Mixing of the Confined Jet of Mist Flow*

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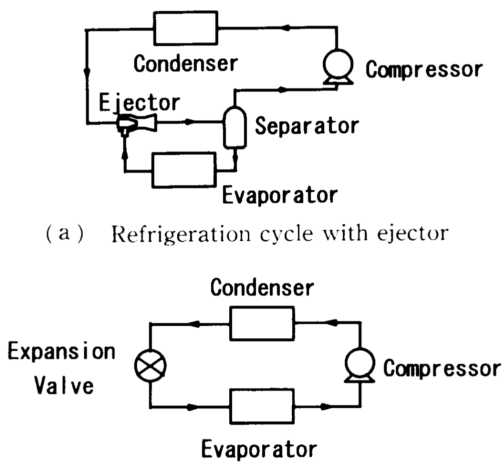
Confined jets of two-phase mist flow are important in the development of a two-phase ejector for the refrigeration cycle. However, the flow characteristics of the two-phase ejector have not been elucidated to date due to the nonequilibrium of velocity and temperature. In this study, the mixing characteristics of two-phase mist flow in the two-phase ejector were investigated experimentally. The pressure increases in the mixing section and the diffuser of the ejector were measured. The following results were obtained by comparison of the measured pressure increase with that calculated using a simple theory. Increasing the length and decreasing the diameter of the mixing section were effective for raising the pressure. The energy efficiency of the two-phase ejector used in this experiment was approximately 10%. This efficiency should be increased by improving the mixing characteristics of the ejector and the nozzle efficiency.

Key Words: Pipe Flow, Mutiphase Flow, Refrigeration, Jet, Diffuser, Nozzle

1. Introduction

Confined jets of steam are of engineering importance in the vacuum pumps of the condensers in a power plant, and in jet pumps for water in boiling water reactors. The mixing performance of the primary and secondary streams in a duct for single phase flow has been investigated[1),(2)]. Recently, confined jets of two-phase mist flow are also of importance in developing two-phase ejectors used in the refrigeration cycle. For this new cycle[(3)], the two-phase ejector is replaced by an expansion valve. Schematics of this cycle (a) and the conventional one (b) are shown in Fig. 1. A refrigerant is expanded isoenthalpically in an expansion valve in the conventional cycle, while it expands almost isoentropically in the nozzle of an ejector. Moreover, the energy dissipated in the conventional cycle can be used to pump vapor from the evaporator by the ejector. All dissipated effective kinetic energy, which is about 15% of the work done by the compressor in the conventional cycle, is not recovered by the ejector in practice.



(a) Refrigeration cycle with ejector

(b) Conventional refrigeration cycle

Fig. 1 Refrigeration cycle

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However, the efficiency of the refrigeration cycle can be increased by recovering the dissipated effective energy.

To develop a two-phase ejector, the performance of the mixing process in the mixing section and the diffuser and the expansion in the nozzle need to be elucidated. Nakagawa et al.[4],[5] have investigated the performance of a two-phase nozzle for a refrigerant theoretically and experimentally. Few studies on the mixing of a high speed two-phase flow in a two-phase ejector have been reported, because the flow is complicatd by the nonequilibrium of velocity and temperature between the liquid and gas phases. The purpose of the present study is to elucidate the performance of the mixing of a confined two-phase mist flow. The mixing performance in the mixing section and the diffuser of a two-phase ejector is investigated experimentally and compared with a theoretical model.

Nomenclature

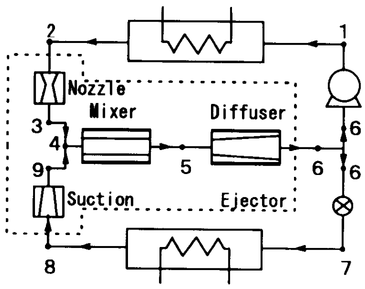
A : cross-sectional area m^2
D : diameter m
d : diameter of droplet m
g : $=G_d/G_s$ -
G : mass flow rate kg/s
h : specific enthalpy J/kg
L : length m
p : pressure Pa
u : velocity m/s
s : specific entropy J/kg
z : coordinate along flow direction m
η : efficiency of ejector -
λ : friction factor -
λ_i : traction factor -
μ : viscosity Pas
ρ : density kg/m^3
ε : coefficient of performance -

