AN EXPERIMENTAL STUDY ON HEAT PUMP CYCLE USING ZEOTROPIC BINARY REFRIGERANT OF HFO-1234ze(E) AND HFC-32

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Abstract: In the present study the performance tests on heat pump cycle have been carried out for a near-azeotropic refrigerant R410A,a pure refrigerant HFO-1234ze (E) and zeotropic binary refrigerant mixtures of HFO-1234ze (E) and HFC-32 at heating mode, using a compressor developed for R410A. It is confirmed that the COP value and the heating capacity of pure HFO-1234ze(E) are the lowest among the refrigerants tested in the present study due to its low vapor density and latent heat. Also, due to the same reason, the pressure drops in condenser and evaporator of HFO-1234ze(E) are the highest among the tested refrigerants. It is also found that adding HFC-32 into HFO-1234ze(E) dramatically improves not only the COP value but also heating capacities; the COP values of 20mass%HFO-1234ze/80mass%HFC-32 mixture are almost the same as those of R410A at the same heating load. As a result, the present cycle performance tests prove that mixtures of HFO-1234ze(E) and HFC-32 are strong candidates for replacing R410A in domestic heat pump systems.

Key Words: Experiments, Cycle performance, Binary refrigerant mixture, HFO-1234ze(E), HFC-32, R410A

1 INTRODUCTION

Since Midgeley synthesized chlorofluorocarbons (CFCs) successfully in 1928, CFCs and hydro-chlorofluorocarbons (HCFCs) had been widely used as working fluids (refrigerants) in air-conditioning and refrigeration industry for many years. However, in 1974, It was pointed out that CFCs could cause the depletion of the stratospheric ozone layer (Monica and Roland 1974), and then a trend to control the emission of CFCs and HCFCs internationally had reached a peak. As result, in 1987, the Montréal protocol for the phase–out of ozone depleting substances such as CFCs and HCFCs was concluded. In that situation, the development of new alternative refrigerants without the ozone depletion potential (ODP) have been proceeded, and hydro-fluorocarbons (HFCs) such as R134a, R410A and R407C were developed as alternatives of CFCs and HCFCs in 1990's. However, at the 1997 Kyoto Conference (COP3), it was determined that the product and use of HFCs should be regulated due to their high global warming potential (GWP). This requirement has forced us to develop environmentally acceptable alternative refrigerants with zero ODP and relatively low GWP. In the last decade, natural refrigerants such as i-C_4H_{10}, water, CO_2 and NH_3, have been attracting a great deal of attention all over the world because of their zero ODP and extremely low GWP. Some heat pump and refrigeration systems such as domestic refrigerators using i-C_4H_{10}, heat pump water heaters using CO_2 and refrigerated storehouses using NH_3/CO_2 have been put in practical use. However, appropriate alternatives for domestic air-conditioning systems still can not be found up to now.

In the above mentioned circumstance surrounding the air-conditioning and refrigeration industry, we have noticed from early stage on HFO-1234ze(E) (*Trans-1,3,3,3-Tetrafluoro-propene*) that was developed as a cover gas for casting process of Magnesium alloy because its zero-ODP and extremely low GWP. In the present study, we have carried out the cycle performance test in order to investigate the possibility to introduce HFO-1234ze(E) and/or its mixtures with R32 as low-GWP alternatives for domestic heat pump systems. The tested refrigerants were a near-azeotropic refrigerant R410A, a pure refrigerant HFO-1234ze(E) and binary zeotropic refrigerant mixtures of HFO-1234ze(E) and HFC-32.

2 EXPERIMENTAL APPARATUS AND METHOD

Figure 1 shows the schematic view of an experimental apparatus, which was used for the present experiments on the cycle performance of a domestic heat pump system. The experimental apparatus consists of a refrigerant loop, a sink water loop and a heat source water loop. The refrigerant loop is composed an inverter controlled hermetic type compressor, an oil separator, a double-tube type condenser, a liquid receiver, an electric expansion valve and a double-tube type evaporator. Using constant-temperature baths, the heat sink water and heat source water are supplied to the condenser and the evaporator, respectively. Four mixing chambers are installed between components in the refrigerant loop to measure the refrigerant pressure and temperature. The other four mixing chambers are installed in heat sink and heat source water loops to measure water temperatures at the inlet and outlet of the condenser and the evaporator. The specifications of the condenser and the evaporator are listed in Table 1. Both of the condenser and the evaporator are double-tube type and coiled, where the refrigerant flows inside the inner tube, while heat sink/heat source water flows in the annulus surrounding the inner tube.

