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Performance of the two-phase ejector expansion refrigeration cycle

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Abstract

This paper is a continuation of the authors' previous work. The paper presents new experimental data of the system performance of the two-phase ejector refrigeration cycle (TPERC). The TPERC uses a two-phase ejector as an expansion device while the conventional refrigeration cycle (CRC) uses an expansion valve. The TPERC enables the evaporator to be flooded with refrigerant, resulting in a higher refrigerant-side heat transfer coefficient. The experimental study shows that the TPERC gives a higher cooling capacity and a higher coefficient of performance. Moreover, the pressure ratio and the discharge temperature of the compressor of the TPERC are lower than those of the CRC.

Keywords: Two-phase ejector; Refrigeration; Liquid-recirculation; Coefficient of performance

1. Introduction

An ejector, or jet pump, is a device that uses a high-pressure fluid to pump a low-pressure fluid to a higher pressure at a diffuser outlet. Due to their low cost and absent of moving parts, ejectors have been used in various engineering applications. A well-known application is the use of the ejector in refrigeration systems as a compressor to compress the refrigerant vapor from the evaporator to the condenser. This type of application is explained in [1]. However, in addition to serving as a compressor, Kornhauser [2] proposed the use of the ejector as an expansion device to expand the high-pressure refrigerant from the condenser to reach evaporator pressure.

Over the years, many studies have focused on the former application while the latter application has received comparatively little attention in literature. This paper investigates the latter; the use of an ejector as an expansion device in a refrigeration system.

In a series of studies, Kornhauser [2] analyzed the ejector expansion refrigeration cycle by a thermodynamically analytical method. He found a theoretical COP improvement of up to 21% over the standard cycle under standard conditions: −15 °C and 30 °C for evaporator and condenser temperatures, respectively. This result is based on using R-12 as a refrigerant. Domanski [3] found that the theoretical COP of the ejector expansion refrigeration cycle was very sensitive to the ejector efficiency. Harrell et al. [4] tested a two-phase ejector with R-134a

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as working fluid, and used its performance, obtained from a test rig, to estimate the COP of the refrigeration cycle. It was found that the COP improvement ranged from 3.9% to 7.6%. Menegay et al. [5] tested the air-to-air refrigeration cycle using a two-phase ejector as an expansion device. They found that the COP improvement was poor. This was due to a non-equilibrium effect in the motive nozzle. Consequently, they developed a bubbly flow tube installed upstream of the motive nozzle to reduce the thermodynamic non-equilibrium in the motive nozzle. An ejector using the bubbly flow tube improved up to 3.8% of the COP over the conventional cycle under standard conditions with R-12 as the refrigerant. Nakagawa et al. [6] tested three different motive nozzles and showed that the longer the length of the divergent part of the motive nozzle, the higher the motive nozzle efficiency could be achieved, resulting in a higher ejector efficiency. This was likely caused because the longer divergent part provided a longer period of time for the two-phase flow to achieve equilibrium.

The published papers mentioned above focused on using the two-phase ejector as an expansion device operating with a dry-expansion evaporator in that they still use an expansion valve installed upstream of the evaporator. However, the purpose of using the ejector is to replace the throttling valve. Therefore, any throttling device in the system should be avoided. Up to now, there has been only one work, carried out by Disawas and Wongwises [7], dealing with this issue. In their experimental apparatus, the evaporator was flooded with refrigerant and became a liquid-recirculation system, in which, in addition to serving as an expansion device, the ejector also acted as a refrigerant pump for the low-pressure side of the system.

However, although some information is currently available on the refrigeration cycle using a two-phase ejector as an expansion device, there still remains room for further discussion. This paper is the second in a series and is a continuation of the authors' previous work. The main concern of the present study is to investigate the performance of the two-phase ejector refrigeration cycle (TPERC). The effect of the flow rate of the heat transfer fluid (HTF), which has never before appeared in open literature, is presented.

2. Experimental apparatus

The two-phase ejector refrigeration cycle (TPERC) consists of the basic components of the conventional refrigeration cycle (CRC) with a two-phase ejector used to replace the expansion valve, and with the addition of a liquid–vapor separator to allow only vapor to pass to the compressor.

Fig. 1 shows the schematic diagram of the experimental apparatus. The refrigerant main loop is designed in order to operate in both TPERC and CRC modes. The TPERC essentially consists of a compressor, condenser, thermostatic expansion valve, evaporator, oil separator, liquid receiver, filter/drier, sight glass and an accumulator. The principal modifications from the standard refrigeration system are the addition of a two-phase ejector and a liquid–vapor separator. A commercial R-134a is used as the working fluid. The operating conditions of the apparatus are similar to those in a typical air-conditioning application.

