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Static and Transient Characteristics of a Microchannel Two-phase Looped Thermosyphon Cooler for the sCO_2 Brayton Cycle

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ABSTRACT

The Supercritical Carbon Dioxide (sCO_2) Closed Brayton Cycle has extensive application prospect due to its high thermal efficiency and broad energy adaptability. However, the abrupt change of thermophysical properties in the near-critical region significantly affects the compressor work and operation stability, and causes great difficulty for the off-design duty control. In this work, a cooling system of microchannel looped thermosyphon filled with a two-phase refrigerant R134a, is proposed to replace the conventional cooler scheme of water forced convection. Static and transient mathematical models are formulated for the coupled fluid flow, heat transfer and thermodynamic process. A MATLAB code is programed for simulating the temperature control characteristics of the cooling system under various initial and boundary conditions. The models are validated by experiment of a small-scale cooling system composed of a microchannel two-phase looped thermosyphon on a recently built sCO_2-R134a-Air test facility. The results of static simulation show that, the R134a boiling and condensation significantly enhance the heat transfer, compared to the forced convection of single phase water under the same constraint conditions, without the necessity for external pumping power, which contributes to a lower mean temperature of the heat rejection process, therefore improves the cycle efficiency and the system compactness. The transient simulation results show that, the

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phase-change heat transfer of R134a in the looped thermosyphon could more accurately and rapidly maintain the inlet temperature of the sCO_2 compressor close to the critical point, which is favorable for reducing the compressor work, improving its stability, increasing the cycle efficiency and promoting the duty adjustability.

INTRODUCTION

The supercritical CO_2 closed Brayton cycle is well adaptable to a wide variety of heat sources, and becomes a hot area of research in new power generation technologies due to its high working fluid density, high heat absorption temperature, and reduced compression work near the critical point. However, because slight changes in temperature and pressure near the critical point can cause large fluctuation of physical properties of the working fluid, which may significantly affect the compressor power consumption and working stability. This brings difficulties for the power generation system during transient operations. The cooling process in the supercritical CO_2 Brayton cycle is conventionally realized by the forced convection of single-phase water [1-5]. Due to the limited capability of sensible heat transfer, the requirement for mass flow rate of cooling water is large, and considerable pump power is consumed accordingly. Additionally, owing to the large thermal inertia of liquid water, the temperature response is slow during load variation [6,7].

To address these issues, the forced convection liquid water cooler is replaced by a two-phase microchannel looped thermosyphon in this work to significantly enhance the cooling process. Further, due to the minimal amount of working fluid required for the two-phase circulation inside the loop and the absence of extra circulating pumps, it is expected that the temperature of supercritical CO_2 at the compressor inlet can be precisely and rapidly regulated to the desired temperature, thereby improving the cycle efficiency and system compactness. Loop thermosyphons have been widely utilized in various research and industrial fields, such as in temperature control of space isotope Stirling engines, vehicles, aerospace crafts, heat storage equipment, cooling of optical devices, precise temperature control and other applications, as well as in cooling of electronic devices and waste heat recovery [8-12]. Currently, research on supercritical CO_2 power generation technology is still in its early stage, and to our knowledge, no reports are found up to date for the cooling and temperature control of supercritical CO_2 by a phase-change method.

METHODOLOGY

Figure 1 shows a schematic diagram of the microchannel two-phase looped thermosiphon cooling system. The experimental system consists of an evaporator, a condenser, a vapor line and a condensate return line. The evaporator is a miniature shell-and-tube type heat exchanger with 32 parallel micro-tubes of 2-mm outer diameter and 1-mm inner diameter. A two-phase refrigerant R134 filled in the shell side is in countercurrent arrangement with the CO_2 stream in the micro-tubes side. CO_2 releases heat to the R134a coolant through the micro-tube wall as it flows downward, and the CO_2 temperature is lowered to a desired value before entering the main CO_2 compressor or a storage tank. R134a absorbs heat from CO_2, then boiling phase change occurs in the shell side. The generated bubbles coalesce into a steam flow and enters the condenser through the vapor line under buoyancy force. The R134a vapor dissipates heat to the surroundings through the multiple channel walls in the microchannel plate-fin condenser, and condenses into a liquid state, and then returns to the bottom of evaporator shell side through the condensate return line to complete a two-phase cycle. With repeated self-driven cycling of the two-phase R134a coolant, the CO_2 working fluid can be cooled to a precisely controlled temperature value.

