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Study on a hybrid refrigeration cycle by combining an absorption process with a compression process using Low-GWP refrigerant

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Abstract

This research is intended to evaluate a “hybrid refrigeration cycle” which is combining an absorption process driven by utilizing low-temperature waste heat with a compression process in one circulation cycle. A compressor is installed between an evaporator and an absorber in a hybrid refrigeration cycle. In this research, HFO or HCFO refrigerant and an organic solvent were used as a working pair for the absorption process.

The absorption equilibrium solubilities of the working pairs were measured when the experimental temperature and pressure conditions were changed. The parameters of NRTL model were determined by the experimental absorption equilibrium solubilities, and the PTx lines of the working pairs were calculated. Further, the enthalpy-concentration diagrams of the working pairs were calculated by using the material properties of the refrigerants and the absorbents.

It was confirmed that there are the conditions that the estimated cooling COP_c based on the power in the hybrid refrigeration cycle can be more than twice as high as that in the compression refrigeration cycle.

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Keywords: hybrid refrigeration cycle; compression; absorption; low-GWP refrigerant; waste heat

1. Introduction

Utilization of waste heat is regarded as one of the effective means to achieve carbon neutrality by 2050. This research is intended to evaluate a “hybrid refrigeration cycle” which is combining an absorption process driven by utilizing low-temperature heat (lower than 100°C) with a compression process in one circulation cycle.

Figure 1 shows the schematic flow diagram of the hybrid refrigeration cycle. A compressor is installed between an evaporator and an absorber. Although the power of a solution pump is necessary, the power of a compressor is reduced more than the increased power for a solution pump. Therefore, it is said that cooling COP_c of the hybrid refrigeration cycle based on power is significantly improved.

Recently, the use of HFCs in air conditioning and refrigeration equipment has been phased out due to their high GWP, and the transition to low GWP refrigerants HFO and HCFO is progressing. As for absorption chillers, working pairs using low GWP refrigerants are being developed, and HFO and ionic liquid working pairs have been proposed [1-2].

In this research, HFO refrigerant (GWP=1 or less) and an organic solvent (glycol ether) were used as a working pair for the absorption process for the purpose of improving the circulation ratio. The absorption equilibrium solubilities of the working pairs were measured when the experimental temperature and pressure conditions were changed. The parameters of Non-Random Two Liquid (NRTL) model were determined by the experimental absorption equilibrium solubilities, and PTx lines of the working pairs were calculated.

The measurement temperature is from 35°C to 80°C. T1 and T2 are K sheath type thermocouples (1.6mm), and P1 and P2 are the pressure sensors (GP-M025, KEYENCE).

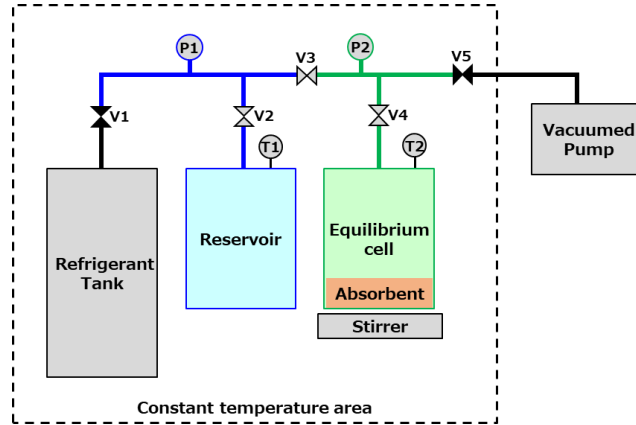


Fig. 2. Outline of apparatus for equilibrium absorption solubilities.

Figure 3 shows the experimental results of the relation between the refrigerant concentration and the pressure at different temperatures. The left of Fig. 3 indicates the result in case of [HMIM][Tf₂N] and the right of Fig. 3 indicates the result in case of Tetraethylene Glycol Dimethyl Ether. The refrigerant concentration is the mass ratio to the sum of the refrigerant and the absorbent. The dotted line in the right of Fig. 3 is fitted to the experimental data by quadratic function.

The refrigerant concentration in case of Tetraethylene Glycol Dimethyl Ether is much higher than that in case of [HMIM][Tf₂N] at the same temperature.

