12th IEA Heat Pump Conference 2017

Development of centrifugal chiller and heat pump using low GWP refrigerant

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Abstract

The manufacture of HFCs has been restricted by increasingly stringent regulations, with the aim of reducing global warming. There is a need for low GWP refrigerants that can be used in air-conditioning and refrigeration equipment. The refrigerant R-1233zd (E) has low toxicity, non-flammable, and a GWP of approximately 1, and these characteristics make it suitable for use in centrifugal chillers and heat pumps. We have developed a centrifugal chiller using R-1233zd (E). It is required that the compressor, heat exchanger, and other components are optimized, and the size of the chiller is minimized, but the specific gas volume of R-1233zd (E) is much larger than that of R-134a. We have designed a chiller whose size was almost the same as that of an R-134a chiller of between 150 and 700 tons. A COP 2% higher than that of the R-134a chiller was achieved. We also have been developing a centrifugal heat pump system using a low GWP refrigerant to heat pressurized hot water to 200°C, with a COP of 3.5. The most important criteria for refrigerant selection are low GWP, toxicity, flammability, and stability at high temperature. We will distribute a practical high temperature heat pump by 2023.

Keywords: Low GWP refrigerant; R-1233zd (E); Centrifugal chiller; Centrifugal heat pump

1. Introduction

In December 2015, the 21st Session of the Conference of the Parties to the United Nations (UN) Framework Convention on Climate Change (COP21) adopted the Paris Agreement, requiring each country to undertake long term action on worldwide decarbonization. For refrigeration and air conditioning equipment, the fluorinated greenhouse gases (F-gases) regulations in Europe, and the Act for Rationalized Use and Proper Management of CFCs and HFCs in Japan, were designed to reduce the environmental impacts of these gases.

HFC refrigerants with high global warming potential (GWP), such as R245fa and R134a, have been replaced by new types of low GWP refrigerant. Several aspects of these low GWP refrigerants must be verified, including their physical properties, stability, toxicity, flammability, and cost effectiveness. The choice of refrigerant depends on the capacity, compressor type, operating temperature, and conditions of the refrigeration and air conditioning equipment.

We have developed and manufactured a centrifugal chiller using the low GWP refrigerant R-1233zd(E), which is suitable for large capacity air conditioning systems. We also have been developing a heat pump system heating high temperature water.

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1.1. Selection of alternative refrigerant

The alternative refrigerant requires the following items to be used in the centrifugal chiller:

- Environmental factors: ODP ≤ 0.001, GWP ≤ 150, long term toxic exposure ≥ 800 ppm.
- Physical properties: Cycle efficiency equivalent to that of existing refrigerants. Design pressure not to be excessively high.
- Low toxicity and low flammability.
- Cost effectiveness.

A range of double bond olefin refrigerants were considered. Table 1 shows a comparison of HFC and olefin refrigerants for chiller applications. The GWPs of the olefin refrigerants range from 0 to 1 (very low), and their cycle efficiencies are equivalent to those of the existing refrigerants.

Table 1. Comparison of refrigerants for chiller applications

Refrigerant	HFCs			Olefins		
	R-245fa	R-134a	R-32	R-1234yf	R-1234ze(E)	R-1233zd(E)
Global Warming Potential (GWP)*1	858	1300	677	<1	<1	1
Ozone Depletion Potential (ODP)	0	0	0	0	0	0
Flammability	non	non	lower	lower	lower	Non
Toxicity	toxicity	low	low	low	low	low
Safety class*2	B1	A1	A2L	A2L	A2L	A1
Atmospheric lifetime	7.6 years	13.8 years	5.2 years	10.5 days	16.4 days	26 days
Allowable concentration [ppm]	300	1000	1000	500	1000	800
Standard boiling point [°C]	15.1	−26.1	−51.7	−29.4	−19.0	18.3
Saturated pressure (@6°C)*3[kPaG]	−32.1	260.7	879.8	283.9	167.3	−39.1
Saturated pressure (@38°C)*3[kPaG]	133.1	861.9	2258	866.4	624.3	100.8
Saturated liquid density (@6°C)*3[kg/m^3]	1389	1275	1034	1157	1222	1308
Saturated liquid density (@38°C)*3[kg/m^3]	1302	1155	902.8	1042	1119	1231
Saturated vapor specific volume (@6°C)*3[m^3/kg]	0.241	0.056	0.037	0.047	0.069	0.277
Saturated vapor specific volume (@38°C)*3[m^3/kg]	0.075	0.021	0.014	0.018	0.026	0.091
Theoretical COP*4	6.86	6.58	6.38	6.31	6.56	6.93

