

CHARACTERISTICS OF PRESSURE RECOVERY IN TWO-PHASE EJECTOR APPLIED TO CARBON DIOXIDE HEAT PUMP CYCLE

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Abstract: The aim of this research is to elucidate the characteristics of pressure recovery of a two-phase ejector applied to a carbon dioxide heat pump cycle. The effects of the geometric parameters of the ejector on pressure recovery are investigated in this study. The diameter and length of the constant-area mixing tube are varied within ranges of 2.0–3.6 mm and 2–40 mm, respectively. The maximum pressure recovery is achieved at a distance of approximately 20 mm from the inlet of the mixing tube, irrespective of the diameter of the mixing tube, when the diameter of the nozzle throat is 0.82 mm. In addition, pressure recovery in the mixing section with a diameter of 3.6 mm is less in comparison to those with diameters of 2.0 and 2.5 mm; in the two latter cases, almost the same pressure recovery is observed. The clearance between the nozzle outlet and the inlet of the constant-area mixing tube is varied to compare the constant-area mixing ejector and the constant-pressure mixing ejector. Pressure recovery decreases when the clearance between the nozzle and the inlet of the mixing tube is large. Therefore, the constant-area mixing ejector is preferable for the carbon dioxide heat pump cycle. Finally, the performance of the ejector equipped with a diffuser is investigated; the measured ejector efficiency is 46%, which corresponds to a COP improvement of approximately 23–24%.

Key Words: *Carbon dioxide, Two-phase ejector, Heat pump cycle*

Nomenclature

A : Cross sectional area [m^2]; D : Diameter [m]; h : Enthalpy [J/kg];
 m : Mass flow rate [kg/s]; P : Pressure [Pa]; q_{in} : Cooling capacity per unit mass flow rate [J/kg];
 u : Velocity [m/s]; w_{in} : Compression work per unit mass flow rate [J/kg]; Z : Axial displacement [m];
 η : Isentropic efficiency [-]; ρ : Density [kg/m^3]; ω : Entrainment ratio [-];

Subscripts

cmp: Compressor; cnv: Conventional; d: Driving flow;
 eje: Ejector; ejo: Ejector outlet; ev: Evaporator;
 mx: Mixing section; mxi: Mixing tube inlet; mxo: Mixing tube outlet;
 s: Suction flow.

1 INTRODUCTION

After the commercialization of “Eco Cute” in 2001, the shipment of carbon dioxide heat pump water heaters has increased steadily. The total domestic shipment volume reached 1 million units in September 2007, and the government expects that this volume will increase up to 5.2 million units till 2010. The coefficient of performance (COP) of the heat pump unit increased from 3.46 in the first generation to 4.9 toward the end of 2007. The improvement in the COP is attributed to the improvement in the efficiencies of the components of the heat pump unit, including the compressor and heat exchangers. However, these improvements in the

efficiencies proved to be expensive, and it is not possible to improve the COP significantly. Since the carbon dioxide heat pump is operated under a high-pressure condition, the energy lost during the expansion process is considerably larger than that of heat pump cycles with conventional refrigerant. The introduction of an ejector in the cycle is greatly beneficial to the recovery of the energy lost during the expansion process. In addition, since the density ratio of vapor to liquid is close to unity, the two-phase flow ejector applied to a transcritical carbon dioxide heat pump cycle can easily achieve a high efficiency. Therefore, the introduction of a two-phase flow ejector is expected to be a cost-effective approach for improving the COP of carbon dioxide heat pumps.

