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Developing a Two-Stage Rotary Compressor for CO₂ Heat Pump Systems with Refrigerant Injection

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ABSTRACT

The reduction of greenhouse gas emissions has been the focus of global attention. As an alternative refrigerant to HFC, CO₂ has been considered. As another promising possibility of energy conservation, use of heat pump systems has also been spreading. However, conventional heat pump systems have problems remaining, such as inadequate heating capacity or overheating of the compressor due to reduced performance under ambient low temperature condition. To solve these problems, we developed a two-stage rotary compressor for CO₂ heat pump systems, examined the effects of the injection into an intermediate pressure connector under the condition of a high pressure-ratio and offered proof of its high efficiency. We also examined unique performance characteristics of the two-stage rotary compressor by its loss analysis. We clarified the merits of the two-stage rotary compressor's performance compared to those of a single-stage rotary compressor.

1. INTRODUCTION

The reduction of greenhouse gas emissions involving HFC refrigerants, e.g. by shifting to natural refrigerants, has been the focus of global attention since the Kyoto Protocol was concluded in 1997 and promulgated in 2005. As an alternative refrigerant to HFC, CO₂ has been considered. As another promising possibility of energy conservation, use of heat pump systems has also been spreading.

However, further development of conventional heat pump systems is necessary because certain problems remain, such as inadequate heating capacity or overheating of the compressor due to reduced performance under ambient low temperature conditions. To solve these problems, a two-stage compression cycle with refrigerant injection is used (Saito, 2007). Unfortunately, other problems arise, such as mixture loss on refrigerant injection into the compression chamber of a single-stage compressor. In the case of CO₂ refrigerant, there is the disadvantage of being more likely to leak compression gas because of operation under high-pressure conditions (Yokoyama *et al.*, 2006).

To resolve these issues, a two-stage rotary compressor is very useful. We designed and manufactured it for use in household applications of 2HP (1.5kW) input. The following experiment was performed to clarify the merits and weak points of the two-stage rotary compressor in comparison to a single-stage type (Maeyama *et al.*, 2006). We

- analyzed compressor loss by measuring the indicated work in the cylinder, and improved its performance;
- measured its performance at various rotational speeds and various pressure ratios;
- measured its performance with refrigerant injection under the condition of a high pressure-ratio equivalent under ambient low temperature (below zero).

2. PROBLEM DESCRIPTION

2.1 Leakage Characteristics of CO2 Refrigerant

We estimated leakage mass flow rate of ideal refrigerant gas through the radial clearance of its rolling piston by the isentropic flow model through a convergent nozzle. It is assumed that a rotary compressor is operated in the condition of ASHRAE-T; that the radial clearance gap of its rolling piston is 10[μm] constant; that the leakage cross-section area A_{leak} is proportional to its representative length equal to $V_{st}^{1/3}$; and that the leakage speed equals sound speed of the critical flow. Table 1 shows the leakage characteristics of CO2 refrigerant with a single type, compared to conventional refrigerants, (R22, R410A), and compared to the 1st-stage cylinder of a two-stage type.

- The estimated CO2 leakage percent of the ideal mass flow rate G_{leak}/G_{s_i} is about three times that of R22 or R410A. CO2 refrigerant has characteristics to leak more easily than conventional refrigerants due to high operating pressure. Especially its small displacement for the same indicated power causes that disadvantage.
- It is expected that the two-stage type decreases the CO2 leakage percent of the ideal mass flow rate G_{leak}/G_{s_i} by a 2/3 ratio compared to a single type.

Table 1: Leakage characteristics of CO2 refrigerant with a single type, compared to those of R22 and R410A, and compared to a two-stage type in the condition of ASHRAE-T (CT/ET=54.4/7.2[deg C], SC/SH=8.3/27.8[K]), but in the case of CO2 P_d is estimated in the condition of maximum COP)

