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Thermodynamic Analysis on High Temperature Heat Pump cycles using Low-GWP refrigerants for Heat recovery

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Abstract

To reduce energy consumption utilizing heat recovery systems is increasingly important in industrial applications. This study presents an exploratory assessment of heat pump type heat recovery systems using environmentally friendly refrigerants. Tested refrigerants are R1234ze(E), R1234ze(Z) and R365mfc. The coefficient of performance of two cycle configurations used to raise the temperature of heat media to 160 °C with a waste heat at 80 °C is calculated. The calculated cycle configurations are single-stage compression cycle, two-stage compression extraction cycle. Additionally, irreversible loss of each cycles are calculated to clarify that cycle characteristic of each cycle. As results, the coefficient of performance of two-stage compression extraction cycle is the highest among calculated cycle. In addition, each irreversible loss in two-stage compressed extraction cycle expect for evaporator is lower than that in single-stage compressed cycle. In all refrigerants, the decrease amounts of irreversible losses in condenser are the largest among each irreversible loss because of using several condensers and extraction. Finally, R1234ze(Z) is appropriate in this calculation condition at single-stage compressed cycle and two-stage compressed extraction cycle among test refrigerants.

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1. Introduction

Steam boilers are often used for the dry process, and the cleaning process. In boiler systems, heat loss from a large steam pipe and the emissions of greenhouse gases from fossil fuel combustion are, however, considerable. In addition, the heat exhaust from these relatively high-temperature processes is not utilized in many cases.

A large portion of unrecovered waste heat is low quality, i.e., at temperature below 200 °C, which is barely within the technical limitation of heat pumps. Therefore, recently, attempts to introduce industrial heat pumps to recover waste heat and reduce primary energy consumption have attracted significant attention.

On other hands, the global warming potential of refrigerants used in heat pumps is one problem. In the past few years, R1234ze(Z) was nominated as a low-GWP alternative to R245fa ($GWP_{100} = 858$) because of its very similar thermodynamic properties and extremely low-GWP ($GWP_{100} < 1$) by [1], [2]. Similarly, R1233zd(E) with a GWP_{100} of less than 1 [3], has been nominated as an alternative for R245fa with Organic Rankine Cycle system [4].

Therefore, in this study, candidate low-GWP refrigerants with different levels of critical temperature are selected, and two different cycle configurations are proposed for a case study. The COP and irreversible losses of

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these cycles using the selected refrigerants are calculated for the case of raising the temperature of pressurized water as a heat media up to 160 °C with the waste heat at 80 °C. From the calculated results, characteristic of the proposed cycles and the optimum refrigerant for target temperature are discussed in this paper.

2. Calculation method

In the following case study, the performance of an industrial heat pump system to recover waste is calculated. Utilizing the waste heat of 80 °C, the heat media of compressed water is preheated to 70 °C. Then the compressed water at 1 MPa is heated 70 °C to 160 °C by a heat pump system and delivers the heat to the usage site. The waste heat is, of course, used as the heat source of the heat pump.

2.1. Cycle configurations

Fig.1 shows two cycle configuration of the heat pump system for heat recovery. Fig.1 (a) and (b) correspond to the proposed: a single-stage compressed cycle and two-stage compressed extraction cycle, respectively. The single-stage compressed cycle in Fig.1 (a) has four compression processes because of reducing pressure ratio per one compression process. The two-stage compressed extraction cycle in Fig.1 (b) has extraction process to extract the vapor from the compressor. The extracted vapor rejects heat in a condenser, and then it converges with the liquid that flowed through a condenser and an expansion valve on the higher pressure side. After the conversion, the enthalpy and mass flow rate are increased by the liquid from the higher pressure side, and then the heat is rejected to the pressurized water in a subcooler. By converging the extracted vapor and the liquid from the higher pressure side, the internal energy remaining in the liquid is utilized in the subcooler instead of losing it as the throttling loss in an expansion valve returning to the evaporator.

