CHARACTERISTICS OF TWO-PHASE EJECTOR IN CARBON DIOXIDE REFRIGERATION CYCLE

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ABSTRACT

Recently, carbon dioxide has gained much attention as a natural refrigerant. Installation of two-phase ejector is one of the most promising ways to improve the COP (Coefficient of Performance) of carbon dioxide refrigeration cycle. However the flow inside two-phase ejector, in which working fluid undergoes phase change, has not been clarified yet. The purpose of this research is to clarify the characteristics of the supersonic two-phase flow of carbon dioxide in ejector, and improve the performance of ejector. In this paper, it is shown that performance of ejector in compression refrigeration cycle is decided by the characteristics of pressure recovery through ejector and pressure drop through suction flow passage. The effect of mixing section diameter and the expansion valve was clarified experimentally, and 11% of COP value was improved in this work.

Key Words: ***carbon dioxide, ejector, compression refrigeration cycle.***

1 INTRODUCTION

Recently, natural refrigerants have gained much attention due to global environment problems, such as ozone depletion and global warming. Carbon dioxide is especially gathering attention because of its incombustibility and low GWP (Global Warming Potential). However, the COP of CO_2 refrigeration cycle is lower than those of other natural refrigerants. The purpose of this research is to improve the COP of CO_2 refrigeration cycle. In conventional cycle, there is a waste of effective energy during the expansion process. Since CO_2 cycle has a larger pressure difference across the expansion process than other refrigerants, more effective energy is wasted. Ejector is expected to improve the COP of refrigeration cycle by recovering this effective energy. (Jeong et al. 2004) investigated the theoretical COP of ejector-equipped compression refrigeration cycle using ammonia and CO_2 as refrigerants. (Ozaki et al. 2004) compared the effectiveness of ejector to compression refrigeration cycle with that of expander using CO_2 refrigerant. (Disawas and Wongwises 2004) examined COP improvement using R-134a refrigerant experimentally. (Nakagawa et al. 1996) investigated the influences of geometric parameters on the ejector performance using R-12 refrigerant. However, characteristics of ejector-equipped compression refrigeration cycle with CO_2 refrigerant have not been sufficiently investigated. In this paper the effect of mixing section diameter and the expansion valve on ejector-equipped compression refrigeration cycle is clarified experimentally.

2 EJECTOR-EQUIPPED COMPRESSION REFRIGERATION CYCLE WITH CO_2 REFRIGERANT

Figure 1 shows the schematic of ejector, and Fig. 2 shows the schematic of ejector-equipped compression refrigeration cycle with CO_2 refrigerant. Figure 3 shows the P-h diagram of the cycle. After compressed with compressor and cooled in gas cooler, super-critical CO_2 is led to the primary nozzle. Flowing through the primary nozzle, super-critical CO_2 is accelerated to supersonic speed. This supersonic two-phase flow is directed to mixing chamber together with the suction flow from evaporator. These two flows are mixed in mixing chamber and then shock wave takes place in order to turn the supersonic speed into the pressure recovery. Following that, in the diffuser section, the remaining speed is further changed into pressure recovery. The two-phase flow at the outlet of ejector gets into separator, where the vapor phase is sucked into the compressor, and the liquid phase is depressurized and led to the evaporator. In this way, unlike the conventional cycle, the suction pressure of compressor can be higher than the pressure at the outlet of evaporator, so the COP of ejector-equipped compression refrigeration cycle can be improved compared with the conventional one. The simulation model for each section of this cycle is as follows.

2.1 Nozzle

The velocity at the nozzle outlet, u_{no}, is calculated by Eq.(1), where η_{nzl} means nozzle efficiency and Δh_{nzl} means the enthalpy difference between nozzle inlet and nozzle outlet assuming isentropic process.

