Development of ejector cycle for solar air-conditioning system powered by stored thermal energy
Part 1. Experimental results

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1. Introduction

From a viewpoint of solving the global warming and the depletion of fossil energy resources, renewable energy utilization, which can reduce CO_2 emission, is quite important. In addition, some countries including Germany are moving toward denuclearization due to nuclear accident at Fukushima. Therefore, renewable energy utilization become of particular importance. On the other hand, air-conditioning demand is increasing in many Asian countries like China, India, Indonesia…etc., especially in the areas of the tropical region because of economy developing. Furthermore, air-conditioning demand is increasing in cities, because of heat island phenomenon caused by releases of higher temperature exhaust heat to atmosphere. Therefore, development of the air-conditioning systems working with renewable energy including solar thermal energy is strongly promising.

An ejector refrigeration cycle can provide cooling by using solar thermal energy with small consumption of electricity, which can be powered by photovoltaic cells. The ejector refrigeration cycle resembles the vapor compression refrigeration cycle, but it uses a vapor generator and an ejector instead of a mechanical compressor to create compression effect. Figure 1 shows the schematic diagram of the system. The ejector refrigeration cycle was known as early as 1901 with the development of a steam jet refrigerator by Le Blanc and Parson (Sokolov and Hershgal. 1989). The cycle is mainly activated by heating the liquid refrigerant in vapor generator to produce system primary energy in form of high pressure and temperature vapor. The performance on P-h diagram of a simple ejector refrigerant cycle is schematically shown in Fig. 2.

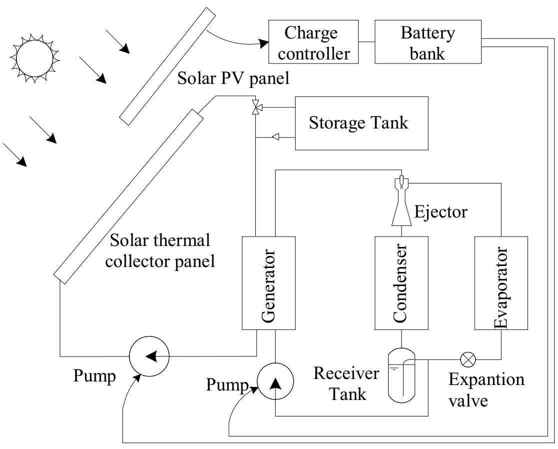

Fig. 1: Solar ejector refrigerant system

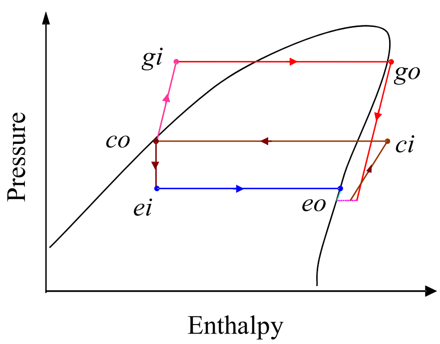

Fig. 2: P-h diagram of simple ejector refrigerant cycle

The design of ejector is explained in detail in another report (Chan et al. 2011).

The objectives of this study are to confirm the actual performance of an ejector refrigeration cycle by using indoor test equipment under stable conditions and to clarify the individual performance of designed ejector itself.

The test equipment was set up and experiment was carried out at 58 – 69 °C of generating temperature T_g and 5 – 15 °C of evaporating temperature T_e.