Figure 1 shows a schematic of the carbon dioxide heat pump cycle with a two-phase flow ejector, and Fig. 2 illustrates the P-h diagram of the cycle. In this ejector cycle, supercritical carbon dioxide is compressed in the compressor and cooled in the gas cooler, and then it is led to the ejector through a driving nozzle. Inside the driving nozzle, it is depressurized into a two-phase flow and accelerated at a supersonic speed. The supersonic two-phase flow is directed to the mixing section together with the vapor aspirated from the evaporator. These two flows are mixed in the mixing section and the exchange of momentum takes place, and pressure recovery is achieved due to the deceleration of the driving flow. In the diffuser section, the remaining momentum further leads to pressure recovery. The two-phase flow from the outlet of the ejector enters a separator, from which the vapor phase aspirates into the compressor; the liquid phase is depressurized by using an expansion valve and flowed to the evaporator. Due to the mass balance with regard to both the liquid and vapor phases, the entrainment ratio ω —the ratio of the suction flow rate to the driving flow rate—must satisfy Eq.1, where x_{ejo} indicates the quality at the outlet of the ejector.

$$\omega = \left(\frac{1}{x_{ejo}} \right) - 1 \quad (1)$$

Because the suction pressure of the compressor is higher than the pressure at the outlet of the evaporator in the ejector cycle, the COP of the ejector cycle can be improved in comparison to that of the conventional cycle. The cooling COP of the ejector cycle is defined as follows.

$$COP = \frac{m_s q_{out,eje}}{m_d w_{cmp,eje}} \quad (2)$$

Kornhauser [1] analyzed the improvement in theoretical COP of an ejector cycle using R-12 refrigerant. It was found that up to 21% of the conventional cycle can be realized under standard conditions, i.e., the evaporator and condenser temperatures of -15°C and 30°C , respectively, assuming constant-pressure mixing and isentropic components. Nakagawa et al. [2] investigated the influence of geometric parameters on the ejector performance using

