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An Experimental Investigation of Steam Ejector Refrigeration System Powered by Extra Low
Temperature Heat Source

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Abstract

A steam ejector refrigeration system is a low capital cost solution for utilizing industrial waste heat or solar energy. When the heat source temperature is lower than 80 °C, the utilization of the thermal energy from such a low-temperature heat source can be a considerable challenge. In this investigation, an experimental prototype for the steam ejector refrigeration system was designed and manufactured, which can operate using extra low-temperature heat source below 80 °C. The effects of the operation temperature, the nozzle exit position (NXP) and the diameter of the constant area section on the working performance of the steam ejector were investigated at generating temperatures ranging from 40 °C to 70 °C. Three ejectors with a same de Laval nozzle for the primary nozzle and three different constant-area sections were designed and fabricated. The experimental results show that a steam ejector can function for a certain configuration size of the steam ejector with a generating temperature ranging from 40 °C to 70 °C and an evaporating temperature of 10 °C. For a given NXP, the system COP and cooling capacity of the steam ejector decreased until inoperative as the diameter of the constant area section reduced. The results of this investigation provided a good solution for the refrigeration application of the steam ejector refrigeration system powered by an extra low-temperature heat source.

Keywords: steam ejector; refrigeration; low-temperature heat source; working performance; nozzle exit position;

Introduction

The steam ejector has a history of more than 100 years. It was invented by Charles Peterson as

early as 1901 in order to remove air from a steam engine's condenser [1]. The advantages of the steam ejector includes utilizing low-grade thermal energy, using water, an environmentally friendly material, as the working medium, simplicity in construction, reliability in operation and low capital cost. These advantages caused the steam ejector refrigeration system to be widely applied in the air conditioning of large buildings in the early 1930s [2]. However, it was superseded by the vapor compression refrigeration system due to its higher COP, smaller volume and convenient operation. With enhanced environmental protection concerns and the growth in demand for air conditioning and refrigeration in daily life, the steam ejector refrigeration once again has been investigated in research. In theoretical research, Keenan et al. [3, 4] first established a one-dimensional model to predict the performance of the ejector based on mass, momentum and energy equation with ideal gas, which ignored friction losses. In the following decades, Munday et al. [5], Huang et al. [6] and Chen et al. [7] improved the one-dimensional model developed by Keenan et al. [3, 4]. At the same time, the empirical and semi-empirical correlations were derived based on the experiment results [8-11]. With the rapid development of computational fluid dynamics (CFD), numerical simulation became a popular method to resolve mixing and shock wave problems and to study the effects of steam ejector geometries on the COP in the steam ejector refrigeration system [12-25]. The realizable k - ϵ model was chosen as the turbulence model to study fluid flow in steam ejectors. In the experimental aspect, Eames et al. [26-28], Sun [29-30], Chen [31], Aphornratana et al. [15, 32-34], Dong et al. [35-37], Riffat et al. [38-39], Meyer et al. [2] and Vineet et al. [19] conducted experimental investigations to determine the effects of working temperature, nozzle structure and NXP on the performance of the steam ejector with generating temperatures ranging from 84 °C to 140 °C, respectively. The experiment results contributed by Meyer et al. [2] and Riffat et al. [38] implied that the steam

ejector can function at a generating temperature below 100 °C for refrigeration application, and the optimum performance took place at a generating temperature of around 90 °C to 95 °C. In this study, a steam ejector refrigeration system was developed and tested in order to find out whether a steam ejector can function for refrigeration application at an extra low-temperature heat source below 80 °C. The effects of the operation temperatures and structure parameters on the working performance of the steam ejector were investigated at generating temperatures ranging from 40 °C to 70 °C.

Experimental Setup

The experimental system of the steam ejector refrigeration, as shown in Fig. 1, mainly consists of a generator, an evaporator, an ejector, a condenser, a reservoir, a circulating pump, measuring devices, and two electrical heaters. The generator and evaporator were fabricated from a stainless steel cylinder (Φ 273 mm $\times$ 1020 mm) with flanges welded on the top. The bottom of each stainless steel cylinder was equipped with an electrical heater. The electrical heaters were used to simulate the low-temperature heat source in the generator and thermal load in the evaporator. For each electrical heater, the maximum heating capacity was 5 kW and the heating power was controlled by means of a transformer. The generator and evaporator were carefully insulated by thermal insulation materials. The condenser was a shell and coil condenser and was cooled by chilled water from a cooling bath. The cooling capacity of the cooling bath was 4000 W at 20 °C. A diaphragm pump was used as the circulating pump.

