Performance of centrifugal chiller and development of heat pump using a low-GWP refrigerant

Ryosuke Suemitsu, Mitsubishi Heavy Industries Thermal Systems, Japan

The manufacture of HFCs has been restricted by increasingly stringent regulations, with the aim of reducing global warming. Low-GWP refrigerants are needed for air-conditioning and refrigeration equipment. We have developed and commercialized centrifugal chillers using the low-GWP refrigerant R-1234ze(E) for capacities from 1055 to 17581 kW, and using R-1233zd(E) for capacities from 527 to 2461 kW. We have also developed a centrifugal heat pump system that is capable of using a low-GWP refrigerant to produce pressurized hot water at 200 °C, with a coefficient of performance (COP) of 3.5.

Introduction

In December 2015, the 21st Session of the Conference of the Parties to the United Nations (UN) Framework Convention on Climate Change (COP 21) adopted the Paris Agreement, according to which every country is to update and submit its own reduction target every five years. Furthermore, in October 2016, the Kigali Amendment to the Montreal Protocol was ratified, obliging signatory countries to curb the production of hydrofluorocarbons (HFC) and gradually reduce their consumption. For refrigeration and air-conditioning equipment, Regulation (EU) No 517/2014 on fluorinated greenhouse gases (the F-gas Regulation) in Europe, and the Act for Rationalized Use and Proper Management of CFCs and HFCs in Japan, were designed to reduce the environmental impact of these gases.

HFC refrigerants with a high GWP, such as R-134a, R-245fa, etc., have been replaced by new types of low-GWP refrigerants. We use R-134a as the refrigerant of centrifugal chillers and heat pumps, so there is a need to switch to low GWP refrigerants in air-conditioning and refrigeration equipment. In order to investigate whether the low GWP refrigerants are suitable for the refrigeration and air conditioning equipment, we must verify several aspects such as their physical properties, stability, toxicity, flammability, and availability. The choice of refrigerant depends on the capacity, compressor type, operating temperature, and other conditions of the refrigeration and air-conditioning equipment, and the alternative refrigerant must have the following characteristics:

• Environmental factors: Non-ozone depleting substance, GWP ≤ 100.
• Physical properties: Cycle efficiency equivalent to that of HFC refrigerants. Design pressure should not be excessively high.
• Low toxicity, and no or mild flammability.
• Availability: Low-GWP refrigerants have applications other than refrigeration and air-conditioning equipment, so there must be an adequate level of production and cost-effectiveness .

A range of double-bond olefin refrigerants was considered. Table 1 shows a comparison of HFC and olefin refrigerants. The GWPs of the olefin refrigerants range from 0 to 2, and their cycle efficiencies are equivalent to those of HFC refrigerants.

Design of centrifugal chiller using low GWP refrigerant

Currently, R-134a is used as refrigerant for centrifugal chillers. Refrigerant R-1234ze(E) is mildly flammable, R-1233zd(E) is non-flammable, and both are of low toxicity. R-1234ze(E) and R-1233zd(E) can be used as a foaming agent and an aerosol, so these offer availability and cost-effectiveness, and the physical properties of R-1234ze(E) are similar to those of R-134a. We therefore selected R-1234ze(E) as the refrigerant for the large-capacity systems from 1055 to 17581 kW.

R-1233zd(E) has a better cycle performance than R-134a and other olefins in Table 1, and its physical properties are similar to those of R-245fa. However, the gas specific volume is about five times that of R-134a – requiring increased volume to accommodate the compressor, evaporator, and condenser. A more advanced and compact chiller design would allow its replacement with R-1233zd(E) centrifugal chillers.

Centrifugal compressor

To minimize the compressor volume, the compressor was designed for a large gas flow rate. A large gas flow rate tends to lower the adiabatic efficiency. The optimal shape of the leading/trailing edge of the impeller blade, the blade angle distribution, the shape of the flow path of the impeller inlet portion, and the shape of the inlet guide vane were therefore optimally determined by a Computational Fluid Dynamics (CFD) analysis. As a result, the adiabatic efficiency was equal to or higher than that of R-134a, and the gas flow rate could be increased by about 30 % with the same impeller diameter.