*1: 5th IPCC *2: ASHRAE34 *3: RefProp Ver9.1

*4: Single stage cycle; evaporating temperature of 6°C, condensing temperature of 38°C, compressor efficiency of 90%

R-1233zd(E) is non-flammable and of low toxicity. It can be used as a forming agent, and it offers availability, and cost effectiveness. Its physical properties are similar to R-245fa.

However, the gas specific volume is about five times that of R-134a, requiring increased volume to accommodate the compressor, evaporator, condenser, and gas piping. Currently, centrifugal chillers use R-134a widely as refrigerants. A more advanced and compact chiller design would allow its replacement with centrifugal chillers of R-1233zd(E).

2. Design of centrifugal chiller using R-1233zd(E)

We have improved the efficiency and reduced the size.

2.1. Compressor and motor

- Improved aerodynamic shape of compressor

The compressor was designed for the large gas flow rate. Since a large gas flow rate tends to lower the adiabatic efficiency, computational fluid dynamics (CFD) analysis was performed to optimize the leading/trailing edges of the impeller, the blade angle distribution, the path form of the impeller inlet, and the form of the inlet guide vane.

Compared with systems designed for R-134a, the gas flow rate at the same impeller diameter was increased by approximately 30% and the adiabatic efficiency was improved by 3%. The volume of compressor unit was increased by approximately 40%.

- Direct-connected motor

For R-134a, the impellers are rotated by a motor via a step-up gear, whereas in our design for R-1233zd(E), the impellers were directly mounted on the motor shaft. Since the vapor sound speed of R-1233zd(E) is lower than R-134a, the impeller diameter and the circumferential velocity is larger and the rotation speed is lower than R-134a, even if the capacity is same. The direct connected motor allows compressor unit with motor to be made more compact, and improves the performance by reducing loss as a result of eliminating the step-up gear and minimizing the number of compressor bearings.

2.2. Evaporator and condenser

A shell and tube heat exchanger was adopted, and we adopted flooded type as an evaporator. Since the specific volume is larger and the differential pressure between the condenser and evaporator smaller than R-134a, the pressure drops should be cared, including (i) low temperature loss caused by refrigerant depth interrupting the evaporation, (ii) dry-out and carry-over from the liquid surface in the evaporator caused by the large gas flow, and (iii) pressure loss caused by the large refrigerant gas flow entering into the condenser. To suppress these losses, at an early stage in of the design process, we analyzed the actual chiller and measured the verification test. As a result, we achieved a smaller size and a higher performance by reducing the gas velocity in the tube bundle and arranging the tubes to control the direction of water flow.

Figures 1, 2, and 3 show examples of the analysis. In the evaporator, local dry-out and carry-over were eliminated by averaging the void fraction distribution and gas velocity distribution in the tube bundle. The equivalent temperature loss due to liquid depth was suppressed by increasing the average void fraction in the tube bundle, and improved the heat exchanger. In the condenser, space was added in the tube bundle for removal of the refrigerant gas. This improved the efficiency of heat exchange with the condenser tubes. Since the evaporator pressure is below atmospheric pressure, the incoming air needed to be considered. The low velocity area was determined and a bleed pipe installed to remove these gases.