Subscripts

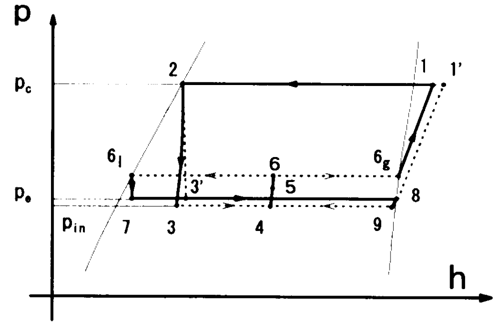
c : condenser
d : primary stream
e : evaporator
g : vapor in primary stream
in : inlet of mixing section
l : droplet in primary stream
md : mixing section and diffuser
mix : mixing section
out : outlet of diffuser
s : secondary stream
sin : inlet of secondary stream
v : vapor in secondary stream

2. Refrigeration Cycle with Ideal Two-Phase Ejector

An ideal two-phase ejector is considered here to explain its basic function in the refrigeration cycle.



(a) Schematic arrangement of cycle



(b) Pressure-enthalpy diagram

Fig. 2 Ideal cycle with ejector

The refrigeration cycle with a two-phase ejector which utilizes the expansion energy is different from the conventional cycle. A schematic arrangement and the pressure-enthalpy diagram are shown in Fig. 2. The refrigerant cooled in the condenser (: 1-2) expands not in the valve but in the nozzle (: 2-3) and is accelerated to give a high-speed two-phase flow. For ideal isoentropic change, the velocity of the primary stream of the ejector u_d is given as

$$\frac{u_d^2}{2}=h_2-h_3, \quad s_3=s_2. \tag{1}$$

The accelerated two-phase flow is mixed with the secondary stream in the mixing section (: 3, 9-5). The flow in the mixing section is complicated. However, if ideal mixing takes place, the mass, momentum and energy conservation equations for the constant flowing cross sectional area at the mixing section A_{mix} are

$$\frac{G_d+G_s}{\rho_5 u_5}=\frac{G_d}{\rho_3 u_d}+\frac{G_s}{\rho_9 u_s}=A_{mix} \tag{2}$$

$$p_5 A_{mix}+(G_d+G_s)u_5=p_{mix}A_{mix}+G_d u_d+G_s u_s \tag{3}$$

$$(G_d+G_s)\left(\frac{u_5^2}{2}+h_5\right)=G_d\left(\frac{u_d^2}{2}+h_3\right)+G_s\left(\frac{u_s^2}{2}+h_9\right). \tag{4}$$

Here, G_d and G_s are the velocities of the primary and the secondary stream, respectively. The velocity u_5 and the pressure p_5 at the outlet of the mixing section for a given inlet (: 3, 9) condition are determined using these equations and the saturation relation.

The high speed flow from the mixing section is decelerated and compressed to a pressure of p_6 which is determined by the enthalpy h_6 and entropy s_6 for an ideal diffuser (: 5-6).

$$h_6 = \frac{u_5^2}{2} + h_5, \quad s_6 = s_5 \tag{5}$$

The compressed two-phase flow is separated into vapor and liquid. The former is extracted by the compressor (: 6-1) and the latter is decompressed (: 6-7) in the evapolator (: 7-8).

The vapor from the evaporator is accelerated at the suction section (: 8-9) of the ejector and becomes the secondary stream in the mixing section. The velocity of the secondary stream is obtained in the same manner as that for the nozzle.

$$\frac{u_s^2}{2} = h_8 - h_9, \quad s_9 = s_8 \tag{6}$$

Since the outlet conditions for each section are determined from the inlet conditions using these equations, the thermodynamic variables at any point can be calculated by equating the quality (dryness fraction) at the outlet of the diffuser and the distribution of the mass flow rates $G_d/(G_d+G_s)$.