Nomenclature

P	pressure	Pa	Subscripts	
T	temperature	K, °C	g	generator
A	exergy	J	c	condenser
Q	heat rate	W	e	evaporator
W	Power, electricity	W	i	inlet
m	mass flow rate	$kg\cdot s^{-1}$	o	outlet
h	enthalpy	$J\cdot kg^{-1}$	p	pump
d	distance	mm		
COP	coefficient of performance	–	Superscripts	
η	efficiency	–	*	critical or optimum point

2. Experimental set up

2.1 Experimental system

Figure 3 shows an actual test equipment and Fig. 4 shows a schematic diagram of the experimental set up, which consists of vapor generator, condenser, evaporator, receiver and separator tanks, pump, heat source and heat sink simulators and two ejectors. As mentioned earlier, the system is designed to operate as double or single-ejector cycle and the cycle can be switched by using valves. The present study didn't use the second ejector, which was designed to use liquid working fluid and enhance much lower pressure condition in evaporator. The vapor generator has been designed to operate at pressure up to 5 MPa while the actual maximum operating pressure is expected to be lower than 4 MPa corresponding to the vapor pressure of propane at about 93 °C. A pressure relief valve is installed and set to open at 4 MPa. Due to high operating pressure, helicoidally tubing coil made from stainless steel is used. The liquid receiver and vapor-liquid separator are also made of stainless steel and equipped with level glass for tracking the liquid level in the receiver and separator. Brazed-plate heat exchangers are used for condenser and evaporator because they are compact and suitable for small scaled laboratory test equipment.

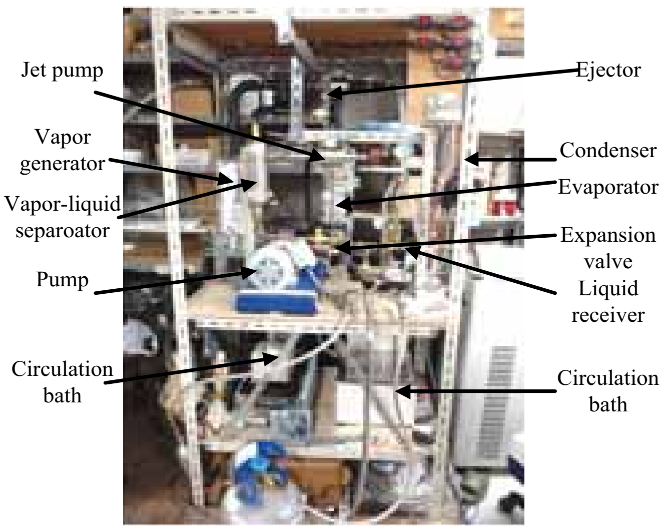

Fig. 3: Experimental set up

The vapor generator heat source, which will be supplied by solar collector in the actual case, is simulated with a circulating bath that can supply heat at controllable temperature up to 200 °C and 0.01 °C of precision. In similar manner, the cooling load is also simulated by a circulating bath. The heat from condenser is removed by a cooling bath.

All the temperature measurements at "T" in Fig. 4 are made by using K-type thermocouples. The flow rates of heat transfer fluids, water, for circulating baths of vapor generator and evaporator are measured by using Keyence FD-V70 series flow sensors with sensor head FD-P20 having the resolution of 0.1 liter and uncertainties of 0.1 % F.S. The pressures of working fluids are measured at "P" in Fig. 4 by using Keyence AP-V80 series pressure sensors and the sensor heads AP-14S and AP-13S. The uncertainties of pressure sensors are 0.1 % F.S.