$$u_{no} = \sqrt{2\Delta h_{nzl} \cdot \eta_{nzl}} \tag{1}$$

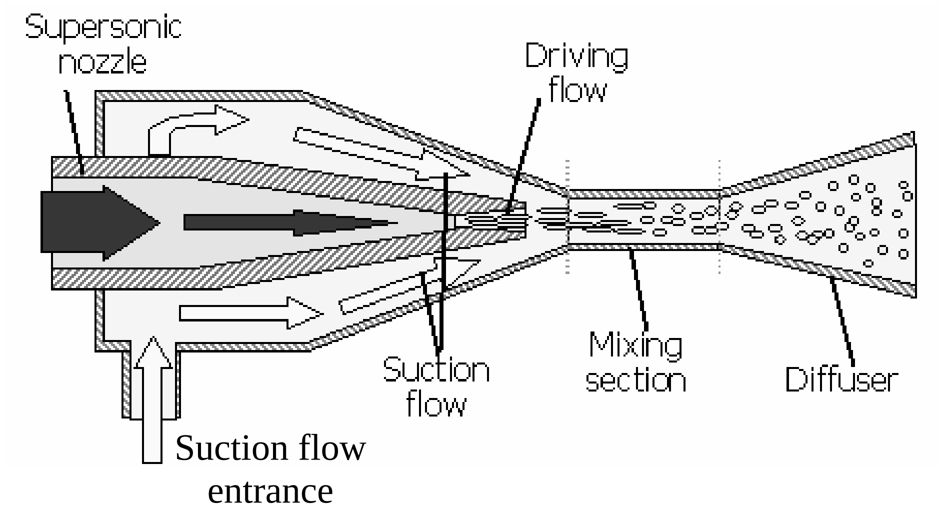

Fig. 1. Schematic of ejector

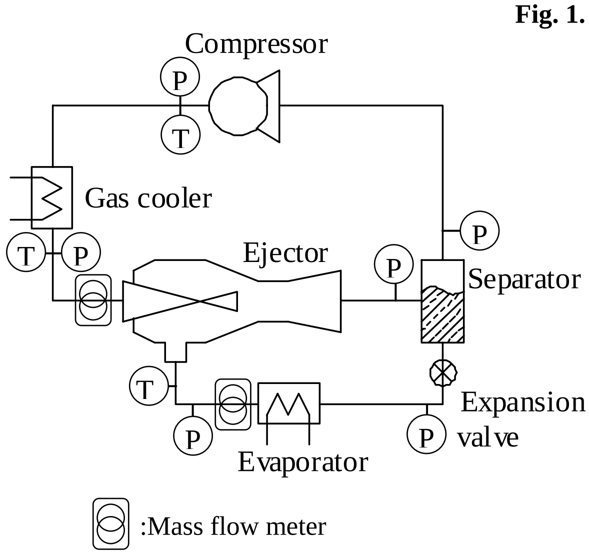

Fig. 2. Schematic of ejector cycle and experimental setup

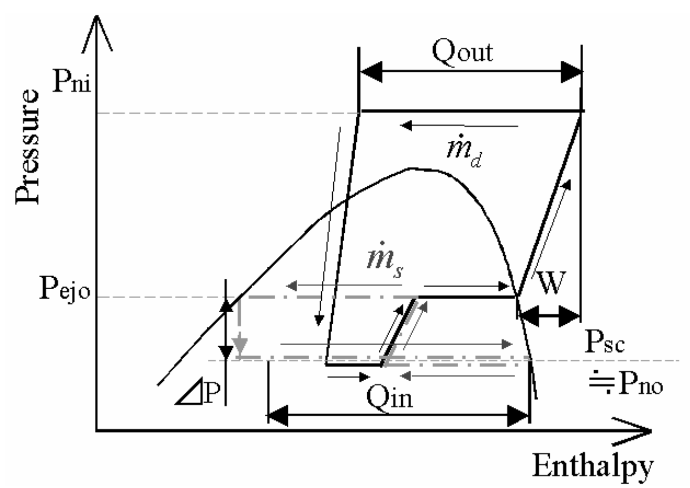

Fig. 3. P-h diagram of ejector cycle

2.2 Mixing Section

Equations (2)–(4) are the basic equations of the model for mixing section described in Fig. 4.

$$\dot{m}_d + \dot{m}_s = \rho_{mo} A_{mx} u_{mo} \tag{2}$$

$$\dot{m}_d u_{no} + P_{no} A_{no} + \dot{m}_s u_{sc} + P_{sc}\left(A_{mx} - A_{no}\right) = \left(\dot{m}_d + \dot{m}_s\right) u_{mo} + P_{mo} A_{mx} \tag{3}$$

$$\dot{m}_d \left(h_{no} + u_{no}^2/2\right) + \dot{m}_s \left(h_{sc} + u_{sc}^2/2\right) = \left(\dot{m}_d + \dot{m}_s\right)\left(h_{mo} + u_{mo}^2/2\right) \tag{4}$$

In this model, pressure from nozzle outlet plane to mixing section inlet plane is assumed to be constant.