A two-cylinder single stage reciprocating compressor (Bitzer, Model III), driven by an electric motor, is used to convey the refrigerant. The speed of the motor is varied by an inverter (Yaskawa, CIMR-G5A47P5). Compact plate heat exchangers (SWEP, CBE-B8-24/C) are used as condenser and evaporators. The evaporator referred to in this paper is the main evaporator shown in Fig. 1.

The motive and the suction mass flow rates are measured by volumetric flow meters (Bailey F&P, 10A3225) located downstream of the sight glass and of the liquid–vapor separator, respectively. All flow meters are specially calibrated for R-134a from the manufacturer. The total capacity of all refrigerant flow meters is 0.3–3.3 l/min. The manufacturer's listed accuracy is 0.1% of the full scale. The temperatures are measured by T-type thermocouples having accuracy of 0.1 °C. Bourdon gauges, calibrated against the dead weight tester, are used to measure the pressures. All static pressure taps are mounted flush in the tube wall.

The heat load is supplied to the evaporator by using a hot-water loop. The tank water is heated with a 4.5 kW electric heater and supplied through the evaporator by a circulating pump. The condenser emits heat to the water coming from a cold water tank. The water is cooled by a separated 2.6 ton refrigeration system using R-22 as refrigerant.

The ejector assembly is shown in Disawas and Wongwises [7]. The ejector is made of brass and divided into three main parts: the motive nozzle, the suction chamber, and the mixing chamber with diffuser. The motive nozzle throat area is designed according to the Henry and Fauske model [8]. This model is used because it considers the metastable effect of the expansion of saturated liquid into the liquid–vapor mixture region. The remaining cross-sectional areas of the ejector are designed according to the homogeneous equilibrium model (HEM) [2,9]. The HEM is based on the assumption that vapor and liquid are in thermal and mechanical equilibrium. Furthermore, the mixing process is assumed to occur at constant pressure. The ejector dimensions, including the lengths of each section and the convergent and divergent angles, are based on recommendations from the ASHRAE Handbook [1] and from Nakagawa et al. [6].

Experiments were performed for both the TPERC and the CRC systems. Tests were carried out at different