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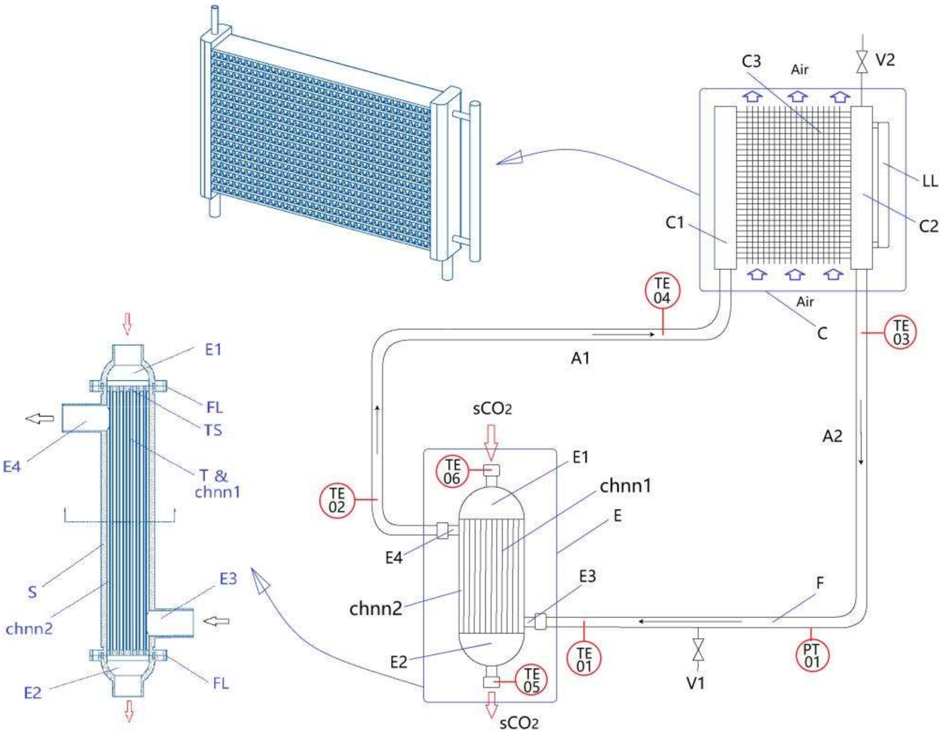

Fig.1 Schematic diagram of supercritical CO_2/R134a microchannel heat pipe cooler

E - Evaporator; E1 - Evaporator CO_2 inlet; E2 - Evaporator CO_2 outlet; E3 - Evaporator R134a inlet; E4 - Evaporator R134a outlet; chnn1 - CO_2 microchannel array; chnn2 - R134a coolant channels; C - Condenser; C1 - Condenser inlet manifold; C2 - Condenser outlet manifold; C3 – Condenser microchannel array; F - Coolant fluid; A1 - R134a vapor line; A2 - R134a condensate return line; V1 - R134a charging valve; V2 - R134a vent valve; LL - R134a liquid level indicator; Red arrow - CO_2 flow direction; Black arrow - R134a flow direction; Blue arrow - air flow direction; T - microtubes; S - Shell; TS - Tube sheet; FL – Flange; TE - Temperature elements; PT - Pressure transducer

One dimensional fluid flow, heat transfer and thermodynamic model is formulated for the cooling system based on the conservation equations of mass, momentum and energy. The entire loop containing the evaporator, the condenser and connecting pipes is discretized into 500~1000 sub-sections. Both steady and dynamic simulation programs are coded by MATLAB, in which variable thermal properties of CO_2, R134a and ambient air are adopted by calling the REFPROP program developed by NIST. Euler implicit method is adopted for the dynamic simulation. The boundary conditions are set as inlet temperature of 50~90℃, outlet pressure of 7.5-8.5MPa and mass flow rates of 0.003-0.01 kg/s for the CO_2 stream. Ambient air temperature of 21℃, ambient pressure of 0.1MPaA, and mass flow rates of 0.001~0.05kg/s are set for the air stream.

The experimental facility for validation of the simulation model is a small-scale cooling system composed of microchannel looped thermosyphon heat exchangers recently assembled in our laboratory, as shown in Fig. 2(a). The temperature measuring positions are denoted in Fig. 1, and some measured temperature data under different air velocity conditions are plotted in Figs. 2(b) and 2(c). We define the thermal resistance of the looped thermosiphon as the temperature difference between the evaporator inlet and the condenser outlet divided by the cooling load. By setting the same boundary conditions for CO_2 and air in simulations as the actual experimental conditions, we compared the calculated thermal resistance with the measured values, and found a deviation of -5.0% ~ +4.8% between them, which is fairly well acceptable for this complicated

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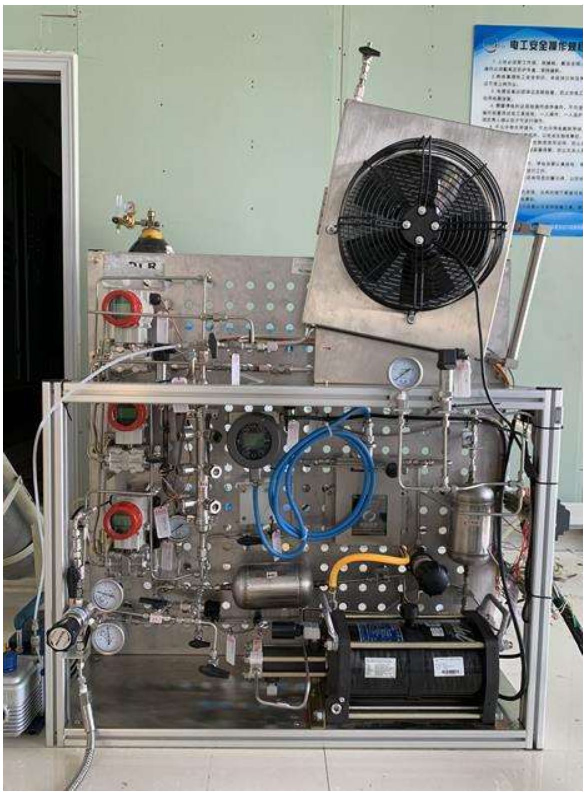
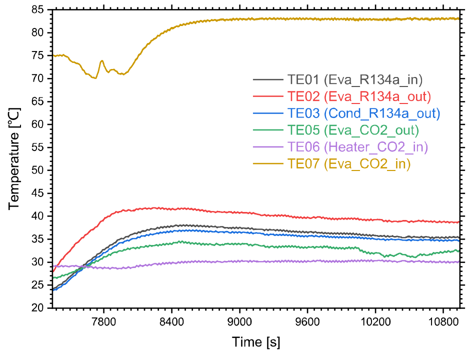
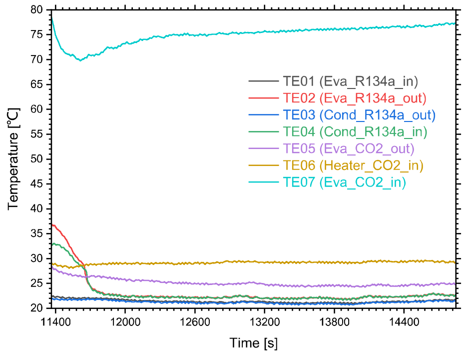
phase change cooling process.