Figure 1 also shows the fabricated stainless steel ejector with movable nozzle. The ejector consists of four components, i.e., a nozzle, a mixing chamber, a constant area section and a diffuser. The dimensions of the experimental steam ejector are shown in Fig. 2. The temperatures of the primary and secondary flows were controlled by the power input of the electrical heaters

in the generator and in the evaporator, respectively. The temperature of the mixed flow in the condenser was controlled by adjusting the outlet temperature of the chilled water from the cooling bath. The installation position of pressure transducers and type T thermocouples are shown in Fig. 1. The accuracy of the pressure transducers (0–5 kPa for evaporator and condenser and 0–100 kPa for the generator) was $\pm 0.08\%$ and the accuracy of the type T thermocouples was $\pm 0.5\text{ }^{\circ}\text{C}$.

Deionized water was used as the working medium charged in the generator and the evaporator. In each test, the test was run at a low condensing temperature, which was then increased gradually by increasing the temperature set-point of the cooling bath. Once the system had reached an equilibrium state for 30 min, all temperatures, pressures, and heat power inputs were recorded for 20 additional minutes via the computer-controlled NI SCXI-1000 DAQ. The system COP was described as the ratio between the thermal load in the evaporator and the heat power input to the generator [35]. The experimental uncertainty analysis showed that the maximum uncertainty of the system COP to be 5.3%.

Results and Discussion

Using the aforementioned experimental setup as shown in Fig. 1, three ejectors as shown in Fig. 2 were investigated to determine the effects of the operation temperatures, nozzle exit position, and the diameter of the constant area section, d_{cas} , on the working performance of the steam ejector powered by extra low-temperature heat source. In this investigation, the working performance of the steam ejector mainly includes system COP, cooling capacity and critical condensing temperature [40], T_c^* . For the test, the generating temperature, T_g , ranged from 40 $^{\circ}\text{C}$ to 70 $^{\circ}\text{C}$, and the evaporating temperature, T_e , was set at 10 $^{\circ}\text{C}$ and 15 $^{\circ}\text{C}$, respectively.

Figure 3 shows the variation of system COP and cooling capacity with the condensing temperature, T_c , for Ejector 1 when NXP was set to 50 mm. As shown, the experimental results imply that the steam ejector can function at $T_g = 40\text{--}70\text{ }^{\circ}\text{C}$ and $T_e = 10\text{ }^{\circ}\text{C}$, respectively. In Fig. 3, it was found that the system COP could reach 0.695 when $T_g = 40\text{ }^{\circ}\text{C}$. For the generating temperatures ranging from $40\text{ }^{\circ}\text{C}$ to $55\text{ }^{\circ}\text{C}$, the system COP decreased with the increase of the generating temperature. When generating temperatures ranged from $60\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$, the system COP slightly increased at first and then decreased as the generating temperature continued to increase. Moreover, it can be seen that the cooling capacity could gain 685.4 W when $T_g = 65\text{ }^{\circ}\text{C}$ and 482.8 W when $T_g = 55\text{ }^{\circ}\text{C}$. With the increase in generating temperature, the cooling capacity decreased at first and then increased when the generating temperatures ranged from $40\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$. The experiment was repeated under the same conditions. Figure 4 illustrates the effect of the generating temperature on the system COP, cooling capacity and critical condensing temperature of the steam ejector when the generating temperatures ranged from $60\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$. As shown in Fig. 4, the system COP increased first and then decreased with the generating temperature increasing and there existed a maximum value of 0.277 at $T_g = 63\text{ }^{\circ}\text{C}$. The variation tendency of the cooling capacity was similar to that of the system COP, but the maximum value of 734.4 W at $T_g = 67\text{ }^{\circ}\text{C}$. The total pressure of the primary flow in the nozzle was relatively low with the generating temperature set under these conditions. The reason for the increase and then decrease in the system COP and cooling capacity at a given NXP for the low generating temperature, as shown in Fig. 4, is not yet fully understood. The possible explanation is that the entrainment effect of the primary flow changed as the generating temperature increased at the given NXP. Furthermore, the critical condensing temperature was $26.8\text{ }^{\circ}\text{C}$ when $T_g = 70\text{ }^{\circ}\text{C}$ in Fig. 4. Results also revealed that there is an approximate linear correlation between the

generating temperature and the critical condensing temperature. With the increasing of the generating temperature, the critical condensing temperature became higher accordingly.