Table 2 shows the experimental conditions at heating mode. The refrigerants tested in the present experiments are R410A, pure HFO-1234ze(E) and mixtures of HFO-1234ze(E)/HFC-

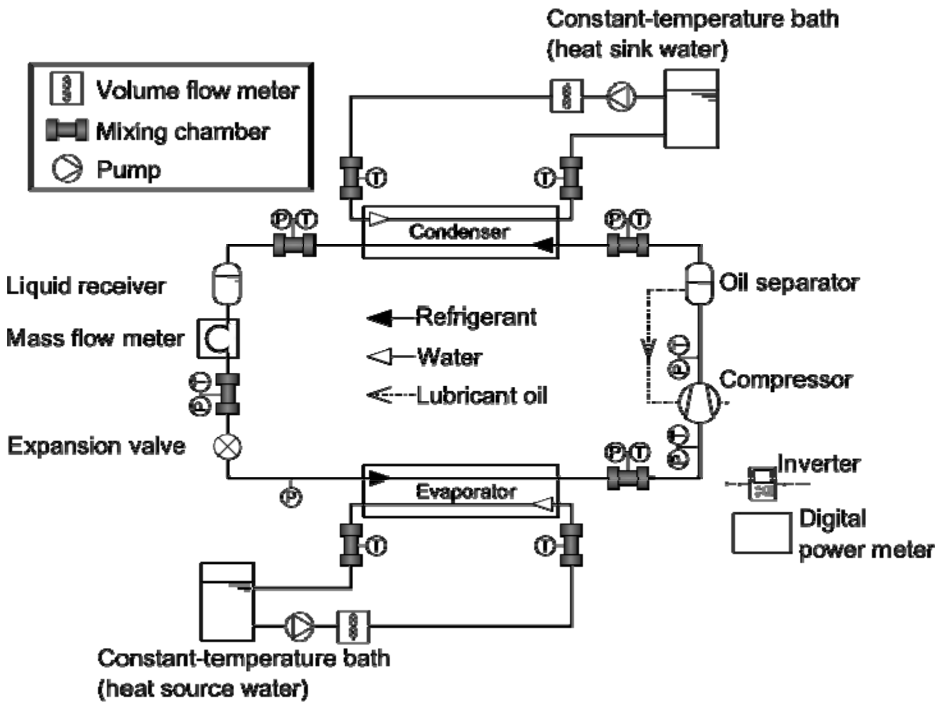


Figure 1: Schematic view of experimental apparatus

Table 1: Specifications of double-tube type heat exchangers

		OD (mm)	ID (mm)	Length (mm)	Type of tube
Condenser	Outer tube	12.7	10.7	5000	smooth tube
	Inner tube	9.52	7.53	5000	micro-fin tube
Evaporator	Outer tube	12.7	10.7	4500	smooth tube
	Inner tube	9.52	7.53	4500	smooth tube

Table 2: Experimental conditions at heating mode

	Refrigerant				
	R410A	HFO-1234ze(E)/HFC-32			
		20/80 mass%	50/50 mass%	80/20 mass%	100/0 mass%
GWP	675	542	342	142	9
Condenser: Heat sink water temperature [^{o}C]	20 → 45				
Evaporator: Heat source water temperature [^{o}C]	15 → 9				
Heating capacity [kW]	1.8~2.8	1.6~2.8	1.8~2.8	1.4~2.8	1.0~1.8
Degree of superheat [K]	3				

-32. The mass fractions of HFO-1234ze(E) in the mixtures are 20, 50 and 80 %. For all experiments the degree of superheat at evaporator outlet is fixed at 3 K. Temperatures of the heat sink water and heat source water are also fixed for all experiments as follows: heat sink water temperatures at inlet and outlet of condenser are kept at 20 ^{o}C and 45 ^{o}C, respectively, and heat source water temperatures at inlet and outlet of evaporator are kept at 15 ^{o}C and 9 ^{o}C, respectively, and the range of the heating capacity changes from 1 kW to 2.8 kW depending on kinds of refrigerants. The refrigerant charge amount is also changed as one of experimental parameters in the discharge-suction pressure range less than 6.

In the experiments, the heat pump system is operated to achieve a specified experimental condition by adjusting the compressor frequency and the opening of the electric expansion valve. As the system reaches a steady state as satisfying an experimental condition, the following physical quantities are measured directly using a data acquisition system:
(1) electric power inputs to the inverter and the compressor,
(2) refrigerant temperature and pressure in every mixing chambers and refrigerant temperature at the discharge port of the compressor,
(3) mass flow rate of refrigerant and volumetric flow rates of heat sink water and heat source water,
(4) temperatures of heat sink water and heat source water at the inlet and outlet of condenser and evaporator.