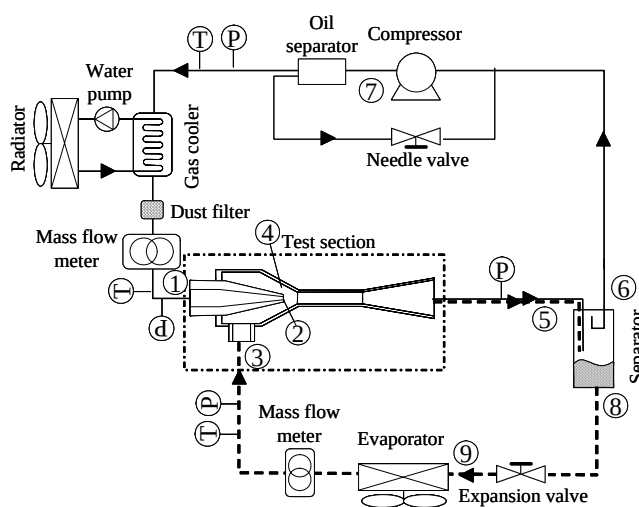


Fig.1: Schematic of the ejector cycle

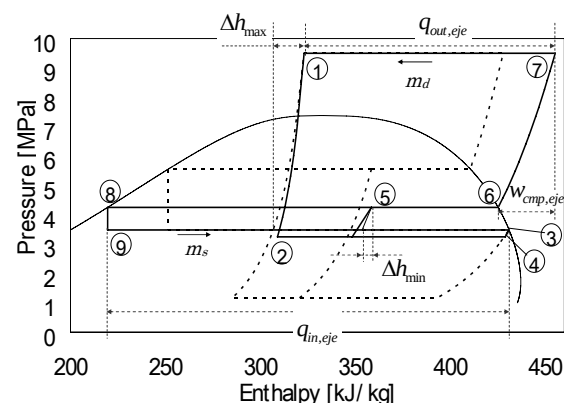


Fig.2: P-h diagram of the ejector cycle

R-12 refrigerant. It was found that the length of the mixing tube must be approximately 200 mm to sufficiently decelerate the liquid phase that had a large amount of inertia under the following conditions: the driving mass flow rate was 240 kg/h; the mixing tube diameter, 9.5 mm; and the entrainment ratio, 0.2. Harrell [3] showed that when the quality at the inlet of the nozzle was 4%, pressure recovery increased with the distance between the nozzle outlet and the constant-area mixing tube in an ejector equipped with a convergent mixing cone, constant-area mixing tube, and divergent diffuser is used. However, when an ejector only equipped with the convergent mixing cone and divergent diffuser was used, pressure recovery decreased by 9%. By using this ejector, pressure recovery decreased as there was an increase in the distance between the nozzle outlet and the constant-area mixing tube. By considering the results of visualization and experiment, it was found that the liquid droplets from the driving nozzle did not enter the surrounding suction flow; however, they remained in the core region, unless recirculation occurred in the mixing section. Jeong et al. [4] investigated the theoretical COP of the ejector-equipped heat pump cycle using ammonia and carbon dioxide as refrigerants. It was found that if the isentropic efficiencies of the driving nozzle, suction nozzle, and diffuser were 0.9, the COP improvement was 5% for the ammonia cycle and 22% for the carbon dioxide cycle. Ozaki et al. [5] compared the effectiveness of the ejector with an expander for the heat pump cycle using carbon dioxide as a refrigerant. It was found that the ejector could improve the COP by a two-stage compression process and recovery of the expansion loss. The maximum improvement in COP was 20%, which was comparable to that of the expander-equipped heat pump cycle. Disawas et al. [6] experimentally examined the improvement in COP using an R-134a refrigerant. It was found that the COP of the ejector cycle was higher than that of the conventional cycle for a wide range of experimental conditions, and the effectiveness increased as the temperature of the heat sink decreased. Li et al. [7] investigated the influence of the design parameters of the ejector using cycle simulation. The isentropic efficiencies of the driving nozzle and suction nozzle were determined from the experimental results according to the constant-pressure mixing model. As a result, the COP and cooling capacity increased by 11% and 9.5%, respectively.

An experimental study has been conducted on the characteristics of the carbon dioxide ejector cycle in our laboratory. A previous report [8] revealed the relationship between the suction flow rate and the pressure drop throughout the suction flow passage. The influence of mixing section diameter and the expansion valve were examined; the optimum diameter of the mixing section was found to be between 2 mm and 2.5 mm. The tested ejector was found to enhance the COP of the cycle up to 11%. In this report, more experimental results obtained with regard to pressure recovery in two-phase flow ejectors are discussed by focusing on the effect of the lengths of mixing section and nozzle outlet on pressure recovery.

2 OUTLINE OF EXPERIMENTS

Figure 3 shows a schematic diagram of a test ejector, in which the nozzle and mixing section can be changed. The pressure, temperature, and mass flow rate are measured to determine pressure recovery and ejector efficiency. The uncertainties of the mass flow rate, pressure, and temperature are 0.2%, 0.2%, and 0.1°C, respectively.

In this paper, unless otherwise stated, the driving nozzle is a convergent nozzle with a throat diameter of 0.82 mm; the pressures at the nozzle inlet and entrance for suction flow are 9 MPa and 4.5 MPa, while the temperatures are 35°C and 12°C, respectively. The distance between the nozzle outlet and the mixing tube inlet is 2.0 mm, and the expansion valve in the suction flow passage is fully opened.

In this experiment, three mixing sections with different mixing tube diameters (2.0, 2.5, and 3.6 mm) are examined to investigate the effect of the mixing tube diameter on pressure recovery. The length of the mixing section is varied within the range of 2–40 mm in order to determine the optimum length of the mixing tube. Subsequently, the nozzle position is varied

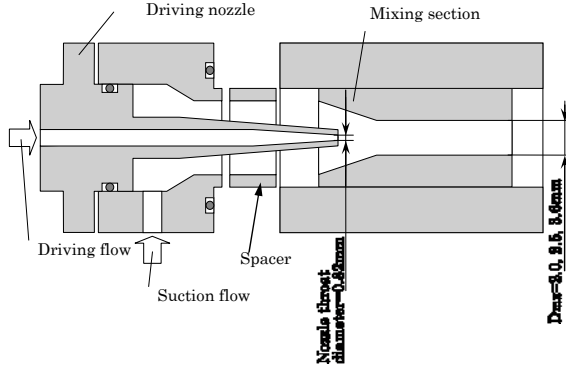


Fig.3: Experimental apparatus of the test

measured using the diffuser, and the ejector efficiency and COP improvement are estimated. In this paper, the ejector efficiency η_{eje} is defined in Eq.3. $\Delta h_{\max}$ denotes the maximum enthalpy change assuming isentropic expansion from the nozzle inlet to the evaporation pressure. $\Delta h_{\min}$ denotes the minimum enthalpy change required to reach the condition of the ejector outlet from the evaporation pressure assuming isentropic compression.