Figures 4 and 5 show the performance of the heat exchanger. Compared with R-134a, the volume of the evaporator and the condenser was increased by up to 20%. The performance under the rated conditions was 10% lower for the evaporator and 20% lower for the condenser, but the improvement in the theoretical COP and the aerodynamic shape compensated for this.

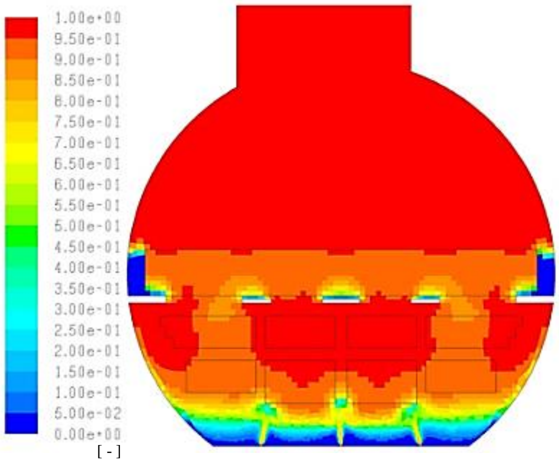


Fig. 1. Void fraction distribution of evaporator

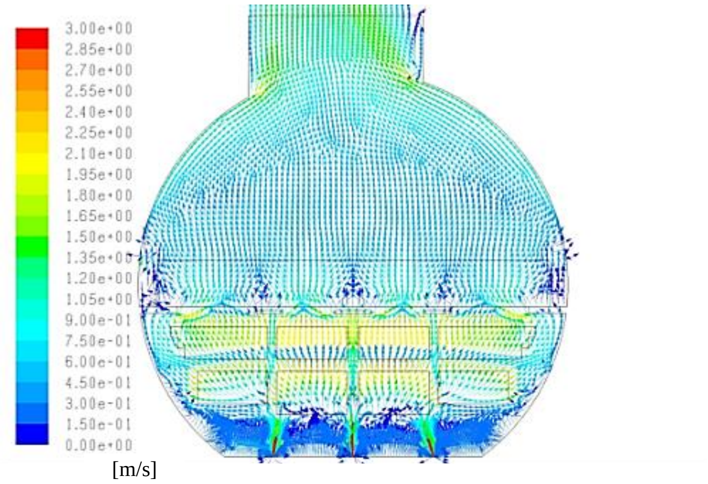


Fig. 2. Gas flow velocity distribution of evaporator

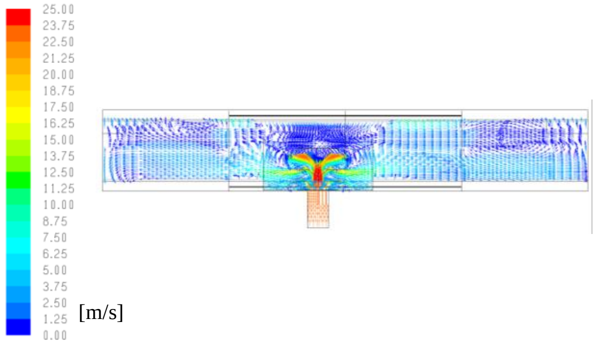


Fig. 3. Gas flow velocity distribution of condenser

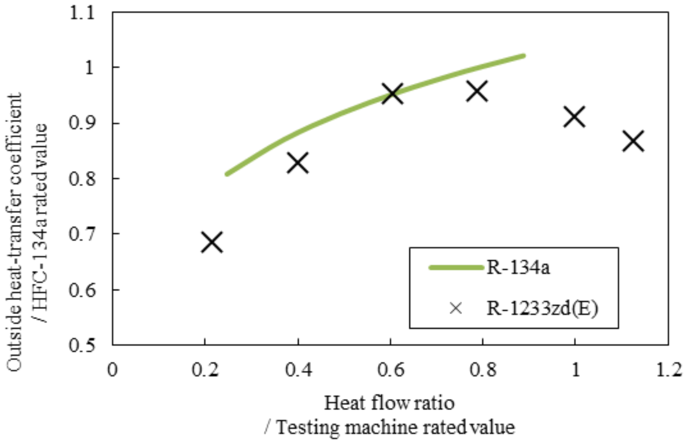


Fig. 4. Overall outside heat transfer coefficient of evaporator

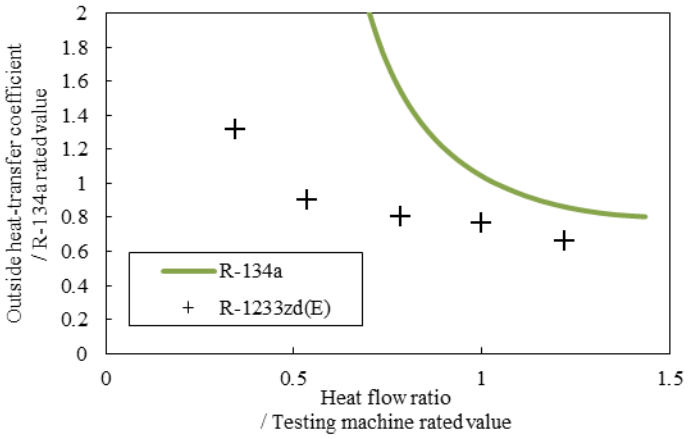


Fig. 5. Overall outside heat transfer coefficient of condenser

2.3. Subcooler

In the subcooler, R-134a equipment uses a brazing type heat exchanger, but a shell and tube type was adopted to reduce pressure loss on the refrigerant liquid, as R-1233zd(E) has a smaller differential pressure. The degree of sub cooling was expected to be equal to that of R-134a equipment, and the heat capacity of the subcooler was 20 kW, for a 200 tons chiller. Space was saved by laying out the tubes in a bottom of the condenser.

2.4. Economizer