The coefficient of performance (COP) of a refrigeration cycle with an ideal two-phase ejector can be estimated from these thermodynamic variables.

$$\varepsilon = \frac{G_s(h_7 - h_8)}{G_d(h_1 - h_{6g})} \tag{7}$$

By changing the pressure difference between the inlet of the mixing section and the evaporator $p_e - p_{\text{in}}$, the COP of a cycle with a two-phase ejector is plotted in Fig. 3 for refrigerant R-12 when the pressure in the condenser is 1 MPa and that in the evaporator is 0.4 MPa. The COP of the conventional refrigeration cycle (: 1′-2-3′-8) with an expansion valve is 7.2, and is shown by the horizontal line in Fig. 3. The COP of the refrigeration cycle with a two-phase ejector is greater than that of the conventional cycle.

The reason why the COP varies with the pressure difference $p_e - p_{\text{in}}$ is related to the change in the performance of the ejector. Since the function of the ejector is to transfer the kinetic energy of the primary stream to the secondary stream, the efficiency of the ejector η is defined as the conversion efficiency of thermal energy.

$$\eta = \frac{G_s\{(h_{1'} - h_8) - (h_1 - h_{6g})\}}{G_d(h_2 - h_3)} \tag{8}$$

Here, the compression work done on the secondary stream equals the pumping work done in the cycle with an ejector subtracted from that of the conventional cycle. The efficiency is plotted in Fig. 3. Although the individual processes are ideal, the total efficiency of this cycle is not unity. This is because the entropy increases when the primary and secondary streams, which have different velocities, are mixed. When an ejector, which has no complicated part like a rotating blade, is used in the cycle, it is impossible to attain an efficiency of unity.

Many of the characteristics of the two-phase flow in an ejector such as that shown by the dotted line in Fig. 2(a) remain unknown. However, it is thought that, the large energy losses occur in the nozzle, the mixing section and the diffuser. The acceleration energy loss in the nozzle has been studied, whereas the mixing performance of two-phase flow has not. Flows in the mixing section and the diffuser are modeled and investigated in the next section.

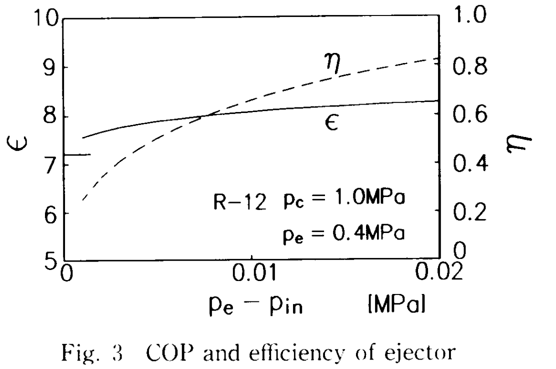


Fig. 3 COP and efficiency of ejector

3. Model for Mixing Jet of Mist Flow

The stream in the mixing section and the diffuser is divided into three parts. The first and second are the primary streams of droplets and vapor in the jet which is ejected from the nozzle. The third is the secondary stream of vapor from the evaporator. It is postulated that the primary stream flows inside a cylindrical volume with the secondary stream outside it. The momentum exchange between the primary and secondary streams takes place at the surface of this cylinder. For simplicity, we assume that there is no condensation or evaporation of the vapor, because the change in the pressure and temperature due to the pressure recovery is small. We also assume that the diameter of the droplets is constant throughout the stream.

The mass conservation equation for the stream in the mixing section and the diffuser is expressed as

$$\frac{G_{dl}}{\rho_l u_l} + \frac{G_{dg}}{\rho_g u_g} + \frac{G_s}{\rho_v u_v} = A(z). \tag{9}$$

Here, the gaseous vapor in the primary stream is denoted by the suffix g, the liquid droplets by l, and the vapor in the secondary stream by v. The flow cross sectional area A is a function of the coordinates z along the flow direction. It is constant in the mixing section and increases in the diffuser.

