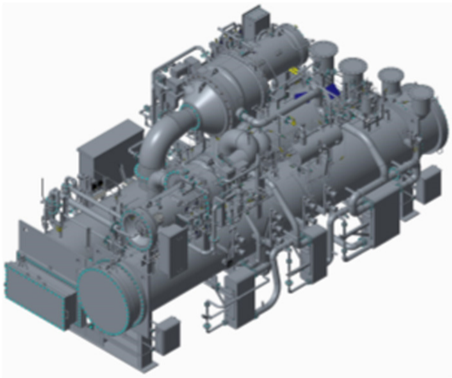
Development of Large Capacity Compound Type Centrifugal Chiller and Centrifugal Heat Pump Using R1234ze(E)



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Conventionally, heat source equipment for low-temperature processes in the industrial field has used propane (R290) and ammonia (R717, NH3), but they place a heavy burden on customers for safe operation due to their flammability and toxicity. Therefore, alternative products with low environmental impact are being demanded. Currently, the move toward terminating the use of HFCs is being accelerated with the agreement of countries around the world, such as the adoption of the Kigali Amendment to the Montreal Protocol. In the air conditioning field, R1234ze(E), which has low toxicity and a GWP (Global Warming Potential) of 1 or less, has been used for centrifugal chillers used as heat source equipment for large-scale facilities. To expand the application range of low-GWP refrigerants in the industrial field, we have established a technology for the multi-stage compression refrigeration cycle using R1234ze(E) that enables the supply of -20°C or lower-temperature brine and 60°C hot water. This technology can also be used for heating tower applications.

1. Introduction

Heat source equipment in the air conditioning field has mainly used HFCs, while natural refrigerants have mainly been adopted in the industrial field where environmental protection is emphasized. Heat source equipment for low-temperature processes has used R290, R717, etc. R290 has flammability and R717 is toxic and slightly flammable, so they place a heavy burden on customers in terms of safety in operation.

On the other hand, at the 28th Meeting of the Parties to the Montreal Protocol (MOP28) held in October 2016, the Kigali Amendment to the Protocol was adopted to prescribe the obligatory gradual reduction of HFC (hydrofluorocarbon) production and consumption, etc., thus clearly setting the course toward terminating the use of HFCs. To prevent global warming, refrigerants for heat source equipment in the air conditioning field are rapidly being switched to low-GWP refrigerants. Centrifugal chillers are cooling and heating sources that are applied to district heating and cooling, air conditioning for buildings and factories and process cooling for chemical and food factories. Centrifugal chillers produce chilled water using a centrifugal compressor and have a cooling capacity of 150 USRt (US refrigeration tons) to 5000 USRt. Currently, centrifugal chillers in the air conditioning field use R1234ze(E), which has a low toxicity and a GWP (Global Warming Potential) of 1 or less. To expand the application range of low-GWP refrigerants in the industrial field, we have established a technology for the multistage compression refrigeration cycle using R1234ze(E) that enables the supply of -20°C or lower-temperature brine and 60°C hot water. This technology can also be used for heating tower applications.

This paper introduces the equipment that has started operation as a heat source for low temperature processes.

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2. Specifications of developed equipment

2.1 Refrigerant

As a refrigerant for the developed equipment, we selected one that had a theoretical COP equivalent to that of the conventional refrigerant and did not significantly increase the equipment design pressure in the supply range from brine of -20°C or lower to hot water of 60°C. In addition, the selection criteria included that the refrigerant to be used was not a flammable gas or toxic gas in the refrigeration safety regulations of the High Pressure Gas Safety Act in terms of manageability. The criteria also included that the refrigerant to be used had applications other than in centrifugal chillers and a reasonable amount of production was expected in terms of availability. As a result of comparing the physical properties of the candidate refrigerants shown in **Table 1**, we selected R1234ze(E), which had already been adopted for heat source equipment in the air conditioning field and as a magnesium masking gas.