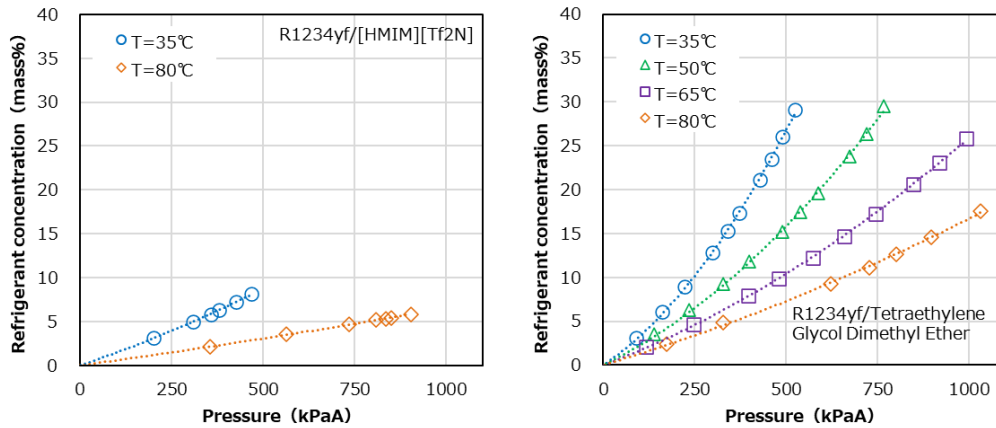


Fig. 3. Experimental results of the relation between the refrigerant concentration and the pressure at different temperatures. The left figure indicates the result in case of [HMIM][Tf₂N] and the right figure indicates the result in case of Tetraethylene Glycol Dimethyl Ether.

In order to validate the measured absorption equilibrium solubilities, the data of ref [1] in case of R1234yf/[HMIM][Tf₂N] is used.

The temperature conditions for the validation are follows.

- Evaporation temperature T_e is 10°C
- Condensation temperature T_c is 35°C
- Absorption temperature T_a is 35°C
- Generation temperature T_g is 80°C

Table 2 shows the refrigerant concentration of the strong solution x_s and that of the weak solution x_w and the circulation ratio in the condition above including in case of R1234yf/Tetraethylene Glycol Dimethyl Ether. The circulation ratio is defined to be the mass flow ratio of the strong solution to that of the absorbed refrigerant and can be also calculated to be $(100\% - x_w) / (x_s - x_w)$. The refrigerant pressure is estimated using Refprop

10.0. In case of R1234yf, the absorption (=evaporation) pressure is 438kPaA, and the generation (=condensation) pressure is 895kPaA. The pressure loss is not considered.

It is confirmed that the refrigerant concentration of the strong and weak solutions and the circulation ratio of ref [1] in case of R1234yf/[HMIM] [Tf₂N] is almost same as this work. Further, the refrigerant concentration difference between the strong and weak solutions in case of R1234yf/Tetraethylene Glycol Dimethyl Ether is increased, and the circulation ratio is significantly improved.

Table 2. Refrigerant concentration of the strong and weak solutions and the circulation ratio at $T_c=10^\circ\text{C}$, $T_e=35^\circ\text{C}$, $T_a=35^\circ\text{C}$, $T_g=80^\circ\text{C}$.

Refrigerant/Absorbent		Refrigerant concentration (mass%)			circulation ratio (-)
		Strong solution	Weak solution	$x_s - x_w$	
		x_s	x_w		
R1234yf/[HMIM][Tf ₂ N]	Ref. [1]	7.3	5.5	1.8	53
	This work	7.5	5.7	1.7	52
R1234yf/Tetraethylene Glycol Dimethyl Ether	This work	22.0	14.7	7.4	12

2.3. PTx lines and enthalpy-concentration diagrams

The Non-Random Two Liquid (NRTL) equation is applied to the experimental results of the relation between the refrigerant concentration and the pressure at different temperatures shown in the right of Fig. 3 for R1234yf/Tetraethylene Glycol Dimethyl Ether. The parameter α of the NRTL equation is set to be constant ($\alpha=0.2$) in this work [3]. It is confirmed that the deviations of the calculated refrigerant concentration using the NRTL equation from the experimental refrigerant concentration for R1234yf/Tetraethylene Glycol Dimethyl Ether are very small.