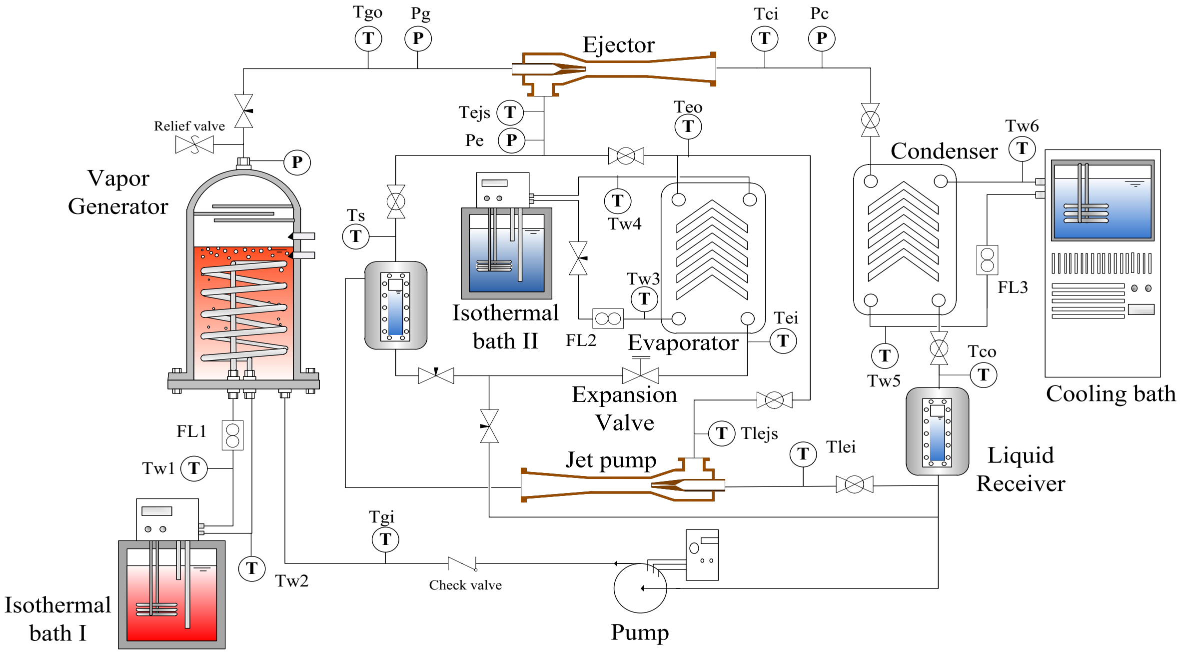

Fig. 4: Schematic diagram of a whole set of test equipment

Figure 5 shows some important dimensions of the ejector used in the test equipment. The ejector is designed to allow change of nozzle position along axial direction to alter distanced between nozzle exit and the inlet of mixing section. The nozzle throat and the mixing section constant-area diameters are important in the ejector to be designed for operating at a required condition. The theoretical predictions of these dimensions were reported in Chan et al. 2011.

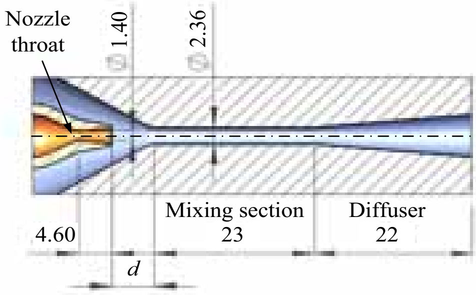

Fig. 5: Ejector important dimensions mm

The refrigerant used in the system was determined based on the thermodynamic properties of four different refrigerants of propane, isobutene, and n-butane, and a hydro-fluorocarbon of R134a as explained in a previous paper by Chan et al, 2011. From the calculation results, propane shows the better performance as a working fluid than R134a. However, R134a was selected to be used in test equipment because propane is a flammable gas, which is not suitable for indoor test equipment due to safety reason. In the generator, R134a is boiled by electric heater. In experiments the generator was operated at temperatures between 58 and 69 °C instead of the stored thermal energy in the actual case. In the condenser, R134a vapor condenses by using cooling water cooled by cooling bath, which will be replaced by a cooling tower in case of an actual system.

2.2 Experimental Procedure

The operating procedure of the developed ejector system is simple. Firstly, the circulating thermostat baths for vapor generator and evaporator and also the cooling bath for condenser are switched on. When heat is provided to liquid refrigerant in generator, boiling occurs and vapor generation rate is getting higher than the vapor flow rate through the nozzle of ejector and the pressure in generator rises. When pressure reaches an equilibrium condition, the vapor flow rate at the nozzle also reaches at chocking flow. The vapor-generator pressure and temperature can be controlled by the temperature. On the other hand, the evaporator pressure does not depend on the generator temperature, while it can be controlled by adjusting the opening of expansion valve. The super-heated vapor at evaporator exit can be controlled by temperature. The condenser pressure can be changed by varying temperature. In the design, liquid level in generator can be detected by two optical level switches and the signals are used for controlling the operation of the pump. However, the level switches did not work correctly in this study; therefore the pump operation was controlled manually with an inverter speed controller by observing the liquid level in receiver tank through a level glass.