Table 1 Comparison of refrigerants

Refrigerant	R290 propane	R717 ammonia	R134a	R1234ze (E)	R1234yf	R1233zd (E)
GWP	3	0	1300	<1	<1	1
Handling in refrigeration safety regulations of High Pressure Gas Safety Act	Flammable gas	Flammable gas Toxic gas	Inert gas	Specified inert gas	Specified inert gas	Inert gas*1
Classification in ASHRAE Standards 34	A3	B2L	A1	A2L	A2L	A1
Saturation pressure (-25°C) [MPa.abs]	0.203	0.151	0.106	0.077	0.123	0.014
Saturated gas density (-25°C) [kg/m^3]	4.63	1.30	5.51	4.43	7.17	0.88
Saturated liquid density (-25°C) [kg/m^3]	560.6	671.5	1373.4	1309.9	1250.5	1376.6
Saturation density (60°C) [MPa.abs]	2.117	2.616	1.682	1.277	1.642	0.391
Theoretical COP*2 [-]	2.68	2.73	2.69	2.67	2.60	2.77

*1: When R1233zd (E) is used at 0.2MPaG or less, the High Pressure Gas Safety Act does not apply.
*2: Two-stage expansion cycle, ET: -25°C, CT: 37°C, Eff.: 80%

2.2 Refrigeration cycle

In the case of equipment in the industrial field, in comparison with that in the air conditioning field, there is an issue that the difference in fluid temperature between the evaporator side and the condenser side is significant and the pressure ratio (the ratio of the condenser pressure to the evaporator pressure) is high, thereby reducing the efficiency of the compressor and limiting the operation range. **Figure 1** shows typical fluid temperatures on the evaporator side and the condenser side in the air conditioning field and the industrial field, as well as the pressure ratio thereof. In the figure, the upper end of the bar indicates the fluid temperature on the condenser side and the lower end the evaporator side.

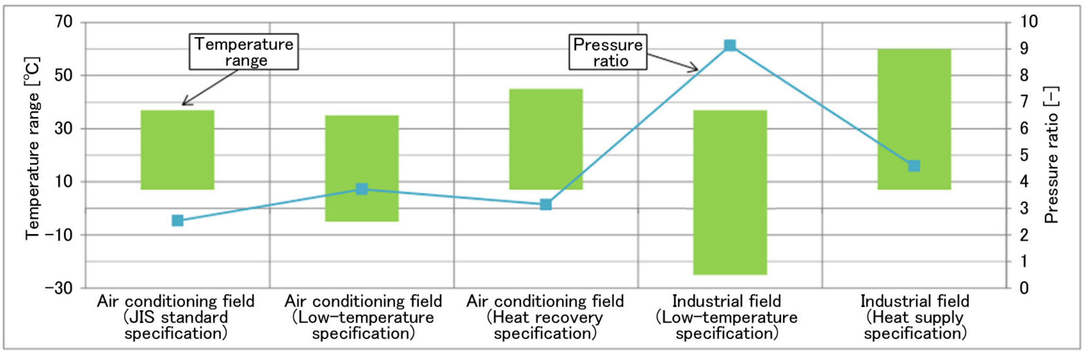

Figure 1 Fluid temperature range and pressure ratio in air conditioning and industrial fields
* The pressure ratio was converted using the R1234ze(E) saturation pressure at the fluid temperature on the condenser and evaporator sides.

To deal with a high pressure ratio, we developed a multistage compression cycle in which two compressors are connected in series, based on a two-stage compression compressor structure that had been proven in centrifugal chillers in the air conditioning field. In the case of a multistage compression cycle, the air volume of the low-pressure side compressor is higher due to its larger suction gas specific volume compared with the high-pressure side compressor, so we adopted a large-diameter impeller, considering the operable range. We examined the compressor's operability in a high-efficiency area within the actual operating temperature and capacity ranges and its controllability to avoid rotating stalls and choke areas. In addition, we adopted a 4-stage compression 1-stage expansion economizer cycle to downsize the unit and the refrigerant charge amount. **Figure 2** is a schematic diagram of the refrigeration cycle.

