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Heat transfer characteristics of a natural circulation separate heat pipe under various operating conditions

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ABSTRACT

In order to determine the heat transfer performance and the mechanisms of separate heat pipe (SHP) under various operating conditions covering two-phase states and transcritical states, experimental investigation was carried out in this study. The effect of charging mass, cold bath temperature, heat load and height between evaporator and condenser on heat pipe was analyzed and the thermal resistance was calculated to characterize the heat transfer performance. Results showed that the increase of head load, the decrease of height difference and the increase of condenser temperature led to increases in thermal resistance. Appropriate charging mass was seen to be the most important factor in terms of optimum heat transfer performance, whereas too much or too little charging mass led to the subcooling or superheating in the evaporator, which was found to have a direct impact on heat transfer performance.

1. Introduction

The heat pipe, firstly conceived by Gaugler [1], has been widely applied as a high efficiency heat transfer device able to transfer heat across long distances with small temperature difference and simple structure; additionally, external pumping power is unnecessary. With the development of heat pipe technologies, several kinds of heat pipe, such as the conventional heat pipe [2,3], the conventional gravity-assisted loop heat pipe [4,5], the separate heat pipe [6–8] and the oscillating heat pipe [9,10] have been developed. Among these, the separate heat pipe (SHP), also called closed loop two-phase thermosyphon (CLTPT), has received extensive attention due to its wickless structure. There are four main components in the SHP system: an evaporator, where the working fluid boils; a condenser, where the boiled fluid condenses; a riser and a downcomer. When the evaporator is heated, the boiling fluid flows up to the condenser through the riser while the condensed fluid flows back to evaporator through the downcomer under the force of gravity.

In the past decades, much investigation has been done on the heat transfer performance of separate heat pipe with various refrigerants, both experimentally and theoretically. Zhang et al. [11] experimentally investigated the effect of several parameters namely temperature difference, liquid charge, height difference and flow resistance on the characteristics of the two-phase thermosyphon loop. They found that larger temperature difference, lower flow resistance and optimum charge might help to achieve better performance. Chen et al. [12] studied overall heat transfer coefficients of SHP using water as the working fluid and overheat phenomenon at low liquid charge level was observed. Franco et al. [13] analyzed the relationship between mass flow rate and heat flow rate using water and ethanol as the working fluid for SHP, and found that mass flow rate increased with the increase of heat load firstly when the force of gravity dominated, which was followed by the decrease of mass flow rate when friction dominated. Haider et al. [8] proposed a mathematical model to analyze the two-phase flow and heat transfer in SHP, which was validated by experimental data with PF-5060 as working fluid. Franco and Filippeschi [7] carried out a review of the experimental apparatus and results of SHP with small dimensions (~several millimeters) and low heat load (<1 kW) where the correlation between mass flow rate and heat load in the loop was analyzed.

Additionally, the separate heat pipe has been applied widely as a practical and effective heat transfer device. Na et al. [14] presented a cooling module to cool multichip modules with SHP, and found that the size of the condenser and liquid charge level were the main parameters influencing the heat transfer performance of SHP. Samba et al. [15] developed a SHP system to cool telecommunication equipment with n-pentane as the working fluid. The result showed that the maximum limited heat load of

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SHP cooling system was two times larger than that of a traditional cooling system (air convection). Yan et al. [6] designed a separate heat pipe system for seasonal cold storage using ice as the storage material; the results showed SHP could effectively enhance the heat transfer performance of phase change material [16].

The effect of various parameters on the heat transfer characteristics of separate heat pipe has been widely investigated [11,12,15]. However, during the charging and discharging process of thermal energy storage, SHP working conditions differ, even including a joint subcooled-supercritical state, which has seldom been discussed in the previous studies. Furthermore, the heat transfer mechanism is still poorly understood. In this study, experiments were conducted to analyze the heat transfer characteristics and mechanism of SHP under the conditions of differing charging masses, heat loads, cold bath temperatures, and height differences between evaporator and condenser covering the two-phase states and the transcritical states.

2. Experiment description

2.1. Experiment apparatus

A test rig (Fig. 1), was designed to study the heat transfer behavior of the separate heat pipe (SHP), which mainly included four parts, namely an evaporator, a condenser, a riser and a downcomer. The evaporator was a metal block with a curved fluid tube inside, heated by heating plate which was connected to a DC power. The condenser was also a metal block with a curved tube inside cooled by another metal block in which the cooling fluid flows through. These three metal blocks share same structure and dimensions. The inner view and dimensions of metal block are illustrated in Fig. 2. The evaporator and condenser were connected by 2 long tubes, the riser and the downcomer. The closed loop was made of a 6 m long stainless steel tube: 1.0 m for the evaporator, 1.0 m for the condenser, 2.0 m for the riser and 2.0 m for the downcomer. The inner diameter of the tube was 6 mm and thickness of the tube wall was 1 mm. A set of tubes and a valve were set in the riser for the vacuuming of the tube and charging of the working fluid. The whole exterior of the SHP was covered with glass fiber for insulation.

