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Experimental study on performance of a hybrid ejector-vapor compression cycle

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ABSTRACT

Improving the Coefficient of Performance (COP) of the vapor compression refrigeration cycle (VCRC) is one of the primary objectives in the HVAC&R field. In this paper, a hybrid ejector-vapor compression cycle (EVCC) was presented to improve the COP of the vapor compression sub-cycle (VCSC). That is to say, the sub-cooling degree of the VCSC was improved by using the cooling effect of the ejector refrigeration sub-cycle (ERSC). R134a was employed in both sub-cycles. Specific experimental studies conducted were: (1) seeking the relations between the optimal nozzle exit positions and area ratio of the ejector to improve the performance of the ERSC, (2) investigating the influence of the compressor frequency on the capability of the VCSC, and (3) evaluating the effect of the compressor frequency as well as the evaporation temperature of the ERSC on the characteristics of the EVCC. As a result, two significant findings were obtained: (1) the range of optimal NXPs for higher ARs is quite narrow and vice versa for lower ARs, (2) on average, the COP improvement of the EVCC over the VCSC reaches 19.4%.

1. Introduction

Ejector based refrigeration cycles which use low grade heat source to generate cooling effect have been deeply concerned by both academic and industrial fields for many years [1–3]. Their strengths are as followings: simple structure, high consistency and low maintenance costs. Furthermore, their driving heat sources can be easily and freely collected from solar heat, industrial waste heat and so on [4,5]. Latest studies regarding to ejector based refrigeration cycles were reported in Refs. [6–9]. However, the systems have some apparent deficiencies: relatively low COP and required on-design operating conditions to obtain fairly steady performances [10–12].

Vapor compression refrigeration cycles are currently much popular in both domestic and industrial applications [13–15]. Such systems mainly consist of four components such as a compressor, a condenser, a throttling valve and an evaporator. The advantages of the systems are relatively high coefficient of performance (COP) as compared to many other types of refrigeration systems [15]. However, their disadvantages are evident as they consume large amount of electricity to drive the compressor, so that they contribute a considerable share of energy consumption [15–17]. The growth of the vapor compression refrigeration cycles was predominantly substantial, which accounts for 52% of building consumption and more than 20% of total consumption in Singapore [18]. Therefore, improving the energy efficiency of the vapor compression refrigeration cycles is one of the primary targets.

In recent years, researchers proposed some kinds of hybrid ejector-vapor compression cycle (EVCC) to exploit the advantages and eliminate the disadvantages of both vapor compression and ejector-based refrigeration sub-cycles [16,19]. In general, three configurations were claimed as followings:

- The first configuration is called ejector expansion refrigeration system. In which, by using an ejector as the expansion device, the performance of a compression refrigeration cycle can be improved [20,21]. The reason is that the ejector can reduce both expansion irreversibility and raise the suction pressure, the COP of the system can be improved by more than 10% [22].
- The second configuration is presented by Sokolov [23], Mansour et al. [24] and Dorantes and Pilatowsky [25]. In this configuration, a booster is employed in an ejector based refrigeration system to increase the secondary flow pressure of the ejector. The value of the COP is improved to 0.767 which is more than double the COP of the single ejector based refrigeration systems as claimed by Dorantes et al., however, the coupling of the booster and ejector in series may cause control issues.

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Nomenclature

COP	coefficient of performance
AR	area ratio of the ejector
NXP	primary nozzle position, mm
$\dot{m}_p$	mass flow rate of primary flow, kg/s
$\dot{m}_s$	mass flow rate of secondary flow, kg/s
P	pressure, bar
T	temperature, °C
λ	entrainment ratio, $\dot{m}_s/\dot{m}_p$
ERSC	ejector based refrigeration sub-cycle
VCSC	vapor compression sub-cycle
VCRC	vapor compression refrigeration cycle
EVCC	hybrid ejector-vapor compression cycle
$\dot{W}_{comp}$	the power input of the compressor
$\dot{W}_{pump}$	the power input of the liquid pump
$\dot{Q}_{e\_e}$	the cooling capacity of the ERSC
$\dot{Q}_{e\_v}$	the cooling capacity of the VCSC
$\dot{Q}_{e\_c}$	the cooling capacity of the EVCC
$\dot{Q}'_{e\_v}$	the cooling capacity of the EVCC

Subscripts

g	generator
v	the abbreviation of the VCSC
e	evaporator, The abbreviation of the ERSC
c	condenser, The abbreviation of the EVCC

- The third configuration proposed by Vidal and Colle [26], Hernández et al. [27] and Huang et al. [28,29] is a coupled ejector-compressor refrigeration cycle. The bottoming sub-cycle is a conventional ejector based refrigeration cycle or a booster ejector based refrigeration cycle, while the topping sub-cycle is a vapor compression cycle driven by a compressor. In this configuration, the heat is transferred between the two sub-cycles in an inter-cooler, which act as the evaporator of the ejector based sub-cycle. This placement can meet the variability of the working conditions and produce more stable performances.

