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Thermal Performance of Cylindrical Heat Pipes with Varying Evaporator Lengths

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Abstract:

The thermal characteristics of heat pipes with varying lengths of evaporator section was investigated keeping condenser and adiabatic section length constant. Three heat pipes with evaporator length of 0.05 m, 0.010 m and 0.015 m respectively were fabricated. Both adiabatic and condenser section length was kept constant at 0.01 m respectively. Studies were carried out separately using die-ionized (DI) water and methanol as working fluid. The heat input at the evaporator was varied from 50 to 300W in heat pipes charged with DI water and from 25 to 200 W in the heat pipes charged with methanol. Total thermal resistances of the heat pipes as well as those of both evaporator and condenser resistances separately were measured and results were analyzed. Methanol charged heat pipes showed the best performance when the evaporator length was longer than adiabatic and condenser section. But for heat pipes with deionized water, the heat pipes with equal length of evaporator and condenser showed the best performance. The length of evaporator section strongly affects the heat transport limit and thermal resistance of the heat pipe. Thermal resistance of the heat pipe varies with evaporator length and the working fluid used.

***Keywords:** cylindricalheat pipe, thermal resistance, heat transport limit, evaporator, condenser & working fluid*

INTRODUCTION

Heat pipe is a simple passive heat transfer device with the simple working principle. It is a cylindrical pipe in which a wick structure is placed in contact with the wall. It consists of an evaporator, condenser and an adiabatic section. As heat is absorbed at the evaporator of the heat pipe, the working fluid in the evaporator section evaporates. During this phase change latent heat of vaporization is absorbed from the surrounding. Due to this evaporation, a pressure difference develops between the two ends of the pipe. The working fluid is driven by this pressure gradient and it flows as a vapor towards the other end of the pipe, which is the condenser. At the condenser end, heat is removed from the heat pipe and the vapor is condensed as a liquid. Capillary forces cause the liquid to flow back to the evaporator through the wick structure and the cycle is repeated.

The working principle and the various limitations of the heat pipe have been explained in the literature [I-III]. Based on its geometry, heat pipes are classified as cylindrical heat pipes [IV], bent heat pipes [V], annular heat pipes [VI], micro heat pipes [VII] and miniature heat pipes [VIII]. Heat pipes using various working fluids (acetone, methanol, water and nanofluids) have been investigated [9-18]. Performance of the miniature heat pipe was studied using acetone and ethanol [IX]. Similarly the effect of working fluids on the performance was studied using copper-water heat pipe system [X]. Zhang [XI] introduced a new kind of working fluid, which possesses a positive gradient of surface tension with temperature, to improve the performance of capillary-pumping in heat pipe systems and ensure operation and stability of the working fluid. Recently, nanofluids have been used in heat pipes as working fluid in place of traditional working fluids. Kang et al [XII, XIII] studied the thermal performance of heat pipe using nanofluids that contain nano sized solid silver particles. Yang et al [IV] used CuO nanoparticle suspension in flat micro grooved heat pipes. Tsai et al [XIV] investigated the thermal performance of circular heat pipe using nanofluid that contains gold nanoparticles. Naphan et al [XV] analyzed the performance of heat pipes using titanium nanofluids (mixture of titanium with alcohol and water). The same authors [XVI] measured the thermal performance of heat pipe using a mixture of titanium and refrigerant (R11). Do et al [XVII] studied the performance of the flat micro heat pipe using Al_2O_3–Water mixture. Wang et al [XVIII] clarified the operational characteristics of cylindrical miniature grooved heat pipe using CuO-water mixture.

The ability of a heat pipe to transfer heat can be controlled by the properties of the wick structure. Kempers et al studied the thermal performance of heat pipe having a screen mesh [XIX] and investigated the effect of mesh layers in the wick structure on the performance of heat pipe [10]. Belousov et al [XX] used an analytical approach to evaluate the performance of heat pipe with Isotropic-Structure wicks made up of the MR material. Williams et al [XI] examined the heat transfer limit of heat pipe using step graded metal felt. A number of studies [VI, XI, XVII, XVIII, XXII, and XXIII] have been carried out on the performance of heat pipe using microgrooves as a wick structure. Similarly several investigators have [ex. XIII, XIV] used sintered wick in heat pipes.

From the literature it is understood that the parameters like geometry of the heat pipe, working fluid and wick material are strong candidates to affect the performance of the heat pipe. Though the effect of these parameters on the performance of the heat pipe is studied, the effects of length of the evaporator, condenser and adiabatic section in a given type of the heat pipe are not sufficiently understood. In this work, the effects of length of the evaporator, condenser and adiabatic section on heat transport limit and thermal resistance of the cylindrical heat pipe as well as temperature of the evaporator are investigated for two different working fluids.

