EXPERIMENTAL STUDIES ON HEAT TRANSFER CHARACTERISTICS OF SS304 SCREEN MESH WICK HEAT PIPE

by

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In this paper, the experimental performance of a SS304 heat pipe under natural and forced convection modes using two different wick mesh sizes is reported. Deionised water was used as a working fluid with different heat inputs. The wick material is made up of SS303 with two mesh sizes (such as, 100 and 200 mesh). The heat pipe was positioned at three orientations (such as, horizontal, inclined, and vertical). During experiments, the ambient temperature was maintained at 30 ± 2 °C. The results showed that the heat pipe at an inclined position of 45° has about 71% higher heat transfer coefficient with minimum thermal resistance when compared to the horizontal and vertical positions. Moreover, the results confirmed that, the heat pipe using wick with 200 mesh specification at inclined position is an optimal configuration when compared to other configurations.

Key words: *heat pipe, heat transfer characteristics, SS304 wick mesh*

Introduction

Heat pipes are the passive heat transport devices used to pick up the heat from source and transfer it to the heat sinks [1]. The heat pipe consists of an evacuated copper tube partly filled with a working fluid, which exists in liquid as well as vapor phases. Heat pipes absorb the heat during evaporation of the working fluid at one end (evaporator) and allowing the vapor to pass along an adiabatic section and get condensed at another end (condenser) [2]. The working fluid is further returned back to the evaporator for further cycle. The wick materials packed in evaporator ensures good liquid distribution and also enhances the performance of the heat pipes [3]. Moreover, the performance of the heat pipes are enhanced by changing the orientation, working fluid, wick material, operating temperature and heat inputs [4]. During last decade, many research and development studies have been reported on heat pipes. In a related work, Abdel *et al.* [5] investigated the performance of a gravity assisted heat pipe with 32.5 cm length and 7 mm diameter at different inclinations using hydrogen as working fluid. It was reported that heat pipes using hydrogen in vertical position has gas fast transient response at different heat loads. In another work, Yousefi *et al.* [6] reported that heat pipe orientation has significantly influenced the cooling performance of CPU. In their work, Al_2O_3 nanofluid was used as working fluid. It was reported that the thermal resistance of the heat pipe was reduced by about 22% at a heat input rate of 25 W.

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Morawietz *et al.* [7] established a research work that it elaborates the operating limit measurement carried out during experiment for horizontal and slightly inclined two-phase thermo-syphons. The experiment has results the self-fabricated test sample (49% filling ratio) heat transfer was limited by entrainment, dry-out phenomena (rise in temperature) occurred for the commercial solar heat pipe (16% - 25% filling ratio) at 0° and 1°. Mameli *et al.* [8] analyzed a compact closed loop pulsating heat pipe (CLPHP) filled with FC-72 and it was designed to operate in the surface tension dominated regime in all the tested conditions. Performed the testing at different heat input levels, tilting the device from the vertical to the horizontal position with steps of 15° and at different filling ratios (0.5, 0.7, and 0.9). Chien *et al.* [9] tested a PHP having a non-uniform channel configuration. Test results showed that the thermal resistance decreases with the rise of heating power due to the rise of circulating speed. Zhihu *et al.* [10] developed an ammonia CLPHP and found that it has good start up performance, the ammonia CLPHP has an ideal candidate for electrical cooling with very low thermal resistance also stated that CLPHP performance was not dependent on any orientation. Khandekar and Groll [11] clarified the role of gravity in the operation characteristics of a Closed Loop Pulsating Heat Pipe and its performance (*i. e.*, overall thermal resistance) is strongly dependent on the flow pattern existing inside the tubes. The suitability of nanofluid in heat pipe applications was comprehensively reviewed by Brahim *et al.* [12]. In their work, the recent developments have been reviewed. Ranjith and Shaji [13] developed a twisted tape in double pipe heat exchanger. It was reported with improved the heat transfer coefficient on both tube side and annulus side of heat exchanger. Remeli *et al.* [14] reported the ways of waste heat recovery and conversion to electricity using a thermoelectric generator assisted by heat pipes. It was reported that, the heat pipe assisted heat recovery system has significant performance improvement when compared to conventional configurations.

