EXPERIMENTAL RESEARCH ON THE TWO-PHASE FLOW NOZZLE PERFORMANCE OF THE EJECTOR FOR CARBON DIOXIDE

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ABSTRACT

Natural refrigerant such as carbon dioxide which is a safe refrigerant without toxicity and flammability attracts attention from a viewpoint of environmental load-reducing. However, the energy loss at the expansion valve in carbon dioxide refrigerant cycle is three times larger than that of the usual cycle using refrigerant R-134a. The two-phase ejector is one of those devices which improve Coefficient Of Performance (COP) by using the expansion energy of the refrigerant.

The object of the present paper is to elucidate the two-phase flow nozzle performance of carbon dioxide from the experiment using the precise measurement of the pressure along the nozzle.

It is found by the present experiment that the maximum energy conversion efficiency of two-phase flow nozzle for carbon dioxide is about the order of 95%. The experimental results show that there is some suitable length in divergent section of two-phase flow nozzle. In the shortest nozzle, the pressure decreases from the saturation are measured at the nozzle throat because of requirements for large superheat of liquid refrigerants. Those experimental results will be utilized for the design of the two-phase flow ejector using carbon dioxide as a refrigerant.

INTRODUCTION

In recent years, environmental problems, such as global warming, and energy saving have been attracting our increasing attention. An alternative refrigerant used for air-conditioner such as R134a has a large Global Warming Potential which indicates a coefficient for each substance that constitutes a greenhouse gas as a ratio to that of carbon dioxide. By the way, natural refrigerant such as carbon dioxide which is a safe refrigerant without toxicity and flammability attracts attention from a viewpoint of environmental load-reducing. However, the energy loss at the expansion valve in carbon dioxide refrigerant cycle is three times larger than that of the usual cycle using refrigerant R-134a, since the compressibility of carbon dioxide is low. Then the needed compressor work becomes greater, and the expansion energy loss turns out to be larger. As the results, the COP of the refrigeration cycle became considerably low. In order to use carbon dioxide as a refrigerant, new device must be developed to raise the COP.

The two-phase ejector [1] is one of those devices which improve COP by using the expansion energy of the refrigerant. We have been researching on the two-phase ejector that recovers the energy lost during expansion process for a carbon dioxide refrigerant cycle [2]. It has been elucidated from the previous study that the conversion efficiency of the nozzle in the two-phase ejector is important.

For the heat pump hot-water supply system in which the carbon dioxide is used as refrigerant, the high efficiency ejector is hoped. The super critical water cooler is indispensable in this refrigeration device, because of avoidance from pinch point at heat exchanger. The research and development of carbon dioxide ejector system are expected in this field. High efficiency super critical nozzle is also needed in this system.

In previous experiments, the temperature of pressure taps had been measured, and converted to the pressure by using the saturation relationship between the pressure and the temperature [3][4][5]. It had been postulated that two-phase equilibrium is maintained in the pressure taps where both liquid and vapor exist. However, the response and the accuracy of the thermocouples are not so high, and it is thought some error occurs from the conversion process.

The object of the present paper is to elucidate the two-phase flow nozzle performance of carbon dioxide from the experiment using the precise measurement of the pressure along the nozzle.

NOMENCLATURE

A	[m^2]	Flow area
$A*$	[m^2]	Flow area at throat
C_D	[-]	Coefficient of discharge
COP	[-]	Coefficient of performance
D	[m]	Characteristic Diameter
f	[-]	Friction factor
G	[kg/s]	Mass flow rate
Δh	[J/kg]	Specific enthalpy difference
L	[m]	Length of the divergent part of the nozzle
L	[m]	Perimeter
P	[Pa]	Pressure
Re	[-]	Reynolds number
S	[m]	Specific entropy
T	[K]	Temperature
U	[m/s]	velocity
Z	[m]	Coordinate along flow direction

Special characters

ρ	[kg/m^3]	Density
μ	[Pa s]	Viscosity
τ_f	[N/ m^2]	Frictional stress at wall
η	[m^2/s]	Kinetic viscosity

Subscripts

g	Gas phase
l	Liquid phase
in	Inlet
out	Out let
th	Throat
z	At arbitrary position in the nozzle