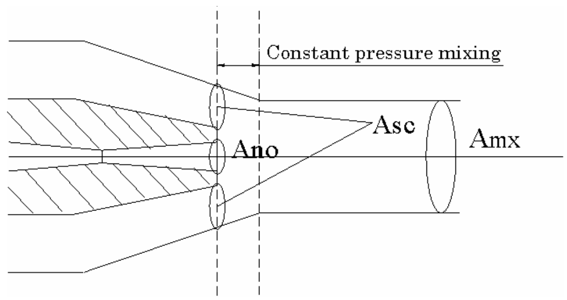

Fig. 4. Model of mixing section

2.3 Diffuser

Diffuser efficiency, η_{dif}, is defined as $\eta_{dif} = \Delta h_{dif1} / \Delta h_{dif2}$, where Δh_{dif1} means enthalpy difference between diffuser inlet and diffuser outlet assuming isentropic process ,and Δh_{dif2} means actual enthalpy difference. The state of diffuser outlet is calculated, when diffuser efficiency is given.

2.4 Separator

Entrainment ratio, which is defined as $\omega = m_s / m_d$, needs to be equal to $(1/x_{ejo}) - 1$, where x_{ejo} means quality at ejector outlet, in order to satisfy mass balance regarding to both liquid and vapor on the assumption that the separator is thermally insulated and that two phase flow is completely separated by the separator.

2.5 COP

The cooling COP of ejector-equipped cycle is defined as (5).

$$\mathrm{COP}_{\mathrm{eje\_cooling}} = \frac{Q_{in}}{W} \times \frac{\dot{m}_s}{\dot{m}_d} \tag{5}$$

Calculated results for cooling COP of ejector-equipped cycle are shown in Fig. 5. In this calculation both suction flow entrance for ejector-equipped cycle and suction pressure of compressor for conventional cycle are assumed to be 4.5 MPa, 12 °C.

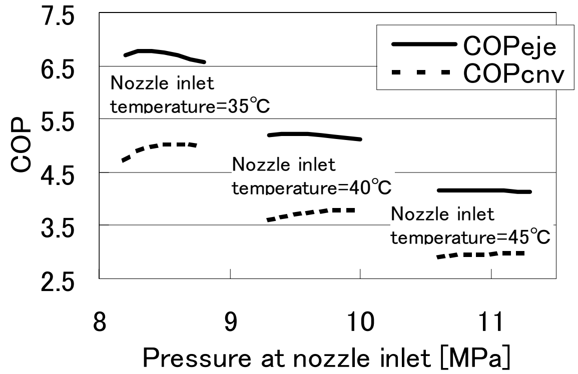

Fig. 5. Theoretical cooling COP of ejector-equipped compression refrigeration

3 EXPERIMENTAL SETUP

The schematic of experimental setup in this research is shown in Fig. 2. This research is aimed to investigate the effect of mixing section diameter and expansion valve by measuring pressure and temperature at nozzle inlet, suction flow entrance, ejector outlet, and just after expansion valve, and mass flow rate of driving flow and suction flow. To investigate the effect of only mixing section without the effect of diffuser, all ejectors for this paper do not have diffuser. Nozzle is convergent nozzle, and has 0.82 mm of nozzle outlet diameter. Mixing section diameter is changed from 1.5 mm to 4.5 mm.