The total momentum equation for the primary and secondary streams is

$$G_{dl}\frac{du_l}{dz}+G_{dg}\frac{du_g}{dz}+G_s\frac{du_v}{dz}+A\frac{dp}{dz}$$
$$=-\lambda\frac{\sqrt{\pi A}}{4}\rho_v u_v^2. \qquad (10)$$

Here, λ is the Fanning friction factor, and only the secondary stream is in contact with the duct wall. The vapor exerts a retarding force on the droplets in the primary stream. The friction factor for the droplets is determined using Stokes's law, because it is small.

$$u_l\frac{du_l}{dz}=-18\frac{\mu_g}{\rho_l}\frac{(u_l-u_g)}{d^2} \qquad (11)$$

The primary stream exerts a traction force on the secondary stream in the flow direction.

$$G_s\frac{du_v}{dz}+(A-A_d)\frac{dp}{dz}$$
$$=-\lambda\frac{\sqrt{\pi A}}{4}\rho_v u_v^2+\lambda_i\frac{\sqrt{\pi A_d}}{4}\rho_g(u_g-u_v)^2 \qquad (12)$$

Here, λ_i is the traction factor. A_d is the flow cross-sectional area of the primary stream, which is given by the sum of the first and second terms on the left side of Eq.(9). Since all of the physical properties assumed to be constant, the four unknown variables, the three velocities and the pressure, are determined using these four equations.

Assuming that the diameter of a droplet is 50 μm and the traction factor is 0.12, a calculation is carried out for refrigerant R-12. The inlet pressure p_{in} is set to 0.4 MPa, u_g to 96 m/s, u_l to 48 m/s and u_v to 14 m/s. The velocity and pressure along the flow direction are shown in Fig. 4. The broken line in the upper part of the figure is the diameter of the mixing section and the diffuser. The secondary stream is carried by the primary stream in the mixing section and decelerates in the diffuser, where the pressure increases. It is assumed that the pressure recovery is small and that $\left(\frac{\partial h}{\partial p}\right)_s$ is constant. The efficiency of the ejector in the mixing section and the diffuser is defined as

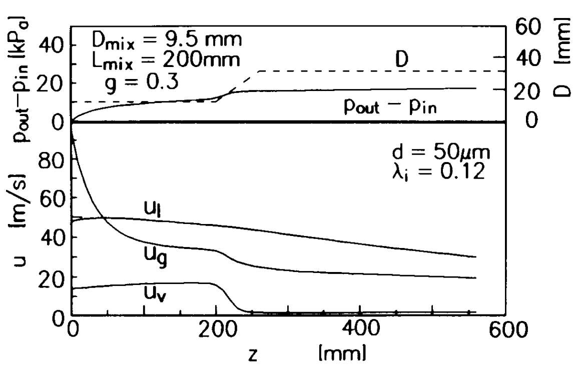


Fig. 4 Calculated velocity and pressure

$$\eta_{md}=\frac{G_s\left\{\left(\frac{\partial h}{\partial p}\right)_s(p_{out}-p_{in})-\frac{u_{vin}^2}{2}\right\}}{G_{dg}\frac{u_g^2}{2}+G_{dl}\frac{u_l^2}{2}}. \qquad (13)$$

The efficiency of the ejector given by this caluculation is 0.11.

4. Experimental Apparatus

A refrigeration cycle with a two-phase ejector whose thermal output is about 1.5 kW is used in this study and shown in Fig. 5. The special feature of this cycle is the subevaporator. The flow rate of the secondary stream which passes through the main evaporator can be changed using this setup. The flow rate of the primary stream is fixed at 240 kg/h, the nozzle inlet temperature at 40°C, and the nozzle back pressure at 0.4 MPa. These conditions are similar to conventional operating conditions.

The two-phase ejector used in this experiment is shown schematically in Fig. 6. It consists of the nozzle, the suction section, the mixing section and the diffuser. The primary stream ejected from the nozzle and the secondary stream from the main evaporator are mixed in the mixing section and the stream is decelerated and compressed in the diffuser. The pressure recovery in the mixing section and the diffuser is measured by changing the diameter D_{mix} and the length L_{mix} of the mixing section and the flow rate of the secondary stream G_s.

