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# Experimental investigation of an adjustable ejector for $CO_2$ heat pump water heaters*

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**Abstract:** An adjustable ejector expansion device for a $CO_2$ heat pump water heater (HPWP) is proposed to improve the system performance. It has been designed to investigate experimentally the effects of the motive nozzle throat area of the ejector, entrained flow pressure, back pressure and primary flow pressure on the entrainment ratio. Experiments based on different motive nozzle throat areas were conducted and the results of the prototype ejector using $CO_2$ as working fluid are presented. The results show that an adjustable ejector can achieve high performance and work well in a wide range of working conditions.

**Key words:** Adjustable ejector, System performance, $CO_2$ heat pump water heater (HPWH), Entrainment ratio
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## INTRODUCTION

$CO_2$ is being advocated as one of the natural refrigerants to replace chloro-fluoron-carbons (CFCs) and hydrogen containing chlorofluorocarbonds (HCFCs) in vapor compression systems because of its excellent thermodynamic transport properties and environmentally friendly characteristics. However, the lower efficiency of the basic transcritical $CO_2$ refrigeration cycle compared to conventional vapor compression systems using HFC and HCFC as refrigerants is an inherent drawback of the technology, hindering progress towards practical applications. To decrease the expansion losses and increase the cycle efficiency, it has been proposed to replace the expansion valve with an ejector expansion device. Its advantages of low cost, no moving parts and an ability to handle two-phase flow without damage make it attractive as an expansion device (Groll, 2006; Hrnjak, 2006; Groll and Kim, 2007).

The ejector cycle was invented by Gay (1931). Initially, ejector design was based mainly on trial-and-error experiments until Keenan *et al*.(1950) published one of the first analytical ejector theories for perfect gases. Stoecker (1958) suggested one of the first iterative ejector design methods involving real property data taken from steam tables. His analysis calculated properties downstream of a normal mixing shock by graphically intersecting the lines of Fanno and Rayleigh flows. Kornhauser (1990) analyzed the thermodynamic performance of the ejector-expansion refrigeration cycle using R12 as a refrigerant based on a constant mixing pressure model and found a coefficient of performance (COP) improvement of up to 21% over the standard cycle. Liu *et al*.(2002) performed a thermodynamic analysis of the transcritical $CO_2$ vapor compression/ejection hybrid refrigeration cycle. Similar numerical work and results were presented by Jeong *et al*.(2004), with COP improvements of up to 22% over the conventional expansion valve cycle, Li and Groll (2005) who found COP improvement of more than 16% for typical air conditioning operation conditions, and Deng *et al*.(2007) who included an internal heat exchanger in their analysis. Furthermore, Boumaraf and Lallemand (2008) developed a whole simulation

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program to investigate the effects of the operating conditions on the characteristics of the ejector with R142b and R600a. Also, Yapıcı (2008) verified that the performance of such an ejector refrigeration system could be improved through better design and manufacture of the ejector. Although a large number of studies on numerical ejectors have been published, very little experimental data on $CO_2$ ejectors is available.

From previous studies, we can see that the performance of a conventional fixed-structure ejector under given working conditions is determined mainly by two dimensionless parameters: the ratio of the section area of the cylindrical mixing-chamber to that of the nozzle throat, that is $A_3/A_{p*}$, and the ratio of the cross-section area of the nozzle exit to that of the nozzle throat, namely $A_{p1}/A_{p*}$. If one of these parameters is adjustable, the performance of the ejector can be optimized by changing the adjustable parameter when the working conditions change.

This paper proposes an adjustable ejector with a variable nozzle throat area, which can function in the range from on-design point to off-design. The main aim was to investigate the effect of the motive nozzle throat area of the ejector on the entrainment ratio and other relevant parameters of a $CO_2$ heat pump water heater (HPWP) system that uses an adjustable ejector as an expansion device.

## EJECTOR

To analyze the parameters of the ejector, an adjustable ejector was designed to be applied to a $CO_2$ HPWP with about 10 kW heating capacity. This specified ejector prototype had an adjustable motive nozzle throat area.

The $CO_2$ adjustable ejector includes a nozzle needle, a motive nozzle, a suction chamber, a mixing chamber and a diffuser (Fig.1). Chunnanond and Aphornratana (2004) pointed out that the constant-pressure mixing ejector, in which the ejector nozzle exit is located in front of the constant area section, had a better performance than the constant-area mixing ejector, in which the ejector nozzle exit is located within the constant-area section. Therefore, in this work, the constant-pressure mixing ejector was used. The divergent and convergent angles, nozzle, constant-pressure chamber, diffuser, and the constant area chamber length were based on the results of Zha and Jakobsen (2007).