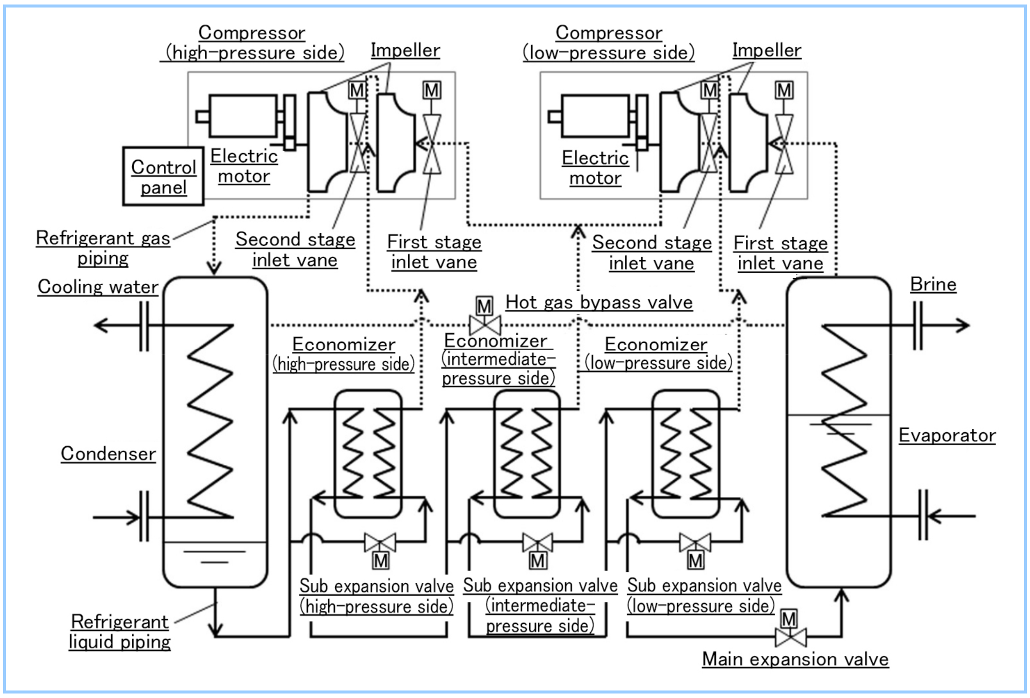

Figure 2 Referigeration cycle

2.3 External view and specifications of developed chiller

The low-pressure side compressor was installed on the evaporator and the high-pressure side on the condenser so that large-diameter gas piping takes the shortest paths with a smaller pressure loss. The economizer and other containers were placed on the condenser side as much as possible to insulate them from the low-temperature evaporator and minimize the cold insulation area. The adopted R1234ze(E) falls under a specified inert gas under the refrigeration safety regulations of the High Pressure Gas Safety Act, so we installed a designated refrigerant detector in the chiller body.

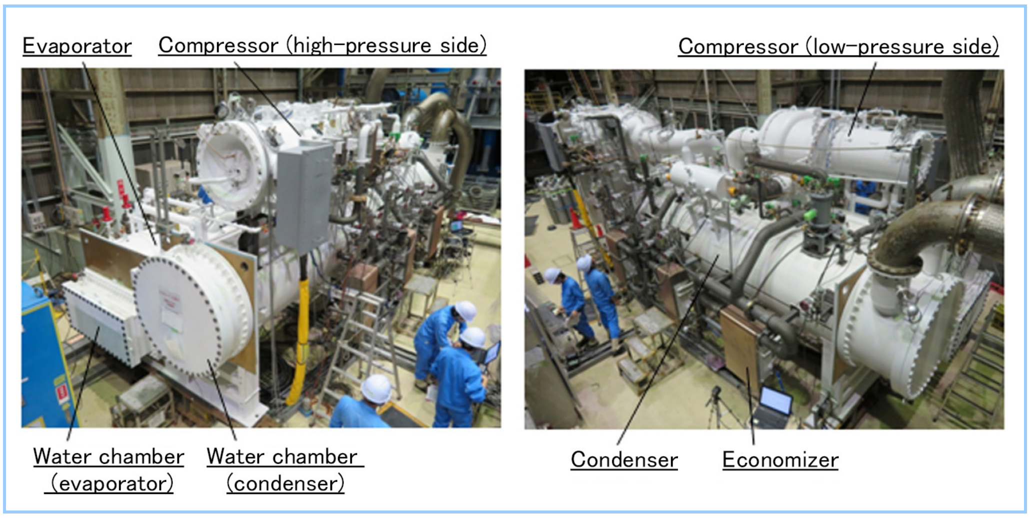

Figure 3 External view (taken at time of factory trial operation)