Regarding the third one, Huang et al. built a prototype of EVCC and tested its performance. The working fluids employed were R22 in the vapor compression sub-cycle and R141b in the ejector based refrigeration sub-cycle. The EVCC has two distinct characteristics: (1) using the wasted heat from the discharged superheated vapor in the vapor compression sub-cycle to preheat the liquid refrigerant of the ejector based sub-cycle, and (2) the cooling product of the ejector based sub-cycle is used to increase the sub-cooling degree of the vapor compression sub-cycle so that the COP of the combined system can be improved. Around 15.4% COP improvement was stated by the authors. R134a has some advantages over other refrigerants [30]: its Ozone Depletion Potential (ODP) is zero, and its condensation pressure (1016.4 kPa) is much higher than those of R245fa (250.5 kPa) and R718 (7.4 kPa) if condensation temperature is set at 40 °C. One more advantage of using R134a is that the on-way resistance of the condenser can be considered negligible. Accordingly, Yan et al. [31] proposed an EVCC in which both sub-cycles use R134a as working fluid. A large amount of experiments were presented to exam the effects of the evaporation, generation and condensation temperature of the ejector based sub-cycle on the performances of the EVCC. The authors claimed that the performance of the combined system is very sensitive to the three temperature of the ejector based sub-cycle. The combined system produced relatively high COP improvements.

However, to the best knowledge of the authors, there have been no attempts to improve the performance of the EVCC by adopting the following two measures simultaneously: (1) seeking the optimum nozzle exit position (NXP) for varied area ratio (AR) of the ejector to raise the ejector sub-cycle performance; and (2) employing compressor with variable frequency to investigate the influence of compressor frequency on the performance of the VCSC and the COP improvement of the EVCC over the VCSC. Thus, to bridge this gap, our current study addresses the above-mentioned issues in the following sections.

2. Experimental rig

The schematic diagram and photo of the experimental rig (EVCC) are illustrated in Figs. 1 and 2, respectively. The EVCC comprises two sub-cycles: an ejector based refrigeration sub-cycle (ERSC) and a vapor compression sub-cycle (VCSC).

There are 7 core devices in the ERSC, such as a gas generator, an ejector, an air-cooled condenser, a receiver, a liquid pump, an Electronic Expansion Valve (EEV) and an electric evaporator.

The gas generator contains a vapor collection tank, a variable electric heater located in the hot water bath with maximum power output of 8 kW and a tubular heat exchanger which is immersed in the vapor collection tank. The function of the variable electric heater can be achieved by using an inverter to tune the voltage. The ejector consists of replaceable nozzles and replaceable main bodies. The key geometric parameters of the ejector were given in Ref. [30]. The EEV can be automatically controlled its expansion rate with the self-developed software. The electric evaporator is applied to simulate the ERSC cooling load, which can be achieved by using a variable electric heater with a maximum power input of 3 kW placed in an insulated tank. The power input of the electric evaporator can be adjusted by using an inverter. Furthermore, an air-cool evaporator is put on the top layer of the rig to demonstrate the cooling effect of the ERSC.

For the VCSC, 10 major elements are deployed as followings: an inverter driven compressor, an air-cooled condenser, a liquid receiver, 3 EEVs, 3 evaporators and an accumulator.

As mentioned in our previous work [31], the compressor has power input of 1.8 kW (50 Hz), and its frequency can be actually varied from 30 Hz to 70 Hz to produce varied cooling capacity according to different requirement. A maximum condensing load of 10 kW air-cooled condenser was utilized. Among the three evaporators, one is a traditional fin-tube heat exchanger with a maximum cooling load of 10 kW which is employed to show the cooling effect of the VCSC. The other two evaporators are located in two individual insulated tanks inserted PID auto-tune controllable electric heater. Henceforth, the cooling load (maximum cooling load of 10 kW) of the VCSC can be balanced with the assistance of the electric heater. Each EEV is motivated by a step motor that is controlled by the self-settled software automatically.

In order to prevent heat losses in the EVCC, both high and low temperature components are well insulated with fiber glass and foam rubber, respectively. All the data acquisitions and control algorithms are developed in National Instruments Labview 8.6. As shown in Table 1, the sensors employed in the rig with their accuracy are specified as below: