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Research Paper

Experimental investigation on a capillary tube based transcritical CO_2 heat pump system

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HIGHLIGHTS

- A capillary tube based transcritical CO_2 heat pump was experimentally studied.
- Outdoor air temperature showed larger effect on performance than indoor temperature.
- The studied system behaved similar to an EEV system based on performance comparison.
- Gas-cooler outlet temperature showed large effect on the system performance.
- The studied system showed very promising results under experimental conditions.

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ABSTRACT

In this study, a transcritical CO_2 heat pump system using a capillary tube as an expansion device was experimentally investigated. A prototype was developed for this purpose. Experimental study was conducted under different outdoor and indoor temperatures. The results showed that the system cooling capacity and coefficient of performance (COP) dropped by 22% and 24%, respectively as the outdoor air temperature increased from 30 to 40 °C and increased by 12% and 15%, respectively as the indoor air temperature increased from 22 to 32 °C. The system performance was further compared with the transcritical CO_2 heat pump using an electronic expansion valve (EEV) at different gas-cooler outlet temperatures. The comparison results showed that the transcritical CO_2 heat pump using the capillary tube is promising and could achieve performance close to the heat pump using EEV over a wide range of gas-cooler outlet temperature.

1. Introduction

As a natural refrigerant, carbon dioxide (CO_2) has lots of technical advantages which include environmental friendliness (zero Ozone Depletion Potential and very low direct Global Warming Potential), low cost, easy availability, non-flammability, non-toxicity, compatibility with various common materials and compactness due to high operating pressures [1]. Some pioneering studies have proved that use of CO_2 as a refrigerant can provide an efficient and environmentally attractive technology in the application areas of air-conditioning, refrigeration and heat pump [2,3].

In the past decade, some theoretical and experimental research to develop an energy efficient CO_2 transcritical system has been carried out by researchers in various applications. Rozhentsev and Wang [4] discussed the special design features of CO_2 air-conditioners in air-conditioning. Richter et al. [5] experimentally compared the commercially available R410A heat pump and prototype CO_2 system in heating mode. The results showed that the CO_2 system operated with a slightly lower heating COP, but its higher capacity at the lower outdoor temperature reduced the need for less efficient supplementary heating capacity. Stene [6] investigated a residential brine-to-water CO_2 heat pump system for combined space heating and hot water heating. The system was tested under different operating modes and working conditions. It was concluded that the CO_2 heat pump system might achieve the same or higher seasonal performance factor than the most energy efficient state-of-the-art brine-to-water heat pump system as long as it met a certain condition. Yokoyama et al. [7] theoretically studied the performance of an air-to-water CO_2 heat pump. The system performance was clarified in consideration of seasonal changes of ambient air and city water temperatures. Sarkar et al. [8] studied the transcritical CO_2 heat pump cycle for simultaneous cooling

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and heating. The optimal COP was found to be a function of the compressor speed, the coolant inlet temperature to the evaporator, the inlet temperature of the fluid to be heated in the gas cooler, and compressor discharge pressure. White et al. [9] experimentally investigated a prototype transcritical CO_2 heat pump operating at high temperature (above 65 °C) and proposed a model to investigate the system performance at hot water temperatures up to 120 °C. Hu et al. [10] experimentally analysed the performance of an air-source transcritical CO_2 heat pump water heater using the hot gas bypass defrosting method. An energy analysis was performed for different system components during the defrosting process.

In the above studies, the electronic expansion valve (EEV) was widely applied in the transcritical CO_2 system to optimize the system COP. Little study investigated the application of capillary tubes in the transcritical CO_2 system although the capillary tube has been extensively used in small size refrigeration and air-conditioning systems. Madsen et al. [11] carried out the first theoretical and experimental study to investigate effects of using adiabatic capillary tubes as an expansion device in a transcritical CO_2 refrigeration system. The results showed that system performance using the capillary tube was promising. However, the simplified study did not consider the heat transfer, fluid flow and internal surface effects of the capillary tube. Królicki and Bialko [12] reported studies on a non-adiabatic capillary tube based transcritical heat pump system. A non-adiabatic homogeneous model of CO_2 flowing through a capillary tube was proposed and the relationships between gas cooler pressure and evaporating temperature was presented. However, issues related to capillary tube optimization and performances of the capillary based system were not well addressed. Agrawal and Bhattacharyya [13] presented characteristics and first law analysis of adiabatic capillary tube flow including choked flow in a transcritical heat pump based on steady flow energy balance. The same authors further theoretically evaluated the performance of a capillary tube based transcritical CO_2 heat pump system for simultaneous heating and cooling at 73 °C and 4 °C, respectively against optimized expansion valve systems [14] and experimentally investigated the system performance using different capillary tubes [15]. The results showed that the capillary tube system is quite flexible in response to changes in ambient temperature, almost behaving to offer an optimal pressure control.

