Two-phase Flow Ejector as Water Refrigerant by Using Waste Heat

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Abstract. Energy saving and the use of clean energy sources have recently become significant issues. It is expected that clean energy sources such as solar panels and fuel cells will be installed in many private dwellings. However, when electrical power is generated, exhaust heat is simultaneously produced. Especially for the summer season, the development of refrigeration systems that can use this waste heat is highly desirable. One approach is an ejector that can reduce the mechanical compression work required in a normal refrigeration cycle.

We focus on the use of water as a refrigerant, since this can be safely implemented in private dwellings. Although the energy conversion efficiency is low, it is promising because it can use heat that would otherwise be discarded. However, a steam ejector refrigeration cycle requires a large amount of energy to change saturated water into vapour. Thus, we propose a more efficient two-phase flow ejector cycle.

Experiments were carried out in which the quality of the two-phase flow from a tank was varied, and the efficiency of the ejector and nozzle was determined. The results show that a vacuum state can be achieved and suction exerted with a two-phase flow state at the ejector nozzle inlet.

1. Introduction

Widespread interest in environmental preservation has grown in recent years. At the same time, concern has also grown about fossil fuel depletion due to rising energy demand, and energy conservation has become a major issue worldwide. In the wake of the Great East Japan Earthquake, moreover, strong doubts have arisen particularly in Japan in regard to the continued use of nuclear energy. For all of these reasons, efforts to develop and use new energy sources and natural energy are expected to expand.

Solar power and solid oxide and other types of fuel cells are expected to proliferate for household use, as new sources of energy. At present, however, their power generation is accompanied by waste heat production, and no related means of efficient energy conversion has as yet been achieved. It is important to recover and use the energy of waste heat, which is currently simply discarded, and thereby increase the total efficiency of household power systems. In winter, this waste heat can be used for home and water heating, but little progress has been made toward its summertime utilization even though its use in air conditioners, which are the largest power consumers in the summer season, would bring a substantial increase in total system efficiency.
In one proposed concept, an ejector[1-2] would be mounted on the air conditioner and utilize the low-temperature waste heat to carry out compression, in place of a dedicated compressor. Water is a safe, shared resource with a small environmental burden, and would represent an ideal refrigerant for home air conditioners.

In existing air conditioners, the refrigerant is compressed by an electrically powered compressor. Ejector systems, which convert heat energy to kinetic energy, could further convert this to compression energy and thus take over the burden of compression work in an air conditioner. Some progress has been made in research on ejector systems[3-5], but the energy conversion efficiency with water as a refrigerant is generally regarded as unacceptably low. Even with a low efficiency, however, its effective utilization of heat that is otherwise simply discarded would constitute a considerable improvement in the overall efficiency of electric power systems.

One way to increase the performance of such an ejector air conditioning system would be to use supersonic two-phase flow in place of vapour. To drive the vapour cycle requires only the addition of heat energy equal to the latent heat, in order to produce vapour. Figure 1 shows the refrigeration cycle of a water refrigerator ejector in a Mollier (p-h) diagram, which represents the conversion of the heat energy of high-temperature high-pressure water to kinetic energy and its return to a low-temperature low-pressure state. If the refrigeration cycle in the two-phase flow state can be accomplished, the required heat quantity for the cycle can be reduced and, in comparison with the vapour ejector cycle, it can serve as an air conditioner with very high efficiency.

2. Objective
Various studies have been performed on the vapour ejector air conditioning cycle with water refrigerants, but the quantity of energy required for a phase change of water from liquid to vapour has been inordinately large. If a two-phase flow ejector cycle can be used instead of a vapour ejector cycle, however, the cycle itself would accomplish the energy saving.

The authors have performed studies on nozzles as a form of basic research on water refrigerant two-phase flow ejectors[6], and have shown that the energy conversion efficiency of the nozzle plays an extremely important role in ejectors using water as a refrigerant. In the present study, the objectives were to verify the achievement of a vacuum state at the outlet of a two-phase flow nozzle, to accomplish input of water refrigerant to the ejector in the two-phase flow state, and to investigate the ejector pressure rise characteristics and suction capability under these circumstances.