3. Experimental results

3.1 COP analysis

Coefficient of performance, COP, is used to verify the performance of a developed ejector refrigeration system. COP is defined in this study as the ratio of the cooling capacity and the total input heat in vapor generator. In ejector refrigerant cycle, the total input energies are the input heat at generator and the energy for driving the pump used for circulating refrigerant. However the power consumed by pump is significantly small as a comparison with the input heat at the vapor generator, therefore COP of ejector refrigerant cycle is defined as

$$\mathrm{COP} = \frac{Q_e}{Q_g} \qquad \text{(eq. 1)}$$

For vapor generator, condenser and evaporator, the pressures and temperatures are used as the values at saturation states. When the quality of available heat sources is concerned, temperature is a better indicator of the system operating condition.

The optimum pressure is commonly defined as the critical pressure. The COP of ejector system becomes constant when the system is operated at condenser pressures lower than the critical point. When the operating condenser pressure is higher than the critical pressure, the COP begins to decrease drastically with increaseing condenser pressure.

Figures 6 shows the relation between COP and condensing temperature T_c at the conditions of T_g = 64 °C, T_e = 15 °C and nozzle position d = 3.0 mm. As shown in Fig. 6, the maximum COP is about 0.35 and the critical condensing temperature T_c* that COP begins to decrease is 29 °C at this condition.

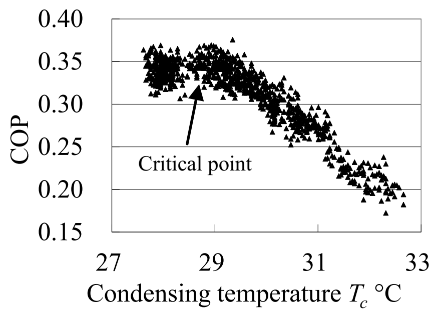

Fig. 6: Results at T_g = 64 °C, T_e = 15 °C and d = 3.0 mm

3.1.1 Nozzle position

Figure 7 shows relation between COP and condensing temperature at the condition of T_g = 64 °C, T_e = 10 °C and d = 1.5, 3.0 and 4.5 mm. As shown in Fig. 7, COP is the highest at d = 3.0 mm and lowest at d = 4.5 mm. Figure 8 shows the relation between maximum COP and nozzle position at the condition of T_g = 64 °C, T_e = 10 °C. As shown in Fig. 8, COP is drastically decreasing when nozzle position is changed from d = 3.0 mm to d = 4.5 mm., It indicates that the peak of COP in designing of present ejector at the condition of T_g = 64 °C and T_e = 10 °C is d = 3.0 mm.