Figure 5 shows the comparison results of the evaporating temperature effect on the system COP, cooling capacity and critical condensing temperature for Ejector 1 with $NXP = 50$ mm with the generating temperature ranging from $40\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$. As shown, the system COP could achieve 1.259 when $T_g = 40\text{ }^{\circ}\text{C}$ and $T_e = 15\text{ }^{\circ}\text{C}$. When the evaporating temperature was $15\text{ }^{\circ}\text{C}$, the system COP was much higher than that when the evaporating temperature was set at $10\text{ }^{\circ}\text{C}$. But the system COPs were almost the same when $T_g = 65\text{ }^{\circ}\text{C}$. In other words, the cooling capacities were nearly equal when $T_e = 10\text{ }^{\circ}\text{C}$ and $T_e = 15\text{ }^{\circ}\text{C}$ at this condition. When $T_e = 15\text{ }^{\circ}\text{C}$, the cooling capacity decreased first and then increased with the generating temperatures ranging from $40\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$. This variation tendency is similar to that when $T_e = 10\text{ }^{\circ}\text{C}$. The worst performance occurred when the generating temperature was $65\text{ }^{\circ}\text{C}$, which was 720.3 W . In Fig. 5, it also can be seen that the critical condensing temperature when $T_e = 15\text{ }^{\circ}\text{C}$ is a little higher than that when $T_e = 10\text{ }^{\circ}\text{C}$ in most cases. The maximum difference between the two conditions is $3.7\text{ }^{\circ}\text{C}$ when $T_g = 55\text{ }^{\circ}\text{C}$.

In order to investigate the effect of NXP on the working performance of the steam ejector with the generating temperature ranging from $40\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$ and evaporating temperature set at $10\text{ }^{\circ}\text{C}$, a movable nozzle mechanism was designed and developed as shown in Figs. 1 and 2. As the nozzle moves to a new location in the mixing chamber, the area of the duct between the nozzle and the mixing chamber changes, which directly affects the mass flow rate of the secondary flow entrained by the primary flow. Figure 6 illustrates the NXP effect on the system COP, cooling capacity and critical condensing temperature for Ejector 1 at the generating temperature ranging from $40\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$ and the evaporating temperature set at $10\text{ }^{\circ}\text{C}$. When

NXP = 100 mm and NXP = 150 mm, the system COP and cooling capacity decreased as the generating temperature increased. Under this condition, the system COP and cooling capacity, when NXP = 100 mm, are much higher than that when NXP = 150 mm. In addition, the differences of the system COP and cooling capacity increased with the generating temperature increased. When $T_g = 70\text{ }^{\circ}\text{C}$, the system COP and cooling capacity when the NXP was 100 mm, reduced by almost 50% than that when the NXP was 150 mm. When NXP = 50 mm, the variation tendencies of the system COP and cooling capacity are quite different from the other two conditions. For the generating temperatures ranging from $40\text{ }^{\circ}\text{C}$ to $55\text{ }^{\circ}\text{C}$, the system COP and cooling capacity, when the NXP was 50 mm, were lower than those at the other two conditions. When generating temperatures ranged from $60\text{ }^{\circ}\text{C}$ to $70\text{ }^{\circ}\text{C}$, the system COP and cooling capacity with the NXP = 50 mm, were higher than those at the other two conditions. The comparison results imply that the possible reason for the increase followed by a decrease in system COP and cooling capacity as shown in Fig. 4 might be related to the nozzle exit position. Results shown in Fig. 6 indicate that the critical condensing temperature almost increased linearly with the increase in the generating temperature. Also, the critical condensing temperature, when the NXP is 50 mm, was inferior to that at other two conditions.

The area ratio between the ejector's constant area section and nozzle throat is a very important factor, which affects the performance of the ejector [41]. Three constant area sections with different diameters as shown in Fig. 2 were designed and fabricated. The effect of the diameter of the constant area section on the working performance of the steam ejector was investigated under the conditions described in Fig. 6. The experiment phenomena were summarized and listed in Table 1. As shown, Ejector 1 can function for all three NXPs, Ejector 2 can function only when NXP = 100 mm and NXP = 150 mm, and Ejector 3 could not function for all three NXPs under

the given conditions. When the malfunction took place, not only did the temperature in the evaporator fail to decrease, but it increased. It should be noticed that at this time the electrical heater in the evaporator was no power input for the cooling load. This implied that the primary flow reversed back into the evaporator caused the ejector to lose its function completely. Figure 7 shows the comparison results of system COP, cooling capacity, and critical condensing temperature for Ejector 2 when the generating temperature ranged from 40 °C to 70 °C, the evaporating temperature was set at 10 °C, NXP = 100 mm, and NXP = 150 mm. As illustrated in Fig. 7, the system COP and cooling capacity decreased as the generating temperature increased. When NXP = 100 mm, the system COP and cooling capacity were much higher than that when NXP = 150 mm in most cases. Moreover, the differences between the system COP and cooling capacity increased first and then decreased with the increasing of the generating temperature. When $T_g = 65$ °C, the system COP and cooling capacity when the NXP was 100 mm, were almost three times less than that when the NXP was 150 mm. Also, it can be seen that the variation tendency of the critical condensing temperature is similar to that shown in Fig. 6. For Ejector 2, the critical condensing temperature, when the NXP was 150 mm, was higher than that when the NXP was 100 mm.