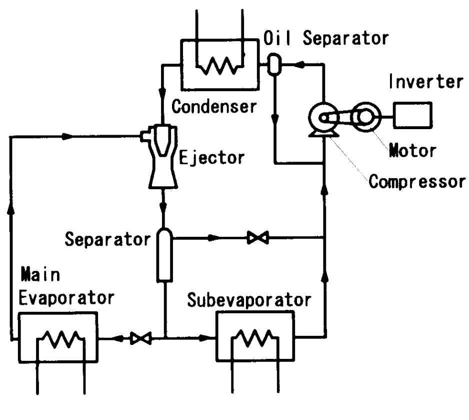


Fig. 5 Experimental apparatus

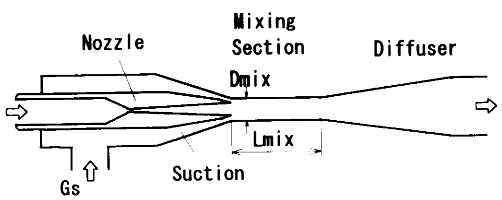


Fig. 6 Two-phase ejector

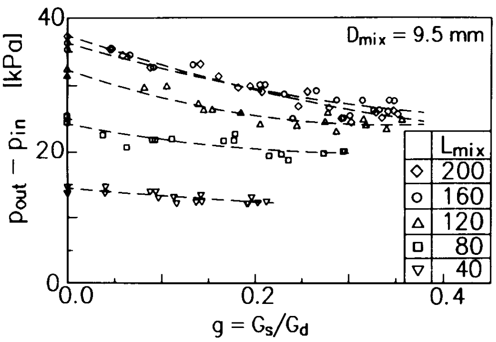


Fig. 7 Presssure recovery

5. Experimental Results

The mixing characteristic of the confined jet of mist flow is evaluated in terms of the pressure recovery, because an important function of the ejector is to compress the secondary stream. The pressure recovery in the mixing section and the diffuser is equal to the sum of the measured pressure difference between the inlet and the outlet of the ejector and the acceleration pressure loss in the suction section.

$$p_{out} - p_{in} = p_{out} - p_{sin} + \frac{\rho_s u_s^2}{2} \tag{14}$$

Experimental results for the pressure recovery versus the ratio of the mass flow rate of the secondary stream to that of the primary stream $g = G_s/G_d$ are shown in Fig. 7. The data represented by ▽ are for a mixing section 40 mm long, □ are one for 80 mm long, △ are for one 120 mm long, ○ are one for 160 mm long and ◇ are one for 200 mm long. The broken curves are drawn by approximating these data. The pressure recovery is the largest when there is no secondary stream $g = 0$, and it decreases as the flow rate of the secondary stream increases. When the mixing section is long, the pressure recovery is large. However, it is almost constant for mixing sections more than 160 mm long.

The experimental results shown in Fig. 7 are also plotted in Fig. 8 versus the length of the mixing section L_{mix} by symbols ● and ○ for mass flow ratios of $g = 0$ and 0.2, respectively. This figure shows that a long mixing length is effective for pressure recovery. This is because the time for which the primary and secondary streams are in contact becomes long, and droplets which have large inertia can be decelerated by the slower secondary stream. A two-phase ejector which has a mixing section long enough in relation to the diameter is a different shape from the conventional one-phase ejector for either vapor or water.

The solid and broken curves in Fig. 8 are the results calculated using the model introduced in the previous section. The diameter of the droplets is