The cited literature on various configurations of heat pipes confirmed that many research and developments have been reported on the performance of heat pipes using different wick materials for different applications. However, there is no specific research work has been reported on the use of heat pipes with SS304 wick materials. Hence, in this research work, the performance of the heat pipe was evaluated using SS304 wire mesh and deionised water as a working fluid.

Experiments

The schematic diagram and photographical view of experimental setup are illustrated in figs. 1 and 2, respectively. The heat pipe was made up of copper tube with outer and inner diameters of 12.7 and 9.52 mm, respectively, with a total length of 450 mm. One end of the heat pipe was closed with 3.2 mm thick copper end cap that was soldered with lead tin solder. Another end of the heat pipe was connected to the tube with pressures gauge and vacuum pump. The wicks for each heat pipe were made up of stainless steel wire screen mesh with two specifications (such as, 100 and 200 mesh). Deionised water is selected as a working fluid, which has relatively large surface tension and latent heat capacity in the temperature range between 30 and 180 °C. The quantity of water used in the heat pipe was optimized to 7 ml based on the trial and error experiments. During experiments the ambient was maintained at 30±2 °C. The heat transfer mechanism in the evaporator is mainly due to the conduction through the liquid filled wick and evaporation at the liquid vapor interface at low heat fluxes. The heat load was applied to evaporator section of the heat pipe for 100 mm length using electrical heater connected with a dimmerstat. The adiabatic section of the heat pipe is covered with zirconia sheet to avoid the heat interaction. The T-type thermocouples with ±0.3°C accuracy were used for measuring the temperatures at typical locations in the heat pipe section. The heater load was varied and the temperature measurements were observed once the heat pipe has attained its steady-state condition. The convection losses between the heater coil and the heat pipe was calculated as less than 5% of the heater load. Hence, the convective losses were neglected for performance evaluation. The output heat transfer rate from the condenser was computed using an energy balance to the condenser heat flow. The influence of heat losses to the ambient and viscous heating effects were characterized independently and were subtracted from the output heat transfer rate. The three way valve is used for removing the removing the non condensable gases from the heat pipe using vacuum pump. The deionised water was selected as a working fluid.

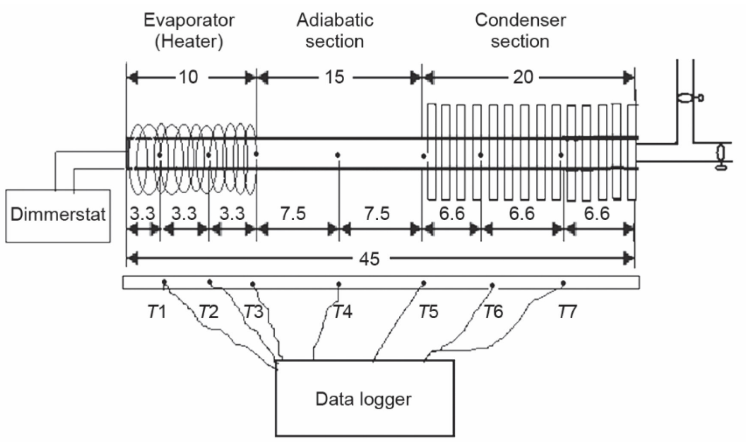

Figure 1. Line diagram of heat pipe experimental set-up

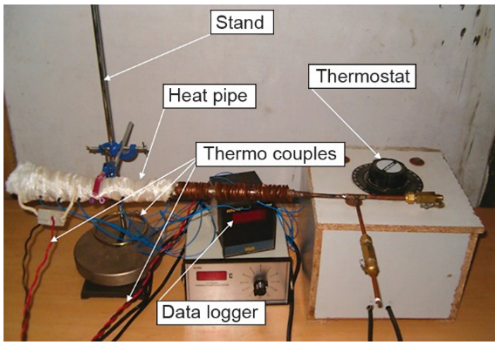

Figure 2. Photograph of experimental set-up

Data reduction

The thermal resistance in the heat pipe is estimated using:

$$R = \frac{T_\mathrm{e} - T_\mathrm{c}}{Q} = \frac{\Delta T}{Q} \tag{1}$$

The thermal conductivity of the heat pipe system is estimated by:

$$k = \frac{LQ}{\Delta T\, A} \tag{2}$$

Uncertainty analysis

The accuracy of measuring instruments will influence the uncertainties in experimental results [14]. The uncertainties in experimental results are predicted using:

$$w_r = \sqrt{\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \ldots\ldots\ldots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2} \tag{3}$$

The uncertainties observed in temperature measurements, input power and in the total resistance are calculated as ±0.2 °C, ±1% and ±2%, respectively.