In the economizer, a brazing type heat exchanger is used in R-134a equipment with a two-stage compression and one-stage expansion refrigeration cycle, but a flash tank heat exchanger and a refrigeration cycle with two-stage compression and two-stage expansion was adopted, again to reduce pressure loss. A larger volume than that of the R-134a chiller is necessary for vapor-liquid separation in a self-inflation heat exchanger. However, the low design pressure allows the form of the tank to be changed freely. Space could therefore be saved by using a shared wall with the condenser shell.

2.5. Oil tank

The reducing pressure of the oil tank, particularly at start-up, requires the gas phase area in the oil tank to have a larger volume, because the volume of gas emitted by the lubricant oil in the tank is greater than that of the R-134a equipment. This allowed space to be saved and the necessary capacity secured by optimizing the ratio of the oil and open space, and by combining the oil tank with the condenser.

2.6. Pipe joint

The saturated vapor temperature of R-1233zd(E) is 18.3°C. The evaporator of a chiller operated under standard air conditions has negative pressure, encouraging the ingress of ambient air. This risk of air ingress was reduced by minimizing the number of joints.

3. Verification of centrifugal chiller

An inverter drive centrifugal chiller was manufactured, based on the principles discussed above, and its performance was tested. Table 2 shows the specifications of the centrifugal chiller and Fig. 6 shows the test equipment.

Table 2. Centrifugal chiller specifications

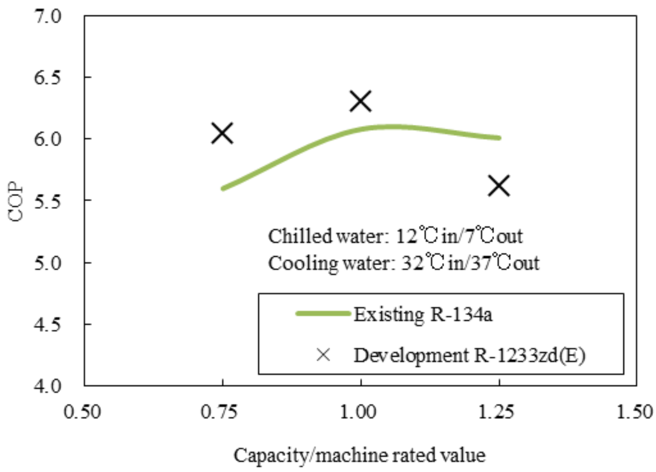
Rated capacity	200 USRt (703 kW)