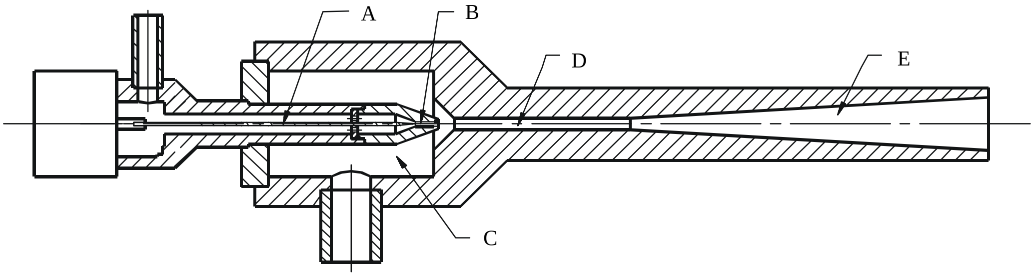
To evaluate and assess the ejector performance, knowledge of the entrainment ratio definition demonstrated by Eq.(1) was required:

$$u = G_H / G_P, \tag{1}$$

where $G_H$ and $G_P$ are the mass flow rates of entrained flow and motive flow, respectively.

## EXPERIMENTAL

The schematic diagram of the experimental apparatus is shown in Fig.2. This system consists of three main loops: the refrigerant loop, the cold-water loop, and the hot-water loop. The refrigerant $CO_2$ loop consists of a compressor, a gas cooler, an ejector, an evaporator, and other accessory parts. All of the components support high-pressure of more than 12.5 MPa, but a safety valve is mounted with 12.5 MPa as release pressure. The main differences between this apparatus and a conventional system are the additions



A: nozzle needle; B: motive nozzle; C: suction chamber; D: mixing chamber; E: diffuser

**Fig.1 The prototype of the adjustable ejector**

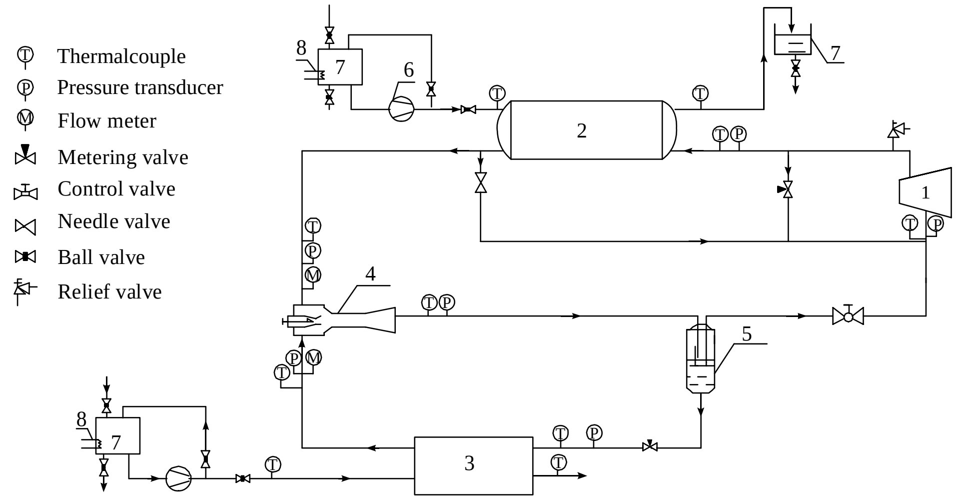


**Fig.2 Schematic diagram of $CO_2$ heat pump water heater with ejector**
1: compressor; 2: gas cooler; 3: evaporator; 4: ejector; 5: liquid-vapor separator; 6: pump; 7: water tank; 8: electrical heater

of a two-phase adjustable ejector with a nozzle needle and a liquid-vapor separator.

The motive and suction mass flow rates were measured using a micro motion mass flow meter with high accuracy. The total range of all refrigerant flow meters was 0~0.2 kg/s. The accuracy of the flow measurement was within ±0.1% of the reading. The temperatures were measured using T-type thermocouples with an accuracy of ±0.5 °C. Absolute pressure transducers were used to measure the pressures. All static pressure taps were mounted flush in the tube wall.

The effect of ejector dimensions on the system performance was studied by changing the motive nozzle throat area. A laterally tapered needle extending into the throat of the motive nozzle was installed in the ejector to enable the motive nozzle throat area to be varied. The needle was connected to a step by step motor (Fig.3) through an electromagnetism loop in the end and was controlled by a programmed driver (Fig.4). When the nozzle needle is moved to the downstream dead end, the motive nozzle throat area reaches a minimum. A maximum is obtained by inputting corresponding electronic pulse quantities.

**Fig.3 Adjustable ejector with nozzle needle**

**Fig.4 Driver of nozzle needle**

In this investigation, the inner diameter of the mixing section of the ejector was 5 mm and the diffuser exit diameter was 16 mm. The motive nozzle throat area could be adjusted from 0 to 2.01 $mm^2$.

Test runs were carried out in different operating conditions which were manipulated using valves including the control valve installed in the suction pipe of the compressor, the metering valve between liquid-vapor separator and evaporator, the needle valve installed in the bypass, as well as the ball valves at the inlet of the cold-water and hot-water systems. Generally, to maintain the operating parameters within the range of experiment conditions, these valves were regulated at the same time.