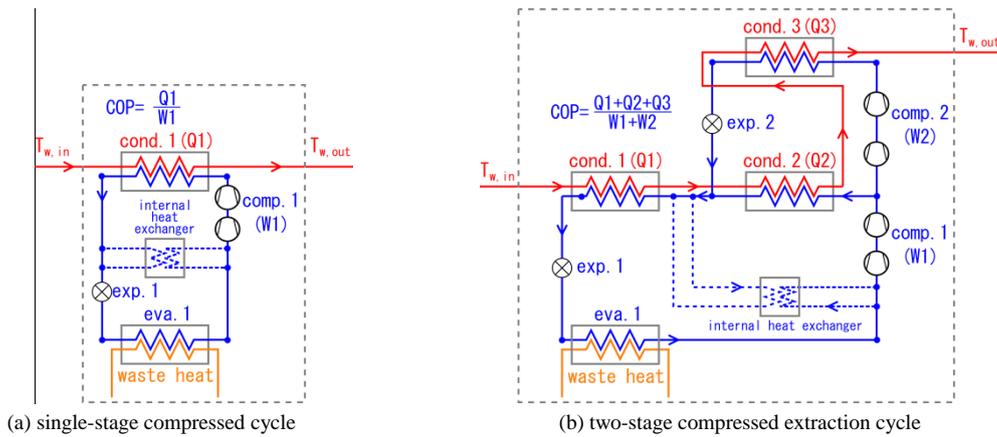


Fig. 1 Proposed cycle configurations for the heat recovery system.

2.2. Calculation condition and models for components

2.2.1 Compressor

Regardless of the pressure ratio or the rotation speed, the isentropic, mechanical, and motor efficiency are given as 0.85, 0.90 and 0.90, respectively, for each compression process. The degree of superheat at the compressor suction side is maintained at 5K. To avoid the wet compression, the degree of superheat is increased excessively by an internal heat exchanger to keep the compressor discharge state superheated.

2.2.2 Heat exchanger (condenser/gas-cooler/subcooler and evaporator)

Fig. 2 illustrates the calculation model of the temperature distribution in the evaporator and the condenser (gas-cooler, subcooler) on the T-Q diagram of the refrigerant. In this model, the pinch temperature (i.e., the minimum approach temperature) is 5 K in the subcool and superheat regions, whereas it is 2 K in the two-phase region (Table 1). The refrigerant temperature is a saturation pressure, which corresponds to the saturation temperature base on temperature difference in Fig. 2. Table 1 listed typical calculation conditions for the case

study. The inlet and outlet temperatures of compressed water are 70 and 160 °C. The inlet and outlet temperatures of heat source fluid (waste heat) are 80 and 70 °C. Compressed water mass flow rate is 0.2 kg/s. Therefore, the refrigerant temperatures and pressures are decided to fulfill above condition using by Refprop ver.9.1.

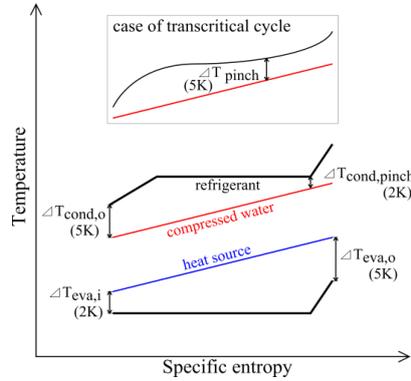


Fig. 2 Model for the approach temperature and pinch point in heat exchanger.

Table 1 Typical calculation conditions for the case study.

	Temperature of waste heat	80 °C
Compressed water	inlet temperature (after pre-heating)	70 °C
	outlet temperature	160 °C
Heat source fluid (waste heat)	inlet temperature	80 °C
	outlet temperature	70 °C
	Compressed water mass flow rate	0.2 kg/s

2.2.3 Internal heat exchanger

As drawn by the dotted lines in Fig., an internal heat exchanger can be considered if it is necessary to keep the refrigerant state superheated at the compressor discharge. The pinch point at either the entrance or the exit is always greater than 2 K.