Accordingly, Fig. 8 compares the results of the system COP, the cooling capacity and the critical condensing temperature for different ejectors at NXP = 100 mm and NXP = 150 mm, when the generating temperature ranged from 40 °C to 70 °C, and the evaporating temperature was set at 10 °C. The system COP and the cooling capacity of Ejector 1 were better than those of Ejector 2 at both positions. Compared with Ejector 2, the maximum value of the system COP and the cooling capacity for Ejector 1 were enhanced by 176% and 186% at NXP = 100 mm and NXP = 150 mm, when $T_g = 70$ °C and $T_g = 65$ °C, respectively. Considering the results illustrated

in Table 1 and Fig. 8, the system COP and cooling capacity of the steam ejector decreased until inoperative as the diameter of the constant area section reduced. Meanwhile, the dimensions of the diameter which rendered the steam ejector inoperative are different at different nozzle exit positions. As Fig. 8 shows, the effects of diameter of the constant area section on critical condensing temperature were different at different nozzle exit positions. When the NXP was 100 mm, the critical condensing temperature of Ejector 1 was higher than that of Ejector 2 with generating temperatures ranging from 40 °C to 60 °C. While, the critical condensing temperature of Ejector 1 was lower than that of Ejector 2 with generating temperatures ranging from 65 °C to 70 °C. When the NXP was 150 mm, the critical condensing temperature of Ejector 2 was higher than that of Ejector 1 with the generating temperature ranging from 40 °C to 70 °C and the maximum value was 38.6 °C when $T_g = 70$ °C.

Conclusions

An experimental prototype for the steam ejector refrigeration system was constructed and powered by an extra low-temperature heat source, i.e., below 80 °C. According to results presented, it can be concluded that a steam ejector can operate successfully at the generating temperature ranging from 40 °C to 70 °C depending on the ejector configuration, evaporating temperature, and NXP. When the system consisted of Ejector 1 with the generating temperatures ranging from 40 °C to 70 °C, the evaporating temperature of 10 °C, and NXP = 50 mm, the cooling capacity decreased first and then increased. When NXP = 100 mm and NXP = 150 mm, the system COP and cooling capacity decreased with the increase of the generating temperatures for both Ejector 1 and Ejector 2. In addition, the system COP and cooling capacity when the NXP = 100 mm were much higher than that when NXP = 150 mm in most cases. For a given

NXP, the system COP and cooling capacity of the steam ejector decreased until inoperative as the diameter of the constant area section reduced. On the other hand, the dimensions of the diameter which caused the steam ejector inoperative were different at different nozzle exit positions. The system COP and cooling capacity of Ejector 1 were superior to those of Ejector 2, which was enhanced by 176 % and 186 % when $NXP = 100$ mm and $NXP = 150$ mm, and when $T_g = 70$ °C and $T_g = 65$ °C, respectively. The critical condensing temperature almost increased linearly with the increasing of the generating temperature, and the maximum value was 38.6 °C for Ejector 2 when $NXP = 150$ mm and $T_g = 70$ °C.

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Table 1 The experiment phenomena for the three ejectors under the given conditions

Ejector No.	NXP = 50 mm	NXP = 100 mm	NXP = 150 mm
1	Function	Function	Function
2	Reversed flow	Function	Function
3	Reversed flow	Reversed flow	Reversed flow

Fig. 1 Schematic diagram of the experimental system for the steam ejector refrigeration system

Fig. 2 Dimensions of the experimental steam ejector

Fig. 3 Variation of system COP and cooling capacity with the condensing temperature for Ejector 1 at different generating temperatures

Fig. 4 Effect of the generating temperature on the system COP, cooling capacity and critical condensing temperature for Ejector 1 at NXP = 50 mm ($T_e = 10$ °C)

Fig. 5 Variation of working performance with the generating temperature for Ejector 1 at different evaporating temperatures

Fig. 6 Variation of working performance with the generating temperature for Ejector 1 at different NXPs ($T_e = 10\text{ }^{\circ}\text{C}$)

Fig. 7 Variation of working performance with the generating temperature for Ejector 2 at different NXPs ($T_e = 10\text{ }^{\circ}\text{C}$)

(a) NXP = 100 mm

(b) NXP = 150 mm

Fig. 8 Variation of working performance with the generating temperature at NXP = 100 mm and NXP = 150 for different diameters of the constant area section ($T_e = 10^\circ\text{C}$)