Experimental details

The experimental set up consists of a test section, cooling unit and data acquisition system (fig.1). To study the effects of evaporator and condenser lengths on the thermal performance of heat pipe, heat pipes with an outer diameter of 0.019 m and length of 0.25m, 0.3 m and 0.35 m were fabricated. In each heat pipe, four layers of 100 mesh copper wire screen (with wire diameter of 0.08 mm) were used.

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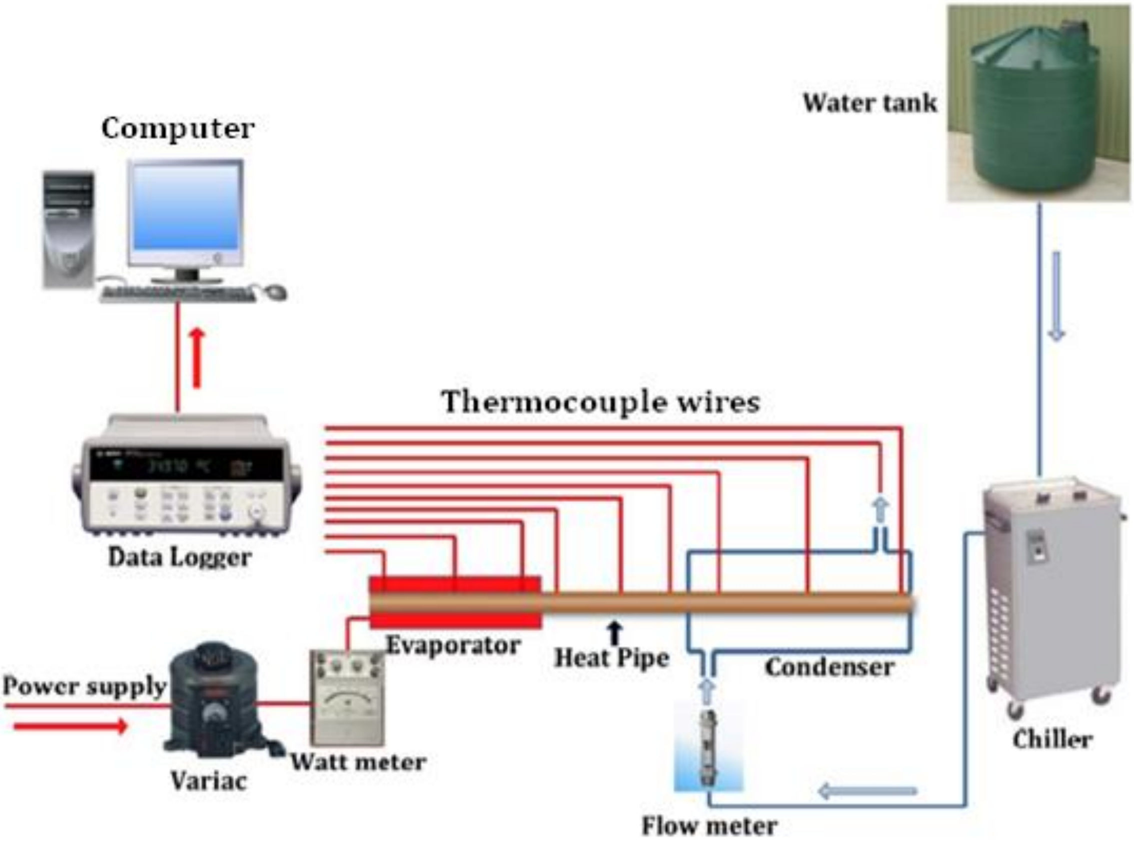

Fig. 1 Experimental set up

The heat pipes were charged with 8, 10 and 12 ml of working fluid respectively for their length of 0.25m, 0.3m and 0.35 m to saturate the wick completely. Two sets of experiments were carried out with two different working fluids, one with methanol and other with deionized water.