Results and discussion

The influence of wick size (100, 200 mesh) and three different heat pipe orientations (0°, 45°, and 90°) under free and forced convection heat transfer modes are investigated in this paper. The results obtained in series of experiments are presented in this section.

The variations of thermal resistances against different heat inputs at three positions under forced convection mode with 200 mesh size are depicted in fig. 3. At 50 W of heat input, the thermal resistances were observed as 0.2726, 0.168, and 0.584 °C/W for 90° (vertical orientation), 45° (inclined orientation), and 0° (horizontal), respectively. The thermal resistance of heat pipe was found to be lowest for 45° inclined orientation. From fig. 3, it is observed that, the thermal resistance of the heat pipe was reduced with increase in heat input. The thermal resistance of a heat pipe positioned at vertical orientation (90°) was about 53% lower than when compared to the horizontal oriented heat pipe operating at 50 W heat input [5, 9]. An additional capillary pressure was developed in a heat pipe inclined at an angle of 45° inclination (with 200 mesh wick), which ensures effective circulation of working fluid from the condenser to the evaporator in forced convection. Furthermore, a large number of nucleation sites were observed in the heat pipe under forced convection model inclined at 45°.

In fig. 4, the variation of heat transfer coefficients at different tilt angles in forced convection heat pipe (using 200 mesh wick size) are depicted. At 50 W of heat input, the heat transfer coefficients are 921.7, 1488.4, and 430.28 W/m^{2}K for 90°, 45°, and 0° inclinations. The high heat transfer coefficient ensures better heat rejection from the heat pipe. The maximum heat rejection was observed at 45° inclination. The heat transfer coefficient of heat pipe (using 200 mesh wick size) operating at 45° inclination is about 71% higher when compared to the heat pipe operating at a 0°. Moreover, it is also observed that, the influence of nucleate boiling enhances heat transfer between the wall (with wick) and the working fluid, which enhances the heat transfer coefficient in both evaporator and condenser of a heat pipe.

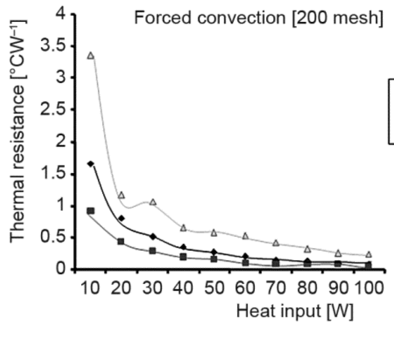

Figure 3. Variation of thermal resistance of heat pipe with heat input

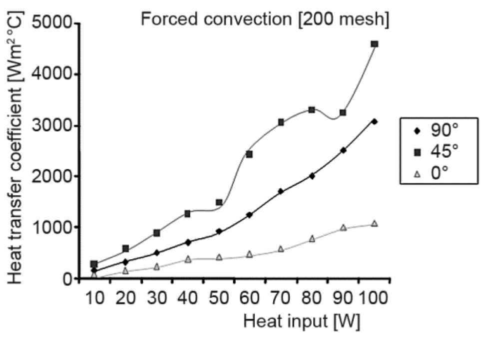

Figure 4. Variation of heat transfer coefficient with heat input

The vapor temperature against time is illustrated in fig. 5. It is observed that the heat pipe has attained the steady-state conditions in 30 minutes with constant heat input of 50 W. All the experimental observations were made after the system has attained steady-state conditions.

Figure 6 expresses the variation of temperature difference with heat input. Temperature difference ΔT increases with increase the heat load at the same condenser temperature. It increases with condenser temperate for the same heat load. The maximum temperature difference of heat pipe operating at horizontal is 72% higher the heat pipe operating at 45° heat input 50 W.