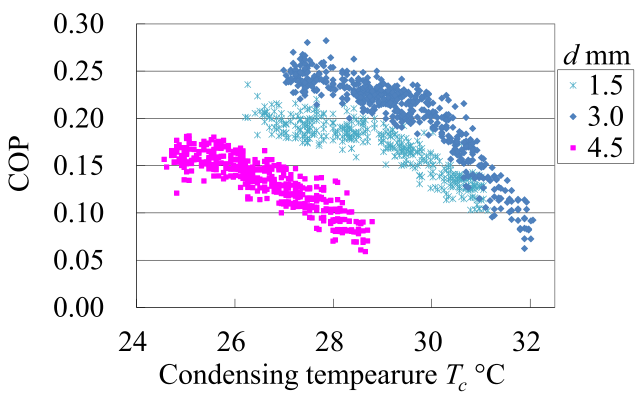

Fig. 7: Results at T_g = 64 °C, T_e = 10 °C and d = 1.5, 3.0, 4.5 mm

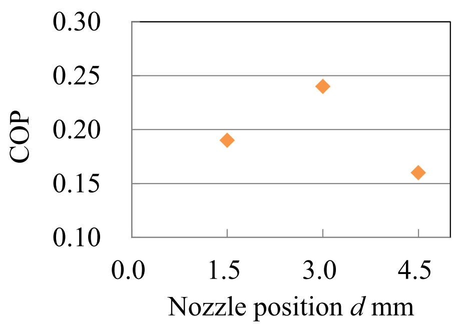

Fig. 8: Relationship between COP and nozzle position d at T_g = 64 °C, T_e = 10 °C

3.1.2 Generating and evaporating temperatures

The performance of the present ejector designed in this study was obtained as a function of generating and evaporating temperatures. Figure 9 shows the relation between COP and T_g at T_e = 10, 15 °C and d = 3.0 mm. As shown in Fig. 9, it was confirmed that COP decreases from 0.33 to 0.20 with increasing in T_g from 58 °C to 69 °C at T_e = 10 °C and from 0.38 to 0.26 with increasing in T_g from 58 °C to 69 °C at T_e = 15 °C. Besides, COP is increasing when T_e is increasing from 10 °C to 15 °C. Figure 10 shows the relation between COP and T_e at T_g = 58, 64, and 69 °C and d = 3.0 mm. As shown in Figs. 10, it was confirmed that COP increases with increasing in T_e. In particular, at the condition of T_g = 64 °C, COP increases to 34 % from 22 % with increasing in T_e from 10 °C to 15 °C.

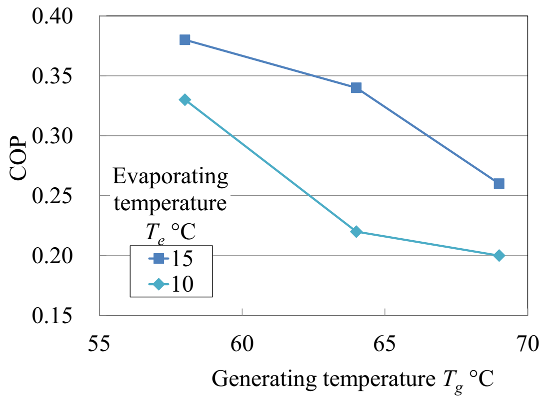

Fig. 9: Relashionship between COP and T_g at T_e = 10, 15 °C and d = 3.0 mm

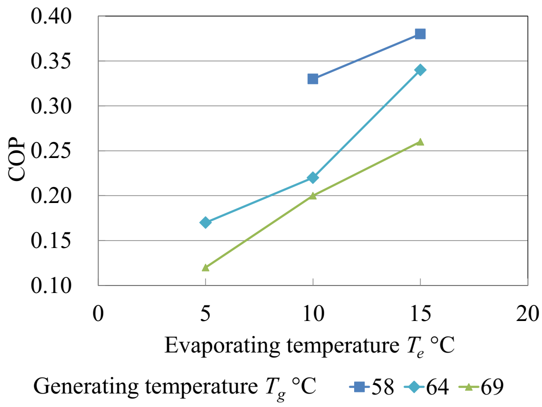

Fig. 10: Relashipnship between COP and T_e at T_g = 58, 64, 69 °C and d = 3.0 mm

3.2 Exergy analysis

Experimental COP is relatively lower than theoretically expected value. The ejector refrigerant cycle is consumes only solar energy at temperatures of intermediate level just like exhaust heat, which is not utilized in many industries. Therefore, it is suitable to be evaluated by exergy efficiency for knowing the reachable highest efficiency and comparing with other energy conversion systems.