EXPERIMENT
Experimental apparatus

The experimental apparatus shown in Figure 1 is fundamentally the same as the conventional refrigerant cycle with expansion valve, but the two-phase nozzle is incorporated into the cycle instead of the expansion valve. It consists of the compressor, the water cooling condenser, the two-phase nozzle and the air heating evaporator.
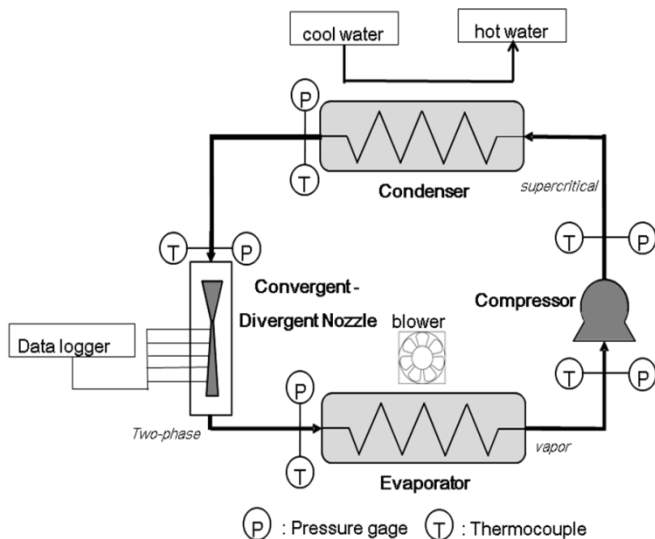

Figure 1 Experimental apparatus

The compressor is controlled by the inverter, and the rotational speed is arbitrary set. The super critical refrigerant is not really condensed in the condenser, but is cooled down by water to get the super critical liquid phase.

Then the temperature of the inlet and the outlet water are measured to obtain the enthalpy drop of refrigerant. The evaporator heated by the atmosphere air using the blower. The rotational speeds of blower are arbitrarily adjusted. The experimental inlet conditions of the nozzle are controlled by those parameters, and are maintained at the constant value.

Two-dimensional convergent-divergent nozzles

Two-phase nozzles with rectangular cross section are designed for the ejector of air conditioning system available to summer season. In this study, three nozzles, of the same throat and outlet areas, but of different length at divergent section, are tested. The throat area is 0.42 [mm^2] and outlet areas are 0.65 [mm^2]. The lengths of divergent section L are 10, 15 and 20 [mm], for Nozzles 1, 2 and 3, respectively. There is 1 pressure tap at convergent section of the nozzle and 5 taps at divergent section.
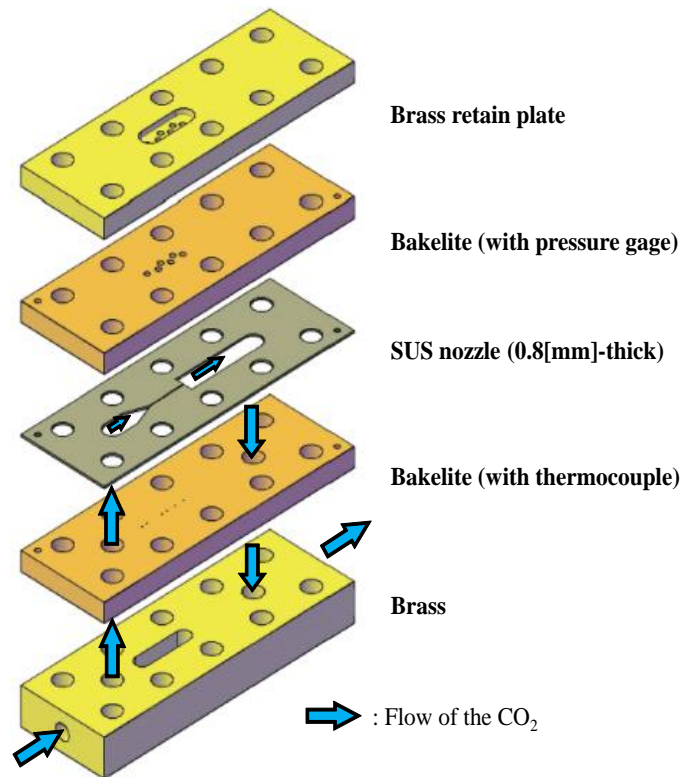