Chilled water temperature	12.0°C →.0°C
Chilled water flow rate	120.7 m3/h
Cooling water temperature	32.0°C → 37.0°C
Cooling water flow rate	139.6 m3/h
power-supply voltage	400V
Starting method	Inverter

Figure 7 shows the performance results under the conditions presented in Table 2 and disaggregated by capacity. Under the rated capacity conditions from Table 2, the COP was found to be 6.3 and the performance was 3% better than that of the existing R-134a type of the same capacity.

Table 3 compares the specifications, and Table 4 the installation area, of the developed equipment and conventional R-134a equipment. The specific volume of the refrigerant gas was approximately five times greater than that of the R-134a type, and the installation area approximately 105% that of the existing type for equipment of between 150 and 700 tons.

Fig. 6. Test equipment



＊machine rated value = 200 USRt

Fig. 7. Equipment performance

Table 3. Comparison of centrifugal chiller specifications

Model	Existing	Developed
Rated capacity	200 USRt (703 kW)	
Refrigerant	R-134a	R-1233zd(E)
Chilled water temperature	12.0°C → 7.0°C	
Chilled water flow rate	120.7 m^3/h	
Cooling water temperature	32.0°C → 37.0°C	
Cooling water flow rate	141.5 m^3/h	139.6 m^3/h
Power consumption	115.0 kW	111.3 kW
COP	6.1	6.3
Dimensions L × W × H	3.7 × 1.5 × 1.8 m	3.8 × 1.6 × 1.7 m
Installation area	5.55 m^2	5.83 m^2
Shipping weight	3.9 ton	4.3 ton

Table 4. Comparison of installation area required

Rated capacity [USRt]		250	350	500	700
Installation area	Existing [m^2]	5.55	6.30	8.36	8.82
	Developed [m^2]	5.83	6.61	8.48	9.10
	Developed/Existing	105.0%	104.9%	101.4%	103.2%

4. Development of centrifugal chiller using R-1233zd(E)

The improvement in performance compared with that of the R-134a equipment suggests that this is an effective approach to reducing global warming through improved energy efficiency. The installation area was similar to that of the existing equipment, allowing its substitution. We expect that conversion to these newer refrigerants will be driven both by demand for new installations, and replacement of existing installations.

5. Design of heat pump using low GWP refrigerant

Heat pumps are increasingly being used for supply of hot water and heating in the household sector, but uptake in the industrial sector has been slower. The demand for high temperature heat and reuse of waste heat is comparatively strong in the 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 the New Energy and Industrial Technology Development Organization (NEDO). These naturally require low GWP refrigerants capable of operating at high temperatures.

5.1. Refrigerant and lubricant oil

In the selection of refrigerant and lubricant oil, the operational temperature range of the heat pump must be considered. This requires the following physical properties:

- Stability at high temperature: Prevention of isomerism and decomposition at the operating temperature of the heat pump.
- Critical point: The critical temperature should be higher than the operating temperature to improve the efficiency of the cycle.
- Standard boiling point: At the operating temperature, the specific volume of the refrigerant gas and the design pressure of the compressor must be optimized. To reduce the design pressure at the high temperature, an appropriate standard boiling point is selected.

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

As a first step, we investigated the replacement of R-134a with a low GWP refrigerant in an exhaust heat-recovery heat pump heating water to 90°C. In the experiments, R-1233zd(E) was selected. Table 5 compares the HFC and olefin refrigerants used in heat pumps. We evaluated the stability of R-1233zd(E) at up to 150°C. The physical properties and temperature conditions of this refrigerant can be compared with those of existing refrigerants using a drop-in test.

