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Research Paper

Analysis for the ejector used as expansion valve in vapor compression refrigeration cycle

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HIGHLIGHTS

- The mixing pressure was determined by using the optimization method.
- Whether the entrained flow would be choked at outlet of suction nozzle was studied.
- Whether a condensation shock would happen at the end of mixing chamber was studied.
- System performance at off-design condition is close to that with optimum ejector.
- No condensation shock happens before the mixed flow flows into the diffuser.

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ABSTRACT

An ejector was used in a compression refrigeration cycle for improving its efficiency. A constant-pressure mixing model was adopted to simulate the ejector. Whether or not the entrained flow would be choked at the outlet of the suction nozzle and whether or not a condensation shock would happen at the end of the mixing chamber were both considered. The effect of the mixing pressure on the performances of the ejector and the hybrid system was evaluated. The mixing pressure was finally determined by using the optimization method. The performances of the ejector and the hybrid system at different operating conditions were studied. Lastly, the performances of the ejector with fixed geometry and the corresponding hybrid system at off-design operating conditions were also theoretically studied. The results indicate that the optimum ejector mixing pressure is a little lower than the entrained fluid's pressure, but far larger than its critical pressure. No condensation shock happens before it flows into the diffuser. The theoretical performances of the hybrid compression refrigeration system with fixed geometry ejector at off-design conditions are very close to that with the optimum geometry ejector.

1. Introduction

Refrigeration system is a kind of large energy-consumption equipment. With the increasing demand of environment protection and energy conservation, it is significant to further improve the efficiency of refrigeration system. Kemper et al. [1] suggested using a two-phase ejector in compression refrigeration system to improve the system performance in 1966. Kornhauser et al. [2] did a first-order analysis for the two-phase ejector in the vapor compression refrigeration cycle with R12. His results show the significant increase in system performance with ejector. Nehdi et al. [3] studied the performance of the improved compression refrigeration cycle with various refrigerants. Ameur et al. [4] proposed a thermodynamic model of the two-phase ejector, in which the nozzle throat can be calculated and the wall friction in the mixing chamber is accounted in the momentum balance equation. Unal [5] determined the dimensions of the ejector in the bus air condition system by the thermodynamic analysis. Lastly, experimental data were used to validate the theoretical results. Liu et al. [6] proposed a modified ejector-expansion refrigeration cycle using mixture R290/R600a for applications in domestic refrigerator/freezers, and theoretically evaluated the cycle performances using energy and exergy methods. Elbel et al. [7] compared the standard two-phase ejector refrigeration cycle with two less commonly considered two-phase ejector refrigeration cycles with first and second laws, and it was suggested that these and other alternate ejector cycles may be worth additional attention in future studies due to their potential practical advantages. Elbel et al. [8] experimentally compared the two-phase ejector cycle and expansion valve cycle with multiple evaporation temperatures. Ersoy et al. [9,10] studied the R134a refrigeration system using a two-phase ejector as an expander. It was found that the COP of the refrigeration system with an ejector as the expander was 6.2–14.5% higher than that of the conventional system, while the exergy efficiency values were 6.6–11.2% higher

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than in the basic system. Hrnjak et al. [11,12] experimentally investigated an R410a vapor compression system working with an ejector. The objective of this study was to quantify separately two major improvements of the ejector system: work recovery and liquid-fed evaporator.

The two-phase ejector is the key component of the vapor compression refrigeration cycle with ejector as the expander. According to the ways to calculate the mixing process in the mixing chamber, the calculation models of the ejector can be classified into two categories, one is constant-pressure mixing model [13,14], and the other is constant-area mixing model [15]. For the constant-pressure mixing model, it is assumed that when the primary and secondary streams first interact and mix together, their pressures are equal. For the constant-area mixing model, it is assumed that the place where the two streams first interact is in the constant-area mixing chamber of ejector, and the pressures of the two streams are different. It was found that the constant-pressure mixing ejector has a better performance than the constant-area one [16]. Therefore, most of current studies about ejector have been focused on the constant-pressure mixing ejector. For the constant-pressure mixing model, there are mainly three types of methods to determine the mixing pressure. The first type is to assume that the mixing pressure is equal to the evaporator pressure [17], the second is to assume that the pressure drop in the receiving section of the ejector is a constant value [13], and the last one is to optimize the mixing pressure by calculation [18]. The suction chamber of ejector is converging, so the entrained flow's velocity at the outlet of the suction nozzle cannot be higher than sonic, that is, the mixing pressure must not be lower than the entrained flow's critical pressure. In addition, affected by the high back pressure of the ejector, the mixed fluid at the inlet of the diffuser should be a stream of subsonic flow. If not, a shock wave would happen. However, these two special phenomena have not been considered for the study of the two-phase ejector in the published literatures.

