[Short Note]

Heat Transport Device with Phase Change Using Two Parallel Tubes

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Abstract

In order to achieve high heat transport performance, a new heat transport device with phase change using two parallel tubes has been developed. The device consisted of two different diameter parallel tubes, which connected an electrically heated evaporator and a water-cooled condenser. Experiments were conducted systematically using water as working fluid. Time-dependent temperatures as well as pressures at the evaporator and the condenser were continuously measured and recorded. The heat transport rate was obtained from the temperature rise and mass flow rate of cooling water in condenser. The present study disclosed that a simple two-different-diameter tube system connecting a heat source and a heat sink achieved re-circulating flow and it gave an extremely high heat transport rate. It was also found in the present study that the orientation of the device had appreciable influence on the performance.

Key Words: *Heat transport device, Two parallel tubes, Phase-change, Re-circulating flow, Inclination angle*

1. Introduction

The rapid development of electronic equipments leads to an increase of energy consumption which demands an efficient cooling system. In order to achieve high heat transport characteristics, many kinds of heat transport devices (HTD) with phase change have been developed. Above all, heat pipes (HP) and thermosiphons are well known [1, 2]. The need for more compact and efficient HTD than conventional types leads to the development of a new type HTD which is described in this paper [3]. The uniqueness of this new HTD exists in the mechanism of fluid flow. The vapor and condensate flow separately through two different parallel tubes realizing one-way re-circulation flow from evaporator to condenser through the larger-diameter tube and condenser to evaporator through the smaller-diameter tube. This is in contrast to the conventional HPs where the flows in opposing directions are coexisting in the same tube causing the entrainment limit.

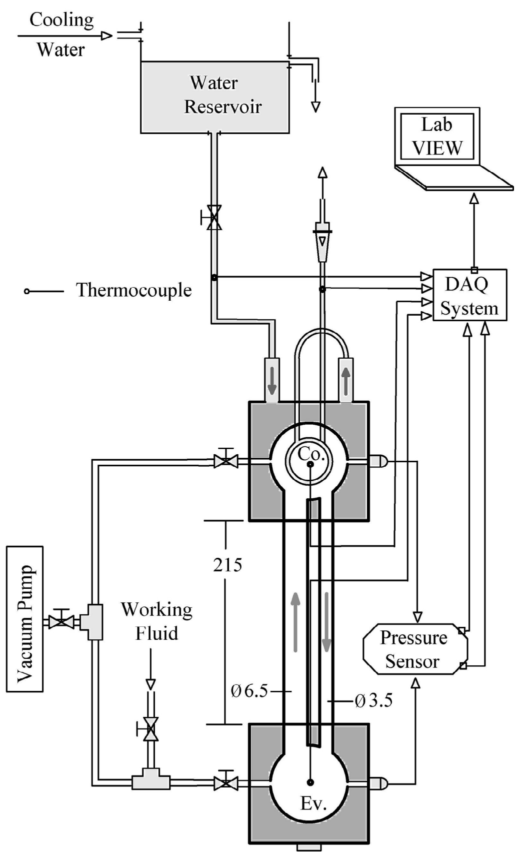

Fig. 1 Schematic of experimental setup (front view).

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2. Experimental setup

A schematic of experimental setup is presented in Fig. 1 and the side view of the test section is shown in Fig. 2. The test section consists of two different inner-diameter copper tubes (6.5 mm and 3.5 mm, respectively, and 215 mm in length) which connect the evaporator (Ev.) and the condenser (Co.) whose shape are cylindrical with 40 mm in diameter and 20 mm in axial length. Before charging the working fluid (pure water) the test section was evacuated to around 2 kPa through a vacuum pump. A given amount of working fluid (40% of the total inner volume of the test section) was charged using a graduated syringe. An electrical heater was placed on the backside of the evaporator and connected to a regulated power supply. A water cooling coil was installed inside the condenser and cooling passages were grooved on the backside of the condenser as it is seen in Fig. 2. Flow behavior was observed through the sight glasses attached to the front side of both evaporator and condenser and it was recorded by digital high-speed video camera.

Thermocouples were used for measuring temperatures of (1) inlet and outlet cooling water at condenser (2) working fluid in the evaporator and condenser and (3) the ceramic heater attached to the evaporator. Two high precision pressure sensors (-100 ~ 100 kPa) were connected to both the evaporator and condenser. The cooling water flow rate was measured using a stop watch and a graduate cylinder. The temperature and the pressure data were fed to a data acquisition card (DAQ) and they were recorded and analyzed using the Lab View software. Each value of $\dot{Q}$ shown in the following figures was obtained from the average of 6000 readings (or more), during the steady state, for an interval of minimum 10 minutes.

3. Procedure

The effect of various parameters such as (a) geometrical properties (diameter and length) of the tubes, (b) orientation of the device (c) amount of charged working fluid (pure water), on the fluid flow and heat transport characteristics were considered.