(a) Experimental cooling system



(b) air velocity = 0.05 m/s



(c) air velocity = 1.0 m/s

Fig. 2 Experimental validation of the simulation model

RESULTS AND DISCUSSION

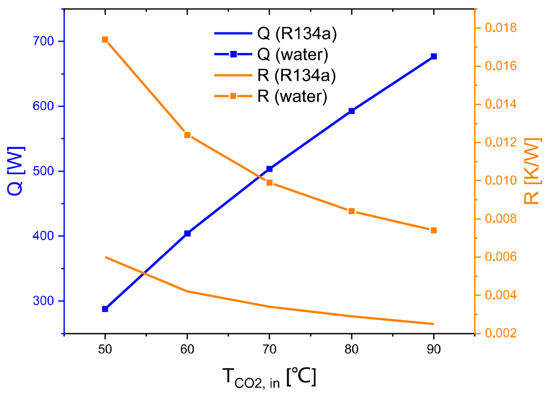


Fig.3 Effect of CO_2 inlet temperature on the loop thermal resistance

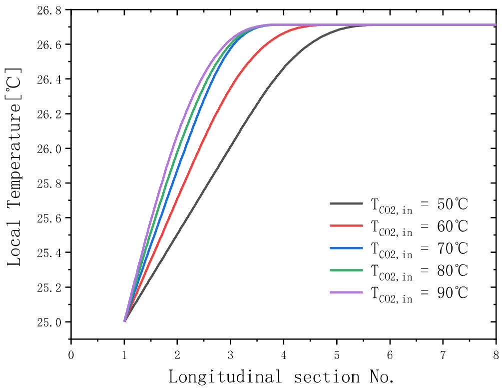


Fig.4 Effect of CO_2 inlet temperature on the R134a temperature distribution in the evaporator

Figure 3 shows the thermal resistance variations with different CO_2 inlet temperature for both R134a and water coolants when the CO_2 inlet pressure is 8 MPa, the mass flow rate is

* Corresponding author

0.01 kg/s, and the outlet temperature is 35 ℃. With the increase of CO_2 inlet temperature, the exchanged heat is correspondingly increased, the R134a coolant more quickly enters the boiling heat transfer stage, and the phase change section occupies a larger proportion of the entire evaporator length, therefore the average heat transfer coefficient is improved and the thermal resistance of the overall loop is reduced. It is also shown that the two-phase R134a loop has higher heat transfer coefficients than that of liquid water under the same boundary conditions, therefore has lower loop thermal resistance.

Figure 4 shows the longitudinal temperature distribution of R134a in the shell side under different CO_2 inlet temperature conditions when the CO_2 inlet pressure is 8 MPa, the mass flow rate is 0.01 kg/s, and the outlet temperature is 35 ℃. It can be seen that, the temperature rising rate slows down with the increase of longitudinal position due to the consecutive transition of the two-phase flow pattern from a liquid flow to a subcooled boiling (tiny bubbly flow), and then to saturated boiling (strong bubble flow or annular flow), and the heat transfer mode changes correspondingly from a sensible heat dominant mode to a latent heat dominant mode. It is also shown that, with the increase of CO_2 inlet temperature, the R134a flow more quickly enters the two-phase region, leading to a smaller loop thermal resistance and a more uniform temperature distribution.

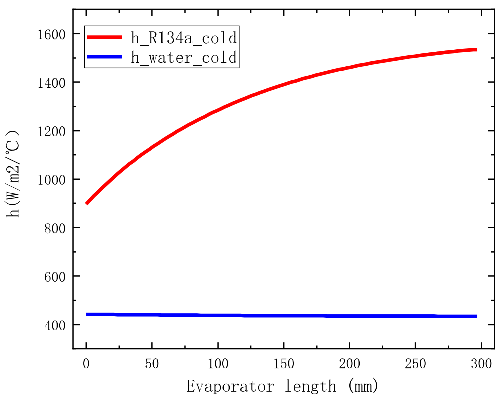


Fig.5 The longitudinal distribution of the coolant heat transfer coefficient

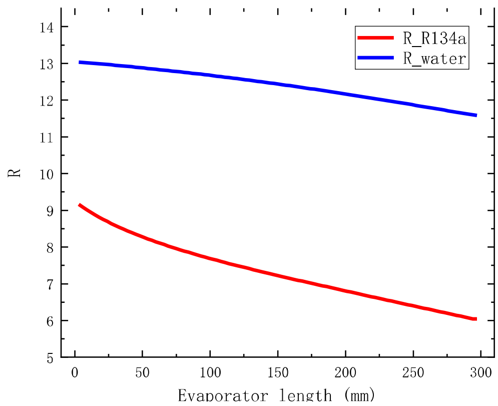


Fig.6 The longitudinal distribution of the evaporator local thermal resistance

When the boundary conditions are set as a CO_2 inlet temperature of 90°C, an inlet pressure of 8 MPa, a mass flow rate of 0.006 kg/s, and an ambient air pressure of 0.1 MPa, an ambient temperature of 21°C, and a mass flow rate of 0.05 kg/s, the simulation results for both R134a and water coolants are shown in Figs. 5 and 6. It is shown that, the boiling heat transfer coefficient of R134a coolant is much higher than that of the single-phase liquid water, therefore resulting in a significantly lower evaporator thermal resistance.