Although the above studies have reported the potential of applying the capillary tube in the transcritical CO_2 heat pump system, there is still a lack of experimental study on the capillary tube based transcritical CO_2 system. In this study, a prototype transcritical CO_2 heat pump system using a capillary tube as expansion device was developed and experimentally studied under different outdoor and indoor conditions. The system performance was experimentally compared with the transcritical CO_2 system using an EEV under different gas-cooler outlet temperatures. The results will provide engineers and researchers important experimental data for future theoretical analysis and design of the capillary tube based transcritical CO_2 system.

2. Experimental setup

Fig. 1 shows a schematic drawing of the capillary tube based transcritical CO_2 heat pump test rig which is established for this study. The experiment is performed in an environmental laboratory, which consists of two environmental rooms to control outdoor and indoor air temperatures, respectively. The dry-bulb temperature of the outdoor environmental room is controlled in the range from 30 to 40 °C while the indoor room temperature is controlled between 20 and 32 °C. The gas-cooler outlet temperature is controlled in the range from 30 to 50 °C by varying the outdoor air flow which is adjusted through the frequency control of the electrical fan.

The test system includes a semi-hermetic reciprocating compressor, a gas cooler (tube-fin configuration), an evaporator (tube-fin configuration), a liquid-vapour separator, an electric expansion valve (EEV), a capillary tube and a set of piping system. The design parameters of the gas cooler and evaporator are listed in Table 1. The test system is well instrumented and the measurement points are indicated in Fig. 1. The compressor power consumption is measured using an electric power meter (8901F, QINGZHI) with an accuracy of ±0.25% of reading. The nominal air flow rate of the gas-cooler and evaporator axial fans is 6000 m^3/h. The mass flow rate of air is measured by using the pressure difference across a nozzle. The pressure difference is measured using a pressure transducer (PTX7500, accuracy of ±0.2% of full scale) with a measuring range of 0–70 mbar. The fan speed can be adjusted via frequency control. The temperatures are measured using Class A type RTD sensors (PT100) with an accuracy of ±0.2 °C. Refrigerant pressures are measured using pressure transducers (accuracy, ±0.25% of reading) with a measuring range of 0–6 MPa for the low-pressure side and 0–16 MPa for the high-pressure side. The mass flow rate of the refrigerant is measured by an electromagnetic flow meter (SITRANS FC of SIMENS) (accuracy, ±0.1% of reading) with a measuring range of 0–125 kg h^{-1} for the mass flow rate and 0–6.5 g cm^{-3} for the density. All the experimental data, such as temperature, pressure, refrigerant mass flow rate and electrical power are collected by a calibrated Agilent HP34970A data acquisition system. The measured data throughout the experimental process are collected and recorded at a time interval of 10 s.

During the experiment, when tests are carried out to evaluate the performance of the capillary tube based transcritical CO_2 heat pump, the EEV loop is isolated from the refrigeration circulation loop. The high pressure CO_2 gas from the gas cooler is throttled by the capillary tube, turning into low pressure and low temperature CO_2 gas entering the evaporator. By contrast, when the experiment is performed to evaluate the performance of the system using an EEV, the capillary tube loop is isolated. The high pressure CO_2 gas is then expanded through the EEV. Both the outdoor and indoor air temperatures are controlled in the environmental room. The indoor air flows through the evaporator and the outdoor air flows through the gas cooler. The air flow rate is controlled by the frequency of the electrical fan.

3. Data reduction and error analysis

3.1. Performance calculation

The measured refrigerant operating pressure and temperature data can be used to determine the specific enthalpy values at the inlet and outlet of each component. Then mass and energy balances can be calculated for the main components of the heat pump system to evaluate their performance with the measured flow rate data.

The cooling capacity $\dot{Q}_c$ is calculated based on the enthalpy change of the air across the evaporator and the air mass flow rate, $\dot{m}_a$ as follows:

$$\dot{Q}_c = \dot{m}_a(h_{e-air,in} - h_{e-air,out}) \quad (1)$$

where $h_{e-air,in}$ and $h_{e-air,out}$ are the enthalpy of the air at the inlet and outlet of the evaporator, respectively. The enthalpy value of each state point is calculated from the measured dry-air, T_a and wet-bulb temperature, T_w using the following equations [16]:

$$h_{air} = 1.005T_a + \omega(2500.9 + 1.82T_a) \quad (2)$$