Figure 2 Assembly of the nozzle

The assembly of two-phase nozzle is composed of one convergent-divergent nozzle plate, two Bakelite plates outside bronze plates. Pressure taps are bored trough the Bakelite plate. The narrow tubes from pressure taps are brazed on the bronze plate and connected to the pressure transducers. The pressure and the temperature at the inlet nozzle and mass flow rates are

adjusted by controlling the rotational speed of compressor by using the inverter, the cooling capacity of condenser by changing water flow rate and the heating capacity of evaporator by changing air flow rate. The inlet pressure ranges from 7 to 10[MPa], and the temperature from 35 to 40[deg C]. The measurements are carried out when the pressure and the temperature at every point became stable for each step. Then, we calculated the energy conversion efficiency of the nozzle from measured pressure distributions. The conversion efficiency of the nozzle is defined as the ratio of the converted kinetic energy at nozzle outlet to the maximum work put out from inlet to outlet condition of nozzle which is equal to isentropic enthalpy drop.

RESULT

The experiments are carried out by using the nozzles which have three different lengths of divergent section. The length of longest nozzle is 20[mm], middle is 15[mm] and shortest is 10[mm]. The pressure distributions along those convergent-divergent nozzles are obtained from the output of the pressure transducer.

Pressure distributions along the nozzle

The measured pressure distributions for the longest nozzle (L=20[mm]) are plotted in Figure 3. The abscissa of the graph indicates the ratio of the flow area to that at throat (A/A^*). Then, A/A^*=1 corresponds to the throat of the nozzle. The inlet temperature of nozzle is set at constant T_{in} = 40[deg C], and the pressures are changed.

The mass flow rate is almost proportionally increased with the inlet pressure. The distribution curves are plotted by changing the mass flow rates [g/s] instead of the inlet pressure.
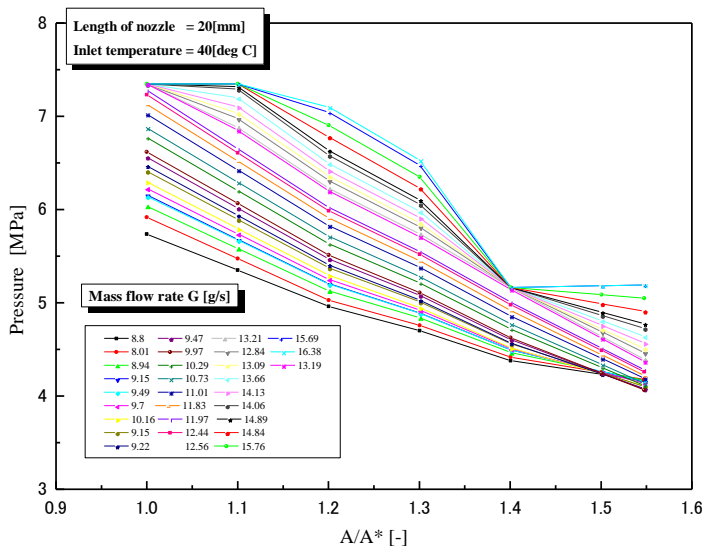
The pressure is almost linearly decreased at small mass flow rate, while approaches horizontal line near the throat at large mass flow rate. This behaviour is same for other refrigerants, R-134a, isobutene and water. The pressure increase at the end of divergent section appears at large mass flow rate. This is thought that two-phase flow shock wave goes up to nozzle.

In Figure 4, the pressure distributions for the nozzle with the middle length (L=15[mm]) are plotted. The pressure distributions near the throat are almost the same as that for longest shown in Figure 3. It looks no shock wave is appeared.

The results of the shortest nozzle are plotted in Figure 5. The pressure distribution near the throat changes almost linear with our experimental condition. In this case, the pressures at the throat are always below the critical pressure 7.37 [MPa].
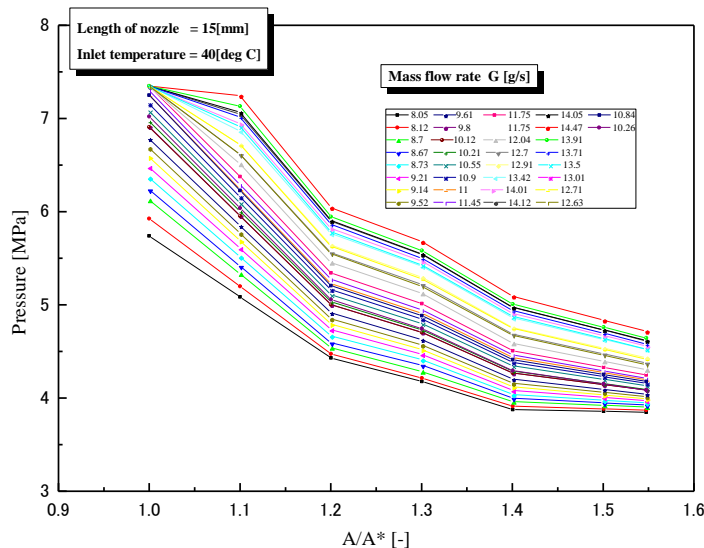