Refrigerant	HFCs		Olefins			
	R-134a	R-245fa	R-1234yf	R-1234ze(E)	R-1233zd(E)	R-1336mzz(Z)
Global warming potential (GWP)*1	1300	858	<1	<1	<1	2
Ozone depleting substance*2	no	no	no	no	no	no
Safety class*3	A1	B1	A2L	A2L	A1	A1
Standard boiling point [°C]*4	-26.1	15.0	-29.5	-19.0	18.3	33.5
Critical point [°C]	101.1	153.9	94.7	109.4	166.5	171.4
Saturated pressure (@6 °C)*4[kPa(G)]	260.7	-314.8	283.9	167.3	-39.1	-68.5
Saturated pressure (@38 °C)*4[kPa(G)]	861.9	133.2	866.4	624.3	100.8	18.0
Saturated vapor specific volume (@6 °C) [m^3/kg]*4	0.0564	0.2409	0.0467	0.0694	0.2770	0.4205
Saturated vapor specific volume (@70 °C) [m^3/kg]*4	0.0087	0.0298	0.0076	0.0109	0.0370	0.0469
Theoretical COP*4*5	7.23	7.14	7.11	7.26	7.47	7.44

Table 1: Comparison of HFCs and olefins
**1: IPCC Fifth Assessment Report *2: Montreal Protocol *3: Refrigerant Safety Classification Standard ASHRAE 34*
**4: Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) Version 10.0 *5: Refrigeration cycle efficiency with single-stage cycle, evaporation temperature 6 °C, condensation temperature 38 °C, supercooling 4 °C, adiabatic efficiency 90 %.*

Evaporator and condenser
We use shell-and-tube type heat exchangers, specifically the flooded type, in our evaporators. As the refrigerant gas flow rate increases, the pressure drop, performance deterioration due to dry-out in the evaporator and carry-over to the compressor should be carefully considered. We therefore conducted thermo-fluid analysis to optimize the arrangement of tube bundles, the direction of water flow, and the shell dimension. By comparing the thermo-fluid analysis at the early design stage with the verification evaluation on the prototype machine, we optimized the arrangement of tube bundles, the direction of water flow, and the shell dimension, and determined a compact shape and their combinations. As a result, the volume of the evaporator and condenser is reduced to 120 % compared with the R-134a heat exchanger, and the outside heat-transfer coefficient of the evaporator and condenser registers no more than a 10 % decrease in the rated condition (703 kW system).

Verification of centrifugal chiller
A centrifugal chiller with the component design described in the previous sections was manufactured, and a verification test was carried out. Table 2 shows the rated COPs and the installation area designed based on the verification experiment results, comparing the developed equipment and the conventional R-134a equipment. Under the rated condition (703 kW system), performance improved by about 3 % compared with the conventional system. Moreover, the refrigerant gas specific volume is about five times that of R-134a, but the installation area was possible to be kept within about 105 % of the conventional throughout the range from 527 to 2461 kW.

Comparison of rated COP and installation area

Chilled water capacity [kW]		527–879	879–1231	1407–1758	1758–2461
Chilled water temperature [°C]		12.0 → 7.0			
Cooling water temperature [°C]		32.0 → 37.0			
Starting method [-]		Inverter			
Rated capacity [kW]		703	1055	1407	2110
Rated COP [-]	Conventional	6.1	6.2	6.1	6.2
	Developed	6.3	6.3	6.3	6.2
	Developed/Conventional	103%	101%	102%	100%
Installation area [m^2]	Conventional	5.55	6.30	8.36	8.82
	Developed	5.83	6.61	8.48	9.10
	Developed/Conventional	105.0%	104.9%	101.4%	103.2%

Table 2.