Then, heat transfer rates of condenser and evaporator are calculated from the water-side energy balance equations using the measured volumetric flow rate and temperatures of heat sink water and heat source water, and then the coefficient of performance (COP) at heating mode is obtained from the electric power input to the inverter and the heat transfer rate of condenser. Thermodynamic properties of R410A are calculated using the program package REFPROP Ver.8 (Lemmon et al. 2007), while those of pure HFO-1234ze(E) and mixtures of HFO-1234ze(E)/HFC-32 are calculated using the program package REFPROP Ver.8 combined with the HFO-1234ze(E) program package (Akasaka 2010). Figure 2 shows the

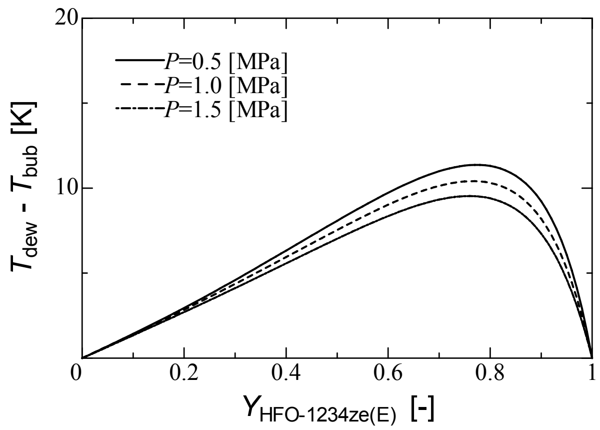


Figure 2: Temperature difference between dew and bubble points of HFO-1234ze(E)/HFC-32 mixture

calculated temperature difference between dew and bubble points of HFO-1234ze(E)/HFC-32 mixture for reference.

3 EXPERIMENTAL RESULTS AND DISCUSSION

Figure 3 shows the experimental results of R410A at heating mode, where the ordinates in Figs. 3(a) and (b) express COP and degree of sub-cooling at condenser outlet, respectively, while the abscissa in all figures represents the heat transfer rate of condenser (heating load). Symbols are employed to distinguish the refrigerant charge amount. In each case of refrigerant charge amount, the value of COP increases with increase of heating load, and it reaches a maximum and then decreases. In case of a constant heating load, the value of COP has a maximum at a certain refrigerant charge amount. The degree of sub-cooling increases with increase of refrigerant charge amount and it almost maintains a constant value when the heating load increases at a constant refrigerant discharge amount.

Figure 4 shows the experimental results of 100mass% HFO-1234ze(E) at heating mode, where the ordinates and abscissa in Figs.4(a) and (b) are the same as in Fig. 3., respectively, and symbols are also employed to distinguish the refrigerant charge amount. In some cases of refrigerant charge amount, the value of COP increases with increase of heating load and then decreases, while in other cases of refrigerant charge amount it decreases with increase of heating load. On the other hand, the degree of sub-cooling increases with increase of heat load in all cases of refrigerant charge amount. It is noted that the COP value has a maximum at a certain refrigerant charge amount. It was also found through the present drop-in experiments that the heating capacity of HFO-1234ze(E) was considerably lower than that of R410A

Figure 5 shows the experimental results of the mixture of 50mass% HFO-1234ze(E)/50 mass% HFC-32 at heating mode, where the ordinates and the abscissa in Figs. 5(a) and (b) are also the same as in Figs. 3(a) and (b), respectively. The relation between COP and heating load of this mixture is almost the same as in case of 100mass% HFO-1234ze(E), but the COP value of this mixture is higher than that of 100mass% HFO-1234ze(E) especially as the heating load increases. This mean that the addition of HFC-32 into HFO-1234-ze(E) improves the cycle performance effectively. In each case of refrigerant charge amount, the degree of sub-cooling increases moderately with the increase of heating load.

Figure 6 shows the experimental results to investigate the effects of the refrigerant charge amount on the cycle performance at the heating mode. The results of R410A, 100mass%