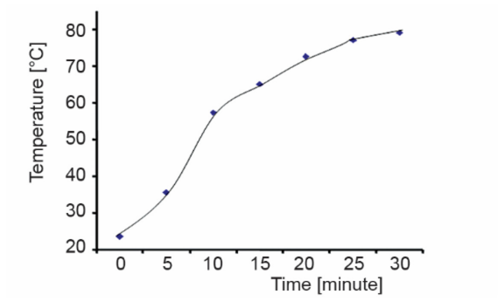

Figure 5. Vapor temperature along the length of heat pipe

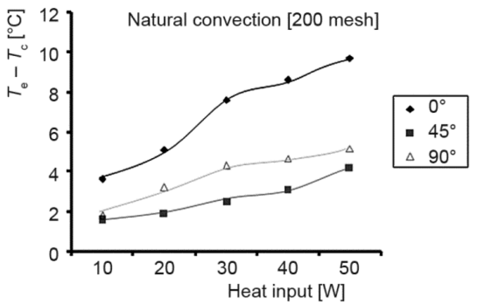

Figure 6. Variation of temperature difference with heat input

The variations of thermal conductivity of the heat pipe are illustrated in fig. 7. The effective thermal conductivity of the heat pipe is a function of the heat load in the condenser at different condenser temperatures and position of heat pipe (inclination angles). The thermal conductivity at 45° inclination was observed to be about 405.5 W/m°C. The effective thermal conductivity of heat pipe operating at 45° (with 200 wick mesh) is about 47% higher when compared to the heat pipe operating with horizontal and vertical positions at 50 W heat input. Thermal resistances of a heat pipe operating with forced convection cooling (using 200 wick mesh size) at 90° inclination was observed about 53% lower than when compared to the heat pipe operating at horizontal and vertical positions. The experimental observations presented in this section confirmed that, heat pipe (with 200 wick mesh size) and 45° inclination is a good option for heat recovery applications.

Figure 8 shows the variation of vapor temperature along the length of pipe. The vapor temperatures are the maximum for natural convection heat pipe. The vapor temperatures of heat pipe operating with forced convection with 200 mesh are 42% lower than heat pipe operating at natural convection 200 mesh at adiabatic section.

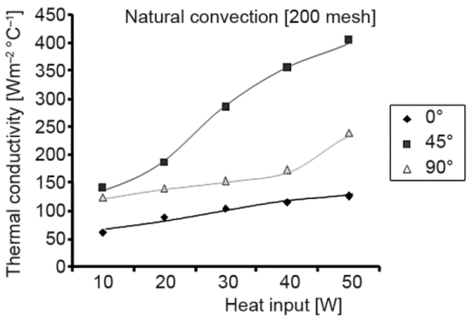

Figure 7. Variation of thermal conductivity of heat pipe with heat input

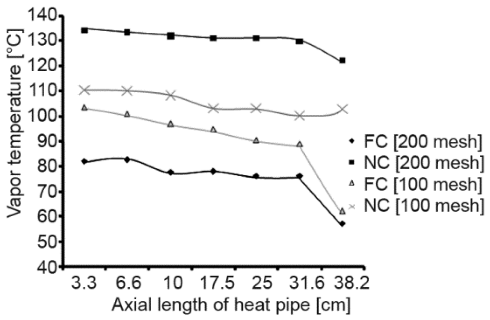

Figure 8. Variation of vapor pressure along the length of the heat pipe

Conclusions

The following conclusions are drawn based on the experimental investigations in a novel wick packed heat pipe using deionised water as working fluid.

- An improved heat transfer coefficient with reduced thermal resistance was observed with 200 wick mesh size.
- The heat pipe orientation was optimized to inclined position at 45°.

- The heat transfer coefficient of heat pipe operating at 45° inclination is about 71% higher when compared to the heat pipe operating at a horizontal position, which ensures better heat interaction in both condenser and evaporator.
- The effective thermal conductivity of the heat pipe increases as the heat input increases. The effective thermal conductivity of heat pipe operating at 45° inclination was about 47% higher when compared to the heat pipe operating at horizontal.

Nomenclature

A – cross-sectional area of the heat pipe, [m^2]
h – heat transfer coefficient, [Wm^{-2} °C^{-1}]
k – thermal conductivity, [Wm^{-2} °C]
l – length of the heat pipe, [m]
Q – heat transfer rate, [W]
R – thermal resistance, [°C W^{-1}]
T – temperature, [°C]
T_c – condenser temperature, [°C]
T_e – evaporator temperature, [°C]
ΔT – temperature difference between evaporator and condenser, [°C]

Abbreviations

c – condenser, [–]
e – evaporator, [–]

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