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Performance Evaluation of Heat Pump Cycle using Low GWP Refrigerant Mixtures of HFC-32 and HFO-1123

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ABSTRACT

In this study, the performance of heat pump cycle using low GWP refrigerant mixtures of HFC-32 and HFO-1123 was evaluated experimentally. The experimental system was a water heat source vapor compression cycle, mainly composed of an inverter-controlled and hermetic-type scroll compressor, an oil separator, a double-tube-type condenser, a liquid receiver, a solenoid expansion valve, and a double-tube-type evaporator. Three kinds of refrigerant mixtures of HFC-32/HFO-1123 with 58/42 mass% (GWP=393), with 46/54 mass% (GWP=312) and with 42/58 mass% (GWP=285) were tested at heating mode as well as at cooling mode. It was found that the COP of HFC-32/HFO-1123 mixture (58/42 mass%) was almost the same as that of R410A, while that of HFC-32/HFO-1123 mixture (42/58 mass%) was a little lower than that of R410A in both heating and cooling modes. It was also shown that the system COP of HFC-32/HFO-1123 mixtures exceeded that of R410A. In conclusion, the tested mixtures of HFC-32/HFO-1123 would be candidates of the alternative to R410A if the design of compressor and heat exchangers were optimized.

1. INTRODUCTION

Since the Montreal Protocol on Substances that deplete the Ozone Layer became effective, hydrofluorocarbons (HFCs) have been a major refrigerant in air-conditioning and refrigeration systems. At Kyoto Conference (COP3) in 1997, however, a worldwide agreement was obtained to regulate the production and use of HFCs due to its high global warming potential (GWP). In 2016, large reduction in production and use of HFCs were agreed at the 28th Meeting of the Parties (MOP 28) to the Montreal Protocol on Substances that deplete the Ozone Layer. In this context, it is necessary to accelerate the change of refrigerants into zero- or low-GWP refrigerants, and hydrofluoroolefins (HFOs), such as HFO-1234yf, HFO-1234ze(E), HFO-1123, are one of the most promising alternative refrigerants.

Our research group intensively studied on binary mixtures of HFO-1234ze(E)/HFC-32 (Koyama et al. 2010, Koyama et al. 2011, Fukuda et al. 2012a, 2012b, 2012c) and HFC-32/HFO-1234yf (Kojima et al., 2015, Kojima et al., 2016), and tertiary mixtures of CO2/HFC-32/HFC-1234ze(E) (Fukuda et al., 2014) and CO2/HFC-32/HFO-1234yf (Kojima et al., 2016). Other researchers also focused on the binary mixtures of HFC-32/HFC-1234ze(E) (Cheng et al., 2017) and HFC-32/HFO-1234yf (Xu et al., 2017). One of the drawbacks of these mixtures is, however, a relatively large temperature glide. It is known that the temperature glide of zeotropic mixture will work to reduce irreversible losses in heat exchangers. When the temperature glide matches the temperature changes of heat transfer media, the cycle is called Lorentz cycle and the COP will be maximized. Nevertheless, the zeotropic mixtures are not so attractive to manufacturers because it will cost for a renewal of heat exchanger design and for control of compositions.

In this study, we focused on a new HFO refrigerant, HFO-1123, which has zero ozone depletion potential (ODP) and negligible GWP. HFO-1123 cannot be used because of the risk of disproportionation reaction. Therefore, three kinds of refrigerant mixtures of HFC-32/HFO-1123 with 58/42 mass% (GWP=393), with 46/54 mass% (GWP=312) and with 42/58 mass% (GWP=285) were used in this study. These three compositions were tested under heating and cooling conditions by using experimental setup of a water heat source heat pump.

2. EXPERIMENT

2.1 Experimental setup

The schematic of the experimental setup is depicted in Fig.1. It is a water heat source vapor compression cycle, and the main components of the setup are an inverter-controlled and hermetic-type scroll compressor, which has a cylinder volume of 11 cm^3, an oil separator, a double-tube-type condenser, a liquid receiver, a solenoid expansion valve, and a double-tube-type evaporator. The specifications of the condenser and the evaporator were identical, and it is given in Table 1.