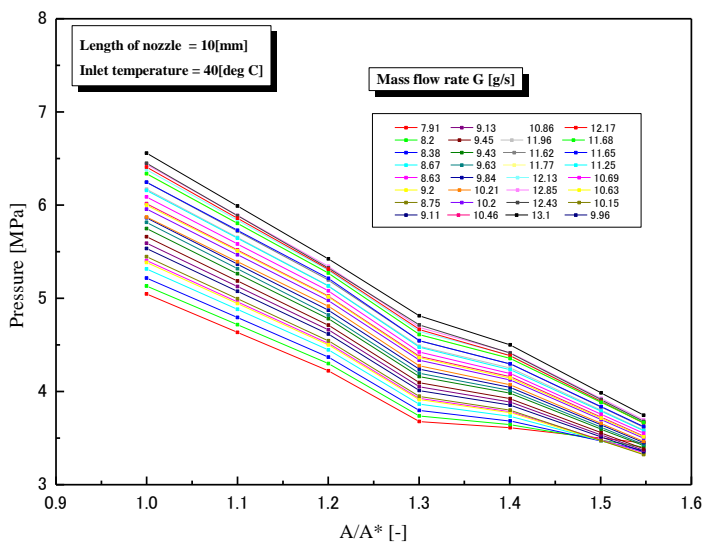**Figure 4** Pressure distributions along the nozzle (L=15[mm])

Figure 3 Pressure distributions along the nozzle (L=20[mm])

Figure 5 Pressure distributions along the nozzle (L=10[mm])

Critical Mass Flow Rate

As seen from the pressure distributions, the pressure at the throat not affect by the outlet pressure of the nozzle. This means that the supersonic flow is established in the divergent section and the mass flow rate is not depending on the outlet pressure. We call this mass flow rates as the critical mass flow rate.

The critical mass flow rates are plotted against the inlet pressure in Figure 6 and 7. All experimental results with the inlet temperature of 40[deg C] are plotted in Figure 6, they look varying along the one curve. Three nozzles are different in divergent section, but same in convergent section and throat. The velocity profiles of those three nozzles are same in convergent section. The critical mass flow rate predicted by Isentropic Homogenous Equilibrium model is plotted by solid curve. The mass flow rate is low compared with this theory.
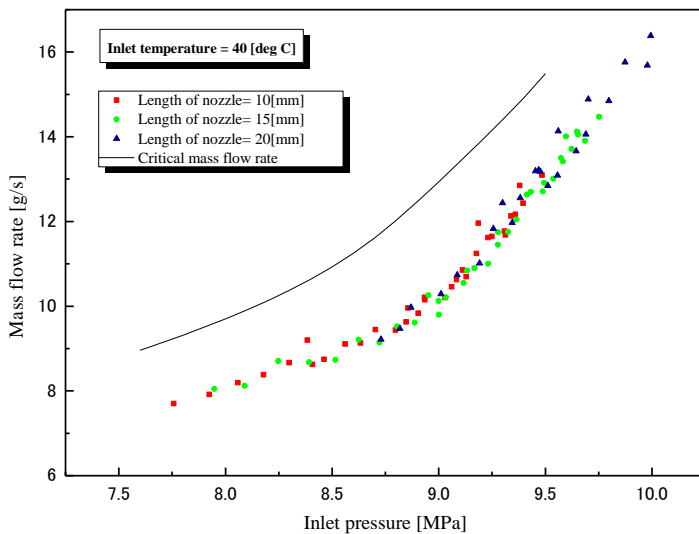
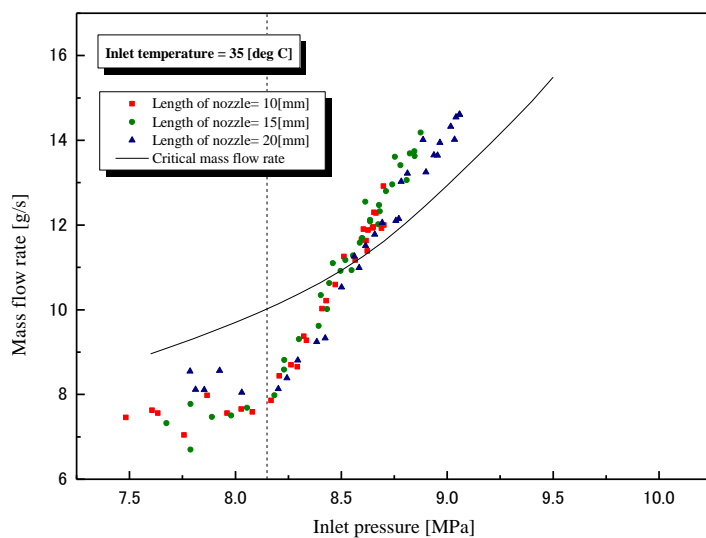In Figure 7, the critical mass flow rate with the inlet temperature of 35[deg C] are plotted. At high pressure, it is expected that the boundary layers for friction is considerably thick at the throat, and flowing cross area became small compare with real one. Experimental results exceed the theoretical one. If the inlet pressure become higher and the inlet temperature became lower at supercritical state, inlet entropy become smaller than that at critical point. In Figure 7, the vertical line shows the condition when the inlet entropy equals to critical one. The experimental curve changes the slope at this point. The inlet conditions in Figure 6 always have smaller entropy than that at critical point. When the heavy liquid phase is produced by the phase change, the liquid phase cannot be accelerated by the decompression boiling, and non-equilibrium phenomena are thought to be promoted.