Exergy efficiency η_E of ejector refrigerant cycle is defined as the ratio of the exergy at evapolator and the total input exergies. In ejector refrigerant cycle, the total input exergies are the input exergy at generator and the exergy for driving the pump used for circulating refrigerant. In case of COP, it can ignore the electricity consumption of the pump can be ignore because of small amount in energy from the view point of heat in the generator but it is not the case for exergy analysis. Therefore exergy efficiency η_E of ejector refrigerant cycle is defined as eq. 2.

$$\eta_E = \frac{A_e}{A_g + W} \qquad \text{(eq. 2)}$$

When temperatures of generator and evaporator are kept at constant, exergy can be simply defiend as eq. 3.

$$A = Q\left(1 - \frac{T_0}{T}\right) \qquad \text{(eq. 3)}$$

Electricity consumption at pump is calculated by eq. 4.

$$W = \frac{m_p (h_{gi} - h_{co})}{\eta_p} \qquad \text{(eq. 4)}$$

From eqs. 2-4 and experimental result, η_E reaches 0.15 at T_g = 64 °C, T_e = 15 °C and T_0 = 30 °C.

If T_e and T_0 of vapor compression refrigerant cycle are the same as this condition, ejector refrigerant cycle is corresponding to the η_E of vapor compression refrigerant cycle whose COP is 2.9.

4. Challenges for the future

The critical condensing temperatures of ejector refrigerant cycle using the present ejector would be relatively low from the consideration of condensing heat being available from ambient air. Figure 13 shows the relation between T_c^* and T_g. T_c^* is the highest at 33 °C in case of $T_g = 69$ °C and $T_e = 15$ °C, it can be easier to get from ambient air. On the other hand, it is possible to condense at lower temperatures, when water is cooled by cooling tower, although it requires additional energy consumption. There are trade-off relation between COP and T_c^* for the present ejector. It is required to investigate the optimum design of ejector itself, which has higher condensing temperature.

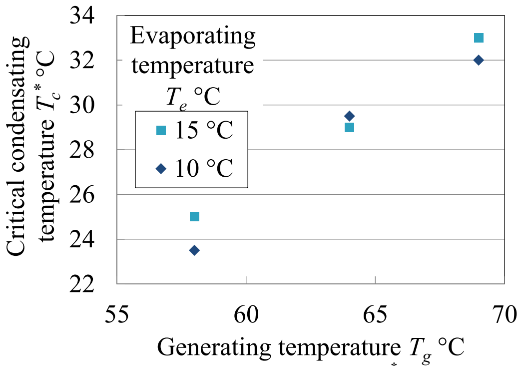

Fig. 13: Relationship between T_c^* and T_g at $T_e = 10$, 15 °C and $d = 3.0$ mm

In our experimental equipment, the heat for generating the vapor is provided from isothermal bath and the cooling water to condense the refrigerant is provided from cooling bath. These will be replaced by solar thermal panel and cooling tower, respectively, for the actual system.

5. Conclusion

The experiments of ejector refrigeration cycle were carried out for a single ejector cycle. The COP reached about 0.35 and exergy efficiency reached 0.15 at $T_g = 64$ °C, $T_e = 15$ °C, $T_c = 29$ °C.

The experiments in order to clarify the characteristics of nozzle position were carried out. From the results, there is a peak of COP when nozzle position is moved. In designing of present ejector, the optimal nozzle position was about 3.0 mm at any conditions.

In design of present ejector, it is confirmed that the COP is decreasing with increasing in generator temperature whereas critical condenser temperature is increasing with increasing in generator temperature. On the other hand, it is expected that COP is increasing with increasing in generator temperature if the design of ejector is optimized in the generator temperature.

From these results, it was confirmed that the actual performance of an ejector refrigeration cycle by using indoor test equipment under stable conditions and clarified the individual performance of designed ejector itself.

6. Acknowledgements

The author would like to thank the Daikin Industries for offering refrigerant for the experiment. This work was supported in part by Grant in Aid for the Global Center of Excellence Program “Center for Education and Research of Symbiotic, Safe and Secure System Design” from the Ministry of Education, Culture, Sport, and Technology in Japan.

7. References

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