![Figure 1. P-h diagram of water ejector refrigeration cycle.](image-url)
3. Theory

3.1. Nomenclature

\( w \) [W/kg]  Specific work needed for compression
\( h \) [J/kg]  Specific enthalpy
\( s \) [J/(kgK)]  Entropy
\( q \) [W/kg]  Specific heat
\( p \) [Pa]  Pressure
\( u \) [m/s]  Velocity
\( A \) [m2]  Area
\( G \) [kg/s]  Mass flow rate
\( z \) [m]  Coordinate along flow from the throat
\( x \) [-]  Quality
\( g \) [-1]  Mass flow ratio
\( COP \) [-]  Coefficient of performance

Special characters
\( \rho \) [kg/m\(^3\)]  Density

Subscripts
\( s \)  Saturated state (for pressure)
\( d \)  Driving flow
\( s \)  Suction flow
\( c \)  Critical condition
\( ex \)  Experimental result
main\text{th}  Isentropic Homogeneous Equilibrium model

3.2. Coefficient of performance

Figure 2 is a schematic of the air conditioning cycle of the water refrigerant ejector investigated in this study. The change of state in pressure and enthalpy is depicted in the Molliere diagram in Figure 1. The numbers shown in this cycle correspond to those shown in the Molliere diagram. The cycle comprises two component cycles. One is the conventional Rankin cycle, moving in turn from the condenser to the pump to the high-temperature evaporator and back to the condenser via the ejector. The other is the vapour-compression refrigeration cycle, moving from the condenser to the expansion valve, the low-temperature evaporator, the ejector, and back to the condenser. In this scheme, the ejector serves the role of a heat engine turbine in the former component cycle and an air conditioner compressor in the latter component cycle. The high-temperature evaporator is heated by heat that would otherwise be discarded as waste heat, and the low-temperature evaporator acts to cool the designated space on the low-temperature side in the refrigeration cycle.

![Figure 2. Ejector refrigeration system in this study.](image-url)
To assess the performance of this water refrigerant ejector refrigeration cycle, the coefficient of performance (COP) was calculated with the following assumptions. Friction loss in the various components of the system was disregarded and the pump, evaporators, and condensers were thus regarded as ideal devices. With subscripts 1 and 2 referring to the states at the pump inlet and outlet, respectively, the pump work \( w_{21} \) per unit mass flow is obtained from the change in enthalpy \( h \) as

\[
w_{21} = h_2 - h_1, \quad s_2 = s_1
\]  

where \( s \) represents entropy.

With the outlet on the high-temperature side designated as 3, and the evaporator inlet as 2, the heat energy absorption quantity \( q_{32} \) is similarly expressed in terms of an increase in enthalpy as

\[
q_{32} = h_3 - h_2, \quad p_3 = p_2
\]  

Ordinarily, the pressure \( p_2 \) at the high-temperature evaporator inlet is taken as being lower than the waste heat saturated vapour pressure \( p_s(100^\circ C) \), but here we assume an ideal heat exchanger, so that \( p_2 = p_s(100^\circ C) \). The saturated vapour pressure \( p_s(T) \) is a function of the temperature \( T \). Similarly, for the low-temperature side evaporator, we have

\[
q_{67} = h_8 - h_7, \quad p_8 = p_7
\]  

The state of the inlet of the evaporator on the low-temperature side is designated by the subscript 7, and at the outlet by the subscript 8. \( p_7 \) is then the saturated vapour pressure \( p_s(10^\circ C) \) of the air conditioner. The condenser inlet section and the ejector diffuser outlet section are represented by the subscript 6, and the quantity of heat \( q_{61} \) released to the atmosphere is

\[
q_{61} = h_6 - h_1, \quad p_6 = p_1
\]  

where \( p_6 \) is based on the assumption of high summer temperatures and is thus taken to be \( p_s(40^\circ C) \).

As in the conventional treatment, the process at the expansion valve is taken as an isenthalpic change and thus

\[
h_7 = h_1
\]  

The ejector consists of four distinct components: the driving flow nozzle, the suction nozzle, the mixer and the diffuser. We assume that the steps in each component are ideal heat exchange processes. The driving flow, suction flow nozzle, and mixer states are indicated by the subscripts 4, 5, and 9, respectively.

\[
\frac{u_4^2}{2} = (h_3 - h_4), \quad s_3 = s_4
\]  

\[
\frac{u_5^2}{2} = (h_8 - h_5), \quad s_9 = s_8
\]  

\[
h_6 = h_5 + \frac{u_5^2}{2}, \quad s_6 = s_5
\]
Since the cross sections of the channels in the ejector inlet and outlet are large in comparison with the ejector’s internal channel cross section, the kinetic energy at the inlet and outlet can be disregarded.