2.3. Data reduction

2.3.1 Coefficient of performance (COP)

The coefficient of performance (COP) of the single-stage compressed cycle is calculated by this equation.

$$COP = \frac{Q_1}{W_1}$$

The COP of the two-stage compressed extraction cycle is calculated by this equation.

$$COP = \frac{Q_1 + Q_2 + Q_3}{W_1 + W_2}$$

2.3.2 Irreversible loss of component

The total irreversible loss during cycling, L_{total} , can be divided into the following irreversible losses of the main elements (e.g., compressor and evaporator), as follows:

$$L_{total} = L_{COND} + L_{EVA} + L_{EXP} + L_{COMPR} + L_H + L_P \quad (3)$$

Fig. 3 illustrates irreversible losses generated in condenser, evaporator, expansion valve, compressor (departure from the isentropic compression), and connecting pipe in a T-s diagram. In the figure, water temperature and refrigerant temperature are plotted against the entropy generation rate. The temperature and the specific entropy are calculated by assuming that the specific enthalpy changes in the heat exchangers are proportional to the pressure. The irreversible losses per refrigerant mass in each component are calculated as follow,

$$L_{COND} = m_{ref} \times \sum_n [(T_{R,i} - T_{W,j}) + (T_{R,i-1} - T_{W,j-1})] s_i / 2$$

$$L_{EVA} = m_{ref} \times \sum_n [(T_{W,i} - T_{R,i}) + (T_{W,i-1} - T_{R,i-1})] \Delta s_i / 2$$

$$L_{EXP} = \int_{s_{EXP, in}}^{s_{EXP, out}} T_R ds$$

$$L_{COMPR} = \int_{s_{COMPR, in}}^{s_{COMPR, out}} T_R ds$$

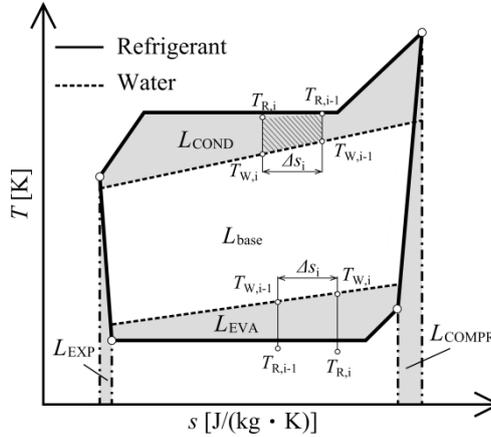


Fig. 3 Irreversible loss in each element.

2.4. Calculated refrigerants

Table 2 compares the characteristics and properties of the selected refrigerants for industrial high-temperature heat pumps. The refrigerants are listed in the order of their critical temperature from top to bottom. R1234zd(Z) have been invested in this decade [5] as alternatives to R245fa. R365mfc has the highest critical temperature among selected refrigerants. Although R365mfc has a relatively high GWP, a low-GWP alternative with similar physical properties, such as HFEs, will likely be found shortly.

Table 2 Fundamental characteristics and properties of the selected refrigerants.

	GWP ₁₀₀ *	Critical Pressure	Critical temp.	NBP**	Latent heat***	Density***	
		[MPa]	[°C]			[°C]	[kJ·kg ⁻¹]
R1234ze(Z)	<1	3.53	150.1	9.7	144.12	982.3	69.39
R1233zd(E)	<1	3.62	166.5	18.3	142.25	1049.7	56.26
R365mfc	794	3.27	186.9	40.2	154.04	1075.5	33.20

*IPCC 5th report **Normal boiling point ***at bulk temperature 100 °C

3. Calculation result

3.1. Results of COP

Table 3 (a) and (b) lists the calculation results of the overall COP and pressure ratio of compressors under the single-stage compressed cycle and two-stage compressed extraction cycle. The values in parenthesis of Table 3 are a compression ratios for a single compressor.

In the single-stage compressed cycle (Table 3 (a)), the COP and pressure ratio of R1234ze(Z) is the highest and the lowest among calculated refrigerants. On other hands, the COP and pressure ratio of R365mfc is the lowest and the highest.

In the two-stage compressed extraction cycle (Table 3 (b)), the tendency of COP by the difference in refrigerant is same in case of single-stage compressed cycle. However, the difference of COP by the difference in refrigerant is smaller. Than that in case of single-stage compressed extraction cycle. The pressure ratio on

higher pressure side of R1234ze(Z) is the highest. On other hands, the pressure ratio on lower pressure side of R1234ze(Z) is lowest.