The lubricant oil was also selected as for the heat pump using R-1233zd(E). The polyol ester (POE) oil used with HFC refrigerants is subject to hydrolysis when used with R-1233zd(E), which includes chlorine. The chlorine also serves as an extreme-pressure additive as a ferric chloride coating is formed. Mineral oil was therefore selected because of the stability of R-1233zd(E). Table 6 shows the results of accelerated thermal stability testing for the 90°C application, demonstrating stability at 200°C, but not enough for the range from 160°C to 200°C applications. We will continue testing long term stability at high temperatures. Moreover, +25% kinetic viscosity and +12% solubility were required for the bearings of heat pumps producing pressurized hot water at 160°C.

Table 5. Comparison of refrigerants for heat pump use

Refrigerant	HFCs		Olefins	
	R-245fa	R-134a	R-1234ze(E)	R-1233zd(E)
Global Warming Potential (GWP)*1	858	1300	<1	1
Standard boiling point [°C]	15.1	−26.1	−19.0	18.3
Critical temperature [°C]	154.0	101.1	109.4	166.5
Saturated pressure (@30°C)*3[MPaA]	0.178	0.770	0.578	0.158
Saturated pressure (@90°C)*3[MPaA]	1.006	3.244	2.476	0.833
Saturated vapor specific volume (@30°C)*3[m^3/kg]	0.0983	0.0266	0.0328	0.1175
Saturated vapor specific volume (@90°C)*3[m^3/kg]	0.0176	0.0046	0.0328	0.0225

Table 6. Thermal stability testing

Test condition				Result
R-1233zd(E)：oil	Temperature [°C]	Duration [h]	air/moisture [ppm]	Acid Value [kOH/g]
50:50	150	168	100/100	<0.01
50:50	150	168	500/100	<0.01
50:50	150	168	1000/1000	0.03
50:50	200	168	500/1000	0.02
50:50	200	336	500/1000	0.01

A heat pump is to be developed that is capable of producing the targeted range from 160°C to 200°C. Since the stability of R-1233zd(E) is poor at 200°C, suitable refrigerants and lubricant oil need to be chosen, taking account of stability, toxicity, flammability, and physical properties at the proposed operating temperatures. We have been selecting a suitable refrigerant from several candidates.

5.1. Equipment design

- Heat pump cycle

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 compression economizer cycle cannot achieve the target COP, 3.5. The two-stage compression bleeding cycle shown by Fig.8 was adopted instead to achieve the target COP. The bleeding cycle is highly efficient as it uses some of the refrigerant gas discharged from the low stage compressor for intermediate heating.

- Aerodynamic shape of compressor

The compressors were designed for a high head and large volume flow rate, to reduce the number and the capacity of the compressors. CFD analysis was used to optimize the leading edge position of the splitter and the blade angle distribution. Figs. 9 and 10 show the results. The compression ratio was larger by approximately 40% compared a compressor of chillers, the flow rate was larger by approximately 39%, and the adiabatic efficiency was improved by approximately 3.5%, for the same impeller diameter.