Coefficient of Discharge of the Nozzle

As mentioned in previous section, the flow patterns in convergent are same for all nozzle. It is well known that the performance of the convergent nozzle is expressed by the coefficient of discharge defined following equation.

$$C_D = \frac{G}{\rho_{th} A \sqrt{2 \Delta h}} \tag{1}$$

Where ρ_{th} and A are the density and the flow area at the throat, respectively. Δh is adiabatic enthalpy drop between inlet and throat. This ideal enthalpy drop is calculated by using the measured inlet and throat pressures. It is maximum work done by the pressure deference between inlet and throat.

In Figure 8, the coefficients of discharge of all data are plotted. Those look to have the mean value about 0.8. But the coefficient of discharge increase slightly with the mass flow rate. The basic textbook for hydrodynamics tells us that the value of the C_D increase with the Reynolds number. And it may be so in this case.
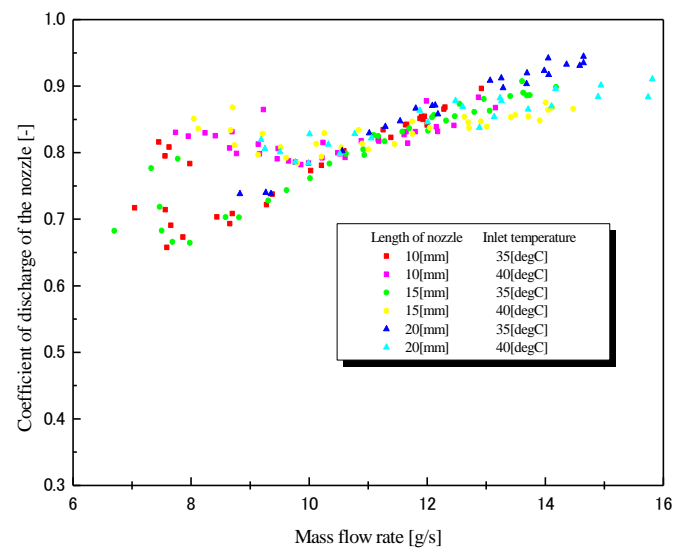

Figure 6 Critical flow rate at inlet temperature of 40[deg C]

Figure 7 Critical flow rate at inlet temperature of 35[deg C]

Figure 8 Coefficient of Discharge of the Nozzle

Energy Conversion Efficiency of the Nozzle

The main function of the nozzle in the ejector system is to convert the thermal energy into the kinetic energy. Therefore, the nozzle with high energy conversion efficiency required for high performance ejector.

