Experimental Studies of A Steam Jet Refrigeration Cycle: Effect of The Primary Nozzle Geometries to System Performance.

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Abstract

This paper describes an experimental investigation of a steam jet refrigeration. A 1 kW cooling capacity experimental refrigerator was constructed and tested. The system was tested with various operating temperatures and various primary nozzles. The boiler saturation temperature ranked from 110 to 150^{O}C. The evaporator temperature was fixed at 7.5^{O}C. Eight primary nozzles with difference geometries were used. Six nozzles have throat diameters ranked from 1.4 to 2.6 mm with exit Mach number of 4.0. Two remained nozzles have equal throat diameter of 1.4 mm but difference exit Mach number, 3.0 and 5.5.

Keywords: Refrigeration, Ejector, Steam-Jet

1. Introduction

In many industrial processes, an amount of heat is rejected to the surroundings as waste. If this waste heat is converted to useful refrigeration by using heat powered refrigeration systems, electricity purchased from utility companies for conventional refrigeration cycles can be reduced. The one widely used heat powered refrigeration cycles is absorption jet refrigeration cycles [1] because the jet refrigeration is relatively simple to construct, operate, and control. It uses only single-component working fluid (refrigerant only). Moreover, the jet refrigeration system is the only refrigeration system that can use water, the most environmentally friendly and cheapest fluid, as the refrigerant. However, this system has a COP less than other kinds of refrigeration system.

Performance of steam jet refrigeration is strongly dependent on the ejector equipped. In the past, a small scale steam jet was studied experimentally. Effects of the operating temperatures and effect of the primary nozzle position [2 and 3] were carried out. Some researchers used CFD technique to explain the process inside the ejector [4].

It this paper, effects of the primary nozzle's geometries and the operating conditions on the ejector performance were concerned. The ejector was tested with eight difference primary nozzles. The nozzle's throat diameters were 1.4, 1.7, and 2.0mm. The nozzle's area ratios were 7:1 (Mach number of 3), 20:1 (Mach number of 4), and 88:1 (Mach number of 5.5). The boiler temperatures were between 110 and 150^{O}C. The evaporator temperatures were fixed at

7.5°C. The tests showed that the throat diameter and the area ratio of the primary nozzle had strong effects to the ejector performance.

2. Background

A schematic view of a steam ejector is shown in Fig.1. A primary fluid expands and accelerates through the primary nozzle. This creates a very low pressure region at the nozzle exit plane and subsequently in the mixing chamber. This low pressure region draws a *secondary fluid* from the evaporator into the mixing chamber. The primary fluid and the secondary fluid then mix together in the mixing chamber. By the end of the mixing chamber, a normal shock of or series of oblique shocks are induced [4 and 5]. The shock causes a major compression effect and a sudden drop in the flow speed from supersonic to subsonic. A further compression of the flow is achieved as it is brought to stagnation through a subsonic diffuser. The ejector is discharged at a pressure (back pressure) equal to the saturation pressure in the condenser.

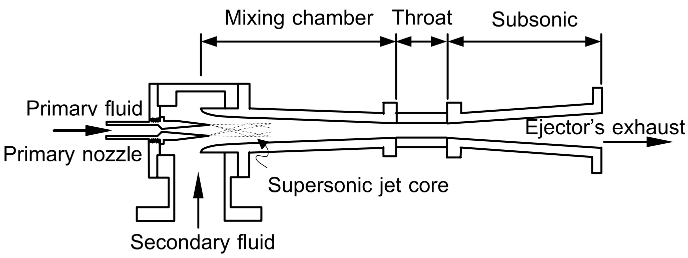


Fig.1 Schematic view of a steam ejector

An important parameter used to describe the performance of an ejector is an *entrainment ratio*:

$$Rm = \frac{\text{mass flow of the primary fluid}}{\text{mass flow of the secondary fluid}} \qquad \text{Eq(1)}$$

In a steam jet refrigeration cycle as shown in Fig. 2, an ejector entrains a low pressure saturated water vapour from the evaporator, where the refrigeration effect is produced, as the secondary fluid. It uses a hot and high pressure saturated steam from the boiler as the primary fluid.

