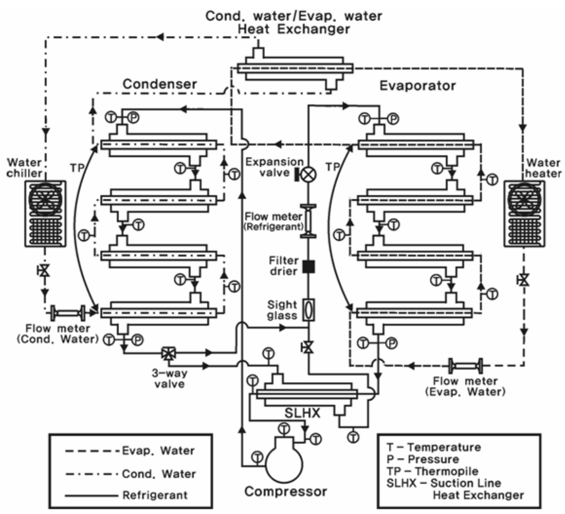

Fig. 1. Schematics of the heat pump bench tester.

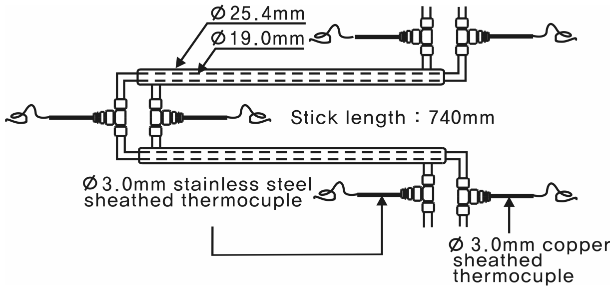

Fig. 2. Details of the evapoator connection.

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. Although a suction line heat exchanger (SLHX) had been installed initially to examine the effect of SLHX, it was not used in this study.

A liquid eye was installed at the exit of the condenser to determine 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. The compressor dome and discharge pipe temperatures were also measured for comparison. All thermocouples were calibrated before their use against a precise RTD thermometer of 0.01°C accuracy. Pressures were measured at the inlets and outlets of the evaporator and condenser using calibrated pressure transducers. Power input to the compressor was measured using a digital power meter of 0.5% accuracy. Finally, mass flow rates of the secondary heat transfer fluid were measured by employing 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 using a thermopile composed of six copper-constantan thermocouples whose performances were calibrated by a set of RTDs of 0.01ºC accuracy. Table 1 lists the uncertainties of some experimental parameters in this study. All data were taken under steady state using a computerized data logging system and were stored for later analysis.

2.3 Test condition

To compare properly the performances of various refrigerants, a fair test condition should be employed. Thus, all tests were conducted with the external HTF (water in this study) temperatures fixed. In this study, tests were performed under two sets of different evaporator/condenser saturation temperatures for R22: 7ºC/45ºC, -7ºC/41ºC. The first condition reflects normal air-conditioning condition during summer, while the second condition reflects normal heat pumping conditions during winter. For a given condition, tests were first carried out for R22 with the adjusted external HTF temperatures to provide the required saturation temperatures in the evaporator and condenser. Table 2 shows the HTF temperatures and mass flow rate under two conditions. Subsequent tests were then performed under the same external conditions for R1270 and R170/R1270 mixture at five compositions. For a given external condition, actual saturation temperatures of the various refrigerants in the evaporator and condenser varied little due to the difference in heat transfer characteristics of these fluids.

Table 1. Uncertainties of experimental parameters.

Parameters	Uncertainty
Temp.(RTD)	±0.01º C
Temp.(Thermocouple)	±0.1º C
Pressure	±3.4kPa
Mass flow rate	±0.2%
Work(Wattmeter)	±0.5%

Table 2. Some variables to set test conditions.

Test condition	$T_{e,w}$ (º C)	$T_{c,w}$ (º C)	$\dot{m}_{e,w}$ (g s^{-1})	$\dot{m}_{c,w}$ (g s^{-1})
A (Summer cooling)	27	29	94	115
B (Winter heating)	9	31	94	115

2.4 Test procedures

The test procedure for a given condition was as follows:

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

(2) Temperatures in the chiller and heating bath were set, and the secondary HTF was pumped into the evaporator and condenser. 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. For the mixture, 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 the amount of charge was simultaneously adjusted to maintain the constant superheat and subcooling, usually 5ºC each, at the exits of the evaporator and condenser.