The laws of conservation of mass, momentum, and energy in the mixer, where the driving flow and suction flow are mixed, are thus expressed as

\[
\rho_d u_d A_d + \rho_s u_s A_s = \rho_d u_d A_d \\
\rho_d u_d^2 A_d + \rho_s u_s^2 A_s + A_d p_d + A_s p_s = \rho_d u_d^2 A_d + A_s p_s \\
\frac{u_d^2}{2} + h_d + \frac{u_s^2}{2} + h_s = \frac{u_s^2}{2} + h_s
\]

and, since the driving flow nozzle and the suction flow nozzle join at the mixer inlet, the following relation applies.

\[
p_d = p_s, \quad A_d + A_s = A_s
\]

The driving flow \( G_d \) and suction flow \( G_s \) are given by

\[
G_d = \rho_d u_d A_d, \quad G_d + \rho_s u_s A_s \quad \text{where } g = G_s / G_d
\]

All of the thermal parameters in this system are thus determined by the flow ratio \( g \) and the nozzle outlet pressure \( p_s \). Using \( \rho_d \), for example, which is determined by \( p_d \), the nozzle outlet cross-sectional area \( A_s \) is determined from the continuity equation and thus depends on \( p_s \).

The \( COP \), which represents the thermal efficiency of this ejector cycle, is defined as the ratio of the heat absorption quantities of the high-temperature and low-temperature evaporators as

\[
COP = \frac{G_d (h_s - h_d)}{G_d (h_s - h_d)}
\]

The flow ratio \( g \) was obtained so that the ejector outlet pressure \( p_0 \) would match the condenser pressure and thus the saturated vapour pressure of the atmosphere. \( p_4 \) was varied to determine the value at which the \( COP \) of this system became maximal.

Based on these assumptions, calculations were performed to obtain the relation between the ejector inlet flow quality and the \( COP \) of the water-refrigerant two-phase flow ejector. The results are depicted in Figure 3. As noted above, these results were calculated with the assumption of summer use, a waste-heat temperature of 100°C, an atmospheric temperature of 40°C, and a cooling temperature of 10°C. As shown in Figure 3, the \( COP \) was high when the inlet flow was low in quality and maximal at an inlet flow quantity of 0.3.

This indicates that the two-phase flow ejector system is more effective than a vapour ejector system as an air conditioner utilizing waste heat.
4. Experimental apparatus

The experimental apparatus is shown schematically in Figure 4. It essentially consists of a high-pressure tank, a heater, a saturated vapour and a saturated liquid mixer, a two-phase flow ejector, a condenser, and a flowmeter. As the purpose of the experiments was to investigate the performance of the ejector, it was not necessary to run the air conditioning cycle. To simplify the apparatus, we assumed a simulated cycle in which the ejector outlet was at atmospheric pressure, and implemented the cycle in an open loop with the outlet open to the atmosphere. The tank was wrapped in thick insulation and the piping and apparatus were wrapped in neoprene rubber, thus insulating the flow channels. The water was heated in the tank by a heater and maintained at a constant temperature with a temperature control unit. The experiment was performed with an internal tank temperature of 150°C, a saturated vapour flow from the top part of the tank and a saturated liquid from the bottom part. The phase flow rates were controlled using flow regulating valves, and the quality of the two-phase flow was arbitrarily controlled. The saturated steam flow was measured using a capillary flowmeter, and the total flow was measured using a coriolis flow meter mounted at the outlet of the apparatus. As it is difficult to use a capillary flow meter with saturated liquids, their flow rates were calculated from the total flow and the saturated vapour flow. It was then possible to obtain the flow quality at the ejector inlet.

As an additional measure to simplify the apparatus, the suction performance of the ejector suction flow was assessed for air suction rather than water vapour refrigerant suction.
As the specific volume of the saturated vapour was nearly 1,000 times that of the saturated liquid, the diameter of the inlet on the liquid side was made small in comparison with that for the vapour inlet so that the flow rate of the phases would be of the same order during the mixing. To obtain a large heat transfer on the droplet surfaces in both phases, it was deemed important to make the droplets small.