In the present study a stress was put on the effect of device orientation on the heat transport performance from evaporator to condenser. Experiments were performed under different orientations of device, with the inclination angle, θ, as indicated in Fig. 2.

The heat transport rate $\dot{Q}$ was directly determined from the mass flow rate and the temperature rise of cooling water flowing through the back side of condenser chamber and through the copper coil installed inside. The transported heat can be expressed as follows:

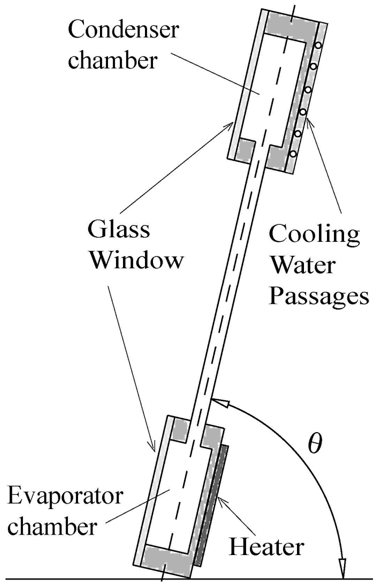

Fig. 2 Representation of the inclination angle, θ.

$$\dot{Q} = \rho \cdot c_p \cdot \dot{V}\left(T_{w,out} - T_{w,in}\right) \tag{1}$$

where $T_{w,,in,}$ $T_{w,,out}$ and $\dot{V}$ are the inlet, outlet temperatures and the volume flow rate of the cooling water, respectively.

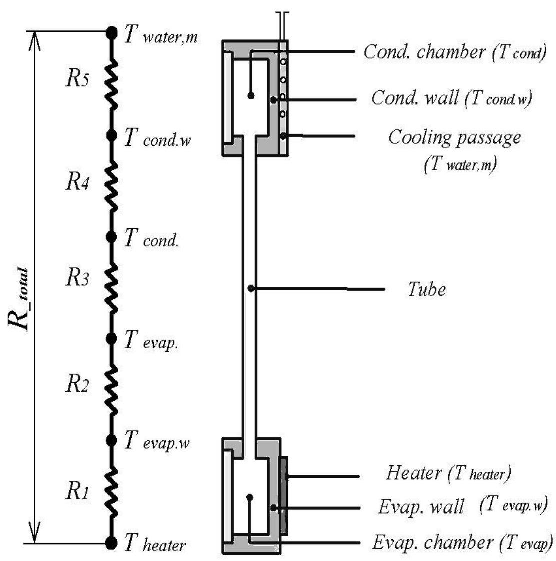

Fig. 3. Overall thermal resistance.

Fig. 3 shows the representation of the thermal resistances through which the heat passes, from the point where the heat is input into the system (the heater) to the one where it is released (the cooling water). The overall thermal resistance, $R_{total,}$ between the heater and the cooling water was calculated using the following relationship:

$$R_{total} = \left[T_{heater} - \left(T_{w,in} + T_{w,out}\right)/2\right]/\dot{Q} \tag{2}$$

4. Measurement uncertainty

The uncertainty was estimated using the method of Kline and McClintock [5]. The error range of the measurements was between 15% and 3%, for low and high heat input respectively. The majority of the experimental errors were less than 10%. Only at the lowest heat input the uncertainty was high as the error range relative to the low temperature difference of the cooling water resulted in a higher percentage error.

5. Results and discussions

Fig. 4 shows the heat transport rate $\dot{Q}$ versus evaporator temperature T_{ev} with orientation angle θ as a parameter. The figure also shows the performance of a conventional HP [4] with water and similar diameter and length. From the figure one can see that:

(1) $\dot{Q}$ increases rapidly with an increase in T_{ev} irrespective of the inclination angles;

(2) Although the conventional HP gives higher $\dot{Q}$ at low T_{ev}, the present HTD shows far better performance at higher T_{ev}, with three to four times higher heat transport rate than the conventional HP.

Fig. 5 shows $\dot{Q}$ vs. θ for three different evaporator temperatures, T_{ev}. One can observe that, in all three cases, the highest value of $\dot{Q}$ is achieved for the vertical orientation (90°). The horizontal orientation (0°) gives the minimum heat transport rate. For the same evaporator temperature, a small increase in the inclination from 0° to 10° brings an abrupt rise in the heat transport rate, $\dot{Q}$ being up to three times higher. After $\dot{Q}$ reaches the maximum at the vertical orientation (θ = 90°), the performance decreases with a further increase in the inclination angle. The difference between the 60° and 120° cases occurs because the structure of

