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Optimized selection of one- and two-stage ejectors under design and off-design conditions

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ABSTRACT

To provide guidance for the optimized selection of one- and two-stage ejectors, the efficiencies of these two ejectors are compared under design and off-design conditions based on a verified mathematic model. Under design condition, the one-stage ejector presents a higher efficiency at low compression ratios but the two-stage ejector is superior at high compression ratios. The superiority of the two-stage ejector is more obvious at low expansion ratios and at higher compression ratio value, due to lesser mixing loss and more boosting loss in the mixing chamber, respectively. The compression ratio of the equivalent entrainment ratio should be defined as the diacritical point between the high and low expansion ratio, whose value is approximately 1.73 for the R141b ejector. The inevitable loss of mechanical energy in the diffuser causes the crossed efficiencies of the one- and two-stage ejectors. Compared with the one-stage ejector, the two-stage ejector possesses more slow change of efficiency under off-design conditions due to a lesser primary vapor passing through the first stage ejector, obtaining the ability to resist the sharp deterioration in efficiency at single chocking mode. For the typical application of ejector refrigeration system with R141b as the working fluid, it would be best to adopt the two-stage ejector.

1. Introduction

Considering current energy shortages, the ejector refrigeration has attracted much attention by virtue of utilizing the low-grade heat [1]. A typical ejector refrigeration system mainly consists of six components—a generator, an evaporator, a condenser, an ejector, an expansion valve and a pump, as shown in Fig. 1 [2]. The ejector, as the core of the system, raises the vapor from the low evaporating pressure to the condensing pressure.

The research conducted by Dong et al. revealed that under the design condition of generating temperature (T_g) of 70 °C, evaporating temperature (T_e) of 15 °C and condensing temperature (T_c) of 31.3 °C, the coefficient of performance (*COP*) could reach 0.3 [3]. The experiment by Pounds et al. indicated that under the working condition given by T_g = 120–135 °C, T_e = 5–15 °C and T_c = 20–35 °C, the *COP* was in the range of 0.38–1.18 [4]. Smierciew et al. pointed out that the system with R1234ze(E) as the working fluid could attained a *COP* more than 0.3 under the condition given by T_g = 56 °C, T_e = 2–5 °C and T_c = 24 °C [5].

To allow the ejector refrigeration system to obtain a relatively high *COP,* most studies have focused on system running at low condensing temperature, high evaporating temperature or high generating temperature. This makes the system lose its ability to utilize waste heat of lower grade or produce a cooling capacity with lower temperature, or it must rely on inconvenient cooling tower. A low expansion ratio (the ratio of the primary pressure to the secondary pressure) or high compression ratio (the ratio of the backpressure pressure to the secondary pressure) forces the ejector to work at low efficiency [6], therein even failing to meet demands [7]. To overcome this drawback, Grazzini et al. introduced a two-stage ejector into the ejector refrigeration system. In a two-stage ejector refrigeration system, the discharge vapor from the first stage ejector is boosted further by the second stage ejector, which is the main difference compared with a one-stage ejector system, as described in Fig. 2 [8]. A two-stage ejector refrigeration system developed by Chen et al. showed that the system could worked at T_g as low as 47–67 °C under the condition given by T_e = 11°C and T_c = 33 °C [9]. Jaruwongwittaya et al. numerically researched a two-stage air-cooled ejector refrigeration system powered by waste heat of an automobile, which showed that the *COP* could reach 0.29 under the condition given by T_g = 100 °C, T_e = 5 °C and T_c = 54 °C, a high

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Nomenclature

a	sonic speed ($m \cdot s^{-1}$)
A	cross section area (m^2)
c_p	specific heat at constant pressure ($J \cdot kg^{-1} \cdot K^{-1}$)
C	volumetric heat capacity ($J \cdot m^{-3} \cdot K^{-1}$)
CR	compression ratio
$CREER$	compression ratio of equivalent entrainment ratio
D	diameter, (m)
ER	expansion ratio
h	specific enthalpy ($J \cdot kg^{-1}$)
m	mass flow rate ($kg \cdot s^{-1}$)
p	pressure (Pa)
s	specific entropy ($J \cdot kg^{-1} \cdot K^{-1}$)
T	temperature (K)
u	entrainment ratio
v	specific volume ($m^3 \cdot kg^{-1}$)
w	velocity ($m \cdot s^{-1}$)