The left of Fig.4 shows PTx lines, and the right of Fig.4 shows the enthalpy-concentration diagrams in case of R1234yf/Tetraethylene Glycol Dimethyl Ether. The enthalpy of the solution which absorbed the refrigerant is calculated based on ref [4]. The enthalpy per unit solution mass is plotted against the refrigerant concentration which is the mass ratio of the refrigerant to the sum of the refrigerant and the absorbent. The enthalpy of the saturated liquid of the refrigerant at 0°C is 200kJ/kg, and the enthalpy of the absorbent at 0°C is 200kJ/kg.

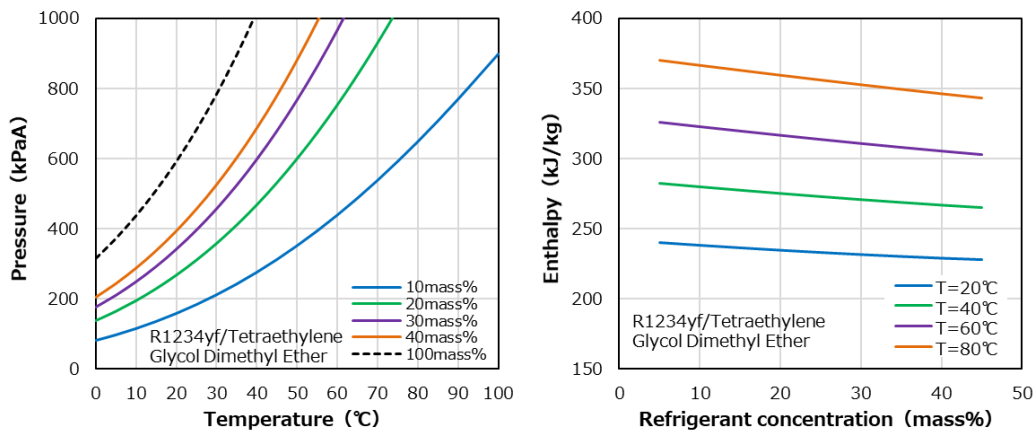


Fig. 4. PTx lines (left) and enthalpy-concentration diagrams (right) in case of R1234yf/Tetraethylene Glycol Dimethyl Ether

3. Process simulation

3.1. Flow and conditions for process simulation

The process simulation in case of R1234yf/Tetraethylene Glycol Dimethyl Ether is conducted for the double expansion flow indicated in Fig. 5.

The refrigerant at the outlet of the evaporator (1) is compressed to the intermediate pressure (2) by the low-stage compressor and is mixed (8) with the saturated vapor refrigerant from the separator (7). The strong solution (10) which absorbs the refrigerant (8) at the absorber is compressed (11) by the solution pump and is heated (12) at the solution heat exchanger and is separated into the refrigerant (9) and the weak solution (13) at the generator. The separated refrigerant (9) is condensed (3) at the condenser and is decompressed (4) by the expansion valve EV2 and is separated into the saturated vapor refrigerant (7) and the saturated liquid refrigerant (5) at the separator. The saturated liquid refrigerant (5) is decompressed (6) by the expansion valve EV1 and is directed to the evaporator. The weak solution (13) is cooled (14) at the solution heat exchanger and is decompressed (15) by the expansion valve EV3 and is directed to the absorber.

The high-stage compressor is provided for operation when the quantity of waste heat is relatively small.

The simulation conditions are follows.

- Compression efficiency is 75%
- Pump efficiency is 50%
- Superheat at the outlet of the evaporator is 5K
- Subcool at the outlet of the condenser is 0K
- Temperature efficiency of the solution heat exchanger is 85% (based on weak solution)
- Pressure and heat loss is none.
- Evaporation temperature T_e is 5°C
- Condensation temperature T_c is 30°C
- Absorption temperature T_a is 30°C