The heaters were designed to have the maximum power capacity of 800 W. An AC power supply was used as the source of power for the electric heater at the evaporator. In order to avoid the thermal resistance between the heater and evaporator surfaces, a thermally conductive paste was applied. A 40 mm thick fiber glass material, which has thermal conductivity of 0.04 W/m-K, was used to avoid the heat loss to the ambient. Condenser section of the heat pipe was made up of acrylic material. Cooling water was supplied from the chiller in to the cooling jacket with a constant flow rate of 350 ml/min and at a constant temperature of 21±0.5°C. Cooling water flow rate was controlled by using a calibrated flow meter. The uncertainty in the cooling water flow rate is ±3%. OMEGA T-type thermocouples with an accuracy of ±0.1°C were used to measure the wall temperatures of the heat pipe as well as to measure temperatures of cooling water at the inlet and outlet of the condenser. The thermocouple positions in each heat pipe were presented in the fig. 2 .

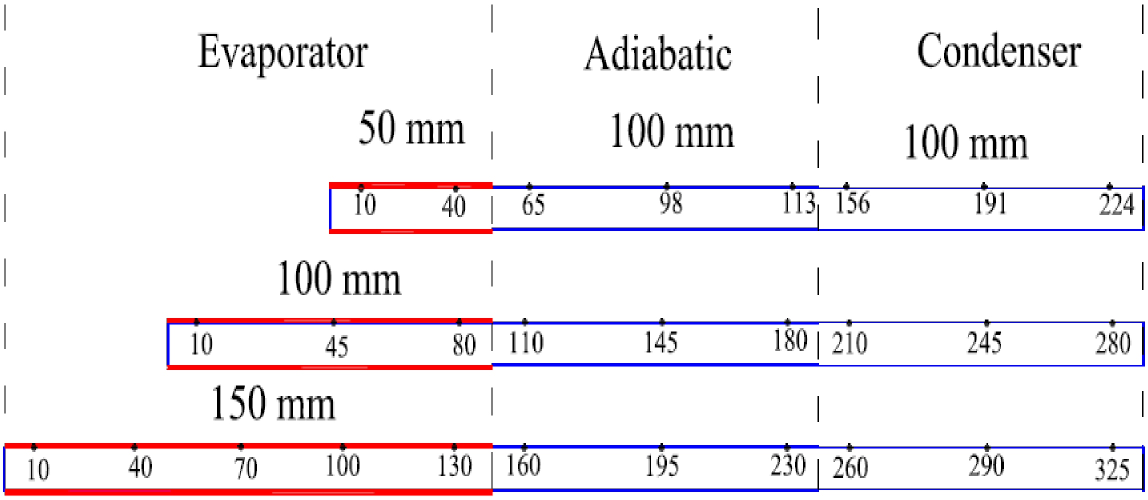

Fig. 2 Heat pipes with various evaporator lengths

The heat transport by the heat pipe under the different heating conditions was calculated using heat balance equation, $Q_t = \dot{m} \times C_p \times \Delta T$, where, $\dot{m}$, C_p and ΔT are mass flow rate, specific heat and difference in temperatures of the cooling water respectively. The calculated uncertainties in heat transfer (Q_t) and total resistance were about 4% and 6% respectively.

Heat pipes were placed in horizontal position ensured by level gauge before experiments were performed. Experiments were carried out in steady state until the dry out take place in the heat pipe. The temperature data were recorded at a time interval of 30 seconds using data logger.

Results and discussion

3.1 Resistances of heat pipe charged with de-ionized water

To study the effect of design parameters on the performance of heat pipe, thermal resistance of each heat pipe was calculated. The total thermal resistance (R_T) of the heat pipe was calculated as $R_T = (T_{e,\,end} - T_{c,\,end})/Q_{in}$. Fig. 3 shows the effect of length of evaporator, adiabatic region and condenser on the total thermal resistance of heat pipes charged with de-ionized water.