## RESULTS AND DISCUSSION

Fig.5 shows the entrainment ratio ($u$) as the function of entrained flow pressure ($P_H$) and motive nozzle throat area ($A_{p*}$). The test was conducted under the conditions that back flow pressure $P_C$ and primary flow pressure $P_P$ were fixed at 3.9 and 9.5 MPa, respectively. It was evident that the entrainment capability of the ejector greatly depends on the motive nozzle throat area. More energetic entrained flow can result in a high entrainment ratio. The entrainment ratio is proportional to entrained flow pressure with a relation of linear approximation and decreases with the motive nozzle throat area. Furthermore, we found that the entrainment ratio had a relatively great dependence on the entrained flow pressure.

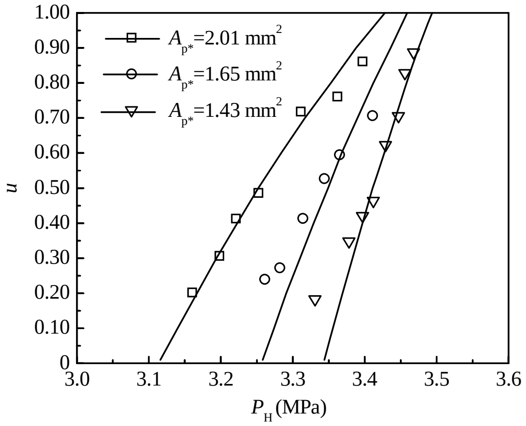


**Fig.5 Effect of entrained flow pressure on entrainment ratio with different motive nozzle throat areas**

Fig.6 shows the curves of $u(P_C, A_{p*})$ when $P_H$=3.4 MPa and $P_P$=9.5 MPa. The entrainment ratio $u$ varied inversely with the increase in mixing flow pressure. A higher back pressure led to a lower entrainment ratio. From an engineering standpoint, it was important to balance carefully the entrainment ratio and the balance of pressure lift. We could not increase the back pressure above a critical level. For example, with the motive nozzle throat area of 2.01 mm$^2$, when the back pressure increased as high as 4.18 MPa, the ejector dramatically deviated from normal conditions. This pressure was considered as the critical back pressure. The results showed that the ejector was able to work under a certain range of back pressure, but when this range was exceeded, the performance of the ejector dropped abruptly. The smaller the motive nozzle throat area, the lower the critical back pressure.

Fig.7 was obtained under an entrained flow pressure of 3.4 MPa and a back pressure of 3.9 MPa. It shows the influences of primary flow pressure and motive nozzle throat area on the entrainment ratio. It is clear that the entrainment ratio increases as the primary flow pressure increases. This indicated that a higher $P_P$ would improve the efficiency of the ejector. However, as the primary flow pressure continuously increased, the rate of increase in the entrainment ratio declined.

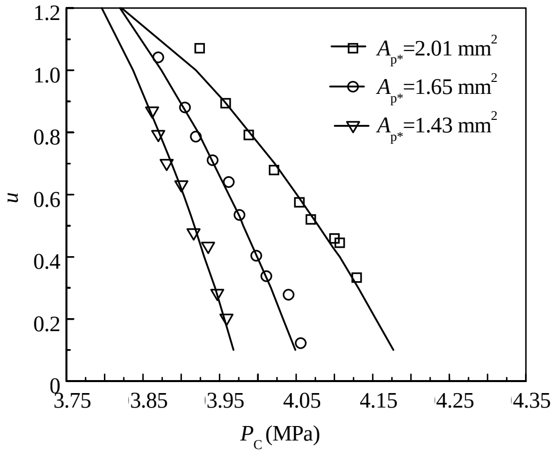


**Fig.6 Effect of back pressure on entrainment ratio with different motive nozzle throat areas**

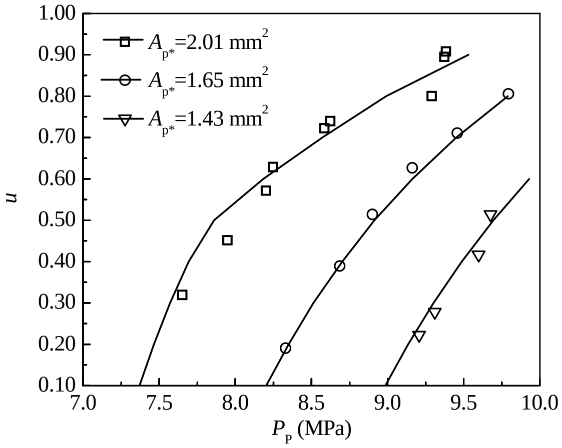


**Fig.7 Effect of primary flow pressure on entrainment ratio with different motive nozzle throat areas**

## CONCLUSION

Experimental studies were performed to investigate the effects of the geometric configurations of an ejector on its performance under different working conditions. By means of a moving nozzle needle, the effects of the motive nozzle throat area on entrained flow pressure, back pressure and primary flow pressure were studied. The results are useful for developing models for the adjustment of the motive nozzle throat area to obtain optimum performance.

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