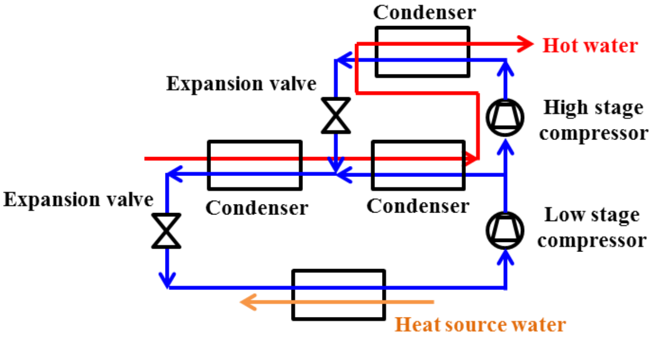


Fig. 8. Two-stage compression bleeding cycle

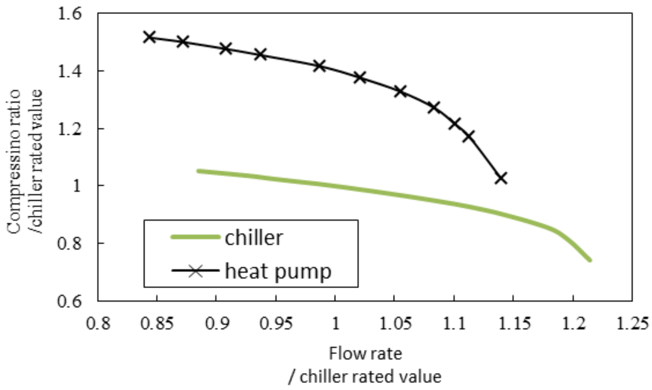


Fig. 9. Compression ratio of compressor

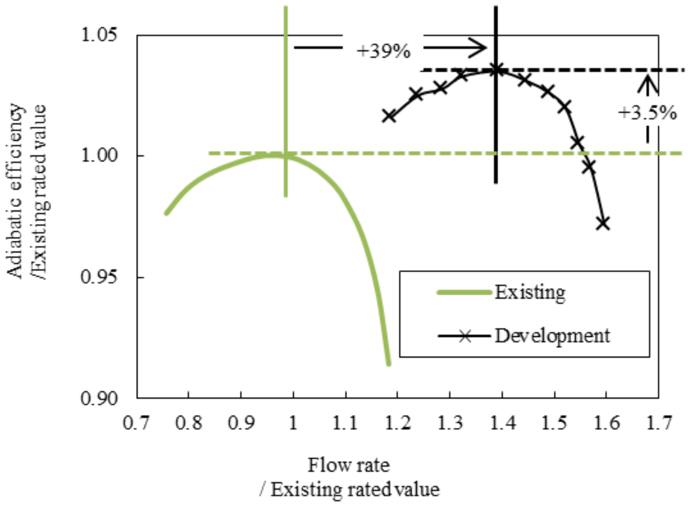


Fig. 10. Adiabatic efficiency of compressor

6. Development of the heat pump using the low GWP refrigerant

The heat pump capable of operating at high temperatures using low GWP refrigerants will be developed by these stages, 90°C, 160°C, 200°C. We focus to introduce a model capable of heating pressurized hot water to

200°C in practical applications by 2023. Heat pumps heating water to 90°C are applied for moderate high temperature, for example, cleaning and food processes. On the other hand, there are thermal demands from 160°C to 200°C in industrial applications, for example, chemical reaction and dried processes, etc. The development of refrigerants and lubricant oil will be conducted in parallel with that of the heat pump, to produce a complete heat system using the proposed heat pumps for those real processes.

7. Conclusions

The development of low GWP refrigerants for use with cooling and heating equipment is essential.

A centrifugal chiller was developed using R-1233zd(E), which has a GWP of 1(equivalent to CO_2). R-1233zd(E) is a suitable refrigerant due to low GWP, nonflammable, low toxicity, cost effective for use in the centrifugal chillers of large heat source equipment. The design achieved an integrated performance better than that of an existing chiller using R-134a. The performance was improved by up to 3%, and the installation area required only an approximately 5% larger than the existing type.

The refrigerant should be selected carefully to allow high temperature exhaust heat recovery.

R-1233zd(E) was selected for heat pumps heating water to 90°C. The design has been complete and we are preparing for drop-in testing. For future work, the developments of heat pumps heating water to 90°C and higher temperatures are going to be carried out, with a final goal of operation at 200°C. We are going to carrying out the development through the measurement of the performance.

Acknowledgements

This work was supported the project of research and development of innovative practical use technologies for unused heat sources by New Energy and Industrial Technology Development Organization.

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