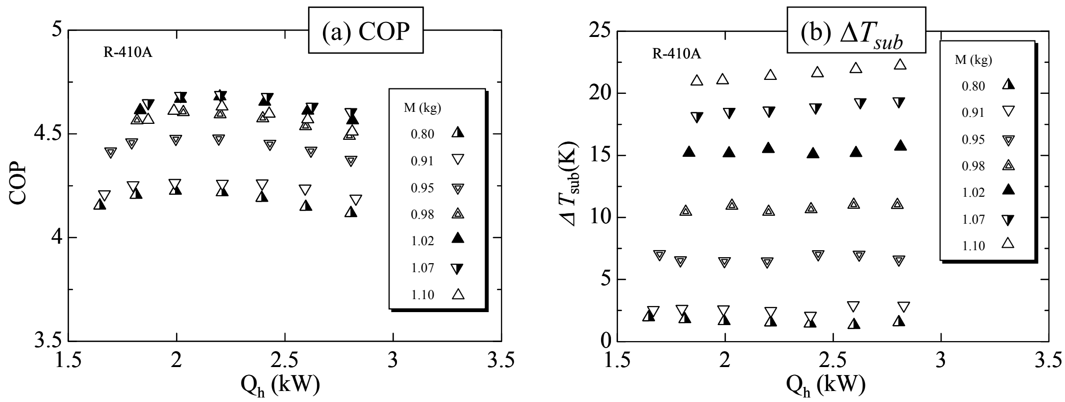


Figure 3: Experimental results of R410A

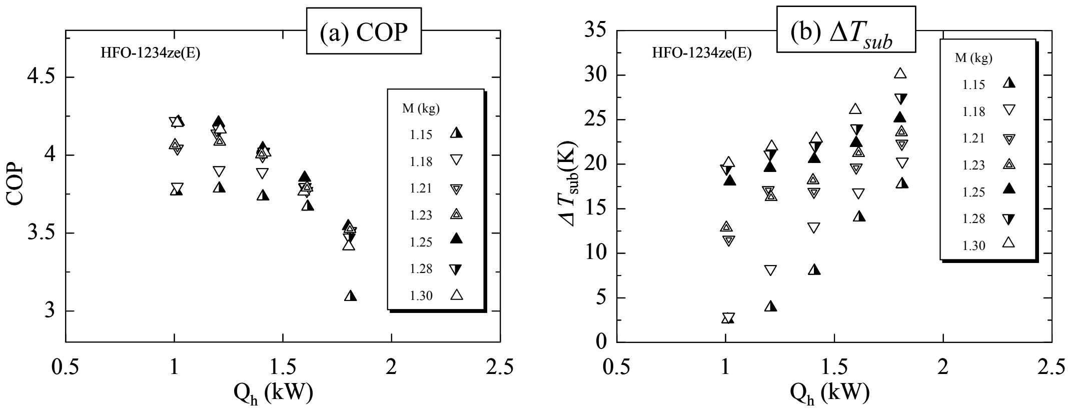


Figure 4: Experimental results of 100mass% HFO-1234ze(E)

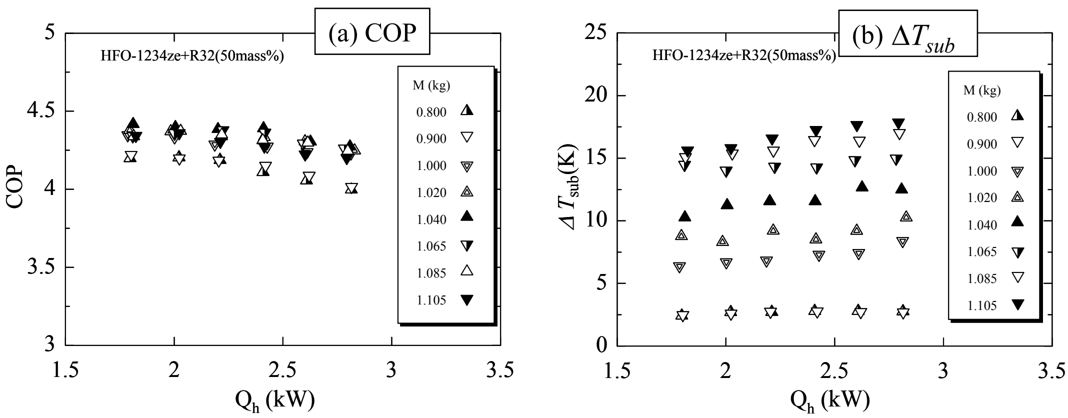
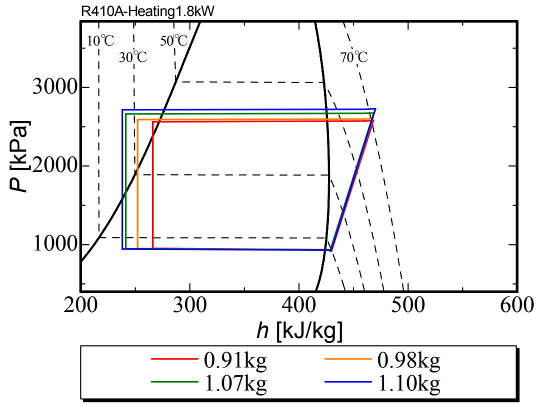


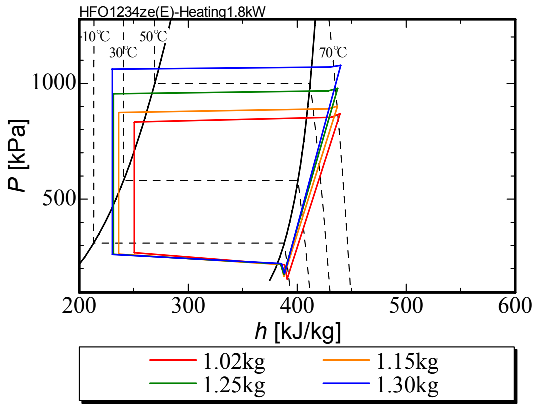
Figure 5: Experimental results of 50mass% HFO-1234ze(E)