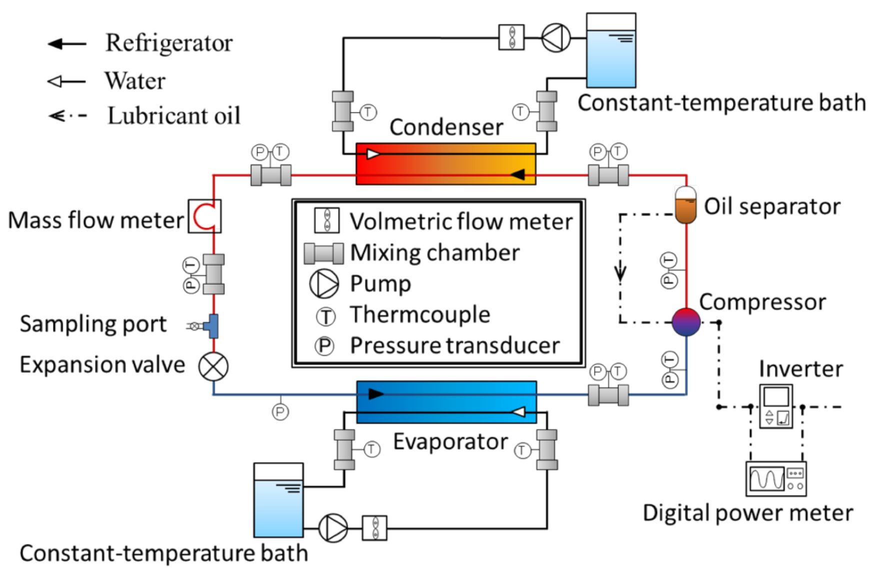


Figure 1: Schematic of the experimental setup

Table 1: Specifications of the heat exchanger for condenser and evaporator

Tube		Outer diameter [mm]	Inner diameter [mm]	Length [mm]	Surface
Double tube type	Outer tube	15.88	13.88	7200	Smooth
	Inner tube	9.53	7.53	7200	Inner grooved

2.2 Measurement

The refrigerant temperature was measured at the inlet and outlet of the compressor, the inlet and outlet of the condenser, the inlet and outlet of the expansion valve, and the outlet of the evaporator. The temperature of the heat source water was measured at the inlet and outlet of the condenser and those of the evaporator. Type K sheathed thermocouple was used as the temperature sensor, which was calibrated between -5 ºC to 70 ºC so that the accuracy was ±0.05 K. The pressure was measured at the inlet and outlet of the condenser, the inlet of the expansion valve, and the outlet of the compressor by absolute pressure sensor with full scale of 5 MPa. The pressure at the inlet and the outlet of the evaporator and at the inlet of the compressor was measured by absolute pressure sensor with full scale of 2 MPa. For both pressure sensors, the accuracies were within ±0.2% of rated output. The mass flow rate of the refrigerant was measured by a Coriolis flow meter, which had a measurement range of 8.1 to 81 kg/h with accuracy of ±0.22%. The volume flow rate of the heat source water was measured by a volume flow meter, which had a range up to 300 L/h with accuracy of ±0.5%. The electricity input to the inverter and that to the compressor were measured by digital power meters. The accuracies of the power meters were ±(0.2% of reading + 0.2% of range) for the former one and ±(0.1% of reading + 0.05% of range) for the latter one. All the data measured were collected to a data acquisition unit. The sampling was performed for 3 minutes with interval of 2 seconds after the system reached steady state.

2.3 Refrigerant

Five different refrigerants, which were R410A, R32 (HFC-32) and three different mass ratios of HFC-32/HFO-1123 mixture, were tested by experiment. R410A was used to measure the baseline performance. The mixtures of HFC-32/HFO-1123 were prepared as 58/42 mass%, 46/54 mass%, and 42/58 mass%. The GWPs of them were 393, 312, 285, respectively.

2.4 Experimental conditions

The experiment was performed under two heating conditions and one cooling condition. The setting of these experimental conditions are summarized in Table 2.