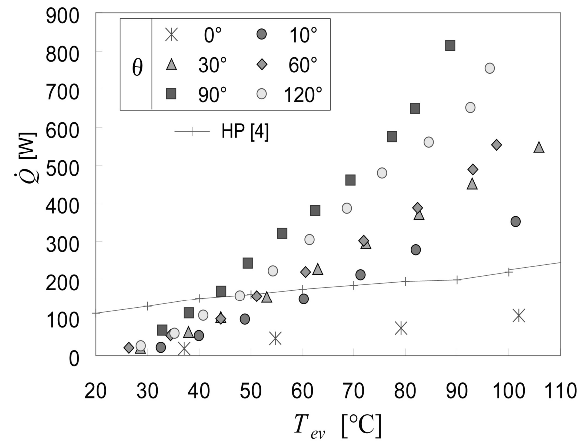

Fig. 4 Heat transport rate vs. evaporator temperature (T_{ev}) for different inclinations angle θ.

the test section is not symmetrical with respect to the center plane as shown in Fig. 2. The contact area between the liquid working fluid and the hot surface of the evaporator, and the one between the vapor and the cold surface of condenser, changes with the inclination angle. This plays a certain role on the characteristic of heat transport phenomenon and influences the performance of the present device. Both, the boiling and the condensation are important in the entire process of transporting the heat efficiently.

The higher performance of the 120° case compared to the 60° case indicates that the condenser is more dominant than the evaporator in the present system.

One can observe in Fig. 6 the results for R_{total} plotted against T_{ev} with the inclination angles as parameter. In this figure it is shown that R_{total} decreases with an increase in T_{ev} and that for an evaporator temperature

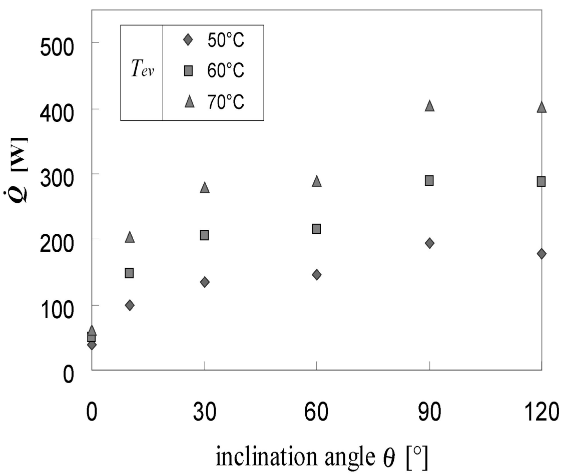

Fig. 5 Heat transport rate $\dot{Q}$, vs. inclination angle θ.

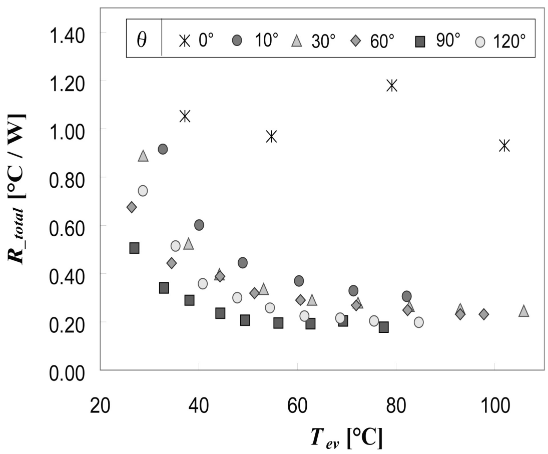

Fig. 6 Total thermal resistance vs. evaporator temperature, with θ as a parameter.

larger than 50 °C, in case of θ =90°, the values of the thermal resistances become as low as 0.2 °C/W. For the horizontal orientation, the thermal resistance is much higher, irrespective of T_{ev} .

The main reason for the high performance of the present device is the unique mechanism of fluid flow. By providing two different-diameter tubes, the flows of vapor and condensate are separated in each tube, and the shear force at the vapor-liquid interface which exists in the conventional HPs is avoided. Therefore, the entrainment limit does not occur in the present HTD.

6. Conclusions

The main results of this study are:

a) It was found that the one way recirculation was maintained to occur and this was consider to be the main cause for the achievement of high heat transport rate.
b) The present device showed better performance than conventional HP.
c) The present device could transport around 800 W for an evaporator temperature being below 100 °C.
d) The vertical orientation with the evaporator at the bottom gave the highest performance.

References

[1] Stauder, F.A. and McDonald, T.W., "Experimental study of a two-ohase flow thermosiphon loop heat exchanger", *ASHREA Trans*. 92, Pt.2A (1986), 486-496.

[2] Zou, L. et.al., Experimental investigation on parallel connection module along the exhaust current direction for gravity type heat pipes , *Proc. of 11th Int. Heat Pipe Conf.*, Vol.1 (1999), 159-163.

[3] Onishi, T. et al., Heat transport device consisting of capillary tubes with boiling driven heat transfer mechanism inside, *1st Int. Symposium on Micro and Nano Technology*, Hawaii (2004).

[4] Japan Association for Heat Pipe ed., Heat Pipe (2001), 104, Nikkan Kogyo Shinbusha.

[5] Kline, S.J. and McClintock, F.A., "Describing uncertainties in single sample experiments", *Mech. Eng. ASME*, 75 (1953), 3-8.