HFO-1234ze(E), the mixture of 80mass% HFO1234ze(E) and the mixture of 20mass% HFO-1234ze(E) are plotted on the pressure–specific enthalpy diagrams shown in Figs. 6(a), (b), (c) and (d), respectively, where the difference of refrigerant charge amount is distinguished by colors of lines. In all cases of refrigerants, the change of specific enthalpy in the compressor, the condenser and the evaporator increases with the increase of refrigerant charge amount. It is also found that the pressure in condenser increases with increase of refrigerant charge amount, while the pressure in evaporator is little affected by the refrigerant charge amount. It is also found that the pressure drop of 100mass% HFO-1234ze(E) in evaporator is the largest among the test refrigerants.

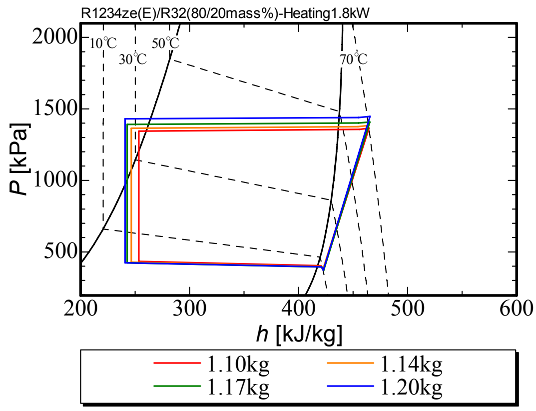
Figure 7 shows the relation between COP and degree of sub-cooling at condenser outlet in case of a heating load (1.8 kW), where symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively. In all cases of test refrigerants, the COP value once increases with increase in the degree of sub-cooling and reaches a maximum. Then, it decreases with increase in degree of sub-cooling.



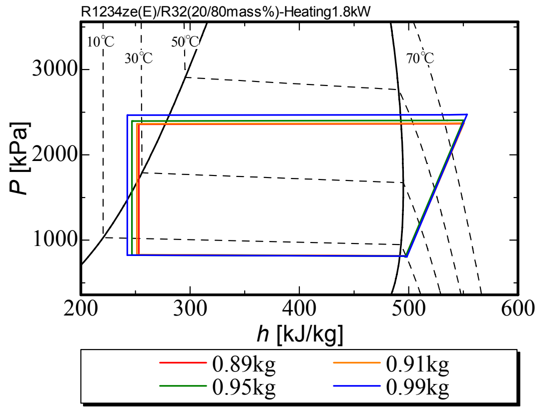
(a) R410A



(b) 100mass% HFO-1234ze(E)



(c) 80mass% HFO-1234ze(E)



(d) 20mass% HFO-1234ze(E)