Table 2: Experimental conditions

Conditions	Heat source water temperature		Heat output [kW]	Degree of superheat [K]
	Condenser	Evaporator		
Heating 1	20 → 30	15 → 9	1.6 to 2.6	4 (±1)
Heating 2	20 → 45	15 → 9	1.6 to 2.6	4 (±1)
Cooling	30 → 45	20 → 10	1.4 to 2.4	4 (±1)

3. DATA REDUCTION

The performance of the heat pump cycle was evaluated by the system COP and the thermodynamic cycle COP. The systems COP was defined as the heating or cooling output divided by the electricity input to the inverter, as given in Eqs.(1) and (2).

$$COP_{\mathrm{h,system}} = \frac{Q_{\mathrm{W,COND}}}{E_{\mathrm{INV}}} \tag{1}$$

$$COP_{\mathrm{c,\,system}} = \frac{Q_{\mathrm{W,EVA}}}{E_{\mathrm{INV}}} \tag{2}$$

Where, E_{INV} denotes the electricity input to the inverter. $Q_{\mathrm{W,COND}}$ and $Q_{\mathrm{W,EVA}}$ are heating and cooling output calculated from the heat source water side. They are given as Eqs.(3) and (4).

$$Q_{\mathrm{W,COND}} = m_{\mathrm{W,COND}} C_{\mathrm{p\ W,COND}} \left(T_{\mathrm{W,COND,out}} - T_{\mathrm{W,COND,in}}\right) + Q_{\mathrm{LOSS}} \tag{3}$$

$$Q_{\mathrm{W,EVA}} = m_{\mathrm{W,EVA}} C_{\mathrm{p\ W,EVA}} \left(T_{\mathrm{W,EVA,in}} - T_{\mathrm{W,EVA,out}}\right) + Q_{\mathrm{LOSS}} \tag{4}$$

The symbols are given in nomenclature. QLOSS represents the heat loss between the heat source water to the ambient air. It was measured by preliminary tests.

The thermodynamic performance of the heat pump cycle was evaluated by cycle COPs, which were defined by Eqs.(5) and (6).

$$COP_{\mathrm{h,cycle}} = \frac{Q_{\mathrm{R,COND}}}{m_{\mathrm{R}}\left(h_{\mathrm{d}} - h_{\mathrm{s}}\right)} = \frac{COP_{\mathrm{h,system}}}{\eta_{\mathrm{INV}} \cdot \eta_{\mathrm{M}}} * \frac{Q_{\mathrm{R,COND}}}{Q_{\mathrm{W,COND}}} \tag{5}$$

$$COP_{\mathrm{c,cycle}} = \frac{Q_{\mathrm{R,EVA}}}{m_{\mathrm{R}}\left(h_{\mathrm{d}} - h_{\mathrm{s}}\right)} = \frac{COP_{\mathrm{c,system}}}{\eta_{\mathrm{INV}} \cdot \eta_{\mathrm{M}}} * \frac{Q_{\mathrm{R,EVA}}}{Q_{\mathrm{W,EVA}}} \tag{6}$$

As shown in Eqs.(5) and (6), the cycle COPs represent the performance from the refrigerant side. The relationship between the system COP and the cycle COP is also given in Eqs.(5) and (6). They are convertible by using the inverter efficiency and the mechanical efficiency of the compressor, which are defined as Eqs.(7) and (8), respectively.

$$\eta_{\mathrm{INV}} = \frac{E_{\mathrm{COMPR}}}{E_{\mathrm{INV}}} \tag{5}$$

$$\eta_{\mathrm{M}} = \frac{m_{\mathrm{R}}\left(h_{\mathrm{d}} - h_{\mathrm{s}}\right)}{E_{\mathrm{COMPR}}} \tag{6}$$

4. Results and Discussion

4.1 Cycle COP

The five refrigerants are compared in terms of COP as a function of heat output for two heating conditions as well as for cooling condition in Fig.2. In these results, the charge amount of the refrigerant was adjusted so that the COP was maximized. R32 showed the highest COPs for all conditions among five refrigerants. The COP of the HFC-32/HFO-1123 mixture was close to that of R410A when the mass ratio of HFC-32 and HFO-1123 was 58% and 42%. By increasing the ratio of HFO-1123, the cycle COP decreased noticeably at the heating condition 1 and the cooling condition. One of the major reasons of the deterioration of cycle COP with less HFC-32 would be the difference in latent heat between HFC-32 and HFO-1123. HFC-32 has larger latent heat than HFO-1123, and therefore, the latent heat of their mixtures decreased with increase of percentage of HFO-1123.