The developed machine was based on centrifugal compressors and heat exchangers of centrifugal chillers that had been proven in the air conditioning field. In addition, to deal with issues in the temperature range below -20°C including the low-temperature brittleness of metal materials and the hardening of resin materials, ensuring compatibility between the lubricating oil and the refrigerant and the increase in the specific volume of the refrigerant gas, we carefully carried out material selection, structural design and control design through elastomer tests and evaluation tests.

Figure 3 and **Table 2** show the external view of the developed machine (taken at the time of factory trial operation) and the main specifications, respectively.

Table 2 Main specifications

Item	Unit	Specification
Refrigerant	-	R1234ze(E)
Refrigeration oil	-	POE
Refrigeration cycle	-	4-stage compression 1-stage expansion economizer cycle (Figure 2)
External dimensions (L x W x H)	m	7.7×4.2×3.4
Rated capacity	USRt	768.5
Brine temperature	°C	-16 (inlet)/-21 (outlet), common for specification 1 and specification 2
Cooling water temperature*1	°C	Specification 1: 32 (inlet)/37 (outlet) Specification 2: 18 (inlet)/23 (outlet)
Compressor	-	Centrifugal type, fixed speed drive, 2 units
Heat exchanger	-	Shell and tube type (evaporator and condenser)
Economizer	-	Plate heat exchanger, 3 units

*1: Cooling water temperature varies depending on the outside air temperature.

2.4 Chiller control

The developed machine uses a multistage compression cycle in which two compressors are connected in series, and it has more control mechanisms than the centrifugal chillers proven in the air conditioning field. To secure the control stability during transients, we optimized the control parameters using dynamic characteristic analysis.

Figure 4 and **Figure 5** give the dynamic characteristic analysis results (typical example) from startup with the brine inlet temperature of -16°C and the cooling water inlet temperature of 32°C to the point where the brine outlet temperature decreased to -21 ° C. We confirmed that the low-pressure side compressor started at the timing of 90 seconds shown in Figure 4, the first-stage inlet vane (IGV) openings of the low-pressure side compressor and the high-pressure side compressor increased and the brine outlet temperature decreased to the target value of -21°C. The air volume and the pressure ratio of both compressors increased after the start of the compressors and did not enter unstable areas such as rotating stalls.

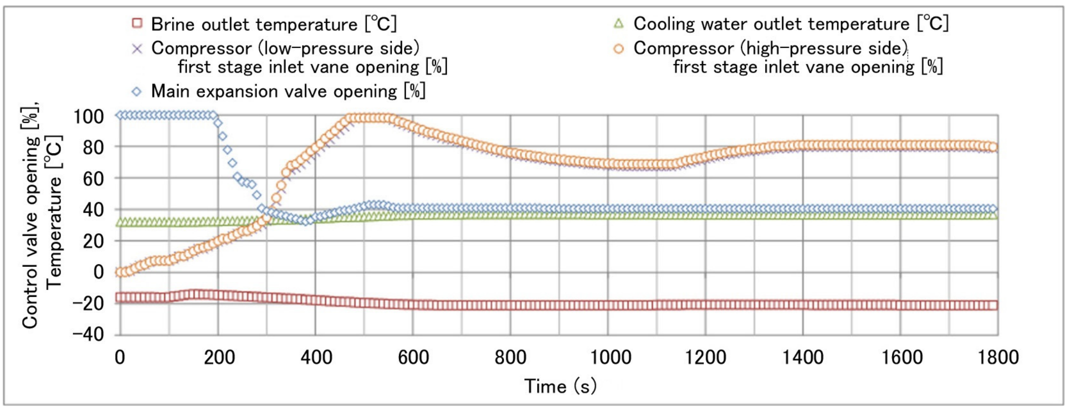

Figure 4 Dynamic characteristic analysis result 1 (control valve opening, temperature)