4 PERFORMANCE OF EJECTOR IN COMPRESSION REFRIGERATION CYCLE

Entrainment ratio ω or suction flow rate m_s, and pressure recovery $\Delta P \equiv P_{ejo} - P_{sc}$, in actual cycle, are determined by the characteristics of pressure recovery through ejector and pressure drop through suction flow passage. Characteristics of pressure recovery through ejector can be calculated with Eq. (2)-(4). The result of this calculation is shown in Fig. 6. This figure shows the relationship between suction flow rate and pressure recovery. From this figure it can be seen that for the same ejector the pressure recovery decreases when the suction flow rate increases. Pressure drop through suction flow passage consists of the pressure decrease from ejector outlet, through separator and evaporator, to the ejector suction entrance. Pressure drop increases with the increase of suction flow rate. Figure 7 shows some experimental results obtained with present experimental setup on the condition that the pressure and temperature at nozzle inlet and suction flow entrance are constant, and expansion valve is fully open. This dashed line can be considered the relationship between pressure drop and suction flow rate at this condition. The reason why this line does not intersect the coordinate origin is the pressure drop of driving flow at separator exists even when there is no suction flow. The intersection of Fig. 6 and Fig. 7 stands for the actual performance of ejector in ejector-equipped compression refrigeration cycle.

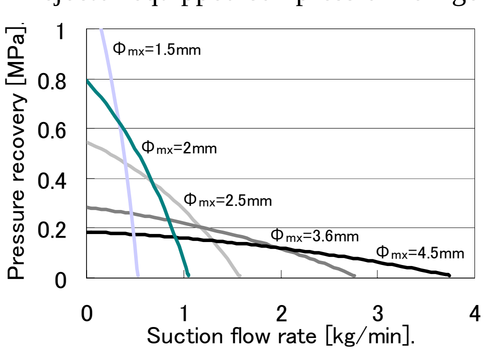

Fig. 6. Calculation results of mixing section

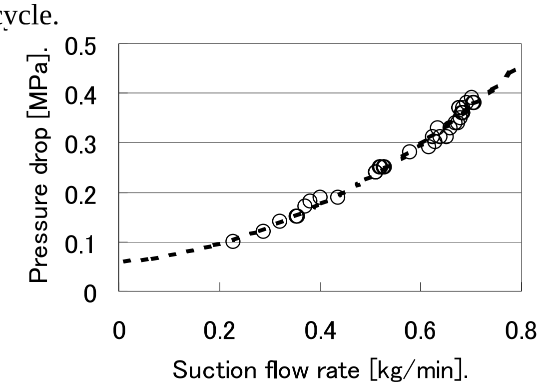

Fig. 7. Pressure drop through suction flow passage

In the actual cycle, ω is not always equal to $(1/x_{ejo})-1$. In general, steady state is not realized if ω is not equal to $(1/x_{ejo})-1$. When ω is less than $(1/x_{ejo})-1$, liquid-phase flow getting into separator is larger than that getting out of separator, then liquid level comes up. When ω is larger than $(1/x_{ejo})-1$, liquid level comes down. There are, however, some ways to realize steady state with ω different from $(1/x_{ejo})-1$. When ω is less than $(1/x_{ejo})-1$, steady state is realized by heating separator to supply the vaporization heat, or is also realized if two-phase flow gets into compressor. When ω is more than $(1/x_{ejo})-1$, steady state is realized if two-phase flow gets into evaporator.

However, if ω is larger than $(1/x_{ejo})-1$, and two-phase flow gets into evaporator, COP decreases compared with COP for $\omega=(1/x_{ejo})-1$, because the gas phase reduces the pressure recovery through ejector although the gas phase does not contribute to the cooling power. If ω is less than $(1/x_{ejo})-1$, and separator is additionally heated, COP is less than COP for $\omega=(1/x_{ejo})-1$. If two-phase flow gets into compressor, compressor would be damaged. So, ejector-equipped compression refrigeration cycle should be operated with ω equal to $(1/x_{ejo})-1$.