Greek letters

β	thermal expansion coefficient
γ	void fraction
η	isentropic efficiency
μ	coefficient of momentum loss
ρ	density ($kg \cdot m^{-3}$)
φ	cross section coefficient
ω	thermodynamic perfectibility of ejector

subscripts

m	mixing vapor
g	primary vapor
e	secondary vapor
c	discharge vapor
cri	critical value
c4	discharge vapor at cross section 4
d	double chocking mode
e1, e2, e3	secondary vapor at cross sections 1, 2, 3
g0, g1, g2, g3	primary vapor at cross sections 0, 1, 2, 3
L	liquid
i	ideal condition
o	design value
s	isentropic process
t	gross value
u	single chocking mode
V	vapor

Roman letters

I	The first stage ejector
II	The second stage ejector

condensing temperature [10]. Peng et al. analyzed the same air-cooled but solar-powered system and discovered the backpressure of the first stage ejector (it serves as the secondary pressure of the second stage ejector at the same time) could affect the system performance [11]. Further, Lu et al. let the system operate as a cold storage with $T_g = 100\,°C$, $T_e = 3\,°C$ and $T_c = 45\,°C$, therein finding that the cold store could reach a *COP* of 0.1 [12]. Xu et al. investigated the system through the methodology of entropy under the condition of $T_g = 71–85\,°C$, $T_e = -15–10\,°C$ and $T_c = 45–65\,°C$, concluding that the entropy generation of two-stage ejector refrigeration system increased with the increasing generating and condensing temperatures but decreased with the increasing evaporating temperature [13]. A simulation performed by Ding et al. showed that the two-stage ejector refrigeration system could create a subzero evaporating temperature as low as $-24\,°C$, although the entrainment ratio was small [14]. Experimentally examining the operational strategy of the system, He et al. noted that two stages of ejector should run at high backpressure, but only the first stage should run at low backpressure [15].

In addition, ejector's efficiency is sensitive to the working condition and seriously deteriorates at the single chocking mode [16,17]. Through experiment, Chen et al. noted that with decreased evaporating temperature the entrainment ratio of the two-stage ejector decreased more slowly than that of the one-stage ejector [18]. It seems that the two-stage ejector may have the potential to work at higher efficiency under off-design conditions, but the influence of the changing primary pressure and backpressure should be analyzed further.

Through a literature research, it can be concluded that the researchers have come to the consensus that the two-stage ejector can effectively run under the design conditions of low expansion ratios or high compression ratios. However, the low expansion ratios and high compression ratios are not perfectly defined, and the analysis of two-stage ejector under off-design conditions is not enough, leading to the vague selection criteria of one- and two-stage ejectors. There is rare research focusing on the optimized selection of one- and two-stage ejectors under design and off-design conditions for ejector refrigeration systems at present. This paper first presents and verifies the analysis and design model for the ejector; then, with the model as tool, the

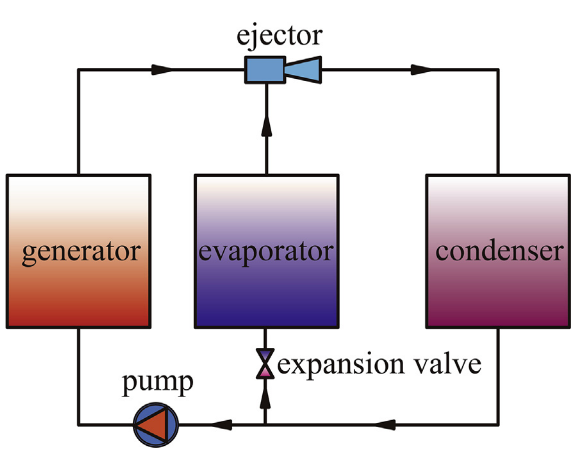

Fig. 1. Flow diagram of one-stage ejector refrigeration system.

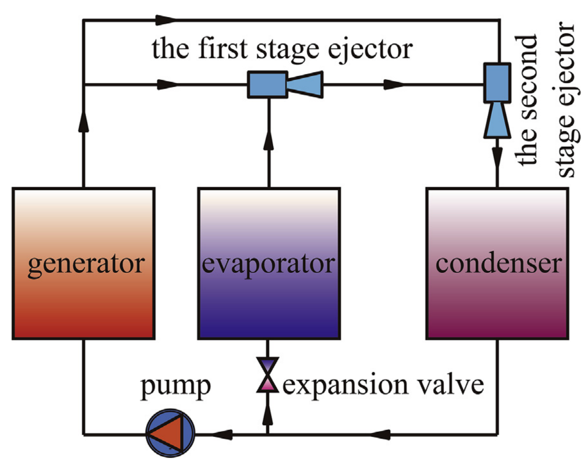

Fig. 2. Flow diagram of two-stage ejector refrigeration system.