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Experimental study of an ammonia loop heat pipe with a flat plate evaporator

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ABSTRACT

In the present paper, the combination of a primary wick and a secondary wick was put forward and applied to the loop heat pipe. The evaporator was constructed from 304L stainless steel, and anhydrous ammonia with a purity of 99.995% was selected as the working fluid. A secondary wick made of 400-mesh stainless steel wire mesh was inserted into the evaporator, and a blind hole was made in its radial direction. Some partial grooves at the end of liquid line served as the returning liquid exit, and it was wrapped by the blind hole for fear that it be directly exposed to the compensation chamber, potentially causing it to be blocked by the vapor. The experimental results demonstrated that the loop could start up successfully and operated without temperature oscillations under the given heat load range from 20 W to 110 W in the horizontal orientation. In addition, the system rapidly responded to heat load cycle. The thermal resistance of the loop and the evaporator heat-transfer capacity were analyzed, and two operation modes existed because of the different distributions of the operating temperature with the evaporator wall temperature above or below 60 °C.

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1. Introduction

Two-phase heat transfer devices have been widely studied owing to their high efficiency, high stability, and good compatibility with electronic equipment [1,2]. Loop heat pipes (LHPs) are two-phase heat transfer device that possess many advantages: long distance heat transport, small thermal resistance, low mass, and simple structure. Therefore, LHPs have great application prospects in the field of thermal control of electronic instruments. The factors affecting the operating performance of LHPs, including evaporator structure, working fluid, charge ratio, and operating conditions, have been investigated both theoretically and experimentally [3–5]. The evaporator plays prominent role in enhancing the heat transfer capacity of the LHPs, and the main evaporator shapes consist of two types: cylindrical and flat [6]. The evaporator structure is closely related to the location of the compensation chamber (CC) and the path of the working fluid supply, further determining the maximum heat transfer capacity.

For the two types of evaporators, there are two different arrangements of the CC and evaporator. For cylindrical evaporators, the CC is located at the end of the evaporator; alternatively, for flat evaporators, the CC is integrated inside the evaporator. Numerous researchers have studied different evaporator structures for gaining the best operating performance. Maydanik et al. [7] tested 15 different variants of miniature LHPs with cylindrical evaporators with ammonia as the working fluid, and achieved a maximum heat flux of 135 W/cm^2. For LHPs with the flat evaporator, Liu et al. [8–10] and Wang et al. [11–13] have conducted numerous fundamental experiments, especially regarding the optimized design of the evaporator structure.

On the other hand, among the loop components, porous wicks are the key to improve the LHP operating performance because the capillary force developed on the evaporating surface of the wick drives the circulation of the working fluid. Numerous investigations about the thermal and hydraulic performance of the wick have been made. Aiming at improving capillary force and effectively organizing two phase flow in porous wick, efforts have been made to fabricate multi-layer wicks and biporous wicks with large pores and small pores in Ref. [14–17]. Xu et al. [18] combined modulated porous wick, secondary monoporous wick and third thermal insulation wick, which could provide good thermal and liquid link between the CC and the evaporating surface, enlarging capillary force and reducing heat leakage.

In existing flat evaporators, most do not contain a secondary wick, and the primary wick is directly placed at the flange of the evaporator envelope [10,19], which increase the thickness of the

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Nomenclature

A	evaporator heating area, m^2
K_e	overall heat-transfer coefficient of the evaporator, $W/(m^2 \cdot °C)$
Q_{app}	heat load applied to the evaporator active zone, W
R_{evap}	evaporator thermal resistance, °C/W
R_{LHP}	thermal resistance of the LHP, °C/W
T_{cond}	average temperature of the condenser inlet and outlet measured by Tc8 and Tc12, °C
T_e	average temperature of the evaporator active zone measured by Tc1–Tc4, °C
T_v	temperature of the vapor exiting the evaporator measured by Tc6, °C

evaporator wall and heat leakage from the active zone. In the present paper, an annular secondary wick was inserted into the evaporator, and a blind hole was made in its radial direction. This idea of implementing a secondary wick into the evaporator of an LHP system was developed and tested. The secondary wick—made of stainless steel wire mesh—contributed not only to the tight contact between the primary wick and the inner surface of the evaporator active zone, but also supported the primary wick while providing a liquid link between the CC and the evaporator. In a horizontal orientation (the evaporator active zone below the CC) under different heat sink temperatures, the startup performance and stability of continuous operation within a heat load cycle were carried out. Specifically, the thermal characteristics of the LHP were analyzed in detail and comparison with other studies was also made.