In this paper, a hybrid compression refrigeration cycle with an ejector was analyzed by using a constant-pressure mixing model for the ejector. The mixing process is also assumed to happen in the constant cross section of the ejector mixing chamber. The mixing pressure was determined by using optimization method. Whether or not the entrained flow would be choked at the outlet of the suction nozzle and whether or not a condensation shock would happen at the end of the mixing chamber were both studied. The performances of the ejector and the hybrid refrigeration system at different condensation temperatures and evaporation temperatures were studied. Lastly, their performances with fixed geometry ejector at off-design condensation temperatures and evaporation temperatures were also studied.

2. Thermodynamic analysis

2.1. Analysis of the ejector at design condition

The schematic diagram of ejector is shown in Fig. 1. The ejector is mainly made up of a supersonic nozzle, a suction chamber, a mixing chamber and a diffuser. It is assumed that the mixing process happens in the constant cross section of the ejector mixing chamber with an identical pressure when they first interact in the constant-area mixing chamber. So the ejector schematized in Fig. 1 is not only a constant-pressure mixing ejector, but also a constant-area mixing ejector. The model of ejector is set up based on the balance of mass, momentum and energy. To simplify the analysis, the other assumptions are made as follows:

(1) The flow in every part of the ejector except the mixing chamber is considered as a one-dimensional homogeneous equilibrium flow.

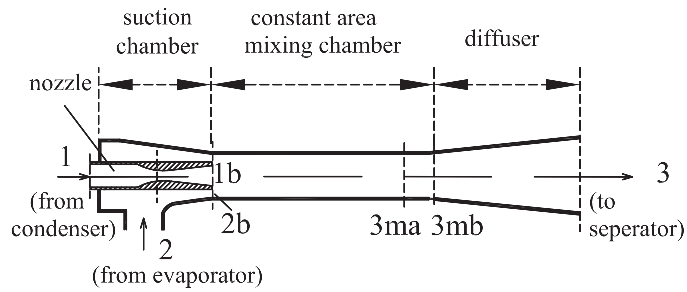

Fig. 1. Schematic diagram of ejector.

(2) The streams from the condenser and the evaporator are both saturated.
(3) The kinetic energy of the fluids flowing into and out of the ejector is neglected.
(4) The ejector is adiabatic.
(5) The wall friction is neglected.

2.1.1. Primary nozzle

The flow parameters at the outlet of the primary nozzle are calculated by defining the isentropic efficiency.

$$h_{1b,is} = f(s_1, P_{1b}) \tag{1}$$

$$w_{1b} = \sqrt{2\eta_n (h_1 - h_{1b,is})} \tag{2}$$

$$h_{1b} = h_1 - \frac{w_{1b}^2}{2} \tag{3}$$

$$v_{1b} = f(h_{1b}, P_{1b}) \tag{4}$$

$$A_{1b} = \frac{\dot{m}_1 v_{1b}}{w_{1b}} \tag{5}$$

2.1.2. Suction nozzle

The suction chamber forms a converging passage like a contraction nozzle for the entrained flow. Similarly, the analysis equations for the entrained flow can be derived as:

$$h_{2b,is} = f(s_2, P_{2b}) \tag{6}$$

$$w_{2b} = \sqrt{2\eta_s (h_2 - h_{2b,is})} \tag{7}$$

$$h_{2b} = h_2 - \frac{w_{2b}^2}{2} \tag{8}$$

$$v_{2b} = f(h_{2b}, P_{2b}) \tag{9}$$

$$A_{2b} = \frac{\omega_{gl} \dot{m}_1 v_{2b}}{w_{2b}} \tag{10}$$

The suction chamber of the ejector is converging, so the entrained flow's velocity at the outlet of the suction chamber cannot be higher than sonic, that is, the outlet pressure of the entrained flow must not be lower than its critical pressure. The critical pressure of the entrained flow is calculated by:

$$P_{2,cr} = P_2 \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \tag{11}$$