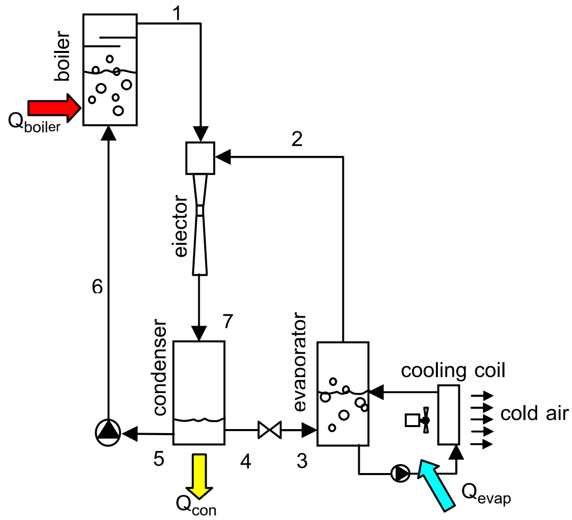


Fig.2 A schematic view of a steam jet refrigeration cycle

The ejector discharges its exhaust to the condenser where the fluid is condensed to liquid by rejecting heat out to the surrounding. Performance of a steam jet refrigeration is defined in term of the Coefficient of Performance for a steam jet refrigeration:

$$COP = Rm \cdot \frac{h_{g-boiler} - h_{f-con}}{h_{g-evap} - h_{f-con}} \qquad \text{Eq(2)}$$

Since the enthalpy change at the boiler is not much different from that at the evaporator, it can be assumed that:

$$COP \approx Rm \qquad \text{Eq(3)}$$

Fig.3 shows a typical performance of a steam ejector. When the boiler and evaporator temperatures are fixed and the condenser pressure is varied, the ejector's performance curve is divided into three regions, *choked flow*, *unchoked flow*, and *reversed flow*[1].

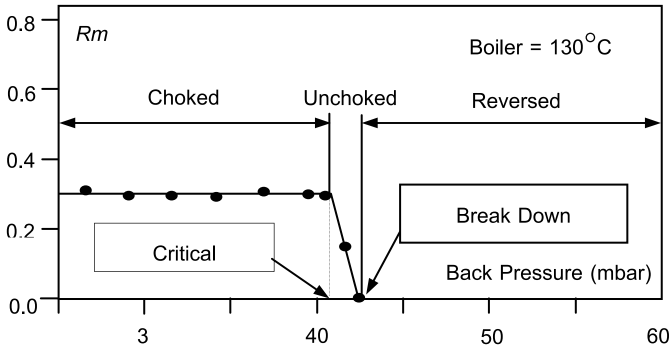


Fig. 3 Performance of a steam jet refrigerator based on experimental data provided by Chunnanond [1]

Fig 4. shows the effect of operating pressures on the performance of the steam ejector based on experimental data [1]. When decrease the primary fluid pressure (boiler saturation temperature), the entrainment ratio will increase but the ejector will operate with a lower critical back pressure. When increase the secondary fluid pressure (evaporator saturation temperature), both the critical back pressure and the entrainment ratio will be increased.

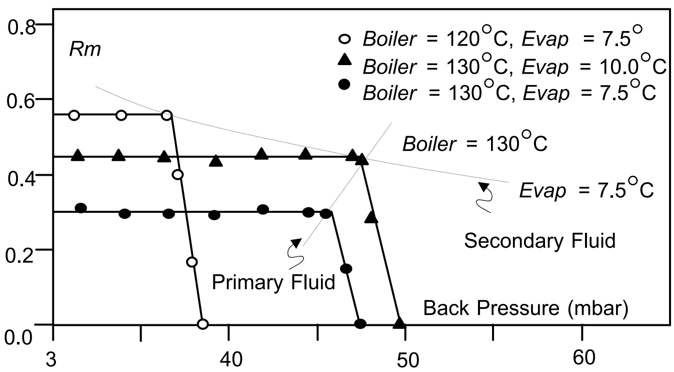


Fig. 4 Performance characteristics of a steam ejector based on experimental data provided by Chunnanond [1]

3. Experimental setup

3.1 Experimental steam jet refrigerator

The schematic diagram of an experimental steam jet refrigerator is shown in Fig 5. In this system, electric heaters were used as simulated heat source and cooling load. The maximum heating capacity at the boiler was 8 kW. A 2 kW heater was used to simulate the cooling load. To ensure that only dry vapour entered the primary nozzle, the saturated steam from the boiler was superheated by 1 to 2°C by using a 500 W (adjustable power) superheater. The condenser used was shell and coil type and was cooled by water. A pneumatic diaphragm pump was used as the boiler feed pump and a magnetic coupled centrifugal pump was used to promote evaporation rate at the evaporator.