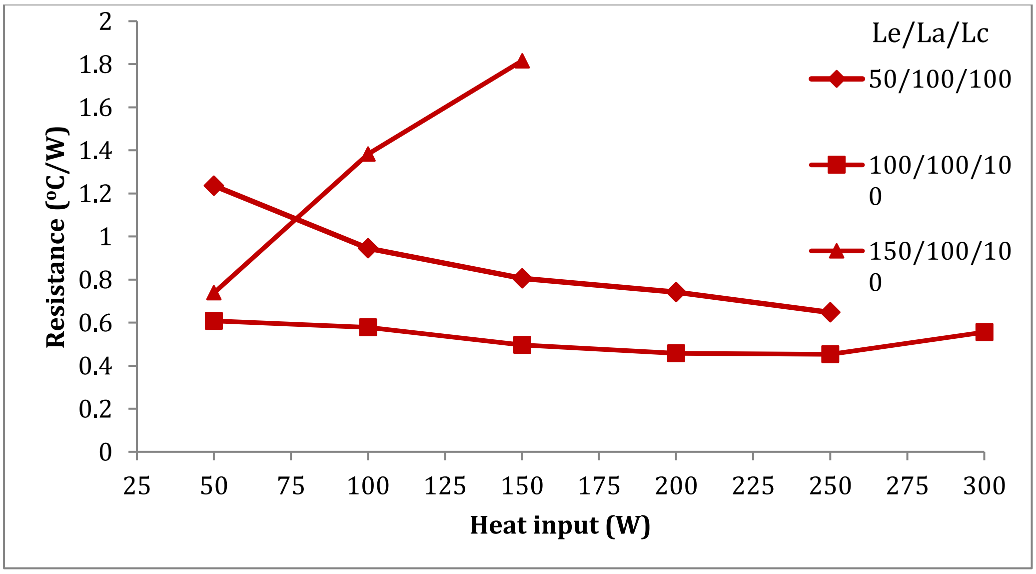

Fig. 3 Total thermal resistance of heat pipes charged with De-ionized water

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When de-ionized water is used, the least thermal resistance is achieved for the heat pipe having equal length of evaporator and condenser section. Heat pipe's with shorter evaporator length than condenser length has thermal resistance slightly higher than with equal evaporator length for lower heat input. Thermal resistance is highest for the heat pipe with evaporator longer than adiabatic and condenser section. Both heat pipe with evaporator length 0.05m and 0.15m shows higher thermal resistance than heat pipe with evaporator length 0.1m. At lower heat inputs the thermal resistance of the heat pipe with longer evaporator and medium are almost same and both are less than of shorter evaporator. When heat input is increased further thermal resistance of the shorter evaporator decreases slightly while that of longer evaporator increase. Around heat input of 75 watts the thermal resistance increases drastically in longer evaporator. The heat pipe having medium length evaporator does not show significant influence of heat input on the thermal resistance until dry out occurs. The increasing thermal resistance at higher input powers indicates the dry out at the evaporator. Since the surface area of the shorter evaporator is smaller than that of medium length evaporator, the temperature of the shorter evaporator is higher than that of medium evaporator at given heat input. This is the reason for higher thermal resistance in the heat pipe having shorter evaporator. Since the heat loss to atmosphere exceeds 20% of the applied heat at higher heat inputs, experiment on the heat pipe having shorter evaporator was stopped at 250 W. The thermal resistance of the heat pipe having longer evaporator is lower than that of heat pipe having shorter evaporator and close to that of heat pipe having medium length evaporator at the lower heat input. Then it increases with increasing heat input rapidly due to the dry out. This dry out is due to the boiling limitation in the evaporator section of the heat pipe. In the longer evaporator, the heat flux and the intensity of boiling may decrease. However, if the input heat flux is sufficient for nucleate boiling, nucleation may occur in the wicking structure and bubbles may become trapped in the wick, blocking the liquid return and resulting in premature evaporator dry out [3].

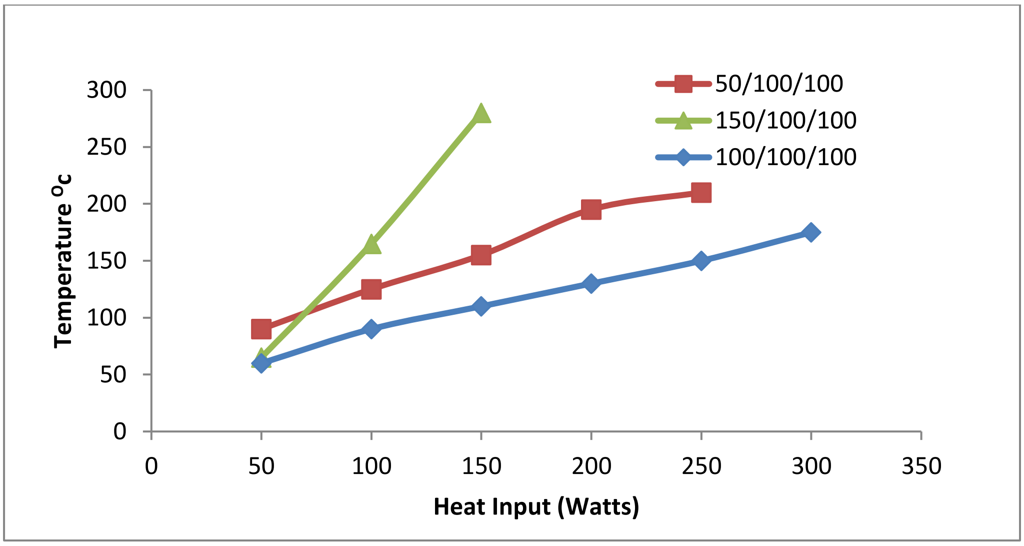

Fig. 4 Evaporator temperature of heat pipe charged with DI-water

Fig. 4 shows the evaporator temperature of heat pipe charged with DI-water for various evaporator, condenser and adiabatic section lengths. Increasing evaporator length decreases the evaporator temperature. However, though longer evaporator shows lower temperature at lower input power, the temperature rapidly increases with increasing input due to the dry out in the evaporator.