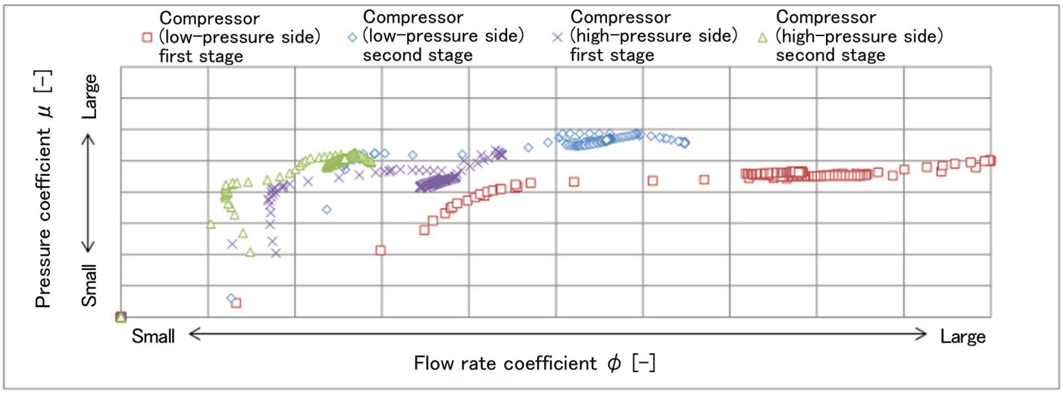

Figure 5 Dynamic characteristic analysis result 2 (flow rate coefficient, pressure coefficient)

The compressor air volume (flow rate coefficient) in Figure 5 was calculated using equation (1), and the pressure ratio (pressure coefficient) equation (2). Q_{st} denotes the compressor air volume, D the impeller diameter, u the impeller peripheral speed and Had the heat insulation head.

$$\phi = \frac{Q_{st}}{\frac{\pi}{4}D^2 u} \qquad \text{Equation (1)}$$

$$\mu_{ad} = \frac{gH_{ad}}{u^2} \qquad \text{Equation (2)}$$

3. Evaluation of operation data

Figure 6 presents the operation data of the developed machine that started operation as heat source equipment for low-temperature processes, from startup (10:10) to shut down (18:30) on a certain day. **Figure 7** plots the data of COP with respect to the load factor for several cooling water inlet temperature ranges in winter.

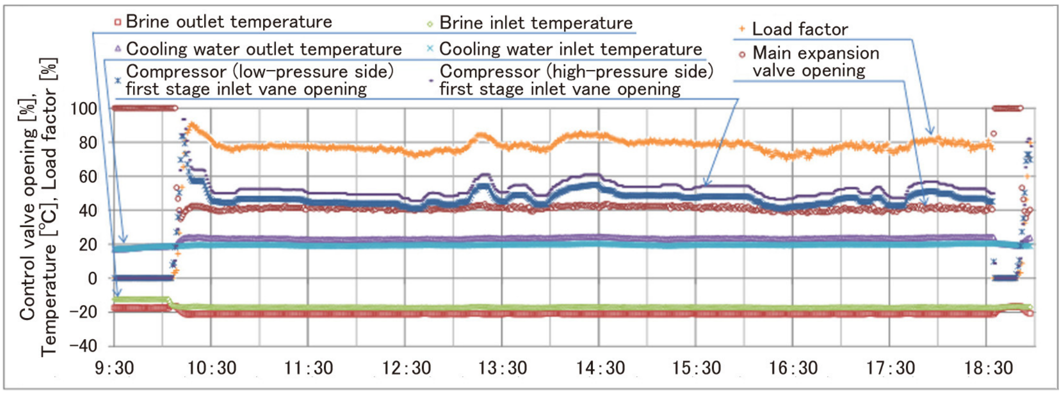

Figure 6 Operation data from startup to shut down

The developed machine was operated as heat source equipment for processes according to the facility load. The load factor (cooling capacity) was stable at around 80% of the rated capacity and the brine outlet temperature was constant at the target value (-21°C). We confirmed that the first stage inlet vane (IGV) and the main expansion valve followed the cooling capacity, brine temperature and cooling water temperature, and that the developed machine could work without carry-over to the compressor and excessive fluctuations in the evaporator pressure.

