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Performance analysis of a new ejector enhanced vapor injection heat pump cycle

Xiao Wang, Jianlin Yu *, Meibo Xing

Department of Refrigeration & Cryogenic Engineering, School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an 710049, China

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ABSTRACT

This paper proposes a novel ejector enhanced vapor injection cycle (EVIC) for air-source heat pumps. In the EVIC system, an ejector associated with an additional flash tank is introduced to enhance the overall system performance at low ambient temperature. The performances of the EVIC using R22, R290 and R32 are evaluated using the developed mathematical model, and then compared with those of the basic vapor injection cycle (BVIC). According to the simulation results, the EVIC with R22, R290 and R32 have 2.6–3.1%, 3.2–3.7% and 2.9–3.1% improvement in coefficient of performance (COP), 6.0–8.4%, 7.3–10.2% and 6.7–8.2% improvement in volumetric heating capacity compared with those of the BVIC under the same given operating conditions. Simultaneously, the EVIC also shows a good performance in reduction of the compressor discharge temperature at the given operating conditions. The performance characteristics of the EVIC may show its promise in heat pump applications.

1. Introduction

Air-source heat pumps (ASHPs) with the vapor-compression heat pump cycle present outstanding performance compared with other conventional residential heating technologies due to higher energy efficiency. However, the performance degradation of the ASHPs at low ambient temperature has been a main issue in practical operations. Thus, many technologies have been proposed and developed to enhance the performance of the ASHPs [1,2]. Among these technologies, the application of vapor injection technology in heat pump systems has proven to be an effective solution to improve the performance at low ambient temperature [3,4]. Over the past years, relevant researches have provided insight into many aspects of this technology.

For recent examples, Heo et al. [5] conducted the comparison of the heating performance of ASHPs using various types of refrigerant injection, and indicated that the vapor injection in the three heat pump cycles could yield the improved heating capacities. Roh et al. [6] experimentally investigated the effects of intermediate pressure on the heating performance of a heat pump system using vapor-injection technique, and revealed that a proper operating strategy was needed for the vapor-injection cycle. Heo et al. [7] proposed an optimum cycle control method for a refrigerant injection heat pump with a double expansion subcooler. Mathison et al. [8] examined the performance limit for economized cycles with continuous refrigerant injection and concluded that implementing a compressor with continuous refrigerant injection would require a much clearer understanding of the physical injection process in the compressor. Other studies have been also reported on refrigerant vapor injection and its applications [9,10]. It is evident that with further development of the vapor injection technology, more relevant applications in the ASHPs can be broadly seen in the near future.

Currently, most vapor-injection cycles for the ASHPs still adopt throttling valves to produce a refrigeration effect, i.e. the heat absorption from air heat source. In principle, the throttling valve limits the potential performance of the cycles due to the inherent thermodynamic loss. To overcome this problem, the idea of using an ejector can be a beneficial way to enhance the performance of a vapor-injection cycle. Xu et al. [11,12] proposed an ejector enhanced vapor injection cycle and confirmed the performance can be further improved under low ambient temperature conditions. In fact, a number of studies have shown the potential of ejectors to drastically improve the performances of vapor compression refrigeration and heat pump systems [13–20]. Thus, the benefits of using the ejector could also make it an attractive alternative to vapor-injection cycles.

In this paper, a flash tank vapor injection cycle with a two-phase ejector is proposed to promote its performance improvement. With the use of the ejector, the thermodynamic loss in the cycle process can be reduced and thereby leading to a better

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* Corresponding author. Tel.: +86 29 82668738; fax: +86 29 82668725.
E-mail address: yujl@mail.xjtu.edu.cn (J. Yu).

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Nomenclature

COP	coefficient of performance
h	specific enthalpy (kJ kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
P	pressure (MPa)
$\dot{Q}$	heating capacity (kW)
q	heating capacity per unit of mass (kJ kg^{-1})
q_{hv}	volumetric heating capacity (kJ m^{-3})
r_{p}	pressure lift ratio of the ejector
t_{c}	condensing temperature (°C)
t_{e}	evaporating temperature (°C)
v	specific volume (m^3 kg^{-1})
$\dot{W}$	compressor input power (kW)
w	velocity (m s^{-1})
x	quality

Greek letters

η	efficiency
μ	entrainment ratio

Subscripts

d	diffuser
is	isentropic
m	mixing chamber
n	nozzle
p	primary fluid
s	secondary fluid
1–10, $1'$, $1''$, $3'$, $4'$	state points of refrigerant

performance for the vapor injection cycle. The aim of this paper is to provide theoretical evaluations on the new proposed ejector enhanced vapor injection cycle, and verify its potential performance enhancement by comparing to the baseline cycle.