$$\eta_{eje} = \frac{(m_d + m_s) \Delta h_{\min}}{m_d \Delta h_{\max}} \quad (3)$$

3 ONE-DIMENSIONAL SIMULATION MODEL OF EJECTOR

The velocities at the outlet of the driving nozzle and suction nozzle are calculated using Eq.4, where $\Delta h_{\text{isentropic}}$ denotes the enthalpy difference, assuming an isentropic process; η_{nzt} is the isentropic efficiency defined as the ratio of the actual enthalpy change to $\Delta h_{\text{isentropic}}$.

$$u = \sqrt{2 \Delta h_{\text{isentropic}} \cdot \eta_{nzt}} \quad (4)$$

For the mixing cone, the exchange of momentum and energy between the driving flow and the suction flow is neglected because of the small clearance of the nozzle outlet, and the condition of the mixing cone outlet is determined by the same method as that used to determine the condition of the nozzle. The isentropic efficiencies of the driving nozzle, suction nozzle, and mixing cone are assumed to be the same.

In the mixing tube, the outlet condition is calculated using Eqs.5–7, assuming that the sufficient mixing of the driving flow and suction flow takes place and the velocity is homogeneous at the mixing tube outlet. Equations 5–7 are the conservation laws of mass, momentum, and energy, respectively. In Eq.6, Z_{mx} denotes the mixing tube length and f_w denotes the frictional force per unit volume obtained using Colburn's equation.

$$\frac{m_d}{\rho_d u_d} + \frac{m_s}{\rho_s u_s} = A_{mx} = \frac{m_d + m_s}{\rho_{mso} u_{mso}} \quad (5)$$

$$m_d u_{d,mxi} + m_s u_{s,mxi} + P_{mxi} A_{mx} + f_w A_{mx} Z_{mx} = (m_d + m_s) u_{mso} + P_{mso} A_{mx} ,$$

$$f_w = - \frac{4}{D_{mx}} \frac{\rho_{mso} u_{mso}^2}{2} \frac{0.046}{Re_{mso}^{0.2}} \quad (6)$$

$$m_d \left(h_{d,mxi} + \frac{u_{d,mxi}^2}{2} \right) + m_s \left(h_{s,mxi} + \frac{u_{s,mxi}^2}{2} \right) = (m_d + m_s) \left(h_{mso} + \frac{u_{mso}^2}{2} \right) \quad (7)$$

The outlet condition of the ejector without the diffuser is also obtained by using the conservation laws of mass, momentum, and energy on the basis of the assumption that the

by replacing the spacer in order to compare the performances of the constant-area mixing ejector and constant-pressure mixing ejector for the same diameter of the mixing tube. In the constant-area mixing ejector, the nozzle outlet is set in the constant-area mixing tube. On the other hand, in the constant-pressure mixing ejector, the nozzle outlet is set in the convergent mixing cone; moreover, the pressure is assumed to be constant between the nozzle outlet and the inlet of the constant-area mixing tube. Finally, pressure recovery can be

pressure at the section in which sudden expansion takes place just behind the mixing tube is constant, the frictional loss is negligible, and the velocity is homogeneous at the ejector outlet.

4 RESULTS AND DISCUSSION

Figures 4 and 5 show the variations in pressure recovery and suction flow rate against mixing tube length for three mixing tube diameters (2.0, 2.5, and 3.6 mm) in the case of ejectors without the diffuser. Figure 6 shows the relation of the suction flow rate with pressure recovery, as obtained from the data shown in Figs. 4 and 5. Because pressure recovery in the ejector is the same as the frictional loss along the suction flow passage, which is the flow path from the ejector outlet to the entrance for suction flow passing through the separator and evaporator. Figure 6 also represents the relation between the suction flow rate and the frictional loss along the suction flow passage.