	Refrigerant	Single type			1st-stage of two-stage type CO2
		R22 (Conventional ref.)	R410A	CO2	
Operating condition	P_d : Discharge Pressure [MPa]	2.15	3.39	11.18	6.85
	P_s : Suction Pressure [MPa]	0.63	1.00	4.20	4.20
	P_d/P_s : Pressure ratio	3.43	3.39	2.66	1.63
	ρ_s : Suction density [kg/m ³]	23.3	32.36	92.4	92.4
Specifications of a rotary compressor	V_{st} : Displacement (@ $W_{in} = 1kW, 60rps$) [cm ³]	20.3	13	3.7	3.7
	Ratio of representative length to CO2= $\{V_{st}/(V_{st} \text{ of CO2})\}^{1/3}$	1.76	1.52	1.00	1.00
	h : Height of cylinder [mm]	17.64	15.20	10.00	10.00
	δ_r : Radial clearance gap of R/P [μm]	10	10	10	10
	$A_{leak} = h \times \delta_r$: Leakage cross-section area of δ_r [mm ²]	1.8E-01	1.5E-01	1.0E-01	1.0E-01
Critical condition and leakage	$n = \ln(P_d/P_s) / \ln(\rho_d/\rho_s)$: Isentropic exponent	1.11	1.08	< 1.31	1.28
	$p_e = p_d \times \{2/(n+1)\}^{n/(n-1)}$: Critical pressure [MPa]	1.25	2.00	6.09	3.76
	ρ_e : Critical density [kg/m ³]	43.38	61.32	< 122.77	84.75
	u_e : Sound speed of the critical flow [m/s]	178.7	187.8	< 252.0	237.4
	$G_{leak} = A_{leak} \times u_e \times \rho_e$: Leakage mass flow rate [μkg/s]	1.4E+03	1.8E+03	3.1E+03	2.0E+03
	$G_{s_i} = \rho_s \times V_{st} \times 60$: Ideal mass flow rate(@60rps) [μkg/s]	2.8E+04	2.5E+04	2.1E+04	2.1E+04
G_{leak} / G_{s_i} : Leakage percent of ideal mass flow rate	4.8%	6.9%	15.1%	9.8%	

2.2 Mixture Loss on Refrigerant Injection

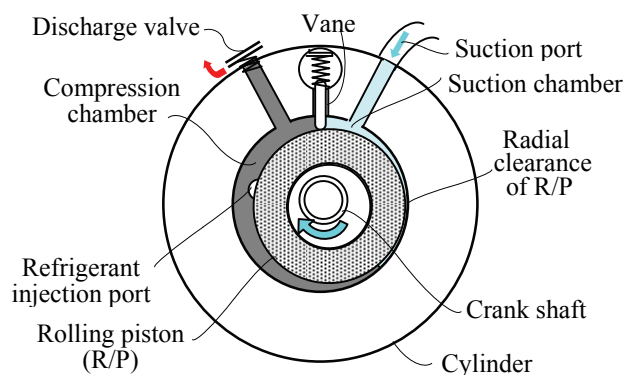


Figure 1: Structure of rotary compression mechanism in single type cylinder with refrigerant injection

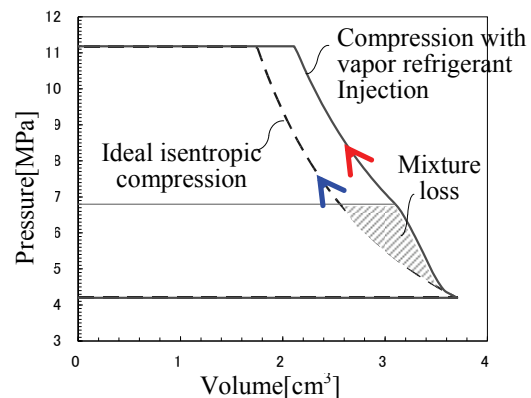


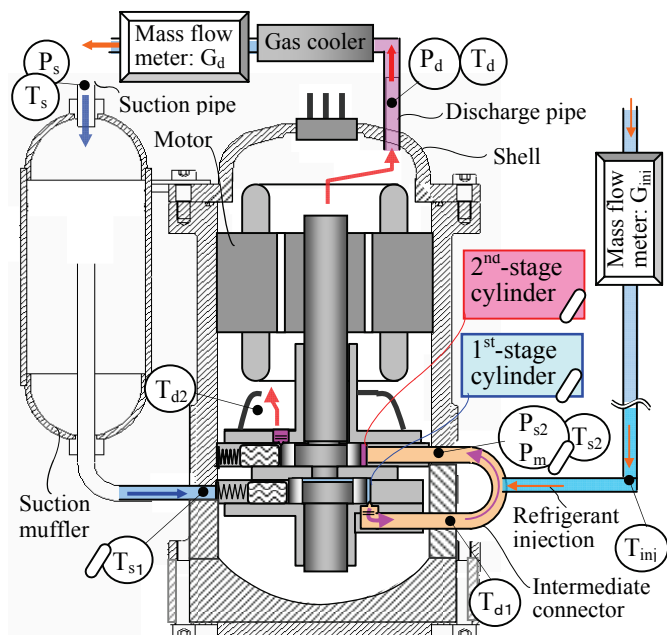
Figure 2: P-V diagram in cylinder of single rotary compressor with vapor refrigerant injection