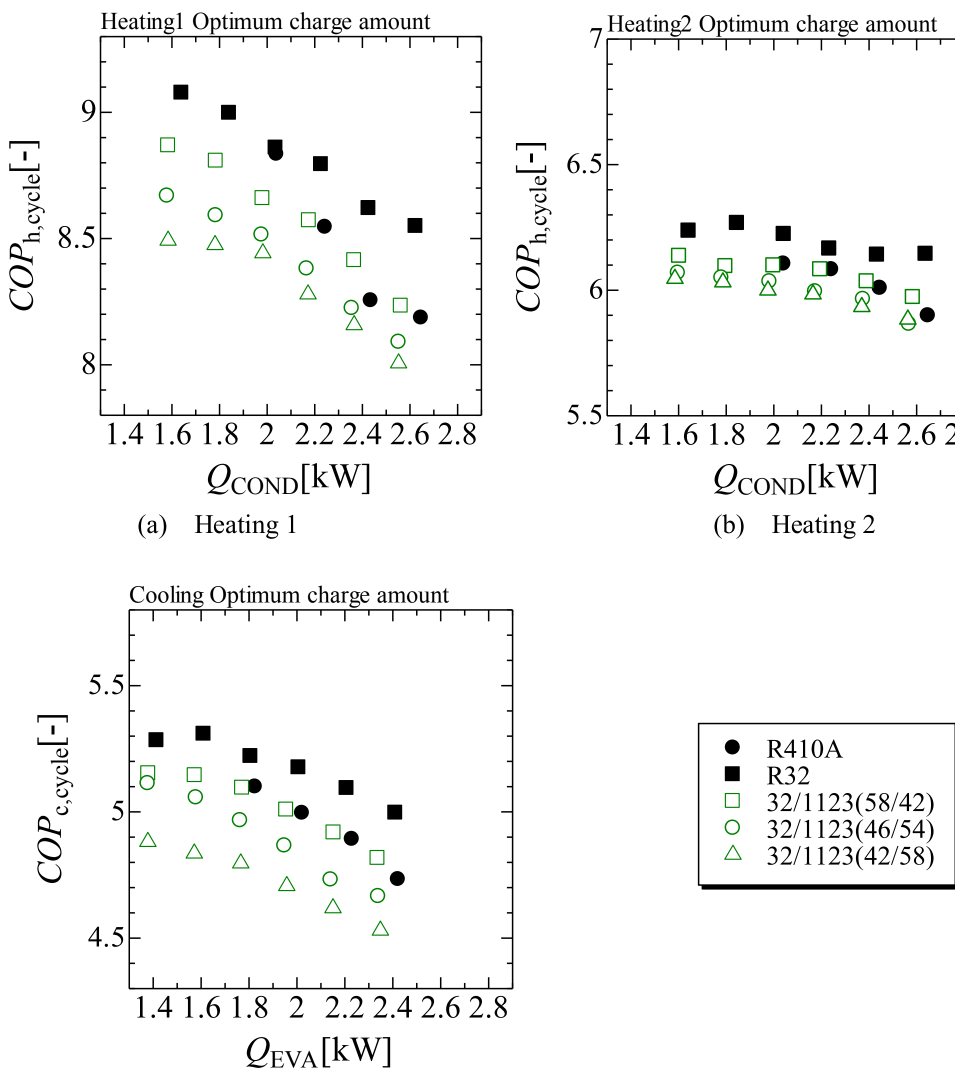
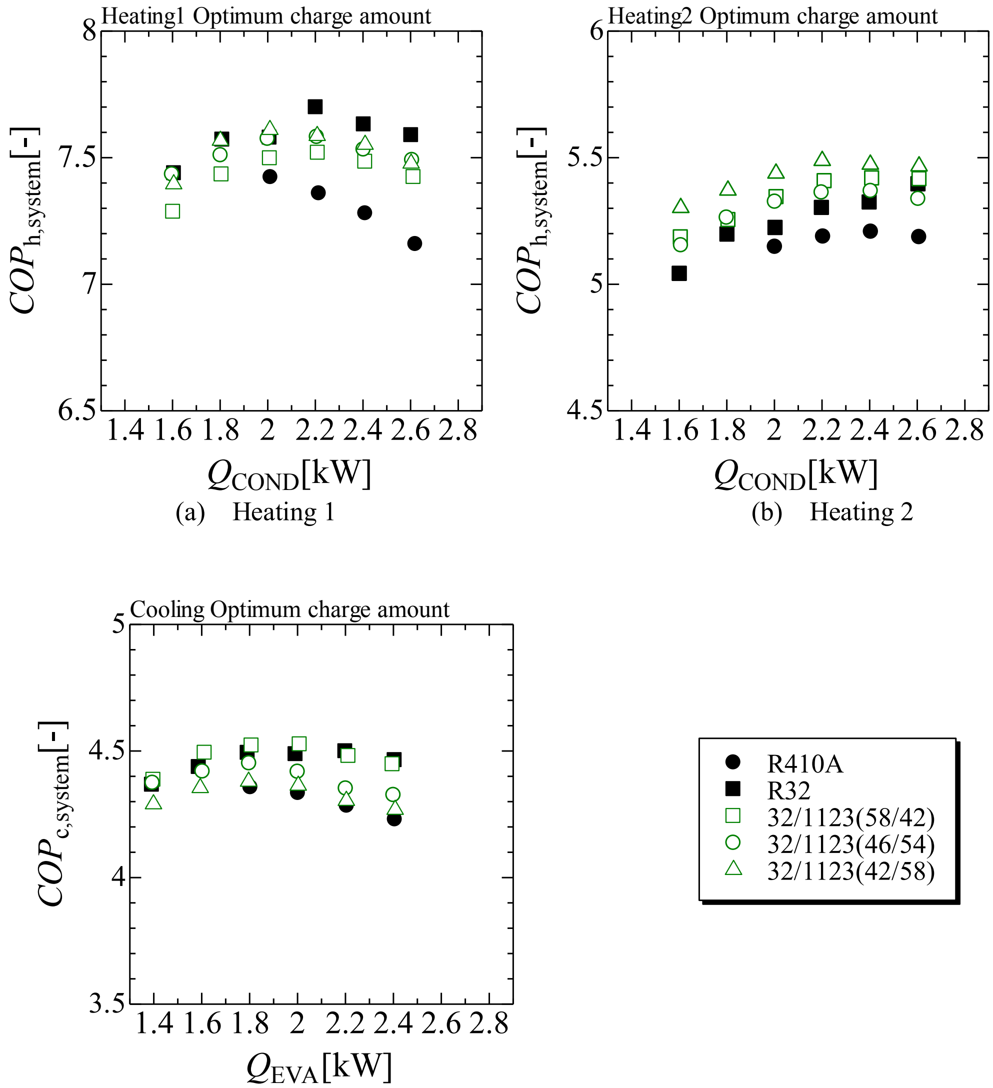