Table 3. Refrigerants tested in this study.

Ref. No.	Refrigerant (Mass fraction)	GTD (º C)
1	R22	0
2	R1270	0
3	2%R170/98%R1270	2.1
4	4%R170/96%R1270	4.1
5	6%R170/94%R1270	5.8
6	8%R170/92%R1270	7.3
7	10%R170/90%R1270	8.7

(4) When the system reached steady state for more than 1 h, data were taken every 30 s for more than 30 min.

2.5 Refrigerants and lubricants

In this study, the thermal performance of R22, R1270, and R170/R1270 mixtures at five compositions was measured. Table 3 lists all the refrigerants tested and their gliding temperature difference (GTD). For a given mixture, GTD is the temperature difference between the beginning and ending temperatures during evaporation. GTD varies with the mixtures and their compositions.

ODP and GWP of R22, which is the reference fluid, were 0.05 and 1700, respectively. ODP and GWP of R170 and R1270 were zero and less than three, respectively. Therefore, R170 and R1270 and their mixtures are environment-friendly and can be good long-term alternatives in residential air-conditioners and heat pumps.

Lubricant is also important in a refrigeration system because it is circulated with the refrigerant inside the refrigeration circuit. As a drop-in replacement is the focus of the present study, conventional mineral oil used with R22 was used for all refrigerants. In fact, one of the advantages of hydrocarbons and their mixtures is that they are completely compatible with mineral oil [14, 15].

3. Thermodynamic cycle simulation

Before the experiments, thermodynamic cycle analysis was carried out for all fluids tested using the Cycle-D program [16] developed by the US National Institute of Standards and Technology to determine the overall trend of COP and discharge temperature of the mixture. In the analysis, the refrigeration capacity and compressor isentropic efficiency were fixed at 3.5 kW and 0.7, respectively. Figs. 3 and 4 show the calculated COP and discharge temperatures of all refrigerants tested. As seen in Fig. 3, the theoretical COP of R1270 is very similar to that of R22. The COP of R170/R1270 mixture, however, decreased as R170 was added to R1270. The compressor discharge temperature of the mixture increased as the amount of R170 increased. All discharge temperatures of the R1270 and R170/R1270 mixtures were 6-19ºC lower than those of R22 under both conditions.

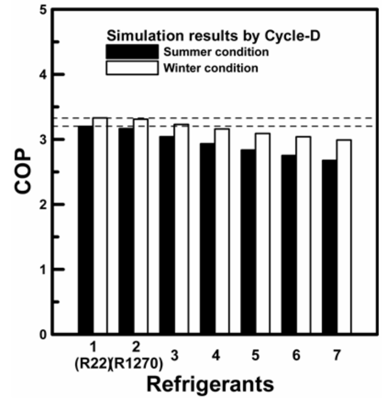

Fig. 3. Calculated COPs of R170/R1270 mixture.

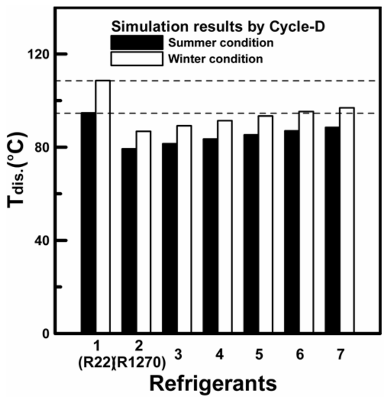

Fig. 4. Calculated discharge temperatures of R170/R1270 mixture.

4. Results and discussion

In this study, the thermodynamic performance of R22, R1270, and R170/R1270 mixtures at five compositions was measured in a heat pump tester equipped with a commercial hermetic rotary compressor under typical air-conditioning and heat pumping conditions. The heat balances between the refrigerant and water sides on the evaporator and condenser were within 3%. For each refrigerant, tests were performed at least twice, and the test results usually agreed within 1% repeatability. Table 4 presents the various measured system parameters such as COP, capacity, discharge temperature, and amount of charge for all refrigerants tested under both cooling and heating conditions.