Figure 1 shows the structure of the compression mechanism in a single type cylinder with a refrigerant injection port. Refrigerant under intermediate pressure is injected into a compression chamber through the port, while the port is opened by eccentric rotation of the rolling piston. Figure 2 shows a P-V diagram in the single rotary compression chamber with refrigerant injection. If a two-stage compression cycle uses a single rotary compressor, there is the problem that refrigerant injection increases the indicated power by mixture loss (Sekiya *et al.*, 2005), compared to that of ideal isentropic compression.

3. EXPERIMENTAL PROCEDURE

3.1 Experimental Model

We made experimental models of the two-stage rotary compressor. Figure 3 shows their structure and measuring instruments. Table 2 shows their specifications. The two-stage type has two cylinders in series, a crankshaft, a motor and a semi-hermetic shell on the high-pressure side. A rolling piston-type compression mechanism in the 1st-stage cylinder compresses CO₂ refrigerant from suction pressure P_s to intermediate pressure P_m and discharges it into its intermediate connector. That in the 2nd-stage cylinder compresses it from P_m to discharge pressure P_d , discharges it into the shell and discharges it out of the shell through flow channels of the motor. Its intermediate connector has a port to inject it under intermediate pressure. We fixed displacement of the 1st-stage cylinder V_{st1} on 4.8[cm³], to target household applications of 2HP; and decided that the basis of displacement ratio V_{st2}/V_{st1} was 0.65, to keep its good performance in the wide range of pressure ratio P_d/P_s (, the experimental range from 1.8 to 6.5).



(T) : Thermocouple (P) : Pressure strain gage (⌒) : Piezometer

Figure 3: Structure of two-stage rotary compressor and measuring instruments

Table 2: Specifications of experimental model of two-stage rotary compressor

Compressor type	Two-stage twin rotary, Rolling piston type
Shell type	Semi-hermetic shell On high pressure side
V_{st1} : Displacement [cm ³]	4.8
V_{st2}/V_{st1} : Displacement ratio	0.65 (basis); 0.6~0.8
Refrigerant	R744 (CO ₂)
Motor	Brushless DC motor
Usage	Heat pump systems
Heating capacity [kW]	6.0 (basis); 4.5~7.5
Input power [HP]	2 (basis); 1~3

3.2 Experimental Measurement and Performance Evaluation Method

Figure 3 shows that we measured refrigerant temperatures and static pressures in a suction pipe of a suction muffer and in a discharge pipe out of the shell, in addition to upstream of the suction port and downstream of the discharge valve of each cylinder. We also measured refrigerant mass flow rates (G_d , G_{inj}) in liquid phase by use of mass flow meters based on Coriolis. They are located in a main flow circuit and in an injection flow circuit downstream of a gas cooler. Moreover, we measured the input power into the motor with a digital power meter. With the above experimental data, we calculated the volumetric efficiency η_v and the compressor efficiency η_c by Eqs. (1) and (2) with a computer. Here we calculated specific enthalpy growth (Δh_{1_i} , Δh_{2_i}) from suction to discharge within the isentropic compression process in each cylinder, by use of experimental data on the temperatures and the pressures and the refrigerant database 'REFPROP' from NIST (Lemmon *et al.*, 2002).

$$\eta_v = \frac{\text{Actual mass flow rate}}{\text{Ideal mass flow rate}} = \frac{G_d - G_{inj}}{\rho_s \times V_{st1} \times (\text{Rotational speed}) \times 3600} \quad (1)$$

$$\eta_c = \frac{\text{Ideal power for actual mass flow rate, } W_{in,i}}{\text{input power, } W_{mot}} = \frac{(G_d - G_{inj}) \times \Delta h_{1,i} + G_d \times \Delta h_{2,i}}{W_{mot} \times 3600} \quad (2)$$

<Ideal specific enthalpy in suction of the 2nd-stage>: $h_{s2,i} = (h_{d1,i} + \alpha \times h_{inj}) / (1 + \alpha)$ (3)

Figure 4 shows the P-h curves in the ideal compression process of the two-stage type, compared to P-h curves of the real compression process of the two-stage type. We calculated the ideal specific enthalpy in suction of the 2nd-stage $h_{s2,i}$ by Eq. (3) assuming isobaric change without mixture loss. Here α is the mass flow ratio of the refrigerant injection; and equals $G_{inj} / (G_d - G_{inj})$.