Figure 2: The cycle COP as a function of heat output

Fig.3 shows the comparison of the five refrigerants in terms of the system COP. It was remarkable that the system COPs of HFC-32/HFO-1123 mixtures achieved higher system COPs compared with R410A at most of the conditions. Moreover, they were close to or superior to that of R32 depending on the conditions. This is mainly due to the better performances in conversion factor between the cycle COP and the system COP, which is $h_{INV}h_M Q_W/Q_R$ from Eqs.(5) and (6), with HFC-32/HFO-1123 mixtures. It was also found that the system COP would be maximized at certain heat output while the cycle COP generally had a decreasing trend. The results suggested that the system performance was dominated by the compressor and the heat exchanger performances, which are represented by $h_{INV}h_M$ and Q_W/Q_R, respectively.



(a) Heating 1

(b) Heating 2

(c) Cooling

Figure 3: The system COP as a function of heat output

5. CONCLUSIONS

The main findings of the study are summarized as follows.

- The cycle COPs of HFC-32/HFO-1123 mixtures were lower than that of R32 for three different mass ratios, and the less percentage of HFC-32 resulted in the lower cycle COP.
- The cycle COP of HFC-32/HFO-1123 58 mass%/42 mass% had a close COP to that of R410A.
- The system COPs of HFC-32/HFO-1123 mixtures were higher than that of R410A at most of the conditions. In addition it exceeded the system COP of R32 at heating condition 2.

NOMENCLATURE

Cp	specific heat	(J kg^{-1} K^{-1})
E	electricity	(W)
h	enthalpy	(J kg^{-1})
m	mass flow rate	(kg s^{-1})
Q	heat	(W)
T	temperature	(K)

Greek Symbols

h	efficiency	(-)

Subscript

COMPR	compressor
COND	condenser
cycle	cycle
d	discharge of compressor
EVA	evaporator
in	inlet
LOSS	loss
M	mechanical
out	outlet
R	refrigerant
s	suction of compressor
W	heat source water

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