In Figs. 4 and 5, the optimum mixing tube length is approximately 20 mm, regardless of the mixing tube diameter. When the mixing tube length is less than the optimum value, pressure recovery drastically decreases because the two streams exit from the mixing section before they mix sufficiently. On the other hand, in the case of a longer mixing tube, pressure recovery gradually decreases due to the frictional loss in the mixing tube.

A one-dimensional simulation is executed, and its results are compared with the experimental results in Fig. 7. The suction flow rate is calculated from pressure recovery using the relation shown in Fig. 6. From the comparison shown in Fig. 7, it is found the calculated results are in good agreement with the experimental results when the comparison is performed under a condition where sufficient mixing takes place ($Z_{mx} \geq 20$ mm) with isentropic efficiencies of the driving nozzle and the suction nozzle assumed in the range of 0.7–0.8. If the mixing tube length is smaller than 20 mm, the calculated results differ from the experimental results due to insufficient mixing.

It is also observed in Figs. 4 and 5 that pressure recovery in the mixing section with a diameter of 3.6 mm is less than that at diameters of 2.0 and 2.5 mm. In the latter two conditions, almost the same pressure recovery is obtained.

The effect of the mixing tube diameter is examined as shown in Figs. 8 and 9. In Fig. 8, the downward-sloping curve represents the variations in pressure recovery with the suction flow rate calculated using the one-dimensional model under the following conditions: the mixing tube length is 40 mm and the isentropic efficiencies of the driving nozzle and suction nozzle are 0.8. The upward-sloping curve represents the relation between the suction flow rate and the frictional loss along the suction flow passage. The intersection of the downward-sloping curve and the upward-sloping curve indicates the predicted pressure recovery. Figure 9 shows the variation in the velocity at the inlet and outlet of the mixing tube with the suction flow rate.

In Fig. 8, it is observed that pressure recovery in the mixing tube reduces when the mixing tube diameter increases for a low suction flow rate. This is based on the conservation law of momentum (cf. Eq.6) as follows. From this equation, it is found that pressure recovery in the mixing tube is the difference between the kinetic momentum of the inlet and outlet of the mixing tube divided by the cross-sectional area of the mixing tube. This is the reason why the mixing section with a large diameter shows low pressure recovery.

In Fig. 9, it is observed that with a decrease in the mixing tube diameter or an increase in the suction flow rate, the velocity of the suction flow increases at the mixing tube inlet and that of the mixed flow increases at the mixing tube outlet. Therefore, there is a decrease in the loss of kinetic momentum by the deceleration of the driving flow; consequently, pressure recovery in the mixing tube reduces. This is the reason why the mixing tube with a diameter of 2.0 mm shows almost the same pressure recovery as that of the mixing tube with a diameter of 2.5 mm.

Figure 10 shows the variations in pressure recovery with the nozzle outlet clearance. From this figure, it is found that pressure recovery reduces with an increase in the nozzle outlet

clearance. Therefore, the constant-area mixing ejector is preferable to the constant-pressure mixing ejector for the same mixing tube. When the nozzle outlet clearance increases, the mixing of the driving flow with the suction flow is not negligible in the mixing cone. As mentioned above, when the loss of kinetic momentum occurs, pressure recovery for a smaller cross-sectional area is larger than that for a larger cross-sectional area. Since the mixing cone has a larger cross-sectional area than that of the mixing tube, the mixing in the mixing cone region—instead of mixing tube—leads to a lower pressure recovery. This is the reason why the increase in the nozzle outlet clearance has a negative effect on pressure recovery. A detailed comparison between the constant-area mixing ejector and constant-pressure mixing ejector requires the two-dimensional simulation of the mixing section, which will be presented in other studies.