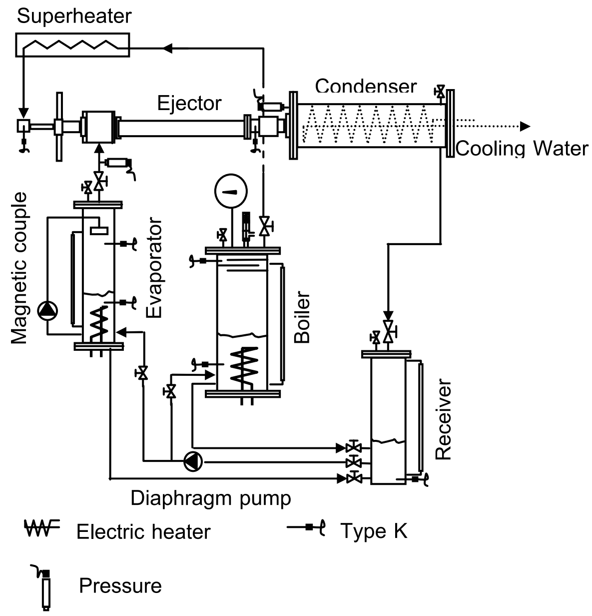
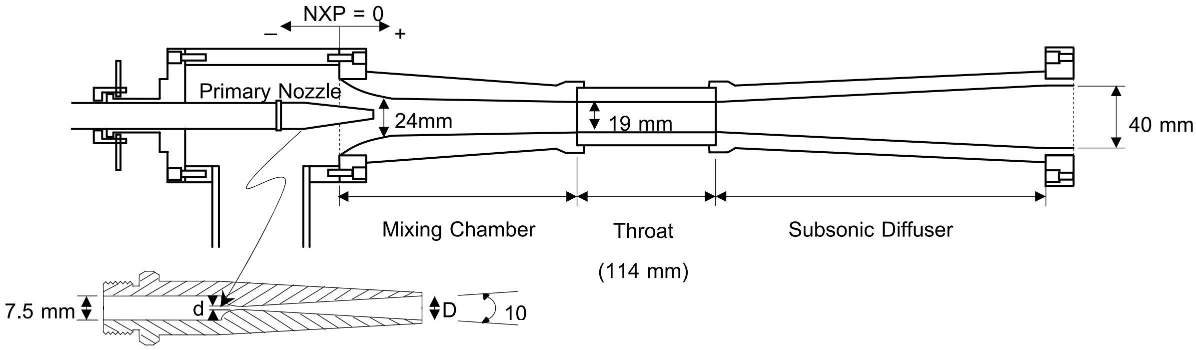


Fig. 5 The experimental steam jet refrigerator

3.2 Experimental steam ejector

In this study, six primary nozzles were used as shown in Fig 6. During the tests, all nozzles were placed at NXP value of +23 mm The NXP (Nozzle Exit Position) was defined as a distance between the primary nozzle exit plane and the mixing chamber inlet plane.



Nozzle code	D (mm)	D : d	Calculated exit Mach number
D1.4 M4	1.4	20:1	4.0
D1.7 M4	1.7		
D2.0 M4	2.0		
D2.6 M4	2.6		
D1.4 M3	1.4	7:1	3.0
D1.4 M5.5	1.4	88: 1	5.5

Fig. 6 The experimental steam ejector

4. Experimental results

4.2 Effect of the primary nozzle's throat diameter

In these tests, the boiler and the evaporator saturation temperatures were fixed at 150°C and 7.5°C respectively. Nozzle D1.4M4, D1.7M4, and D2.0M4 were used. All the nozzles have an area ratio of 20:1 which produce equal exit Mach number of 4.0. Fig 7. shows effects of the primary nozzle's throat diameter to the entrainment ratio.