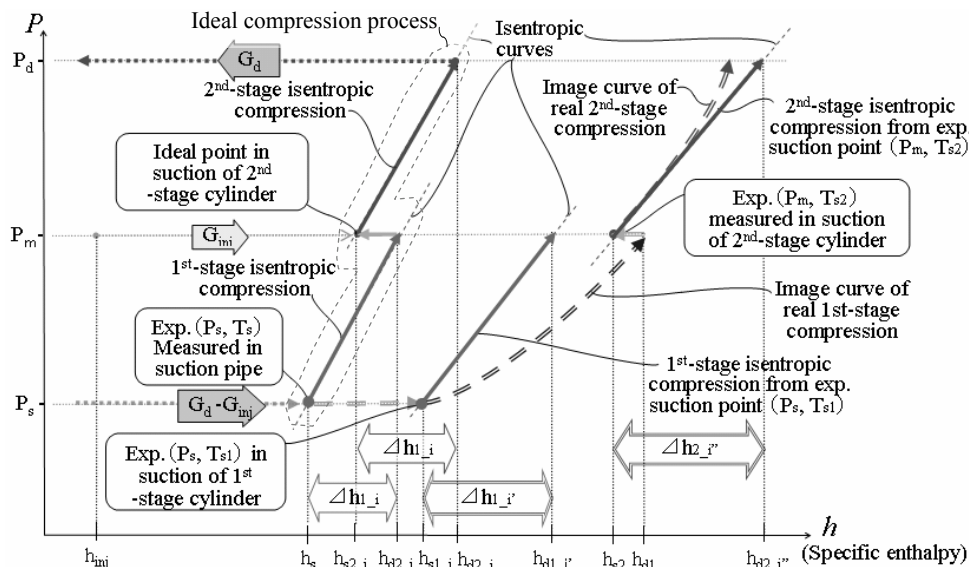


Figure 4: P-h curves in ideal compression process of two-stage type with injection, compared to real process

We calculated the indicated power in each cylinder by measuring pressure fluctuations in the suction port or in the compression chamber of each cylinder by use of piezometers, to analyze compressor losses as in the next section.

3.3 Compressor Loss Analysis Method

Table 3: Classification of loss factors of two-stage rotary compressor

Input power (into Electric motor), W_{mot}	
└ Motor loss	
└ Motor output, W_{out}	└ Mechanical loss
	└ Indicated power, W_{in}
	└ Ideal power for actual mass flow rate, $W_{in,i}$
	└ Indicated loss
	└ 1 st -stage suction undershooting loss
	└ 1 st -stage heat loss
	└ Preheat loss
	└ Heat loss in compressing
	└ 1 st -stage reexpansion loss
	└ 1 st -stage leakage loss
	└ 1 st -stage discharge overshooting loss
	└ 2 nd -stage suction undershooting loss
	└ 2 nd -stage heat loss
	└ Preheat loss, W_{Lpre2}
	└ Heat loss in compressing
	└ 2 nd -stage reexpansion loss
	└ 2 nd -stage leakage loss
	└ 2 nd -stage discharge overshooting loss

} Intermediate pressure pulsation loss, W_{Lpm}

Table 3 shows classification of the loss factors of the two-stage rotary compressor. It is abbreviated to explain the loss analysis method overall, which refers to the conventional method of a single type (Koda *et al.*, 1989). Below are explained the peculiar losses of the two-stage rotary compressor. 1) Intermediate pressure pulsation loss $W_{L,pm}$ is defined as the sum of 1st-stage discharge overshooting loss and the 2nd-stage suction undershooting loss and is strongly affected by pressure drops and fluctuation delay of the intermediate pressure. 2) 2nd-stage preheat loss $W_{L,pre2}$ is defined as Eq. (4). Each specific enthalpy is shown in Figure 4. Δh_{2_i} equals specific enthalpy growth from an experimental suction point to a discharge point within the isentropic compression process in the 2nd-stage cylinder.

$$\langle \text{2}^{\text{nd}}\text{-stage preheat loss} \rangle: W_{L,pre2} = (\Delta h_{2_i} - \Delta h_{2_i}) \times G_d / 3600 \quad (4)$$