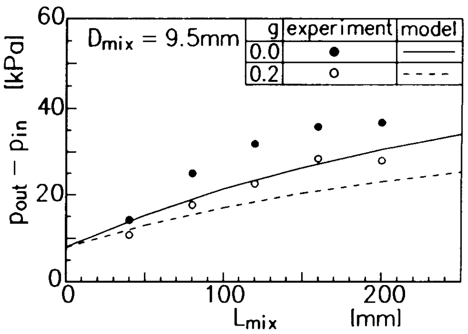


Fig. 8 Effect of length of mixing section

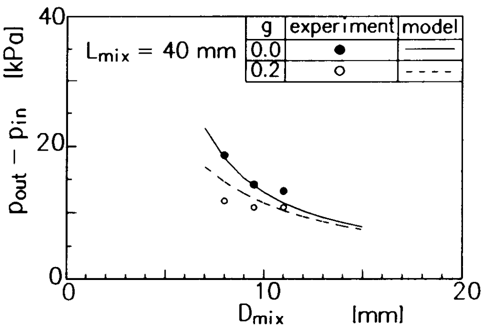


Fig. 9 Effect of diameter of mixing section

taken as 10 μm. The droplet diameter is calculated from the Weber number and the velocity difference between the droplets and the vapor in the nozzle. It is about 10 μm. The traction factor is assumed to be 0.12. This value was used by Ueda[(6)] for the steam ejector. The inlet velocity in the mixing section is that given in the previous section, which was calculated by assuming that the nozzle efficiency is 80%. The nozzle efficiency is obtained by experiment[(5)] using refrigerant R-113. The model cannot be used to estimate precise values of the pressure recovery but it can be used to predict its tendency and order of magnitude.

The pressure recovery versus the diameter of the mixing section D_{mix} is shown in Fig. 9 for a mixing section with a length of $l_{mix} = 40$ mm. the data for mass flow ratios of $g = 0$ and 0.2 are plotted as before. The pressure recovery increases as the diameter decreases. When the velocity of the secondary stream is as high as that of the primary stream, the mixing efficiency is increased as already mentioned. The solid and broken curves in Fig. 9 are calculated using the model, which can also be used to predict the dependence of the pressure recovery on the diameter of the mixing section.

In order to estimate the energy conversion efficiency of the ejector from the experimental results, Eq. (8) is written as

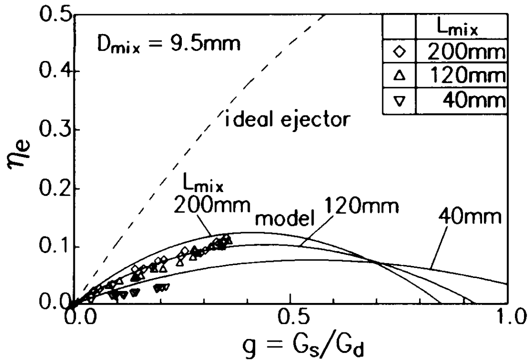


Fig. 10 Energy conversion efficiency

$$\eta = \frac{G_s\left(\frac{\partial h}{\partial p}\right)_s (p_{out} - p_{sin})}{G_d(h_2 - h_3)}, \quad (15)$$

because the pressure recovery is small. This energy conversion efficiency is plotted in Fig. 10 versus the length of the mixing section. The broken curve is the efficiency of an ideal ejector. If an ejector is used in the refrigeration cycle, an efficiency above this curve cannot be realized. The maximum value obtained in experiments is about 0.1. The solid curves which have a maximum at the mass flow ratio of $g=0.4$ are calculated using the model. The value are rather small, but should to be increased by improving the mixing characteristic and the nozzle efficiency.

6. Conclusions

The pressure recovery in the mixing section and the diffuser of a two-phase flow ejector in the refrigeration cycle is investigated to elucidate the mixing characteristics of a confined jet of high-speed mist flow. The following conclusions are reached by comparing the measured pressure recovery with that calculated using the theoretical model.

(1) It is found that a sufficiently long mixing length is necessary to decelerate droplets, which have large inertia. The pressure recovery is increased by increasing the length of the mixing section long.

(2) The velocity difference in the mixing section between the primary and secondary streams must be minimized in order to obtain a high energy conversion efficiency. The diameter of the mixing section must be small in order to increase the pressure recovery.

(3) The influences of the length and diameter of the mixing section and the flow rate on the pressure recovery are predicted using the present model assuming the phase of mist flow does not change.

(4) The energy conversion efficiency of the ejector obtained by experiment is 0.1. This should be increased by improving the mixing characteristics and the nozzle efficiency.

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