4.1 Capacity

Capacity is very important in refrigeration and heat pumping. 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 at least a similar (or higher) capacity to that of the reference fluid. The increase in capacity is especially important during winter season when the building heating load increases

Table 4. Summary of test results for the various refrigerants.

Ref. No	Refrigerants	Condition A (Summer cooling)						Condition B (Winter heating)					
		Q_e(W)	diff. (%)	*COP*	diff. (%)	$T_{dis.}$ (° C)	Charge (g)	Q_c(W)	diff. (%)	*COP*	diff. (%)	$T_{dis.}$ (° C)	Charge (g)
1	R22	3830		3.48		86.7	1300	3325		3.51		97.1	1350
2	R1270	3952	3.2	3.68	3.6	70.6	600	3656	10.0	3.60	2.7	77.5	600
3	2%R170/98%R1270	3991	4.2	3.48	0.1	72.7	600	3690	11.0	3.48	-0.9	78.7	600
4	4%R170/96%R1270	4059	6.0	3.35	-3.6	74.3	600	3788	13.9	3.42	-2.6	80.0	600
5	6%R170/94%R1270	4106	7.2	3.25	-6.7	76.3	600	3871	16.4	3.34	-4.8	81.7	600
6	8%R170/92%R1270	4145	8.2	3.15	-9.5	76.7	600	3921	17.9	3.24	-7.7	83.1	600
7	10%R170/90%R1270	4200	9.7	3.05	-12.4	78.4	600	4002	20.4	3.18	-9.5	85.1	600

as the outdoor temperature decreases.

Fig. 5 shows the refrigeration and heat pumping capacities, Q_e, and Q_c in Table 4 of R22, R1270, and R170/R1270 mixtures under both conditions. As seen in Fig. 5, the capacities of R1270 were 3.2-10.0% higher than those of R22 under both conditions. Therefore, R1270 is a good refrigerant that provides more capacities than those of R22 during both summer and winter seasons.

As for the mixture of R170/R1270, both refrigeration and heating pumping capacities increased as the amount of R170 increased because of the large capacity of R170. The capacities of the mixture were 4.2-20.4% higher than those of R22 in the composition range of 2-10% of R170. This result indicates that R170/R1270 mixture is a good candidate from the viewpoint of capacity, especially for heat pumping application.

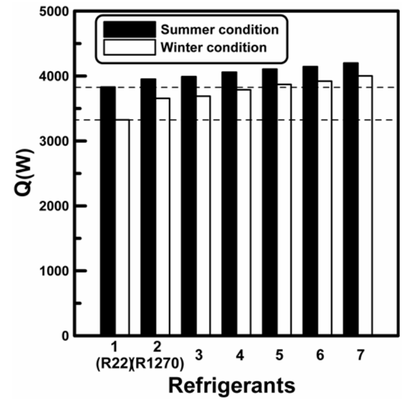

Fig. 5. Measured capacities of R170/R1270 mixture.

4.2 Coefficient of performance

To alleviate greenhouse warming, the energy efficiency of energy conversion devices should be improved. In air-conditioning and heat pumping, the COP is a measure of energy efficiency for a given device charged with a specific refrigerant. Hence, it is important to examine COPs of tested refrigerants against the reference fluid, R22.

Fig. 6 shows the COP of R22, R1270, and R170/R1270 mixtures at various compositions under both conditions. As seen in Fig. 6, the COP of R1270 is 2.7-3.6% higher than that of R22 under both conditions. This result agrees well with the previous results from other studies [5]. As for the mixture of R170/R1270, the COP decreased at a constant rate as R170 was added to R1270 with an interval of 2%. This trend agreed well with the thermodynamic cycle simulation results. In fact, the COP is a strong function of the critical temperature of the refrigerant. Ethane (R170), a very high vapor pressure refrigerant, has very low critical temperature, and thus its COP is inherently low. Hence, adding R170 to R1270 produced an unfavorable result in terms of energy efficiency. In this study, a compressor designed for R22 was used. In the future, for the mixture, the compressor needs to be optimized for better performance. COPs of the mixture will then increase further.