The nozzle with larger throat diameter can provide the higher primary fluid mass flow rate than that for the smaller one. Therefore, less flow area in the mixing chamber for the secondary fluid to be entrained which results in a lower entrainment ratio is induced. However, at the nozzle exit, larger momentum and kinetic energy of the primary fluid is produced. This results in a higher critical condenser pressure which is similar to the case of an increase in the boiler saturation temperature.

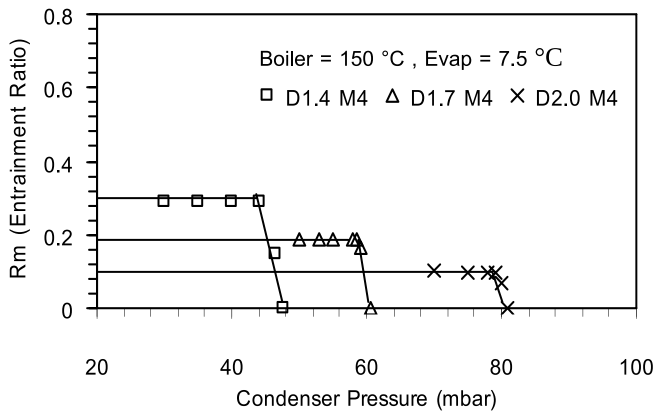


Fig.7 Variation of the entrainment ratio with the nozzle's throat diameter

4.3 Effect of the nozzle's exit Mach number

In these tests, nozzles D1.4M3, D1.4M4, and D1.4M5.5 were used. The boiler saturation temperature and the evaporator temperature were fixed at 150°C and 7.5°C respectively.

The three nozzles have equal throat diameter of 1.4 mm. They provide the same critical mass flow rate but difference exit Mach number. Fig.9 shows effects of the nozzle's exit Mach number to the entrainment ratio.

From Fig. 8, the entrainment ratio in choke flow region is independent from the change of the Mach at the nozzle exit. All nozzles entrain the same amount of the secondary fluid. However, the critical condenser pressure is increased with the Mach number.

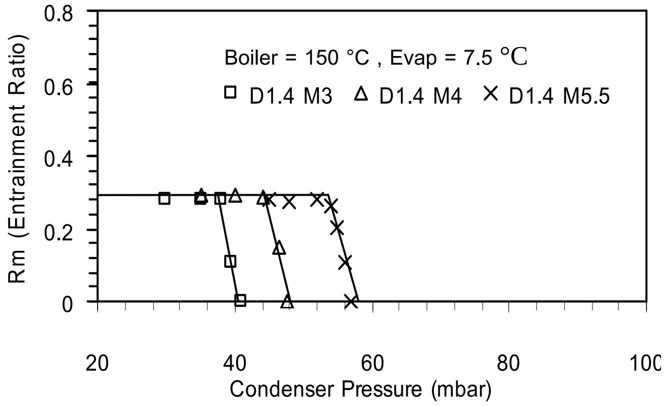


Fig.8 Variation of the entrainment ratio with the nozzle's exit Mach number

This is due to the momentum of the primary fluid, the higher the Mach number results in the higher the momentum of the flow.

4.3 Effect of the primary nozzle's throat diameter with fixed critical mass flow rate

In these tests, evaporator temperature was fixed at 7.5°C. Nozzles D1.4M4, D2.0M4, and D2.6M4 were used. These nozzles had different throat diameter but has the same area ratio. They produced equal exit Mach number of 4.0. During the tests, the boiler saturation temperature was adjusted so that the critical mass flow rate was approximately fixed at 4.6 ±0.05 kg/hour as shown in table 1.

Table 1. Boiler temperature and critical mass flow rate

Nozzle	T_{boiler} (°C)	Critical mass flow (kg/h)
D1.4M4	150.0	4.568
D2.0M4	130.0	4.537
D2.6M4	111.2	4.608

Since the Mach number and mass flow rate of the primary fluid leaving each nozzle was fixed, momentum of the primary flow was the same for all nozzles. One would expect that, both the entrainment ratio and the critical condenser pressure for all nozzles would be very similar. However, from the tests it was not. Fig. 9 shows the variation of the entrainment ratio.