The experimentally used ejector was designed for a vapour flow of 1.0 g/s with an inlet flow quality of 1.0. Figure 5 shows a schematic of the ejector assembly. The heated water flows through the flow channel formed by plates of PES, which has a low thermal conductivity, and stainless steel. In the ejector flow channels, pressure taps were included at 7 points in the nozzle section and 7 points in the mixer and the diffuser. K type thermocouples were inserted into the pressure taps, the ejector internal pressure was measured, and the pressure distribution in the flow direction was obtained.

5. Experimental results
In the experiments, the vapour quality at the ejector inlet was varied for experimental verification of whether or not two-phase flow to the ejector actually yields higher performance than vapour flow as indicated by the theoretical calculation. In each experiment, we measured the flow rate of each phase from the tank, the internal pressure distribution of the ejector in the flow direction, the rate of suction flow into the ejector, and the ejector inlet and outlet pressures. Our previous studies had clearly shown that the nozzle’s energy conversion efficiency greatly affected the ejector performance, and in the present study, therefore, the nozzle performance and ejector performance were obtained for various flow qualities at ejector inlet.

5.1. Verification of nozzle performance

5.1.1. Shock wave in the nozzle. The pressure distribution in the nozzle in the direction of flow was measured for various inlet flow qualities. Figure 6 shows the results at qualities of 0.20 and 0.75. The x-axis represents the distance from the inlet throat in the direction of the flow, with the throat at 0 mm and the nozzle outlet at 10 mm. The distributions shown were obtained with no suction flow for various ejector outlet pressures $P_{out}$, and with suction flow. When the ejector outlet pressure was increased, a moderate or sharp pressure recovery was observed in the nozzle divergence region. The results showed that changes in the outlet pressure and the presence or absence of suction had no effect on the convergent nozzle, thus indicating that the flow in the nozzle was supersonic, with the pressure rise in the ejector representing the shock wave generated by this flow. The gentle pressure rise at low flow quality in comparison with the sharper rise at high vapour quality is considered to be a reflection of the experimental results shown in Figure 6.

![Figure 6. Pressure distribution (x=0.20 and 0.75).](image-url)
5.1.2. Homogeneous flow. As described in the previous section, the state in the convergent nozzle is constant regardless of the state in the nozzle outlet. As the flow rate and the vapour quality in the throat are constant, the flow rate is not changed by the backpressure. This is a critical state, and the flow rate in this state is referred to as the critical flow rate.

In these experiments, as noted above, saturated vapour and saturated liquid were mixed to form the two-phase flow upstream from the ejector. It was therefore difficult to assess the phase distribution and gas-liquid slip. One method that can be applied for this purpose has been described by Isbin et al., Starkman et al., and Akagawa[7] in their assessments of flow conditions, by comparing critical flow rate corrections with homogeneous flow models.

Figure 7 shows the relation between the flow quality at the ejector inlet and the error correction, with the theoretical critical flow rate at the nozzle throat designated as $G_{nth}$ and the measured flow rate as $G_{ex}$. The error tends to increase at low flow qualities. The results are in general agreement with those of other researchers, and the two-phase flow mixed upstream from the ejector may therefore be treated as a homogeneous flow.

![Figure 7](image)

**Figure 7.** Critical flow correction with homogeneous flow model.

5.1.3. Nozzle efficiency. Figure 8 shows the pressure distribution in the nozzle for different inlet flow qualities, using colour-coded symbols. The corresponding curves represent the pressure distribution as calculated using the theoretical IHE model from the inlet conditions for the different flow qualities. Since the experimental distributions are higher than those given by the IHE model, expansion in the nozzle was insufficient. Thus, as described in the previous section for the critical flow correction, this tendency is marked at low flow qualities and indicates the occurrence of larger nonequilibrium phenomena at low qualities.

The two-phase flow velocity at the nozzle outlet can be calculated from the pressure distribution in the nozzle, using the law of conservation of momentum. Figure 9 shows the energy efficiency of the nozzle as derived from the outlet flow velocity and the adiabatic enthalpy drop. The nozzle efficiency declines with decreasing flow quality from a value of about 80% in the vapour state to about 60% for a flow quality of 0.2.
5.2. Assessment of ejector performance

5.2.1. Ejector suction force. The ejector suction performance was investigated experimentally by gradually opening the valve on the suction side of the ejector for suction flow to the ejector mixer and measuring the suction force. The suction flow rate was measured using a capillary flowmeter. Figure 10 shows the relation between the suction flow rate and the suction stream pressure rise $dp$ at the ejector with the driving-flow flow inlet conditions held constant. The pressure rise $dp$ represents the ejector suction force, and the results show that the ejector used in these experiments induced a maximum suction pressure of 0.08 MPa. The pressure difference between the ejector outlet and the suction flow inlet tended to decrease with increasing suction flow rate, and the ejector thus performs the same function as a constant-rotation compressor. The maximum suction flow rate shown in the right of the figure is an important indicator of performance of the ejector. Figure 11 shows the relation between the inlet flow quality and the achieved suction flow rate.