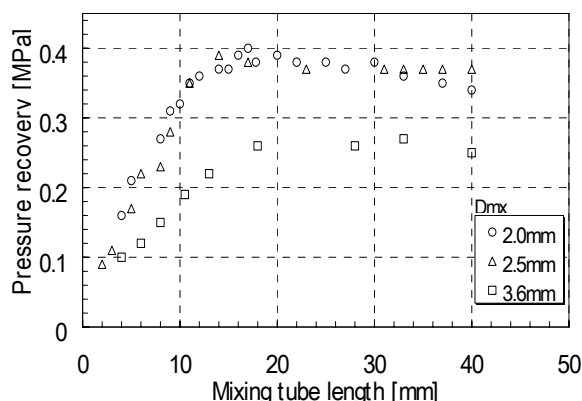


Fig.4: Influence of length of mixing tube on pressure recovery

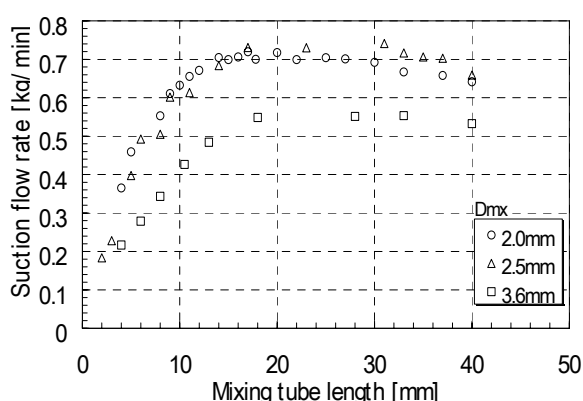


Fig.5: Influence of length of mixing tube on suction flow rate

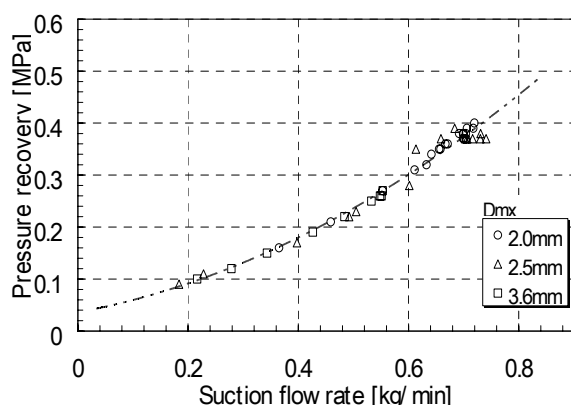


Fig.6: Relation of pressure recovery with suction flow rate with the expansion valve fully opened