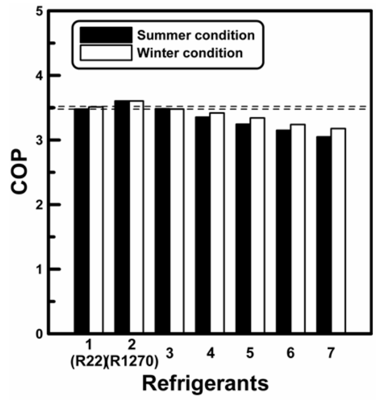

Fig. 6. Measured COPs of R170/R1270 mixture.

4.3 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. 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 3 mm insulation around the sensors, thus making the temperature deviation due to the

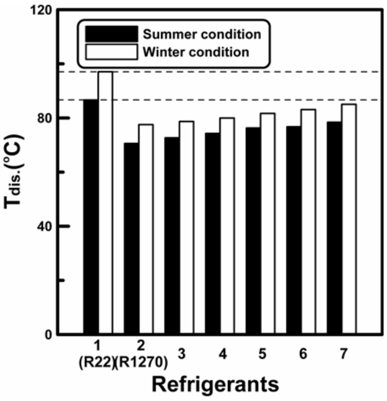

Fig. 7. Measured discharge temperatures of R170/R1270 mixture.

change in surrounding very small.

As seen in Fig. 7, the compressor discharge temperatures of R1270 and the R170/R1270 mixture were 8.3-19.6ºC lower than those of R22 under both conditions. This is a good characteristic and is very beneficial to the manufacturers as it will lead to an improvement in system reliability and lifetime. Based on this observation, the proposed mixture being appropriate from the viewpoint of system reliability and fluid stability can be safely concluded.

As for the mixture of R170/R1270, the compressor discharge temperature increased as the amount of R170 increased. This trend agreed well with the cycle simulation results.

4.4 Refrigerant charge

In this study, the optimum amount of charge was determined when the subcooling at the exit of the condenser was 5°C. Most of the hydrocarbons have smaller liquid densities than those of most of the halocarbons; thus, the amount of charge decreases significantly with hydrocarbons [17]. As seen in Table 4, the amount of charge for all hydrocarbon refrigerants tested decreased up to 55% compared with R22. This will help further alleviate the direct emission of refrigerant, which is responsible for the greenhouse warming.

5. Conclusions

In this study, the thermodynamic performance of two pure refrigerants, R22 and R1270, and the R170/R1270 mixture at five compositions was measured in a breadboard-type heat pump tester equipped with a commercial hermetic rotary compressor under air-conditioning and heat pumping conditions. For the mixture, the composition varied from 2-10% of R170 with an interval of 2%. Various performance parameters were measured. The following conclusions were drawn from the results:

(1) Both cooling and heating capacities of R1270 and R170/R1270 mixture in the composition range of 2-10% of R170 were 3.2-10.0% and 4.2-20.4% higher than those of R22, respectively. Hence, pure propylene and R170/R1270 mixture are good candidates, especially for heat pumping applications.

(2) The COP of R1270 was 2.7-3.6% higher than that of R22. The COP of R170/R1270 mixture decreased at a constant rate as R170 was added to R1270. The COP of R170/R1270 mixture was similar to that of R22 at 2% R170, which then became lower than those of R22 as R170 increased under both conditions.

(3) R170/R1270 mixture had 8.3-19.6ºC lower compressor discharge temperatures compared with that of R22. This is a good characteristic for long-term system reliability.

(4) The amount of charge of R170/R1270 mixture decreased up to 55% compared with that of R22 due to its low liquid density.

(5) Overall, pure propylene and R170/R1270 mixture are good long-term candidates to replace R22 in residential air-conditioners and heat pumps from the viewpoint of energy efficiency and greenhouse warming.

Acknowledgment

This study was supported by Inha University.

Nomenclature

COP : Coefficient of performance
GTD : Gliding temperature difference
GWP : Global warming potential
HTF : Heat transfer fluid
$\dot{m}$: Mass flow rate [kg/s]
ODP : Ozone depletion potential
Q : Capacity [W]
T : Temperature [ºC]

Subscripts

c : Condenser
dis : Discharge
e : Evaporator
w : Water

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