5 RESULTS AND DISCUSSION

All experimental conditions in section (5.1) and section (5.2) are 9 MPa, 35 °C at nozzle inlet, and 4.5 MPa, 12° C at suction flow entrance. In this experiment, ω was not always equal to be $(1/x_{ejo})-1$, mostly less than $(1/x_{ejo})-1$. Therefore, in some experimental conditions, separator was heated. In other conditions, two-phase flow got into compressor. For this experimental condition, $(1/x_{ejo})-1$ is about 0.62. Critical mass flow rate measured for driving nozzle, m_d, was about 1.5 kg/min. m_s needs to be about 0.93 kg/min to satisfy $\omega=(1/x_{ejo})-1$.

5.1 Mixing Section

Figure 8 shows the experimental results and the predicted values. Predicted values were decided as the intersection of Fig. 6 and Fig. 7. From this figure, it can be seen that once the relationship between suction flow rate and pressure drop of suction flow passage is obtained, the performance of ejector in ejector-equipped cycle can be predicted, if an appropriate value of nozzle efficiency is specified. Nozzle efficiency is 0.8 for this prediction. There is the optimum value for mixing section diameter. In this experimental condition, the optimum value exists between 2mm and 2.5mm. Performance of ejector gets worse drastically, if mixing section diameter is changed by no more than 1mm.

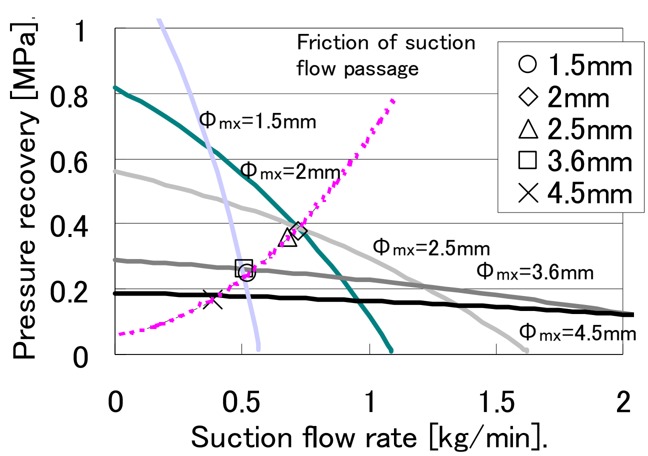

Fig. 8. Predicted and experimental results of ejector

5.2 Expansion Valve

The effect of expansion valve on performance of ejector is investigated. If expansion valve is gradually narrowed, pressure drop through suction flow passage increases, then the slope of the line for pressure drop through suction flow passage is steepened as shown in Fig, 9. Triangle marks in Fig. 9 are experimental results executed for ejector whose mixing section diameter is 2.5mm, where expansion valve is gradually narrowed. When expansion valve is gradually narrowed, suction flow rate decreased and pressure recovery increased according to the calculated results obtained from Eq. (2)-(4). Therefore, when ω is larger than $(1/x_{ejo})-1$, expansion valve should be narrowed to reduce suction flow rate and increase pressure recovery.

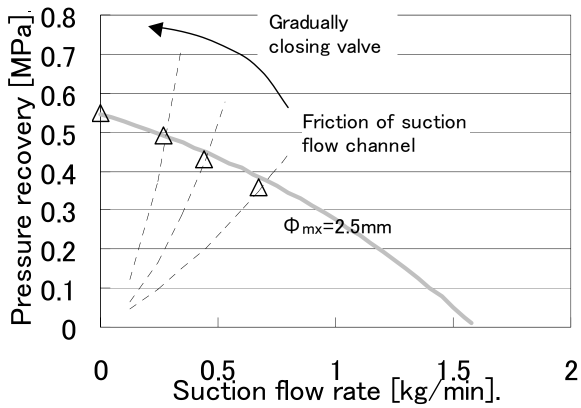

Fig. 9. Experimental results where pressure drop of suction flow passage is increased with expansion valve

5.3 COP Improvement

Estimation of COP improvement with the present experimental setup was executed for three different experimental conditions at nozzle inlet. Driving nozzle was changed to a nozzle with 0.6 mm of outlet diameter for this section, because 0.82 mm of nozzle outlet diameter is too large to satisfy $\omega = (1/x_{ejo})-1$ due to the large pressure drop through suction flow passage. Suction flows were 4.5 MPa 12 °C for all of the three conditions. Mixing section diameter was 2 mm.