The energy conversion efficiency of the nozzle is defined as the ratio of the kinetic energy at outlet to the maximum work expected for the nozzle. This is defined by following equation.

$$\eta = \frac{u_{out}^2/2}{\Delta h} \tag{2}$$

Where u_{out} is nozzle outlet velocity and Δh is adiabatic enthalpy drop between nozzle inlet and outlet. The enthalpy drop is calculated from measured pressure deference by using the thermodynamic table presented by NIST.

The discharge flow from the nozzle outlet is high speed two-phase flow. We estimate the outlet velocity of two-phase flow by momentum increase at the nozzle, because the velocity measurement for two-phase flow is quite difficult. The velocity u_z at arbitrary position z is presented by following momentum equation along the nozzle.

$$u_z = u_{th} - \frac{1}{G}\left\{\int_{z_{th}}^{z}\left(A\frac{dp}{dz} + \tau_f l_z\right)dz\right\} \tag{3}$$

Where u_{th} is the velocity at throat, G is the mass flow rate, A is the flow area of the nozzle, τ_f is the frictional stress on the wall and l_z is the perimeter at z. As the flow is assumed to be homogeneous, the homogeneous friction coefficient for the two-phase flow is adopted in this study. The frictional stress is expressed by following equation.

$$\tau_f = f_z \frac{\rho_z u_z^2}{2D} \tag{4}$$

Where f_z is the friction coefficient, ρ_z is mean density at z and D is the characteristic diameter ($=4A/l_z$). The frictional confident is calculated by

$$f_z = 0.0791\mathrm{Re}_z^{-\frac{1}{4}} \tag{5}$$

And Reynolds number and the mean value at z are estimated by next equations

$$\mathrm{Re}_z = \frac{u_z D \rho_z}{\mu_z} \tag{6}$$

$$\frac{1}{\mu} = \frac{x}{\mu_g} + \frac{1-x}{\mu_l} \tag{7}$$

$$\frac{1}{\rho_z} = x\frac{1}{\rho_g} + (1-x)\frac{1}{\rho_l} \tag{8}$$

$$x = \frac{s(T_{in}, p_{in}) - s_l(T)}{s_g(T) - s_l(T)} \tag{9}$$

The quality needed for the calculation of the mean value is roughly estimated by that of isentropic change. The pressure gradient dp/dz appears in the equation (3). But, it is changed to the slope of area dA/dz by using partial integral, because derivative of the measured value cause large error. The iterating calculations are executed until the steady velocity and pressure profiles are obtained, because u_z is complexly included in every equation.

The energy conversion efficiency obtained from the measured pressure distributions are plotted in Figure 9. They are shown as the function of the mass flow rate. As the maximum are appears for each nozzle, the optimum flow rate exists for each nozzle. Usually, the over expansion is observed for smaller flow rate and the under expansion for larger.

The maximum efficiency of two long nozzles is high respectively, but that of shortest one is considerably small. The rapid expansion occurs in the shortest nozzle. Then the phase change cannot follow the quick change of the saturation temperature. This is obviously shown in the temperature distributions shown in Figure 5. The temperatures at the throat are much lower than others. The large super heat of the refrigerant is required by the rapid phase change.

In Figure 10, the energy conversion efficiencies of the nozzles are shown at inlet temperature of 35 [deg C]. For two longer nozzles, the maximum efficiency is not obtained within our experimental conditions. But at large flow rate, experimental results show the large efficiency, and the maximum efficiency may take place where the flow rate is slightly higher than our experimental range.

The shortest nozzle also has bad efficiency. The transport phenomena with phase change are thought to be late for depression speed.
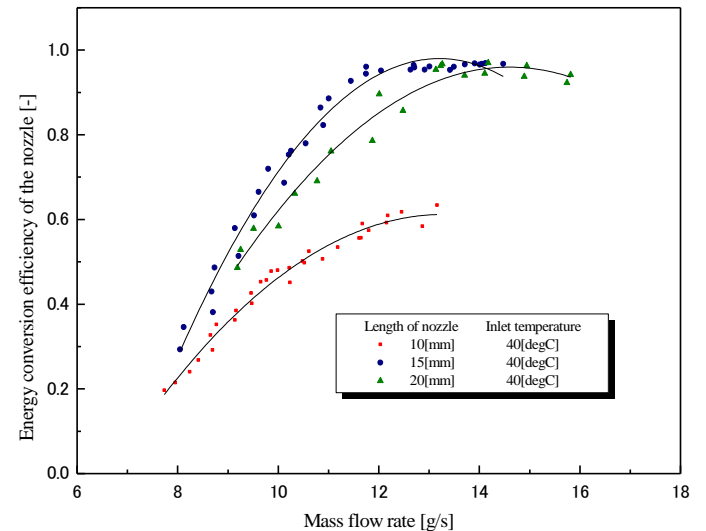