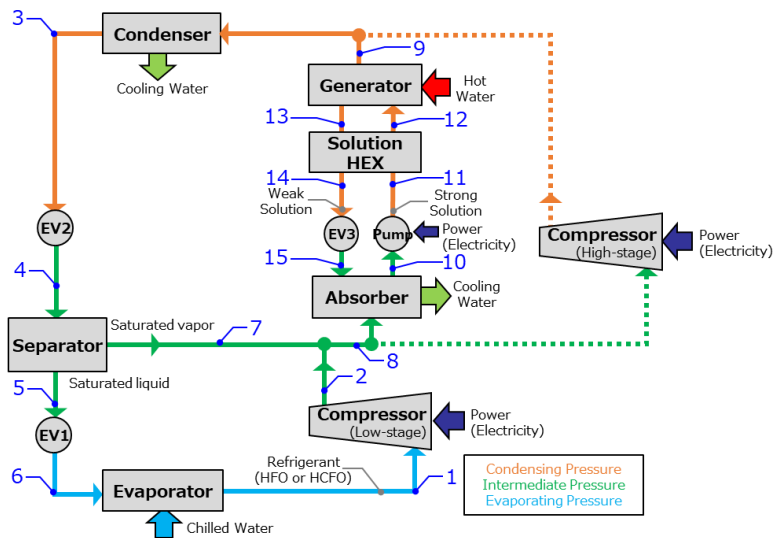


Fig. 5. Flow of the process simulation (double expansion).

3.2. Results of process simulation

Table 3 shows the example of simulation results of the state value at each position and the heat quantity and the power when the high-side compressor is not operated, the intermediate temperature T_m is 12°C, the generation temperature T_g is 80°C and the quantity of evaporation Q_e is 100kW.

The intermediate temperature T_m is defined to be the saturated temperature of the refrigerant at the outlet of the low-stage compressor.

Table 3. Simulation results of the state value at each position (upper) and the heat quantity and the power (lower) at $T_e=5^\circ\text{C}$, $T_c=30^\circ\text{C}$, $T_a=30^\circ\text{C}$, $T_g=80^\circ\text{C}$, $T_m=12^\circ\text{C}$, $Q_e=100\text{kW}$ without high-stage compressor.

Position	Pressure (kPaA)	Temperature (°C)	Enthalpy (kJ/kg)	Flow rate (kg/s)	Refrigerant concentration (mass%)
1	373	10.0	371	0.643	100
2	466	17.9	377	0.643	100

3	784	30.0	241	0.765	100
4	466	12.0	241	0.765	100
5	466	12.0	216	0.643	100
6	373	5.0	216	0.643	100
7	466	12.0	371	0.122	100
8	466	17.0	376	0.765	100
9	784	80.0	435	0.765	100
10	466	30.0	251	3.592	31.1
11	784	30.3	251	3.592	31.1
12	784	66.0	323	3.592	31.1
13	784	80.0	365	2.827	12.5
14	784	37.8	274	2.827	12.5
15	466	-	274	2.827	12.5

Part		Quantity of heat or power (kW)
Evaporation	Q_e	100
Condensation	Q_c	149
Absorption	Q_a	162
Generation	Q_g	205
Solution heat exchanger	Q_s	255
Compressor (low-stage)	W_{cl}	3.5
Solution pump	W_{sp}	2.3

Cooling COP_c based on the power in this hybrid cycle is 17.0 (=100kW / (3.5kW+2.3kW)). On the other hand, although the details are omitted, the compression power in the single compression cycle under the same condition is 14.0kW, so cooling COP_c in the single compression cycle is 7.2 (=100kW / 14.0kW). Therefore, cooling COP_c based on the power in the hybrid cycle corresponds to about 238% of that in the single compression cycle. The required quantity of generation heat corresponds to about 2.1 times of that the evaporation heat. The circulation ratio in the absorption process is about 4.7 (=3.592kg/s / 0.765kg/s).

Cooling COP_c based on the power (left) and the quantity of generation heat Q_g (right) are shown in Fig. 6 when the intermediate temperature T_m and the generation temperature T_g are changed in the condition above. Cooling COP_c is expressed as a ratio to that of single compression cycle in the same condition, and the quantity of generation heat Q_g is expressed as a ratio to that of evaporation heat Q_e . The result in the condition of Table 3 (the intermediate temperature T_m is 12°C and the generation temperature T_g is 80°C) is indicated by “□” in Fig. 6.

It can be seen that COP_c has a local maximum value against the intermediate temperature T_m (indicated by “○”), and the quantity of generation heat Q_g increase as the intermediate temperature T_m decreases. It is considered that although the compression power decreases with the decrease in the intermediate temperature T_m , the power of the solution pump and the quantity of generation heat Q_g increase due to the increase in the circulation ratio.