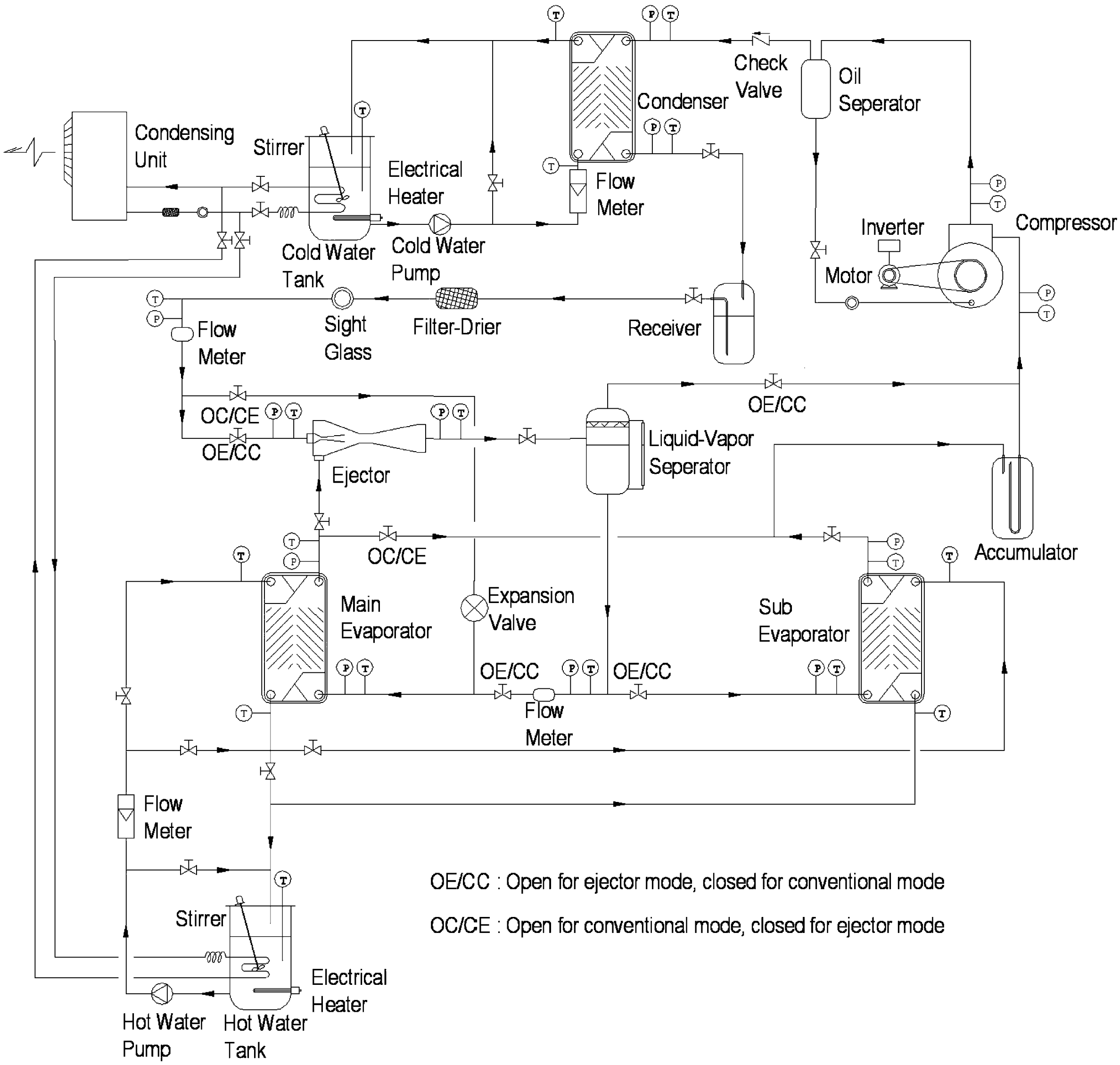

Fig. 1. Schematic diagram of experimental apparatus.

volumetric flow rates of hot water flowing through the evaporator, ranging between 8 and 16 l/min while the volume flow rate of cold water flowing through the condenser was kept constant at 14 l/min. At each value of hot water flow rate, experiments were conducted with the inlet cold water temperature ranging between 26 and 38 °C. The compressor speed was maintained at 450 rpm by controlling the inverter frequency.

3. Results and discussion

This section presents the results obtained from the experimental apparatus explained before. The comparison of the cycle performance from both modes of operation, TPERC and CRC, used in this work is based on the external parameters as described in [10,11].

Figs. 2 and 3 present the variations of the compressor discharge temperature and the compressor pressure ratio with the inlet cold water temperatures, respectively. Considering Fig. 2, for TPERC, at the same hot water flow rate, the discharge temperature at higher inlet cold water temperature is higher than at lower inlet cold water temperature across the range of inlet cold water temperatures. The experimental results obtained from the CRC show a similar trend. However, at the same inlet cold water temperature and the same hot water flow rate, the discharge temperature of the CRC is higher

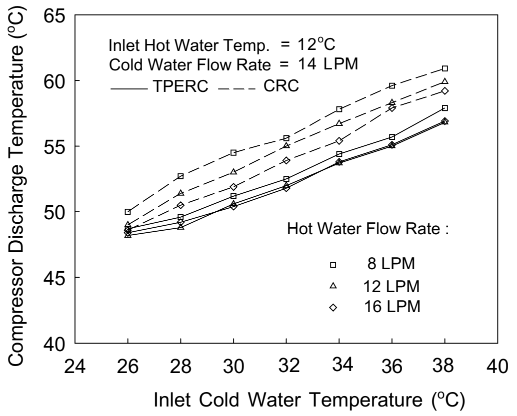

Fig. 2. Comparison of discharge temperature between the TPERC and the CRC as a function of inlet cold water temperature.

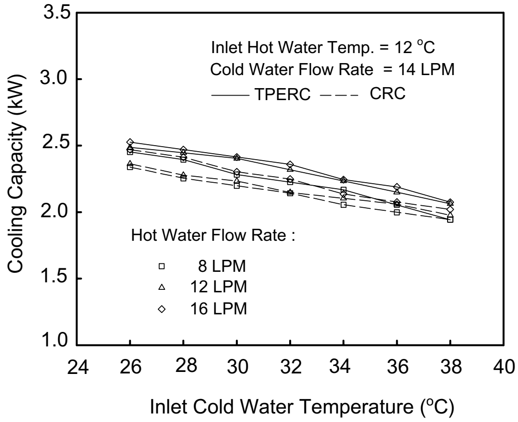

Fig. 4. Comparison of cooling capacity between the TPERC and the CRC as a function of the inlet cold water temperature.