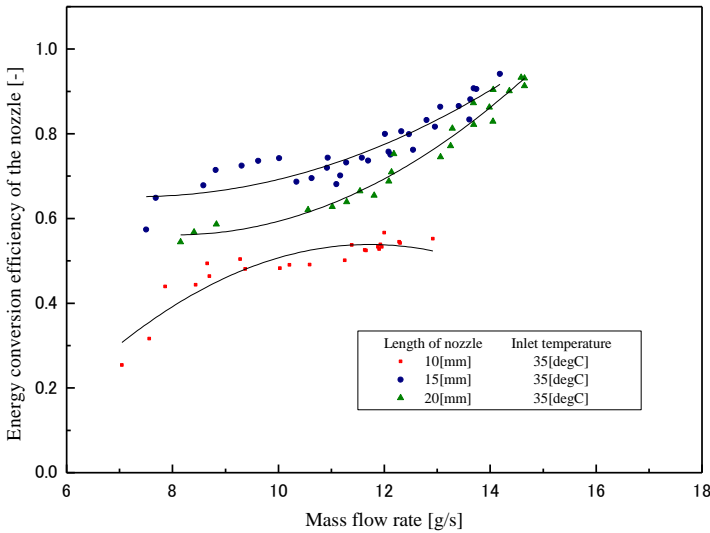**Figure 9** Nozzle efficiency at inlet temperature of 40[deg C]

Figure 10 Nozzle efficiency at inlet temperature of 35[deg C]

Dependence of the nozzle length

The dependence of the nozzle length on the energy conversion efficiency are shown in Figure 11. Though two sets of data at the different inlet temperatures are simultaneously plotted on the same figure, there are no distinction between two data. The maximum efficiency is almost same for the different inlet temperature. Longer two nozzle have large efficiency. The nozzle with some length between 15 and 20 [mm] has the opium efficiency.
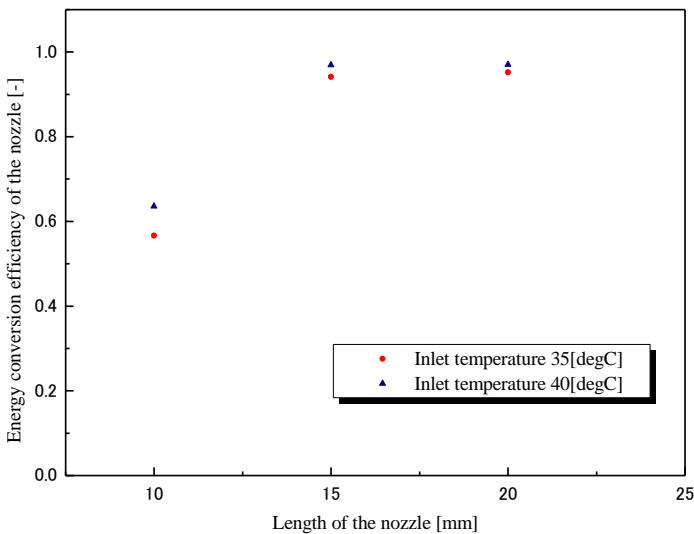

Figure 11 Nozzle efficiency at inlet temperature of 40[deg C]

CONCLUSIONS

The experimental study is carried out to elucidate the two-phase flow nozzle performance of carbon dioxide for the ejector refrigeration cycle.

It is found by the present experiment that the maximum energy conversion efficiency of two-phase flow nozzle for carbon dioxide is about the order of 95%. The experimental results show that there is some suitable length in divergent section of two-phase flow nozzle. In the shortest nozzle, the pressure decreases from the saturation are measured at the nozzle throat because of requirements for large superheat of liquid refrigerants. Those experimental results will be utilized for the design of the two-phase flow ejector using carbon dioxide as a refrigerant.

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