A pressure transducer applied to supervise static pressure of the SHP system with an accuracy of ±1.0 kPa, and 20 calibrated T-type thermocouples applied to measure temperature with an accuracy of ±0.1 °C were attached at different positions on the SHP closed loop as shown in Figs. 1 and 2. Meanwhile, a National Instruments PXI data log system was used to acquire pressure and temperature signals, and then transform electrical signal to pressure and temperature data which was finally recorded in a computer. In this study, R-125 is chosen as the two-phase working fluid due to the merit of low critical pressure, low critical temperature and low viscosity. The condenser was cooled by glycol solution supplied from a low temperature constant bath (SHP DCY-3015) while the evaporator was heated by a plate electric heater controlled by a DC power (Gw INSTEK SPS-3610). The charging mass was measured using an electronic scale (SHP MP100K-1).

2.2. Heat loss to ambient

Although glass fiber with a thickness of 33.5–50 mm was utilized to insulate the whole exterior of the heat pipe, a certain amount of heat loss to the ambient environment is inevitable, especially under high temperature working conditions. Before the SHP experiment, the heat loss of the evaporator without the riser and downcomer was tested solely based on the energy balance of heat load and heat loss when evaporator temperature was adjusted from 45 °C to 152 °C, as shown in Fig. 3.

At the ambient temperature of 20 °C, the linear fit of relation between the heat loss of the evaporator ϕ_{eva} (W) and the evaporator temperature T_{eva} (°C) was as below

$$\phi_{eva} = 0.1319 \cdot T_{eva} - 3.7448 \quad (1)$$

The total heat loss of the SHP was comprised of the heat loss of the evaporator, the condenser, the riser and the downcomer. For the quadrate evaporator and condenser, the heat loss can be calculated from

$$\phi_i = k_i A_i (T_h - T_\infty), \quad \frac{1}{A_i k_i} = \frac{1}{A_1 h} + \frac{\delta}{A_2 \lambda_g} \quad (2)$$

and for cylindrical riser and downcomer, the heat loss can be calculated from

$$\phi_i = k_i A_i (T_h - T_\infty), \quad \frac{1}{A_i k_i} = \frac{1}{A_3 h} + \frac{\ln(d_2/d_1)}{2\pi \lambda_g l} \quad (3)$$

where k_i is overall heat transfer coefficient, T_h is surface temperature of heat pipe, h is coefficient of convection heat transfer, A_i is

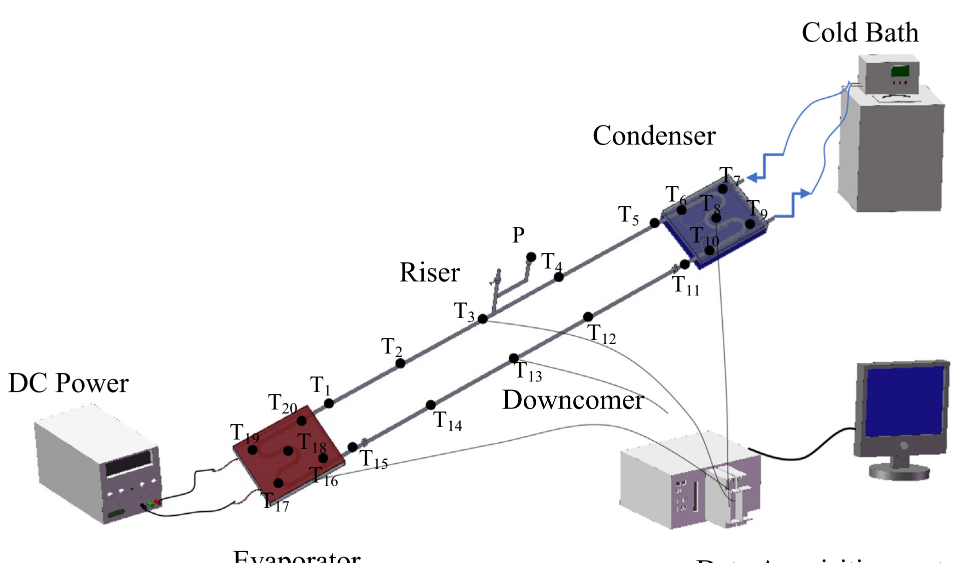

Fig. 1. 3D view of the SHP.