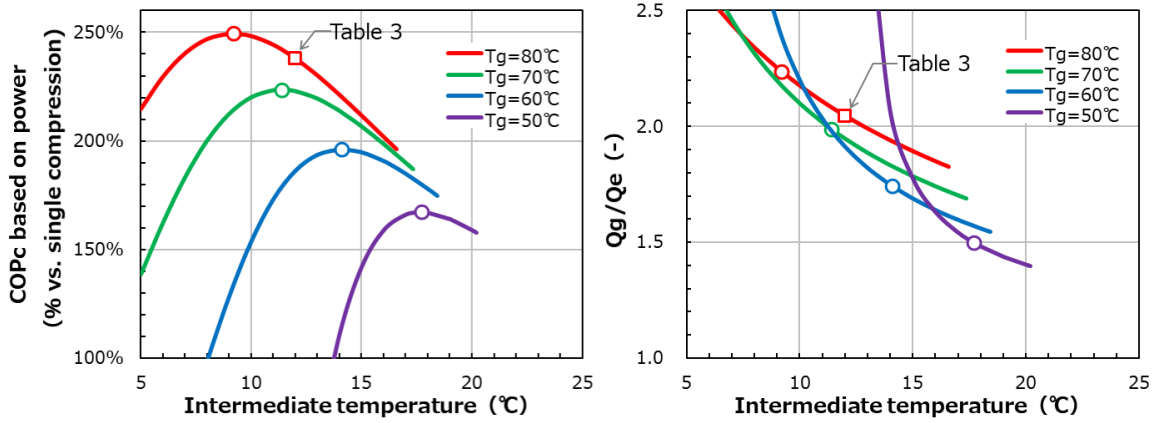


Fig. 6. Cooling COP_c based on the power (left) and the quantity of generation heat Q_g (right) against the immediate temperature T_m at $T_e=5^\circ\text{C}$, $T_c=30^\circ\text{C}$, $T_a=30^\circ\text{C}$, $T_g=50-80^\circ\text{C}$ without high-stage compressor.

Cooling COP_c based on the power against the quantity of generation heat Q_g is shown in Fig. 7 in the condition above. It can be seen that COP_c has a local maximum value against the quantity of generation heat Q_g (indicated by “O”).

When the generation temperature T_g decreases, the maximum value of COP_c decreases. However, it can be seen that the required quantity of generation heat Q_g at that time is also reduced.

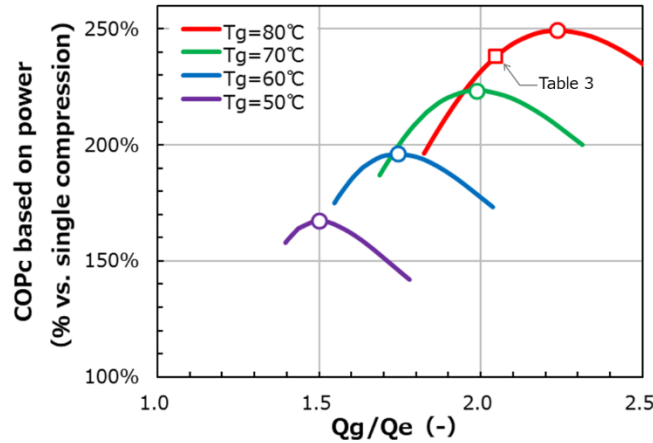


Fig. 7. Cooling COP_c based on the power against the quantity of generation heat Q_g at $T_e=5^\circ\text{C}$, $T_c=30^\circ\text{C}$, $T_a=30^\circ\text{C}$, $T_g=50-80^\circ\text{C}$ without high-stage compressor.

4. Conclusions

After measuring the equilibrium absorption concentration, PTx lines and enthalpy-concentration diagrams were created, and process simulation was performed in the hybrid refrigeration cycle which uses R1234yf as the low-GWP refrigerant and Tetraethylene Glycol Dimethyl Ether (organic solvent) as the absorbent. Below are the conclusions.

- The circulation ratio of the absorption process using Tetraethylene Glycol Dimethyl Ether as the absorbent is significantly lower than that using [HMIM][Tf₂N] (ionic liquid) in case of R1234yf as the refrigerant.
- In the simulation condition of this work, there is a local maximum value of cooling COP_c based on the power of the compressor and the solution pump against the quantity of generation heat. It was confirmed that there are the conditions that the estimated cooling COP_c based on the power in the hybrid refrigeration cycle can be more than twice as high as that in the single compression refrigeration cycle.

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