4. EXPERIMENTAL RESULTS

4.1 Compressor Loss Analysis and Effects of Intermediate Pressure Pulsation

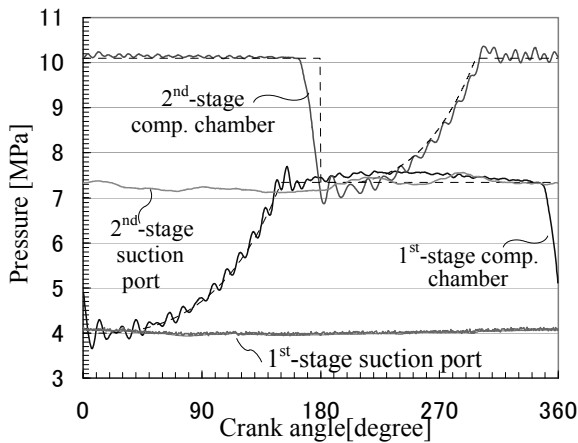


Figure 5: Pressure fluctuations in each cylinder

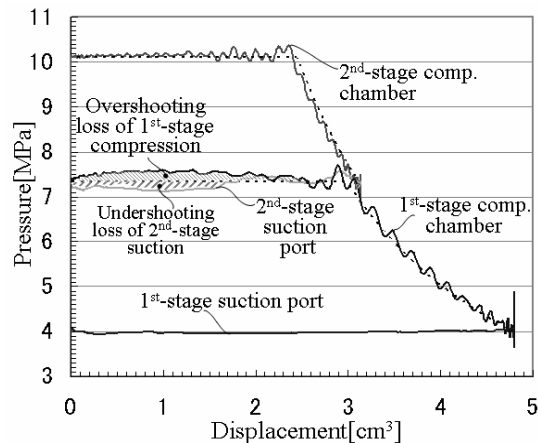


Figure 6: P-V diagram in each cylinder of original two-stage rotary compressor

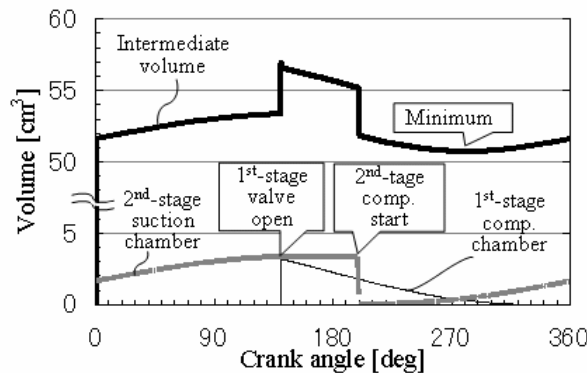


Figure 7: Fluctuation of original intermediate volume

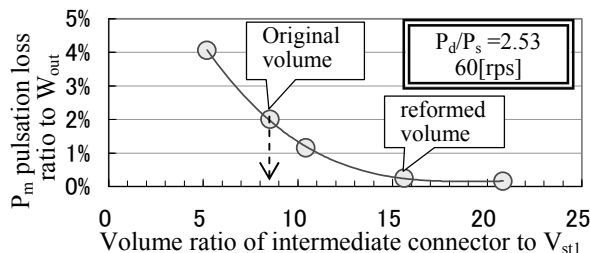


Figure 8: Intermediate pressure pulsation loss at various volumes of intermediate connector

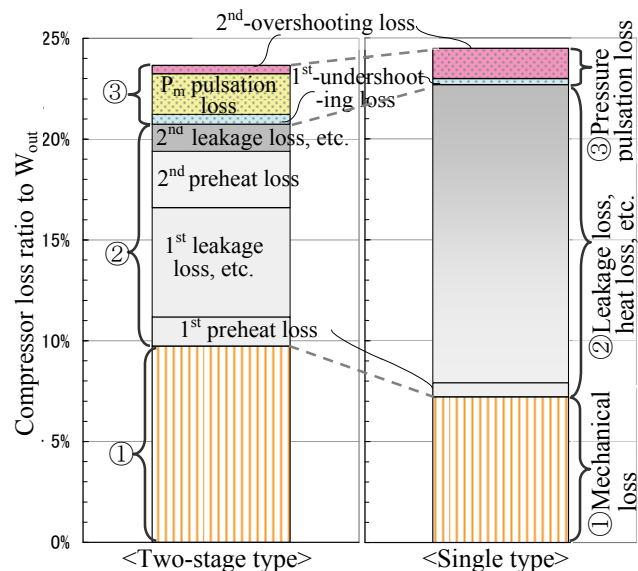


Figure 9: Measured compressor loss ratio compared with original two-stage type and single type, on the base condition ($P_d/P_s = 2.53$, 60[rps], SH=10[K])