Figure 7 shows the longitudinal temperature distribution of the working fluids and metal wall along the evaporator length for both R134a and liquid water coolant cases, where the boundary conditions are the same as in Fig. 6. As can be seen from the figure, when R134a is used as the coolant, the overall temperature of CO_2 on the hot side is lower than that for water coolant, which can be attributed to the higher heat transfer coefficient. The average heat releasing temperature of CO_2 working fluid can be significantly reduced by using the two-phase R134a coolant, which is much favorable for improving the thermal efficiency of the Brayton cycle.

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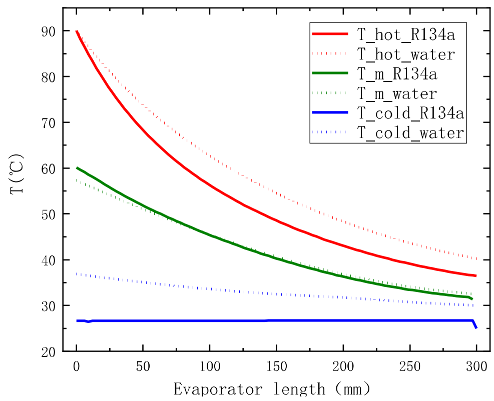

Fig.7 Temperature distribution along the evaporator

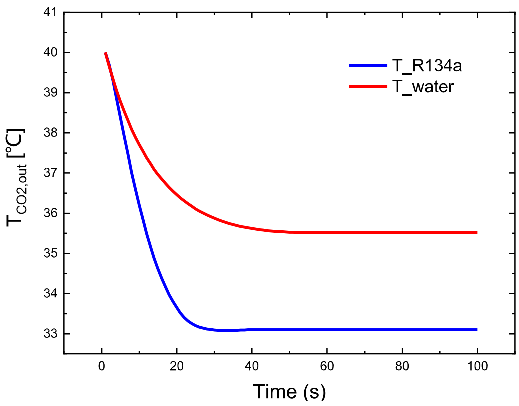

Fig.8 Transient variation of CO_2 outlet temperature with time

For the dynamic simulations, we firstly set the boundary conditions as a CO_2 inlet temperature of 90°C, an inlet pressure of 8 MPa, and a flow rate of 0.006 kg/s, and then we obtain the CO_2 outlet temperature of 40°C when a steady-state simulation converges. With the results of this steady-state calculation as the initial condition, we suddenly reduce the CO_2 inlet temperature to 70°C, with the other boundary conditions unchanged, and then the dynamic process is simulated. Figure 8 shows the CO_2 outlet temperature as a function of time for both R134a and liquid water coolant cases. It can be seen from Fig.8 that the CO_2 outlet temperature undergoes an unsteady process, wherein the CO_2 outlet temperature is finally stabilized at 33°C in the R134a coolant case, while the CO_2 outlet temperature is finally stable at 35.5°C in the liquid water case. The response time of temperature stabilization for R134a is also shorter than that for liquid water, wherein the former needs only 25 seconds to reach stability, while the latter takes up to 45 seconds. This can be explained by the higher heat transfer coefficient, significantly smaller mass flow rate and smaller thermal inertia brought by the phase change coolant.

To sum up, the steady-state simulation results show that, the heat transfer coefficient of the R134a two-phase looped thermosyphon is much higher than that of the forced convection of liquid water, leading to smaller thermal resistance, and thus could significantly reduce the average CO_2 temperature in the heat releasing process, which may contribute to promotion of thermal efficiency of the Brayton cycle. The dynamic simulation results show that, the CO_2 outlet temperature can be more quickly and accurately regulated due to the high phase-change heat transfer capability, smaller mass flow rate, and less thermal inertia.

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