Figure 10 shows experimental results of pressure recovery and suction flow rate, and COP improvement calculated by the results. From this figure, it can be seen that ejector without diffuser can improve COP by 11%, which corresponds to 90% of theoretical value for ejector without diffuser. If ejector with diffuser is used, more improvement rate will be expected.

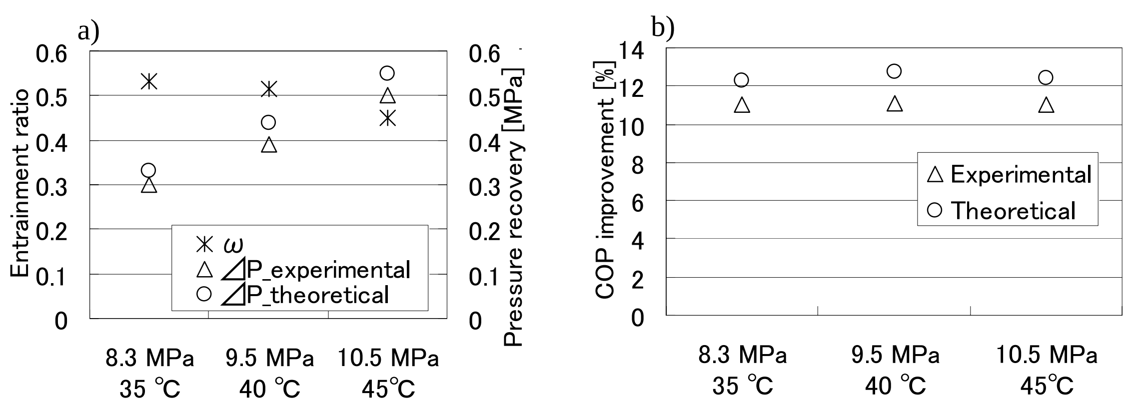

Fig. 10. COP improvement rate
a) Pressure recovery and suction flow rate; b) Calculated COP improvement [%]

6 SUMMARY

Actual performance of ejector in CO_2 compression refrigeration cycle has been investigated. Once the relationship between suction flow rate and pressure drop through suction flow passage is obtained, the performance of ejector in ejector-equipped cycle can be predicted, if an appropriate value is specified as nozzle efficiency.

Effect of mixing section diameter and expansion valve was clarified experimentally. There exists an optimum value for mixing section diameter, which lies between 2 mm and 2.5 mm for present experiment. Expansion valve is useful to adjust ejector performance when ω is larger than $(1/x_{ejo})-1$.

COP improvement of about 11 % was estimated with the present experimental setup, which corresponds to 90 % of the theoretical value.

NOMENCLATURE

A : cross-section area	h : specific enthalpy
$\dot{m}$: mass flow rate	P : pressure
u : velocity	x : quality
ρ : density	ϕ : diameter
ω : entrainment ratio	

SUBSCRIPT

d : driving flow	s, sc : suction flow
no : nozzle outlet	mx : mixing section
mo : mixing section outlet	eo : ejector outlet

REFERENCES

Somjin Disawas, Somchai Wongwises 2004. “Experimental investigation on the performance of the refrigeration cycle using a two-phase ejector as an expansion device,” *International journal of refrigeration*, Vol. 27, pp. 587-594.

Jongsoo Jeong, Kiyoshi Saito, Sunao Kawai, Choiku Yoshikawa, and Kazuhiro Hattori 2004. “Efficiency Enhancement of Vapor Compression Refrigerator Using Natural Working Fluids with Two-phase Flow Ejector,” *Natural Working Fluids Conference*, Glasgow.

Masafumi Nakagawa, Toshiyuki Matumi, Hirotugu Takeuchi, and Naomi Kokubo 1996. “Mixing of the confined jet of mist flow,” *JSME International Journal, Series B*, Vol. 39, No. 2, pp. 381-386.

Yukikatsu Ozaki, Hirotsugu Takeuchi, and Toshio Hirata 2004. “Regeneration of expansion energy by ejector in CO_2 cycle,” *Natural Working Fluids Conference*, Glasgow.