Therefore, R1234ze(Z) is appropriate in this calculation condition at single-stage compressed cycle and two-stage compressed extraction cycle.

Table 3 calculation results of the overall COP and pressure ratio of compressors.
(a) single-stage compressed cycle

Refrigerant	COP	Pd/Ps
R1234ze(Z)	4.24	5.94(2.44)
R1233zd(E)	4.18	6.36(2.52)
R365mfc	3.68	8.30(2.88)

(b) two-stage compressed extraction cycle

Refrigerant	COP	$Pd/Ps1$	$Pd/Ps2$
R1234ze(Z)	4.58	2.65(1.63)	2.26(1.50)
R1233zd(E)	4.55	2.74(1.65)	2.35(1.53)
R365mfc	4.44	2.59(1.61)	3.21(1.79)

3.2. Irreversible loss of single-stage compressed cycle.

Fig 4 (a), (b) and (c) shows state of single-stage compressed cycle with R1234ze(Z), R1233zd(E) and R365mfc in T-s diagram. The cycle using R1234ze(Z) forms transition critical cycle, the cycle using R1233zd(E) and R365mfc form sub critical cycle. In addition, the internal heat exchanger is used in case of cycle with R365mfc.

Table 4 lists the calculation results of the irreversible losses of condenser, evaporator, expansion valve, compressor and all. The irreversible loss trend of each component is different by difference in refrigerants. The irreversible loss in condenser of R365mfc is the highest and that of R1234ze(Z) is the lowest. These causes are that average temperature difference between R365mfc and water is largest due to using internal heat exchanger and average temperature difference between R1234ze(Z) and water is smallest because refrigerant is state of super critical (Fig. 4 (a) and (c)). In the case of irreversible loss in evaporator, even if any pure refrigerant is used, result is almost same. Because evaporation temperature almost never change even if any pure refrigerant is used. The irreversible loss through expansion valve of R1234ze(Z) is the highest and that of R365mfc is the lowest. That is, the irreversible loss though expansion valve is smaller as refrigerant where normal boiling point is higher and subcooled liquid of refrigerant is larger. The irreversible loss in compressor of R365mfc is the highest and that of R1234ze(Z) is the lowest. This cause is that the irreversible loss in compressor is smaller as refrigerant where pressure ratio of compressor is larger.

As results, total irreversible loss of R365mfc is the largest and that of R1234ze(Z) is smallest. Therefore, the COP of R1234ze(Z) is the highest among calculated refrigerants and the COP of R365mfc is the lowest.

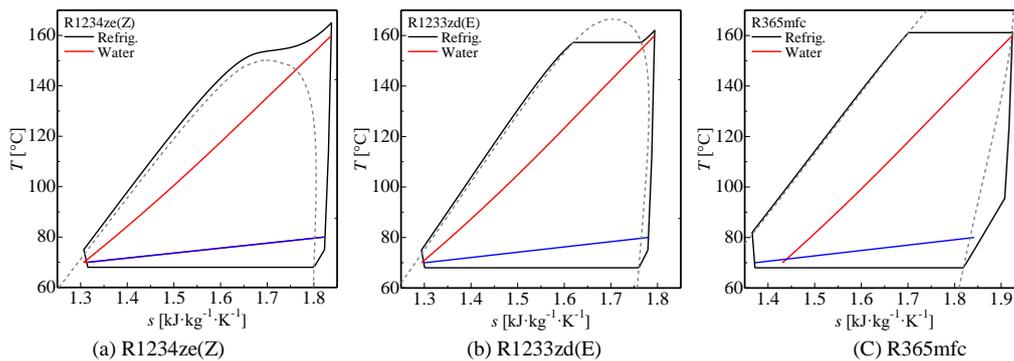


Fig. 4 State of single-phase compressed cycle in T-s diagram

Table 4 irreversible losses of condenser, evaporator, expansion valve, compressor and all in case of single-stage compressed cycle.