2. Description of experimental setup

The investigated secondary wick was inserted into a 304L stainless steel LHP with a disk-shaped evaporator; the working fluid was anhydrous ammonia with a purity of 99.995%. Considering the thermophysical properties of the working fluid, ammonia has the advantages of a steep slope of vapor pressure to temperature and considerable latent heat of vaporization. However, the working pressure of ammonia is much higher than other fluids, such as water, methanol and acetone, which pose a challenge for the fabrication of the flat evaporator.

Table 1 gives design parameters of the loop components. The tested evaporator comprises an upper tube structure, bottom cover plate, primary wick, and secondary wick, as shown in Fig. 1. A sample evaporator was used to test the pressure-bearing ability of the device prior to the beginning of the practical experiment. The upper tube structure and bottom cover plate were soldered by laser weld, which could bear an approximate pressure of 2000 kPa. The secondary wick was made of multiple layers of 400-mesh stainless steel wire mesh using wire-cutting technology, and a blind hole with a length of 32 mm was formed in its radial direction, as shown in Fig. 2. A biporous wick, which was introduced in Ref. [20], was adopted as the primary wick. The primary wick, secondary wick, and the bottom surface of the upper tube structure encircled a cavity as the CC. Some partial grooves at the end of the liquid line formed the returning liquid exit, and the end was inserted into the blind hole for fear that it be directly exposed to the CC, potentially causing it to be blocked by the vapor. A cylinder with three embedded heaters, which were controlled by a voltage regulator and wattmeter, provided power input for the LHP system. A double pipe condenser was used to dissipate heat carried by the vapor. The external loop was made of stainless steel tube and wrapped by adiabatic material. Fig. 3 gives the system diagram of the LHP and the locations of the temperature measuring points.

In order to minimize the effect of non-condensable gas, the loop was evacuated to a pressure of 3.7×10^{-4} Pa before charging with ammonia. The charging ratio was determined at 65% of the total volume of the loop: approximately 10.4 cm^3 at an ambient temperature of 20 °C. During the tests, the temperature in the laboratory room was maintained between 24 °C and 28 °C, and the temperature of vapor exiting the evaporator was set below 35 °C for safety.

3. Test results and discussion

3.1. Startup tests

Successful startup is crucial for any type of LHP and the necessary precondition that demonstrates the LHP's application potential. Startup of an LHP is closely connected with the CC and evaporator structure, initial conditions inside the evaporator, and operation termination prior to startup [21]. Good startup performance guarantees subsequent steady operation. When heat load is applied to the evaporator, the pressure difference across the primary wick drives the circulation of the working fluid in the horizontal orientation. The required pressure difference—affected by heat leakage from the heating zone and vapor penetration—corresponds to the temperature difference between the CC and the heating zone. Well-organized thermal distribution inside the evaporator and appropriate evaporator structure contribute to the establishment of this pressure difference. In this experiment, the CC was separated from the heating zone by a secondary wick that improves sealing quality and makes the primary wick tightly attach to the inner surface of the evaporator active zone, further leading to a fast startup.

Fig. 4 illustrates the startup process with 10 W at a sink temperature of −5 °C. When the heat load was applied to the evaporator

Table 1
Main design parameters of the loop components.

Evaporator	Overall height	12 mm	Compensation chamber	Height	5.1 mm
	Heating surface diameter	36 mm		Volume	4.6 ml
Secondary wick	Thickness	5.1 mm	Primary wick	Thickness	3.83 mm
	I.D./O.D.	27/37.4 mm		Groove width/Groove depth	2 mm/1.5 mm
	Hole depth	32 mm		Diameter	37.4 mm
				Porosity	78.69%
Condenser	Length	910 mm	Liquid/vapor line	Length	515 mm/320 mm
	I.D./O.D.	16 mm/20 mm		I.D./O.D.	2 mm/3 mm

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