Figure 6: Heat pump cycle plotted on P-h diagram

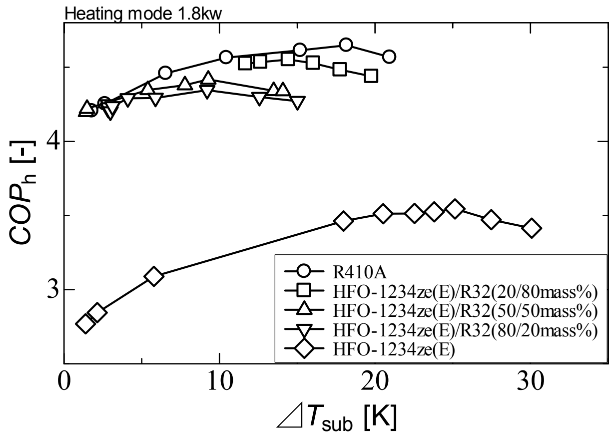
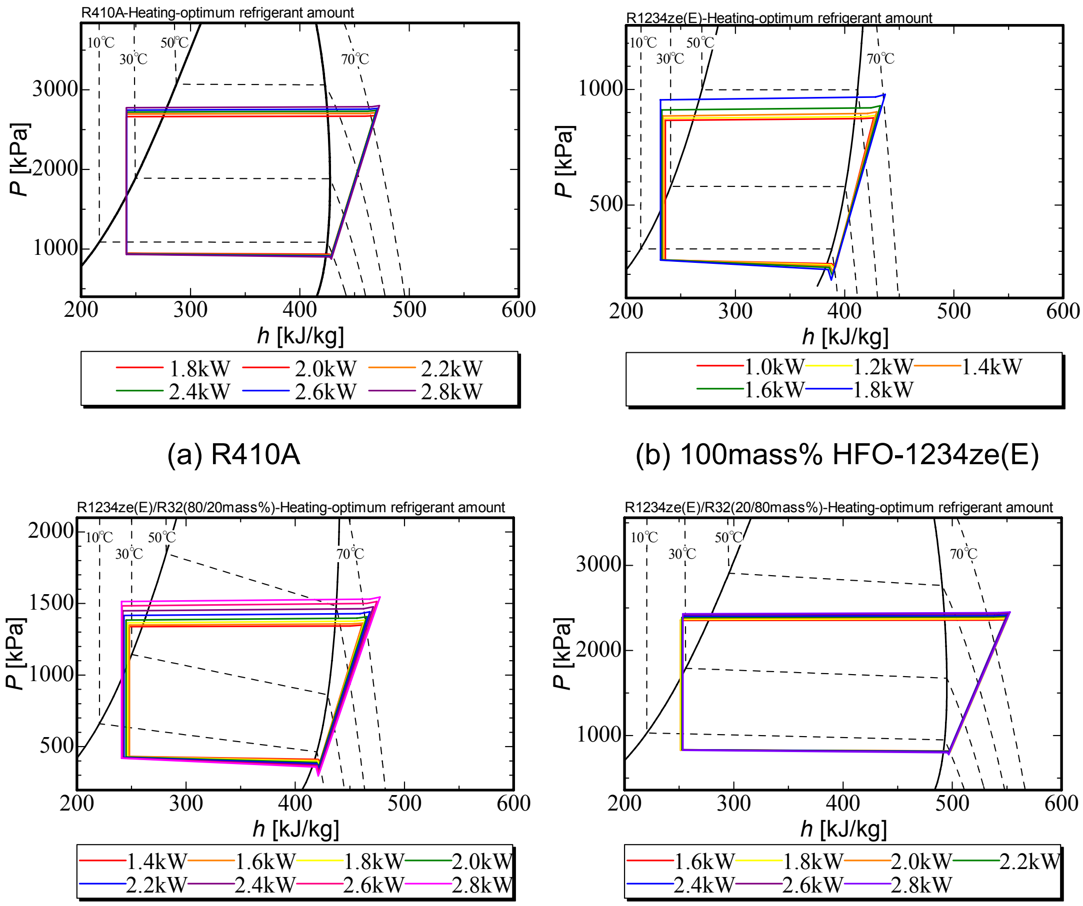


Figure 7: Relation between COP and degree of sub-cooling

The optimum value of refrigerant charge amount is determined as the refrigerant charge amount at a maximum COP value. Compared with the maximum COP values of R410A, those of 100, 80, 50 and 20 mass% HFO-1234ze(E) are about 24 %, 7%, 5% and 1% lower, respectively. This means that 100 mass% HFO-1234ze(E) is not suitable for an alternative of R410A, but mixtures of HFO-1234ze(E) and HFC-32 are available for alternatives of R410A.

Figure 8 shows the heat pump cycles of test refrigerants, plotted on the pressure–specific enthalpy diagrams, where Figs. 8(a), (b), (c) and (d) are results of R410A, 100mass% HFO-1234ze(E), the mixture of 80mass% HFO1234ze(E) and the mixture of 20%mass% HFO-1234ze(E) at optimum refrigerant charge amounts, respectively. In all cases of test refrigerants, the condenser pressure and enthalpy change in condenser and compressor increase with increase of heating load. This trend is remarkable in the cases of 100 and 80 mass% HFO-1234ze(E).

Figure 9 shows the relation between COP and heating load at optimum refrigerant charge amount, where symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively. The COP value of R410A is the highest among test refrigerants and is a little affected by heating load, while that of 100 mass% HFO-1234ze(E) is the lowest among test refrigerants and decreases considerably with increase of heating load. This also proves that 100 mass% HFO-1234ze(E) is not suitable for an alternative of R410A, but addition of HFC-32 into HFO-1234ze(E) is one of



(a) R410A (b) 100mass% HFO-1234ze(E)

(c) 80mass% HFO-1234ze(E) (d) 20mass% HFO-1234ze(E)

Figure 8: Heat pump cycle potted on the P-h diagram

promising methods to introduce a Low GWP refrigerant as an alternative of R410A; the COP value of the 20mass% HFO-1234ze(E) mixture is only 1% lower than that of R410A.

Figure 10 shows the relation between refrigerant pressure drop in heat exchangers and refrigerant flow rate, where Figs.10(a) and (b) are the results of condenser and evaporator, respectively. In each figure symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively, and among of them the symbols red-circled are the results at heating load 1.8 kW. As well known, the refrigerant pressure drop in heat exchangers increases with increase of refrigerant flow rate in all cases of test refrigerants; the increase of refrigerant flow rate corresponds to the increase of heating load. At the same heating load (1.8 kW), the pressure drop of R410A is the smallest among test refrigerants, while that of 100 mass% HFO-1234ze(E) is the largest . The addition of HFC-32 into HFO-1234ze(E) reduces the pressure drop in heat exchangers.