Figure 5 shows pressure fluctuations in the compression chambers and the suction ports in each cylinder. Figure 6 shows the P-V diagram in each cylinder of the two-stage rotary compressor. The two-stage type with the V_{st2}/V_{st1} equal to 0.65 was tested on the base condition ($P_d/P_s=2.53$, 60[rps], SH=10[K]). Figure 7 shows the fluctuation of the intermediate volume V_m which is defined as the sum of the volume of the intermediate connector $V_{m,con}$ and that of the 2nd-stage suction chamber. Additionally, the volume of the 1st-stage compression chamber is added to the sum, while the discharge valve of the 1st-stage compression chamber is open. V_m is smaller than the average volume in the region from 200 to 360 degrees and has the minimum point. Then pressure in the 1st-stage compression chamber tends to overshoot in the degree region, and to cause an increase in $W_{L,pm}$.

Figure 8 shows intermediate pressure pulsation loss at various $V_{m,con}$ ratios to V_{st1} . We thought of a method to decrease intermediate pressure pulsation loss. We tried to damp the pressure pulsation of the intermediate connector and to decrease the 1st-stage discharge overshooting loss by increasing the intermediate volume. It was confirmed that the $W_{L,pm}$ was almost zero, when the $V_{m,con}$ ratio to V_{st1} got as large as about 15. (From the next section, the two-stage type used the reformed intermediate volume.) Figure 9 shows the compressor loss ratio to the motor output of the two-stage type with the original intermediate volume, compared to the single type of almost equal specifications. Compressor loss analysis was operated on the base condition.

- For loss ratio of ①, Mechanical loss, that of the two-stage type is greater.
- For loss ratio of ②, Leakage loss, heat loss, etc. (including the rest not separated by the indicated loss analysis), that of the two-stage type is less.
- For loss ratio of ③, Pressure drop and pulsation loss, that of the two-stage type is greater due to large pressure pulsation in the intermediate pressure.

4.2 Performance at Various Rotational Speeds

Figure 10 shows performance of the two-stage type at various rotational speeds, compared to the single type of almost equal specifications. Volumetric efficiency η_v and compressor efficiency ratio η_c/η_{c0} were measured on a certain constant that $P_d/P_s=2.53$. Here η_{c0} was the base value, η_c of the single type measured on the base condition.

- η_v of the two-stage type had a lead gap of more than 5% over that of the single type. In particular, the lead increased significantly in the low-speed region because the single type it was more likely to leak.
- η_c/η_{c0} of the two-stage type was superior to that of the single type in the speed region lower than 70rps. And it was inferior in the high-speed region because of a greater mechanical loss ratio.

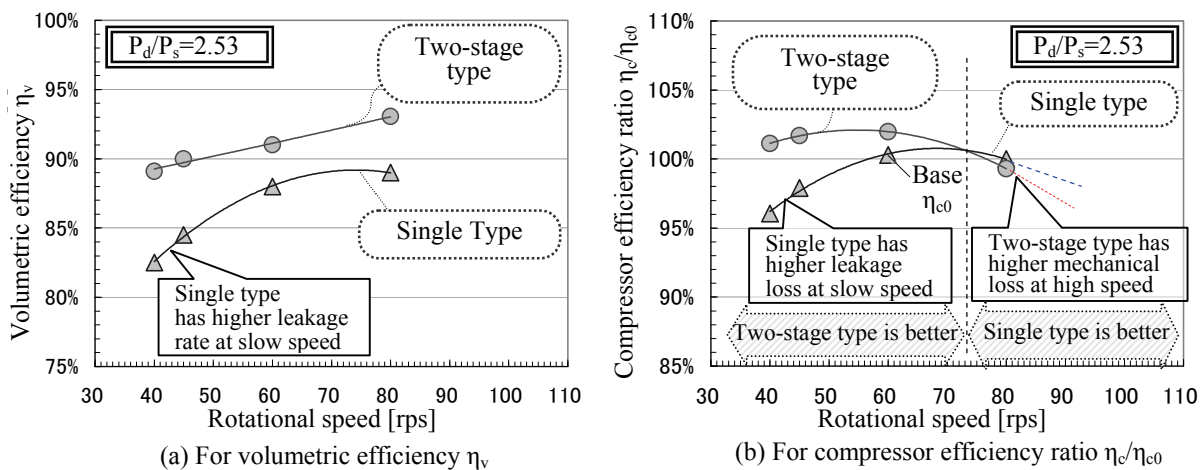


Figure 10: Performance at various rotational speeds, comparing two-stage type and single type

4.3 Performance at Various Pressure Ratios and Effects of Refrigerant Injection

Figure 14 shows performance of the two-stage type at various pressure ratios, compared to that of the single type of almost equal specifications. η_c was measured on a certain constant that the heating capacity equals 6kW. When P_d/P_s was larger than 4.5, refrigerant injection decreased the discharge temperature T_d to equal the allowance temperature (120[deg C]).