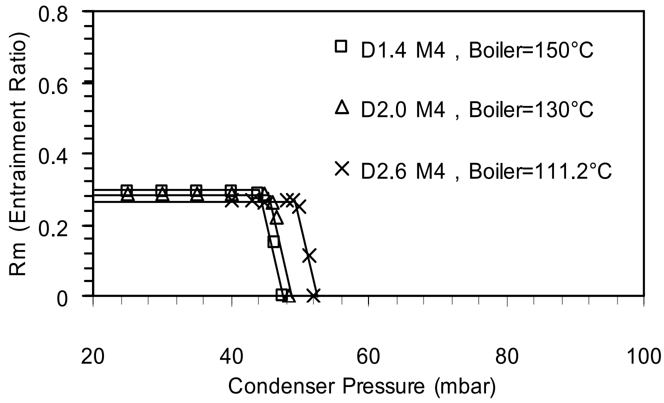


Fig.9 Variation of the entrainment ratio when using various nozzles but fix the critical mass flow rate and the exit Mach number

It can be seen that when using a large nozzle with low boiler pressure, the entrainment ratio is slightly decrease but provides a higher critical condenser pressure.

The slightly reduction in the entrainment ratio may be resulted from a larger core of the primary fluid which results in a smaller effective flow area for the secondary fluid. The raise in the critical condenser pressure may be resulted from a lower secondary fluid entrained. However, the entrainment ratio is only slightly decreased. If this is the main reason, the critical condenser pressure should not be significantly increased. Another reason may be caused by the mixing process between the primary and secondary fluids in the mixing chamber. In order to explain this phenomenon, a further study using CFD technique may be used.

5. Conclusions

In this study, the experimental steam jet refrigerator was tested with 6 different geometries primary nozzles. In all tests, the evaporator saturation temperature was fixed at 7.5°C. The boiler saturation temperature was between 110°C and 150°C. The primary nozzles used had throat diameters between 1.4 mm to 2.6 mm. The nozzles produced the exit Mach number from 4.0 to 5.5.

When the boiler and evaporator saturation temperatures are fixed and use several primary nozzles with different throat diameter but the same area ratio. These nozzles produce different critical mass flow rate but the same Mach number. The entrainment ratio decrease when a larger nozzle is used but the ejector can be operated with a higher condenser pressure, and vice versa.

When the boiler and evaporator saturation temperatures are fixed and use several primary nozzles with the same throat diameter but different area ratio. These nozzles produce the same critical mass flow rate but different exit Mach number. The entrainment is essentially constant and independent from the area ratio of the primary nozzles. However, the nozzle with larger area ratio, which produces a high Mach number, is able to be operated with a higher critical condenser pressure.

When several nozzles with different throat diameter but the same area ratio are used, these nozzles produce the same exit Mach number. If the boiler saturation temperature is allowed to varied so that the critical mass flow rate through each nozzle are constant. Therefore the nozzle with larger throat diameter is operated at a lower boiler saturation temperature, and vice versa. The nozzle with larger throat diameter (with lower boiler saturation temperature), will entrain slightly less amount of the secondary fluid from the evaporator. However it can be operated at a higher condenser pressure.

6. Acknowledgement

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7. References

[1] K. Chunnanond, S. Aphornratana (2004). Ejectors: applications in refrigeration technology, Renewable and Sustainable Energy Reviews, vol. 8(2), April 2004, pp.129-155.

[2] W. Eames, S. Aphornratana, H. Haider, A theoretical and experimental study of a small-scale steam jet refrigerator (1995). International Journal of Refrigeration, vol.18 (6), July 1995, pp. 378-386.

[3] S.Aphornratana, I. W. Eames (1997). A small capacity steam-ejector refrigerator: experimental investigation of a system using ejector with movable primary nozzle International Journal of

Refrigeration, vol.20 (5), August 1997, pp.352-358

[4] T.Sriveerakul,S.Aphornratana,K.Chunnanond (2007). Performance prediction of steam ejector using computational fluid dynamics: Part 1. Validation of the CFD results, International Journal of Thermal Sciences, Vol. 46(8), August 2007,pp.812-822.

[5] Y.A.Cengel, M.A.Boles (2007), Thermodynamics: An engineering approach 6th edition, McGraw-Hill.