2. Cycle description and modeling

There are two basic vapor injection cycle configurations known as the flash tank vapor injection cycle and the internal heat exchanger vapor injection cycle. In the present study, the flash tank vapor injection cycle is taken as a basic vapor injection cycle (BVIC) and its schematic diagram is shown in Fig. 1. The BVIC system consists of a compressor with vapor injection, a condenser, a flash tank, an evaporator and two throttling valves. Obviously, the refrigerant expansion processes in the BVIC are accomplished with the two throttling valves which result in the energy loss in the throttling processes. To partially recover this energy loss, a two-phase ejector can be introduced into the BVIC, which may enhance its performance. The schematic cycle system and corresponding P–h diagrams for the proposed ejector enhanced vapor injection cycle (EVIC) are shown in Fig. 2. Note that in the EVIC configuration, an additional flash tank is used to supply a saturated vapor to the ejector. In the EVIC, the ejector takes the high pressure refrigerant from the condenser to be the primary fluid and the low pressure refrigerant from the second flash tank to be the secondary fluid. The detailed working process of the EVIC is described as follows:

The compressed refrigerant vapor coming from the compressor (point 2) goes through the condenser to the ejector (points 3), in the ejector the high pressure refrigerant further entrains the vapor refrigerant from the second flash tank (point 8); the two-phase refrigerant from the ejector (point 4) is separated into the saturated vapor and the saturated liquid by the first flash tank (points 5 and 6); the saturated vapor refrigerant returns to the compressor through supplementary inlet as the injection vapor, and the saturated liquid refrigerant goes through the first throttling valve to be the two-phase refrigerant (point 7); the two-phase refrigerant from the first throttling valve is separated into the saturated vapor refrigerant and the saturated liquid refrigerant by the second flash tank (points 8 and 9); the saturated vapor refrigerant is entrained by the high pressure refrigerant from the condenser, and the saturated liquid refrigerant transfers through the second throttling valve to the evaporator (point 10) where it is vaporized completely and further returns to the suction line of the compressor (point 1). In the compressor, the vapor refrigerant at the end of the first compression process (point $1'$) mixes with the vapor refrigerant from the supplementary inlet (point 5) in the compression chamber, and then the mixing vapor refrigerant (point $1'$) is continually compressed to the condensation pressure (second compression process). Note that points $3'$ and $4'$ represent the refrigerant states in the ejector, and the working process of each ejector is omitted for simplicity.

A mathematical model of the presented EVIC is developed based on the one-dimensional constant pressure mixing ejector model [21–23,16,24] to investigate the performance characteristics of the EVIC. The additional assumptions of the EVIC mathematical model are given as follows:

(1) The system is running under the steady-state conditions.
(2) The isentropic efficiency of the compressor and the efficiencies of the ejector are given as constant values.
(3) The isenthalpic throttling processes are assumed in capillary tubes.
(4) The refrigerant velocities at the inlet and outlet of the ejector are neglected.

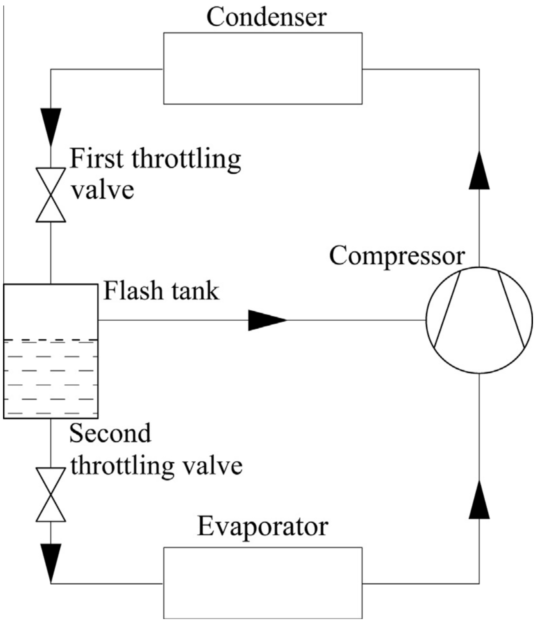

Fig. 1. The schematic diagram for the BVIC.