Figure 8. Pressure distribution for different inlet qualities.

Figure 9. Nozzle energy conversion efficiency.

Figure 10. Pressure recovery of ejector.

Figure 11. Mass flow rate of suction flow.
The suction flow rate is maximal when the flow quality is high. To induce air suction at the nozzle outlet, it is necessary for the fluid to accelerate at the driving flow nozzle and expand sufficiently to drop below atmospheric pressure. Comparison with Figure 8 clearly indicates that the nozzle outlet enters a higher vacuum state and the suction flow rate becomes maximal in the high flow-quality region. Substantial acceleration is therefore essential to induce suction flow at the nozzle by the ejector, and a high-efficiency nozzle is therefore necessary.

5.2.2. Calculation of ejector efficiency. The functions of the ejector are to convert the high-temperature high-pressure heat energy of the driving flow to kinetic energy and thereby induce suction flow and conversion to pressure work. Here we define the ejector efficiency \( \eta_{eje} \), which is an indicator of the performance in terms of the actual pressure rise of the suction flow induced by the driving flow, as

\[
\eta_{eje} = \frac{G_s \frac{P_6 - P_8}{\rho_s}}{G_s dh_n}
\]

with the provision that \( dh_n \) is the adiabatic enthalpy drop from the ejector driving flow inlet to the ejector outlet, \( p_6 - p_8 \) is the pressure rise from the ejector suction flow inlet to the ejector outlet, and \( \rho_s \) is the air density and equals to \( (\hat{c}_p/\hat{c}_h) \).

Typical results are shown in Figure 12. The higher ejector efficiencies were obtained for the case of high in quality at inlet rather than case of low inlet quality. It was also found that there were differences in the ability to raise the pressure of the suction fluid at a given suction flow rate with different inlet flow qualities. Furthermore, it was clear that the efficiency rises toward the right and thus shows that the ejector efficiency increases with suction flow and also, only with high inlet quality, the efficiency had the maximum value.

Figure 13 shows the values at which the ejector efficiency became maximal in the experiments on inlet flow quality. The tendency shown in this figure is similar to that found in the graph for maximal suction flow in Figure 11, which indicates that the ejector efficiency becomes high for high flow quality accompanying high suction flow, and becomes low for low flow quality.

![Figure 12. Ejector efficiency.](image)

![Figure 13. Maximum ejector efficiency.](image)

Although the ejector efficiency values obtained in these experiments may seem rather low, they nevertheless demonstrate that these values can be achieved with no more than a laboratory-scale ejector. This refrigeration cycle is driven by waste heat and thus has considerable potential even at low efficiencies through its capability for utilization of heat energy that would otherwise simply be
discarded. Further studies will be necessary to develop an ejector design that will yield higher efficiencies.

6. Conclusion
In the present study, a two-phase flow ejector was fabricated with the ultimate objective of realizing a two-phase flow ejector air conditioning system that could achieve the performance level shown by theoretical calculations. Its performance was assessed in experimental operation while inputting a water refrigerant in a two-phase flow state, and the following results were obtained:

1. It was found to be possible to obtain a vacuum state at the nozzle outlet under a two-phase flow condition at the ejector driving flow inlet and to induce a suction flow.
2. The energy conversion efficiency of the two-phase flow nozzle of the ejector was 80% at an inlet flow quality $x=1.0$, and declined with decreasing inlet flow quality to a value of 60% at $x=0.2$.
3. The efficiency of the ejector was 0.08 at a nozzle inlet flow quality of $x=1.0$ and declined to 0.04 at $x=0.3$.
4. Although the experimental results showed rather low ejector efficiency, the two-phase flow ejector air conditioning system may nonetheless be expected to provide a useful cooling capacity through its utilization of waste heat that would otherwise be discarded.

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