- η_c/η_{c0} of the two-stage type was superior to that of the single type in the pressure ratio region higher than 3.5,

equivalent under ambient low temperature.

- In the case of the two-stage type, η_c/η_{c0} is improved, if refrigerant in wet or dry condition is injected into its intermediate connector. (The reason is described in the next paragraph.)
- To the contrary, η_c/η_{c0} of the single type is decreased due to mixture loss caused with refrigerant injection.
- η_c/η_{c0} of the two-stage type was also good in the pressure ratio region lower than 3 because the reformed intermediate volume made $W_{L,pm}$ a little.

Here we consider the effects of injecting refrigerant in wet or dry condition (quality > 0.5) into the intermediate connector of the two-stage type. Figures 11 and 12 show performance characteristics examined at various pressure ratios, such as T_d and $P_m/(P_s \times P_d)^{0.5}$. As P_s/P_d gets large, T_d increases and $P_m/(P_s \times P_d)^{0.5}$ decreases to increase the pressure ratio of the 2nd-stage P_d/P_m . Then, wet vapor injection (quality > 0.5) leads not only to decreasing T_d to equal the allowance temperature, but also to increasing $P_m/(P_s \times P_d)^{0.5}$ to approach 1, which is generally the optimal value. Figure 13 shows image P-h curve of the two-stage type with vapor injection compared to that with liquid injection. Defining volumetric efficiencies of the 1st-stage cylinder as η_{v1} and that of the 2nd-stage cylinder as η_{v2} , Eq. (5) approximately relates the density in the 2nd-stage suction point ρ_{s2} with that in the 1st-stage suction point ρ_{s1} .

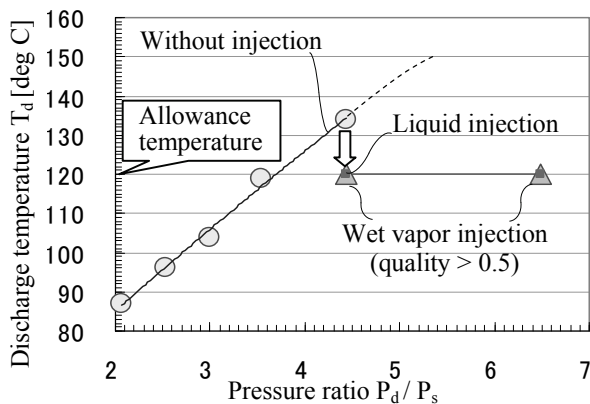


Figure 11: Discharge temperature T_d measured at various pressure ratios

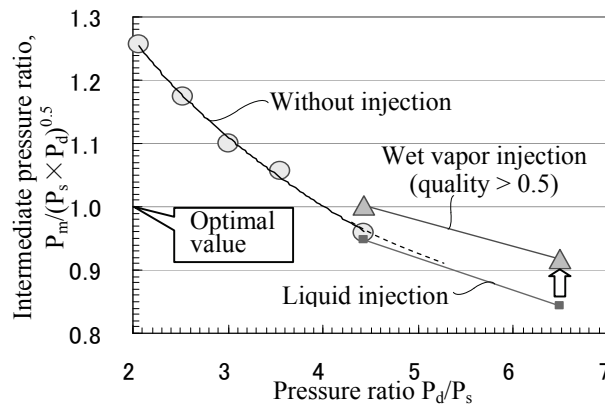


Figure 12: Intermediate pressure ratio, $P_m/(P_s \times P_d)^{0.5}$, measured at various pressure ratios