Figure 11 shows the relation between refrigerant temperature and heat transfer rate in heat exchangers at heating load 1.8 kW, where Figs.11(a) and (b) are the results of condenser and evaporator, respectively. In each figure symbols of circle, square, triangle, inversed triangle and diamond represent the results of R410A, 20mass% HFO1234ze(E), 50mass% HFO-1234ze(E), 80mass% HFO-1234ze(E) and 100mass% HFO-1234ze(E), respectively, and the chain line denotes water temperature. In evaporator, the temperature difference

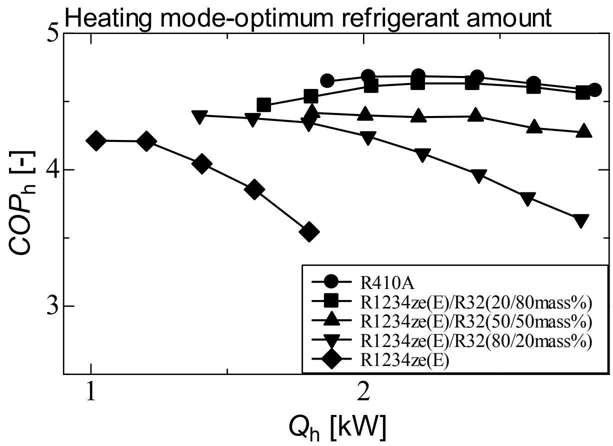
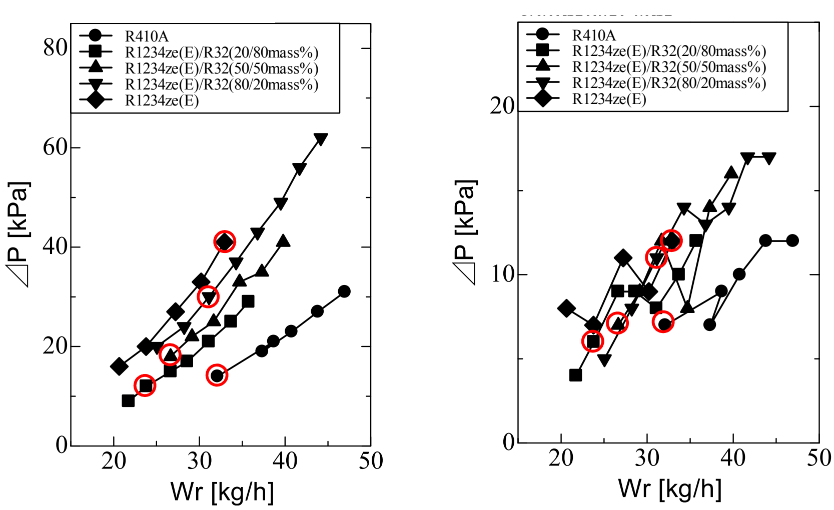


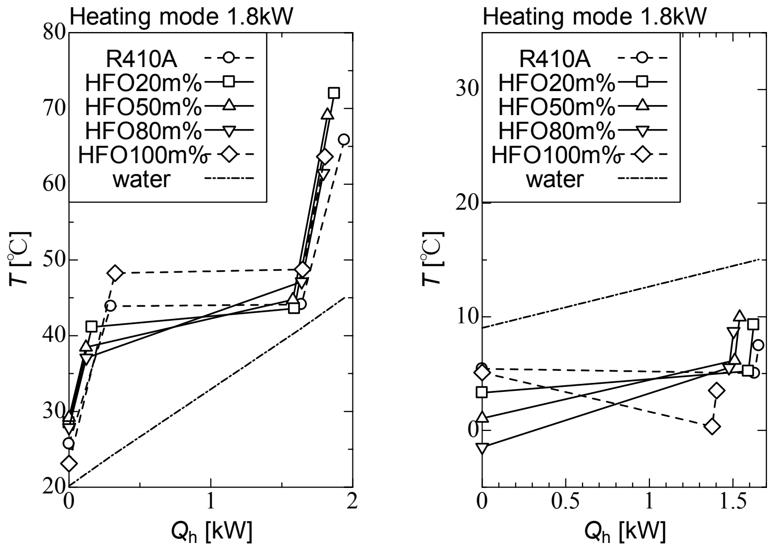
Figure 9: Relation between COP and heating load