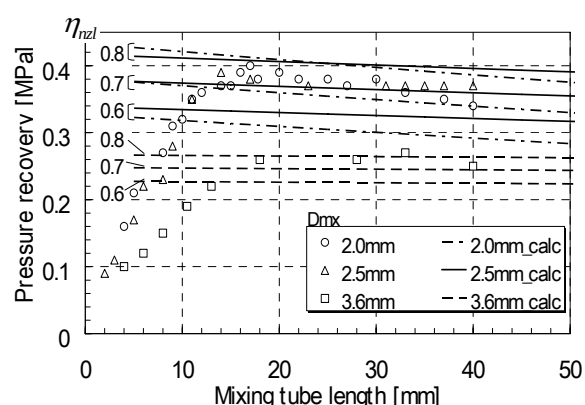


Fig.7: Comparison between experimental and calculated results

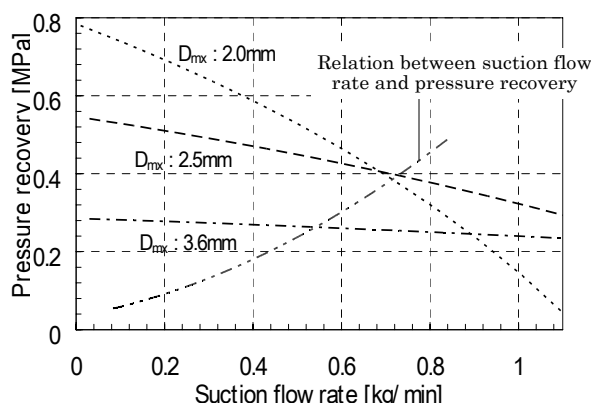


Fig.8: Effect of mixing tube diameter on pressure recovery

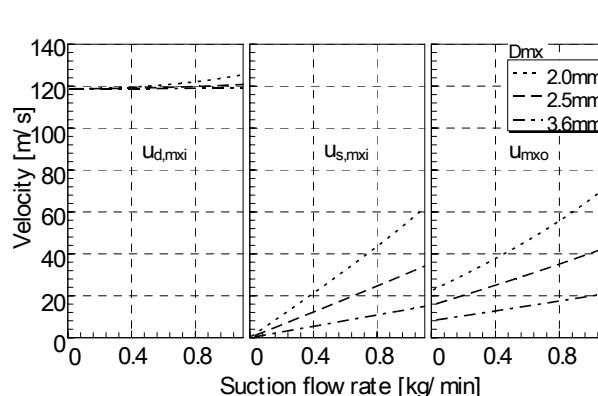


Fig.9: Effect of diameter of mixing tube on velocities at inlet and outlet of mixing tube

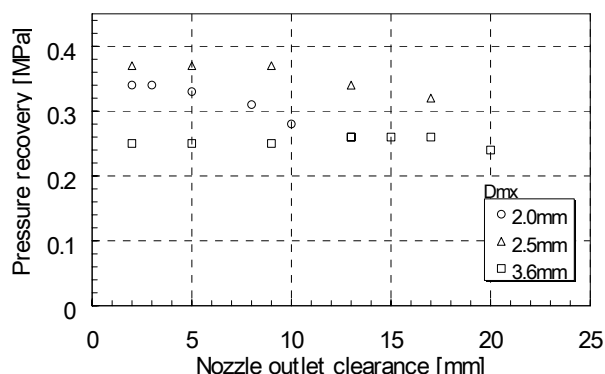


Fig.10: Effect of the nozzle position on pressure recovery

In order to estimate the ejector efficiency and the improvement in COP by using the ejector with the diffuser, pressure recovery is measured using this ejector. The diameter and length of the mixing tube is 2 mm and 23 mm, respectively, and the diffuser angle is 8°. In this experiment, the expansion valve was adjusted in order to satisfy Eq. 1. Table 1 shows the experimental conditions and the results. The COP improvement is estimated on the basis of the assumption that the compressor efficiency is unity; the frictional loss is neglected in the cycle. As shown in this table, the efficiency of the present ejector is 0.46, which corresponds to a COP improvement of 23–24%.

Table 1; Experimental conditions and results

Parameter	Value		
Experimental conditions			
Pressure at nozzle inlet [MPa]	8.3	9.5	10.5
Temperature at nozzle inlet [°C]	35	40	45
Pressure at entrance for suction flow [MPa]	4.5		
Temperature at entrance for suction flow [°C]	12		
Experimental result			
Driving flow rate [kg/min]	1.24	1.44	1.60
Suction flow rate [kg/min]	0.66	0.72	0.70
Entrainment ratio [-]	0.53	0.50	0.44
Pressure recovery [MPa]	0.57	0.75	0.96
Ejector efficiency [-]	0.46	0.46	0.46
Calculated COP of conventional cycle [-]	4.91	3.72	2.88
Calculated COP of ejector-equipped cycle [-]	6.04	4.61	3.57
Calculated COP improvement [%]	23.0	23.9	24.0

5 CONCLUSIONS

Experimental measurements were conducted to investigate the characteristics of pressure recovery in the two-phase flow ejector, and the following conclusions were obtained.

1. Pressure recovery is achieved at a length of approximately 20 mm from the inlet of the mixing tube, irrespective of the mixing tube diameter, when the diameter of the nozzle throat is 0.82 mm. The experimental results satisfactorily fit the one-dimensional simulation results when the efficiencies of the driving nozzle and suction nozzle are assumed to be in the range of 0.7–0.8.
2. Pressure recovery in the mixing section with a diameter of 3.6 mm is lower than that at diameters of 2.0 and 2.5 mm; in the latter two conditions, almost the same pressure recovery are obtained.
3. The constant-area mixing ejector is preferable to the constant-pressure mixing ejector for the two-phase flow ejector applied to the carbon dioxide heat pump cycle.
4. The ejector efficiency and COP improvement obtained by using the carbon dioxide two-phase flow ejector with the diffuser are 0.46 and 23–24%, respectively.

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