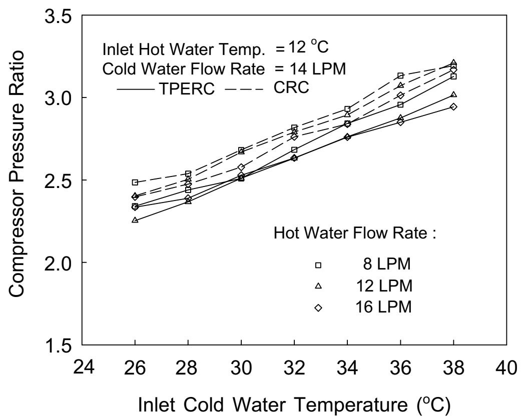

Fig. 3. Comparison of pressure ratio between the TPERC and the CRC as a function of the inlet cold water temperature.

than that of the TPERC. It can also be clearly seen from Fig. 3 that at the same inlet cold water temperature and the same hot water flow rate, the pressure ratio of the TPERC is lower than that of the CRC.

This implies that at the same inlet hot water temperature, the evaporator pressure of the TPERC is higher than that of the CRC. In other words, the evaporator of the TPERC operates at a higher saturated temperature than that of the CRC. This is due to the higher heat transfer coefficient and higher mass flow rate of refrigerant flowing through the evaporator. Consequently, the temperature difference between the refrigerant and the heat transfer fluid (water) in the evaporator of the TPERC is lower than that of the CRC. A lower pressure ratio and lower discharge temperature means better lubrication and that a longer compressor lifespan can be achieved, resulting in a higher compressor reliability.

Fig. 4 shows the variation of the cooling capacity with inlet cold water temperature for an inlet hot water temperature of 12 °C at the different hot water flow rates of 8, 12, and 16 l/min. It can be seen from the figure that as the inlet cold water temperature increases, the cooling capacity for both modes of operation decreases. For the CRC, this results from the increasing condensing temperature which also affects the refrigerating effect.

For the TPERC, although the temperature difference between the refrigerant and the water at the evaporator is lower, its cooling capacity is higher than that of the CRC. This results from the increase of wetted area and mass flow rate in the TPERC evaporator, which causes the overall heat transfer coefficient in the evaporator of the TPERC to be higher than that of the CRC under the same area of the heat exchanger. In other words, the CRC is controlled in order to obtain a completed evaporation of liquid–vapor mixture into a superheated vapor before exiting the evaporator. Therefore, the evaporator of the CRC loses some area at the outlet for superheating, while the evaporator outlet of the TPERC is in a liquid–vapor mixture condition, causing the increase of the overall heat transfer coefficient. With the higher overall heat transfer coefficient, the evaporator of the TPERC can be smaller than that of the CRC, indicating that it can reduce the initial cost of the system.

Fig. 5 shows the variation of the coefficient of performance with inlet cold water temperature at an inlet hot water temperature of 12 °C for the different hot water flow rates of 8, 12 and 16 l/min. It can be seen that the COP decreases with increasing inlet cold water temperature. This is due to the increment of power input to the compressor. At the same conditions, the COP of the TPERC is higher than that of the CRC over the range of low inlet cold water temperatures. However, the

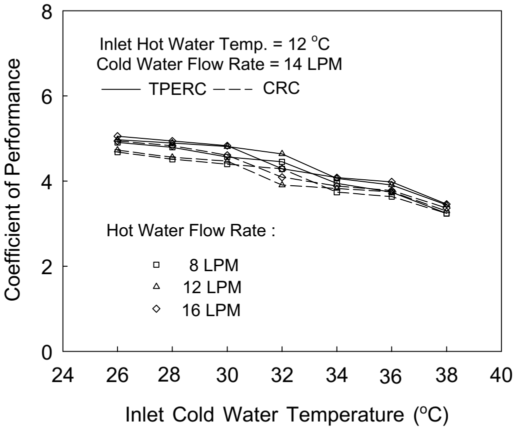

Fig. 5. Comparison of coefficient of performance between the TPERC and the CRC as a function of the inlet cold water temperature.

measured data reveals that the improvement in COP diminishes as the inlet cold water temperature increases.

4. Conclusions

The experimental results show that the TPERC has lower compressor pressure ratio, lower discharge temperature, higher cooling capacity and higher COP than those of the CRC. The COP of the TPERC shows a slight improvement above that of the CRC at low inlet cold water temperatures. The flow rate of heat transfer fluid (hot water) has a significant effect on the relevant parameters, especially for the CRC.

Acknowledgements

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