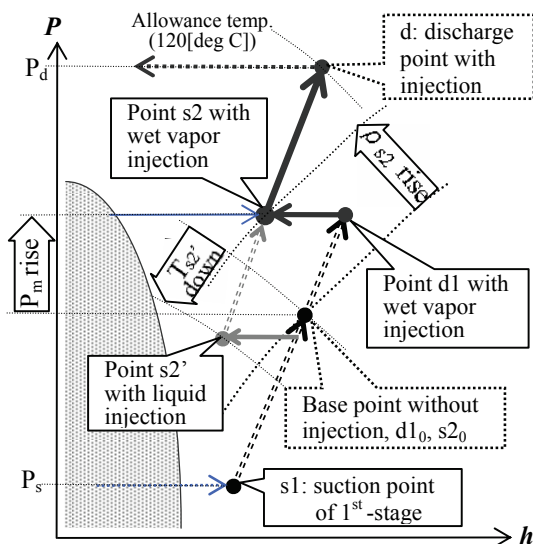


Figure 13: Image P-h curve of two-stage type with wet vapor injection (quality > 0.5) compared to that with liquid injection

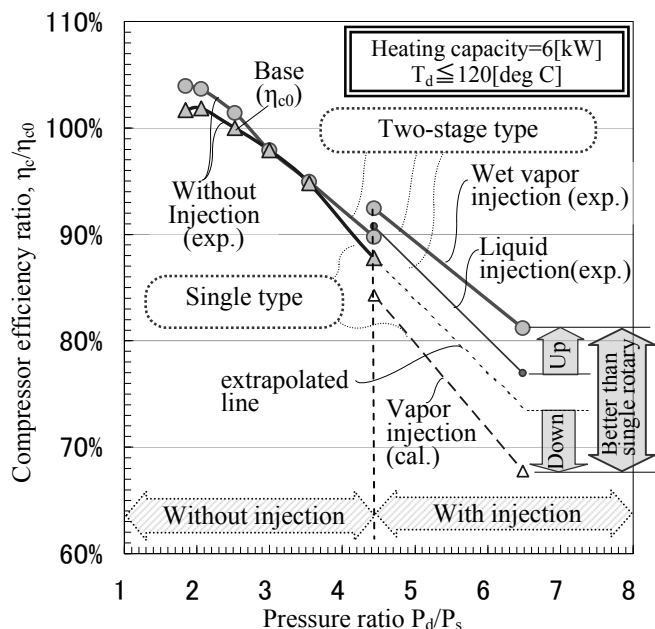


Figure 14: Compressor efficiency at various pressure ratios, in comparison with two-stage type and single type

$$\rho_{s2} = \frac{\eta_{v1} \times V_{st1}}{\eta_{v2} \times V_{st2}} \rho_{s1} \times (1+\alpha) = \rho_{s20} \times (1+\alpha) = \text{function}(P_m, T_{s2}) \quad (5)$$

Injecting refrigerant, ρ_{s2} becomes $1+\alpha$ times larger than the density without injection ρ_{s20} , to hope that P_m rises. The wet vapor injection (quality > 0.5) delivers the expected results. On the other hand, P_m rises little or lower in the case of the liquid injection because α is half less than of the wet vapor injection due to its great cooling effect, to keep T_d the same allowance temperature (120[deg C]). Thus the wet vapor injection can improve the performance by balancing each pressure ratio of 1st-stage and 2nd-stage in addition to by keeping down T_d and increasing G_d and heating capacity under the condition of the high pressure-ratio. It is important that the two-stage type is designed as P_m approaches the optimal value $(P_s \times P_d)^{0.5}$, considering not only V_{st2}/V_{st1} , but also α and its quality.

5. CONCLUSIONS

This paper described how we developed the two-stage rotary compressor for CO2 heat pump systems with refrigerant injection; clarified its unique performance characteristics such as the intermediate pressure pulsation loss by the experimental analysis results; and clarified its merits compared to the single type.

<Merit 1> The two-stage type is superior to the single type with respect to compressor efficiency in the condition of low rotational speed or high pressure-ratio because it has characteristics more unlikely to leak compression gas.

<Merit 2> The two-stage type is superior to the single type with respect to compressor efficiency and heating capacity because it has the ability to improve these performances with refrigerant injection in the condition of the high pressure-ratio.

If we properly design and operate the two-stage rotary compressor with refrigerant injection, we can greatly contribute to energy conservation by use of heat pump systems including CO2 and the other refrigerants.

NOMENCLATURE

G: Mass flow rate [kg/h]

h: specific enthalpy [J/kg]

P: Pressure [MPa]

T: Temperature [deg C] or [K]

W: Power [W]

ρ : Density [kg/m³]

Subscripts

d: Discharge

m: Intermediate

s: Suction

i: Ideal

0: Basic

1: 1st-stage

2: 2nd-stage

inj: Injection

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