The heat exchanger performance of the developed machine and the pressure loss at each part satisfied the planned values and we achieved the planned COP (Table 2, Specifications 2). We confirmed that when the cooling water inlet temperature decreased, the COP increased, and that even when the cooling water inlet temperature decreased to 12°C, the developed machine could maintain the brine outlet temperature stably at the target value (-21°C).

The COP is calculated by equation (3). Where Q_{brine} denotes the cooling capacity (heat exchange amount of the evaporator) and $W_{chiller\ elec\text{-}in}$ the chiller power consumption (sum of high-pressure side compressor power consumption $W_{high\ pressure\ side\ comp\ elec\text{-}in}$ and low-pressure side compressor power consumption $W_{low\ pressure\ side\ comp\ elec\text{-}in}$).

$$COP = Q_{brine} / W_{chiller\ elec\text{-}in} \quad \text{Equation (3)}$$

$$W_{chiller\ elec\text{-}in} = W_{high\ pressure\ side\ comp\ elec\text{-}in} + W_{low\ pressure\ side\ comp\ elec\text{-}in} \quad \text{Equation (4)}$$

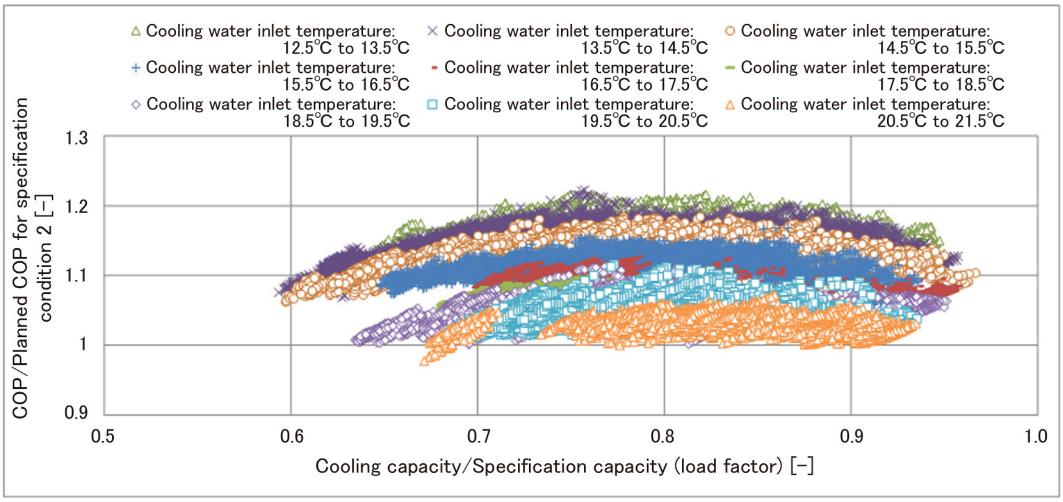

Figure 7 Load factor and COP in winter
*1: The brine outlet temperature is constant at -21°C.
*2: The vertical axis represents the ratio between the measured COP and the rated COP under specification condition 2.

4. Conclusion

We adopted R1234ze(E) with a GWP of 1 or less and the multi-stage compression refrigeration cycle and established the technology for heat source equipment for processes in the industrial field. The developed machine, which started operation as heat source equipment for low-temperature processes, maintained the brine outlet temperature stably at the target value (-21°C) according to the equipment load and achieved the planned COP. In addition, we demonstrated the COP increased when the cooling water inlet temperature decreased, and that the chiller control followed the cooling water inlet temperature stably down to 12°C.

This development enables the expansion of the application range of low-GWP refrigerants in the industrial field. Going forward, we will continue to make efforts to promote low-GWP refrigerants and develop chillers continuously to satisfy our customers.

References

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(2) ASHRAE, 2015 Addenda Supplement to ANSI/ASHRAE Standard 34-2013, Designation and Safety Classification of Refrigerants, 2015.
(3) NIST REFPROP Ver.9.1.1