(a) Evaporator (b) Condenser

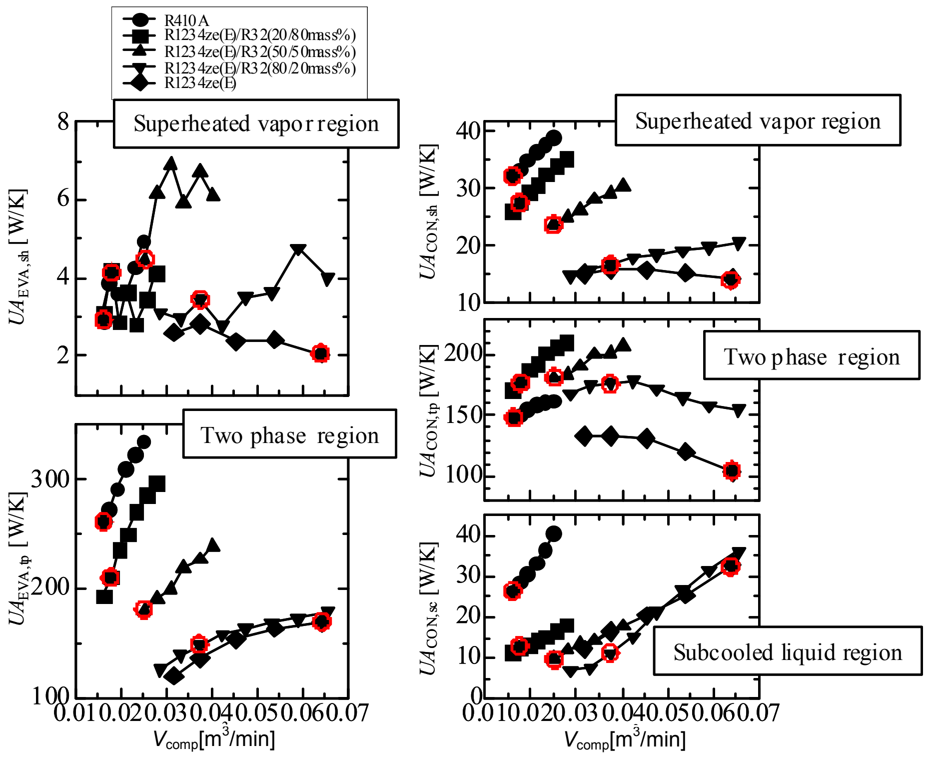
Figure 10: Relation between refrigerant pressure drop and mass flow rate

between refrigerant and water of R410A is the smallest among test refrigerants, and the pressure drop of 100mass% HFO-1234ze(E) is the largest. The refrigerant temperature of test mixtures rises in refrigerant flow direction overcoming the saturation temperature drop due to pressure drop increase. In condenser, the temperature difference between refrigerant and water of 100mass% HFO-1234ze(E) is the largest among test refrigerants, and the refrigerant temperature of test mixtures drops in refrigerant flow direction due to the temperature glide during phase change.



(a) Evaporator side (b) Condenser side

Figure 11: Relation between temperature and heat load in heat exchanger



(a) Evaporator side (b) Condenser side

Figure 12: Relation between UA and V_{comp}

Figures 12(a) and (b) show the heat exchange performance of test refrigerants in evaporator and condenser, respectively, where the ordinate denotes the overall heat transfer coefficient multiplied by the heat transfer area, and the abscissa denotes the volumetric refrigerant flow rate at the suction port of compressor. Symbols in both figures are the same as those in Figure 10, and red-circled symbols are the results at heating load 1.8 kW. Comparison among red-circled symbols suggests us the following heat transfer characteristics in two-phase region:

(1) Heat transfer performance in evaporator decreases, in order, R410A, 20mass%HFO-1234ze(E), 50mass% HFO-1234ze(E), 100mass% HFO-1234ze(E) and 80mass% HFO1234ze(E). It is inferred that the nuclear boiling of 100mass% HFO-1234ze(E) is suppressed by the forced convection effect, and those of HFO1234ze(E) mixtures are suppressed by the mass transfer resistance in the liquid surrounding vapor bubbles.
(2) Heat transfer performance in condenser decreases, in order, 20mass%HFO-1234ze(E), 50mass%HFO-1234ze(E), R410A, 80mass%HFO-1234ze(E), and 100mass% HFO1234ze(E). Heat transfer performances of mixtures are relatively higher than pure refrigerants. This is mainly due to temperature glide effect of mixtures.

4 CONCLUDINGS REMARKS

The main findings in the present study are summarized as follows:

(1) Pure HFO-1234ze(E) is not suitable for an alternative of R410A, but the addition of HFC-32 into HFO-1234ze(E) are one of promising methods to introduce a Low GWP refrigerant as an alternative of R410A; the COP value of the 20mass% HFO-1234ze(E) mixture is only 1% lower than that of R410A.
(2) The addition of HFC-32 into HFO-1234ze(E) is also effective for the reduction of the pressure drop in heat exchangers.
(3) The boiling heat transfer of 100mass% HFO-1234ze(E) may be the lowest among test refrigerant due to suppression of nuclear boiling by the forced convection effect, and the nuclear boiling of HFO1234ze(E) mixtures may be suppressed by the mass transfer resistance in the liquid surrounding vapor bubbles.
(4) Heat transfer performances of mixtures are relatively higher than pure refrigerants. This is mainly due to temperature glide effect of mixtures.

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