Refrigerant	L [kW]				
	COND	EVA	EXP	COMPR	ALL
R1234ze(Z)	3.10	1.24	1.06	2.07	7.48
R1233zd(E)	3.47	1.24	0.90	2.11	7.71
R365mfc	5.48	1.19	0.75	2.43	9.85

3.3. Irreversible loss of two-stage compressed extraction cycle.

Fig. 5 (a), (b) and (c) shows state of two-stage compressed extraction cycle with R1234ze(Z), R1233zd(E) and R365mfc in T-s diagram. The cycle using R1234ze(Z) forms transition critical cycle, the cycle using R1233zd(E) and R365mfc form sub critical cycle. In addition, the internal heat exchanger is used in case of cycle with R365mfc as in the case of single-stage compressed cycle.

Table 5 lists the calculation results of the irreversible losses of condenser, evaporator, expansion valve, compressor and all. The tendency of each irreversible loss by the difference in refrigerant is same in case of single-stage compressed cycle. Each irreversible loss in two-stage compressed extraction cycle expect for evaporator is, however, lower than that in single-stage compressed cycle. This cause is that heat transfer amount of evaporator increase due to COP increase than in case of single-stage compressed cycle. In all refrigerants, the decrease amounts of irreversible losses in condenser are the largest among each irreversible loss because of using several condensers and extraction. The irreversible loss in expansion valve and compressor slightly decrease than in case of single-stage compressed cycle. These causes are that the mass flow in higher pressure side is less than single-stage compressed cycle due to using extraction.

As results, total irreversible loss of R365mfc is the largest and that of R1234ze(Z) is smallest. Therefore, the COP of R1234ze(Z) is the highest among calculated refrigerants and the COP of R365mfc is the lowest.

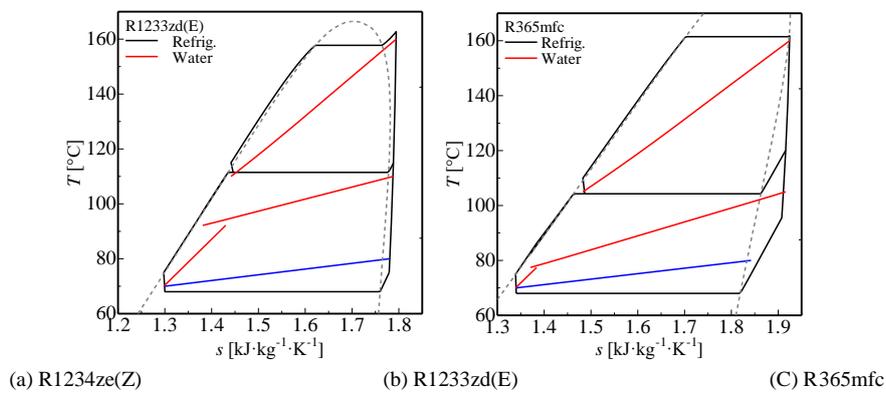


Fig.5 State of two-phase compressed extraction cycle in T-s diagram

Table 5 irreversible losses of condenser, evaporator, expansion valve, compressor and all in case of two-stage compressed extraction cycle.

Refrigerant	L [kW]				
	COND	EVA	EXP	COMPR	ALL
R1234ze(Z)	2.03	1.26	1.02	1.97	6.28
R1233zd(E)	2.29	1.26	0.84	1.99	6.37
R365mfc	2.73	1.25	0.53	2.05	6.55

4. Conclusion

The cycle performances and irreversible loss of three refrigerants, R1234ze(Z), R1233zd(E) and R365mfc, were thermodynamically analyzed by using heat recovery systems. The following conclusions are drawn from the experimental results:

- (1) In both the single-stage compressed cycle and two-stage compressed extraction cycle, the COP of R1234ze(Z) is the highest among calculated refrigerants. On other hands, the COP of R365mfc is the lowest.
- (2) In single-stage compressed cycle, all irreversible loss of R1234ze(Z) is lowest because irreversible loss in condenser is the lowest specifically.
- (3) In two-stage compressed extraction cycle, the tendency of each irreversible loss by the difference in refrigerant is same in case of single-stage compressed cycle. Each irreversible loss in two-stage compressed extraction cycle expect for evaporator is lower than that in single-stage compressed cycle. In all refrigerants, the decrease amounts of irreversible losses in condenser are the largest among each irreversible loss because of using several condensers and extraction.

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