Design of heat pump using low-GWP refrigerant

Heat pumps are increasingly being used to supply hot water and heating in the household sector, but awareness in the industrial sector has been slower. The demand for high-temperature heat and reuse of waste heat is comparatively strong in mechanical and chemical industries, and the adoption of exhaust heat recovery heat pumps offers great potential for improved energy efficiency and a reduction in CO_2 emissions. High temperature heat pump systems capable of producing pressurized hot water at 200 °C with COPs of 3.5 or more are part of the research and development program of Japan's New Energy and Industrial Technology Development Organization (NEDO). These naturally require low-GWP refrigerants capable of operating at high temperatures.

Refrigerant and lubricant oil

When selecting the refrigerant and lubricant oil, the operating temperature range of the heat pump must be considered. This requires the following physical properties:

• Stability at high temperature: Prevention of isomerisation and decomposition at the operating temperature of the heat pump.
• Standard boiling point: The compressor volume should not be too small for the adiabatic efficiency, and the design pressure should not be too high.
• Critical point: The critical temperature should be higher than the operating temperature to improve the efficiency of the cycle.

The lubricant oil must maintain its stability at high temperatures and have the required temperature-dependent solubility in the refrigerant, viscosity, and other factors.

As a first step, we investigated an exhaust heat recovery heat pump which can heat pressurized water to 160 °C using a low-GWP refrigerant. The vapor specific volume of R-1234ze(E) is small and the stability of R-1233zd(E) is poor at 175 °C. Refrigerant R-1336mzz(Z) has a larger vapor specific volume than other olefins in Table 1 and good stability at up to 250 °C, so R-1336mzz(Z) was selected for the 160 °C application.

The lubricant oil was also selected as for the heat pump using R-1336mzz(Z). Synthetic oil is commonly used with non-chlorine refrigerants such as HFCs. The polyol ester (POE) oil was therefore selected because of the stability of R-1336mzz(Z). Table 3 shows the results of accelerated thermal stability testing for the 160 °C application, demonstrating stability at 220 °C. We estimated the replacement period of the lubricant oil, but this is not long enough for 200 °C applications. Moreover, the kinetic viscosity and solubility are 175 % and 167 % compared with the requirements for bearings in applications of up to 160 °C.

We intend to develop heat pumps targeting an outlet temperature of 200 °C. R-1336mzz(Z) does not have a large enough vapor specific volume at that temperature to use with the centrifugal compressor. For higher temperatures, we will take appropriate refrigerants and lubricant oils from among the refrigerant candidates, evaluate their stability, safety and physical properties in parallel, and select the best refrigerants.

Equipment design

A two-stage compression economizer cycle is used to produce water at 90 °C in existing heat pump designs. However, a high temperature heat pump using the two-stage economizer cycle cannot achieve the target COP of 3.5. A two-stage compression bleed cycle shown in Fig. 1 was adopted to achieve the target COP. The bleed cycle is highly efficient as it uses some of the refrigerant gas discharged from the low-stage compressor for intermediate heating. The compressors were designed for a high head and large volume flow rate, to reduce the number and volume of the compressors. CFD analysis was used to optimize the

Accelerated thermal stability testing for 160 °C applications

Test condition				**Result**
R-1336mzz(Z):Oil	Temperature [°C]	Duration [h]	Air/Moisture [ppm]	Acid Value [g KOH/kg]
50:50	200	168	0 / 0	0.01
50:50	200	168	100 / 1000	0.01
50:50	220	168	0 / 0	0.01
50:50	220	168	100 / 1000	0.01
50:50	220	336	0 / 0	0.25
50:50	220	336	100 / 1000	0.23
50:50	220	672	0 / 0	0.55
50:50	220	672	100 / 1000	0.45
50:50	250	168	0 / 0	1.37
50:50	250	168	100 / 1000	2.25

Table 3.

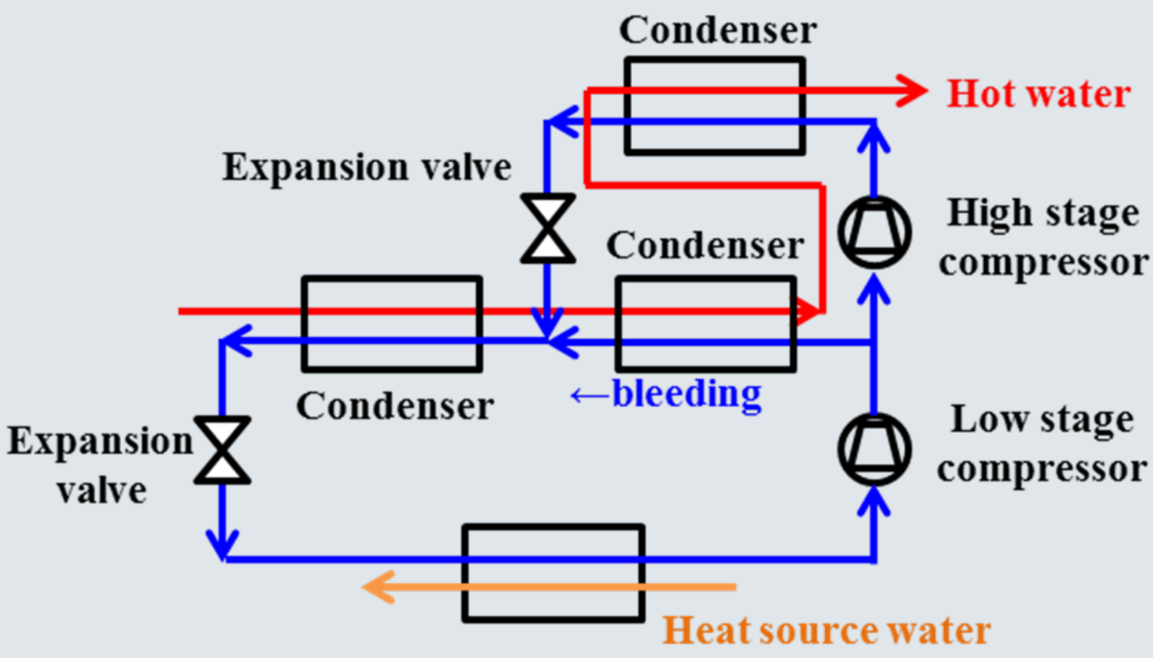

Fig. 1: Two-stage compression bleed cycle

leading-edge position of the splitter and the blade angle distribution. The flow rate is 39 % greater, and the adiabatic efficiency is improved by 3.5 %, for the same impeller diameter.

Conclusions

It is essential that low-GWP refrigerants are developed for use with cooling and heating equipment.

Centrifugal chillers using low-GWP refrigerants have been developed. Refrigerant R-1234ze(E) can be used from 1055 to 17581 kW, and refrigerant R-1233zd(E) from 527 to 2461 kW.

Centrifugal heat pumps using low-GWP refrigerants are in development. R-1336mzz(Z) was selected for a heat pump capable of heating pressurized water to 160 °C. Preparations are in progress to manufacture this and to develop the 160 °C application further, with a final goal of operating at 200 °C.

Acknowledgements

This work was supported by the New Energy and Industrial Technology Development Organization (NEDO) in Japan through its research and development project for innovative practical-use technologies for unused heat sources.

References

[1] Suemitsu, R. et al. Development of centrifugal chiller and heat pump using low-GWP refrigerant. 12th IEA Heat Pump Conference, 2017.

[2] Miyoshi, N. et al. Performance characteristics of centrifugal chiller using HFO-1233zd(E). 52nd Japanese Joint Conference on Air-conditioning and Refrigeration, 2018.

[3] Kontomaris, K. HFO-1336mzz-Z: High Temperature Chemical Stability and Use as A Working Fluid in Organic Rankine Cycles. 2014 Purdue Conferences, 2014.

RYOSUKE SUEMITSU

Mitsubushi Heavy Industries Thermal Systems

Japan

ryosuke_suemitsu@mth.mhi.co.jp

https://doi.org/10.23697/2asr-yp72