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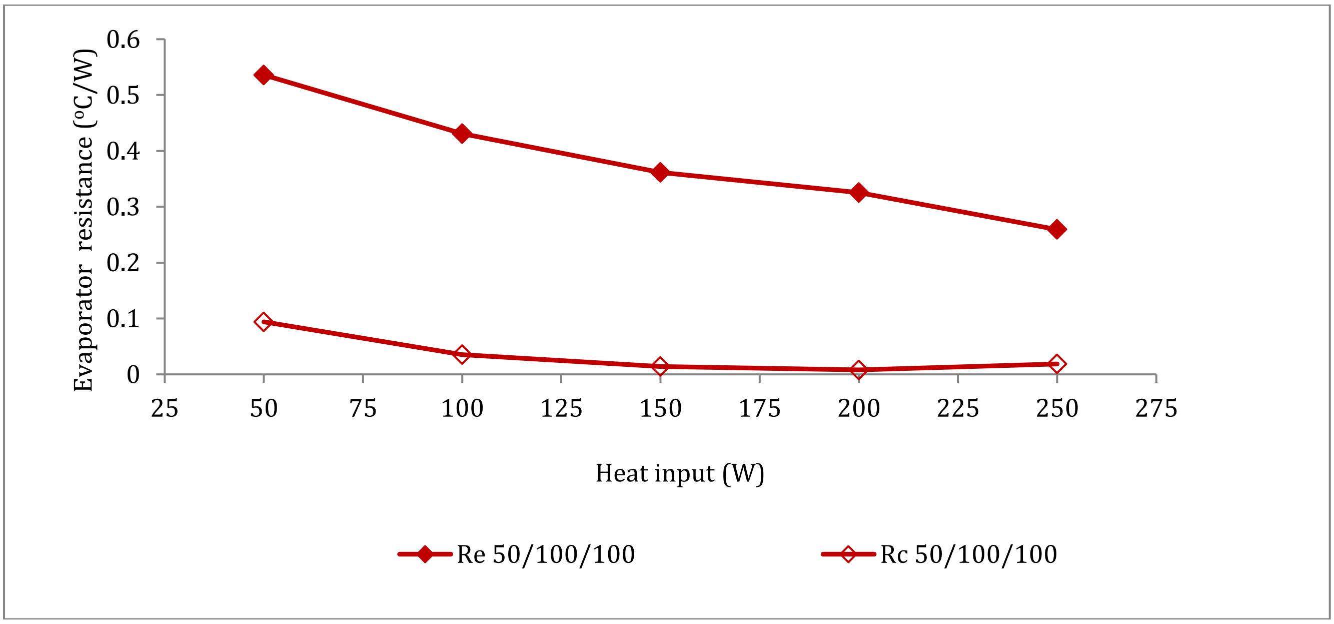

Fig. 5 Evaporator and condenser resistances of heat pipes charged with de-ionized water with 0.05m evaporator length.

Fig. 5 shows the thermal resistance across the evaporator and the condenser sections of heat pipes having same total length (0.25 m). The thermal resistance across the evaporator section (R_e) was calculated as $R_e = (T_{e, end}-T_{e, a})/Q_{in}$ and the thermal resistance across the condenser section (R_c) was calculated as $R_c = (T_{a, c}-T_{c, end})/Q_{in}$. It is interesting to note that thermal resistance across the condenser is lower than that across the evaporator. The thermal resistances across both condenser and evaporator decreases with increasing heat input but the effect of input power on the same diminishes with rising input power if there is no dry out. Heat pipe having equal lengths of condenser and evaporator shows lower thermal resistance across the evaporator than others. Though heat pipe having condenser shorter than evaporator and adiabatic region shows same thermal resistance of the evaporator at lower input powers, the dry out is achieved at the input power of 200W. Thermal resistance of the condenser is lower in the heat pipe having evaporator shorter than condenser and adiabatic region.

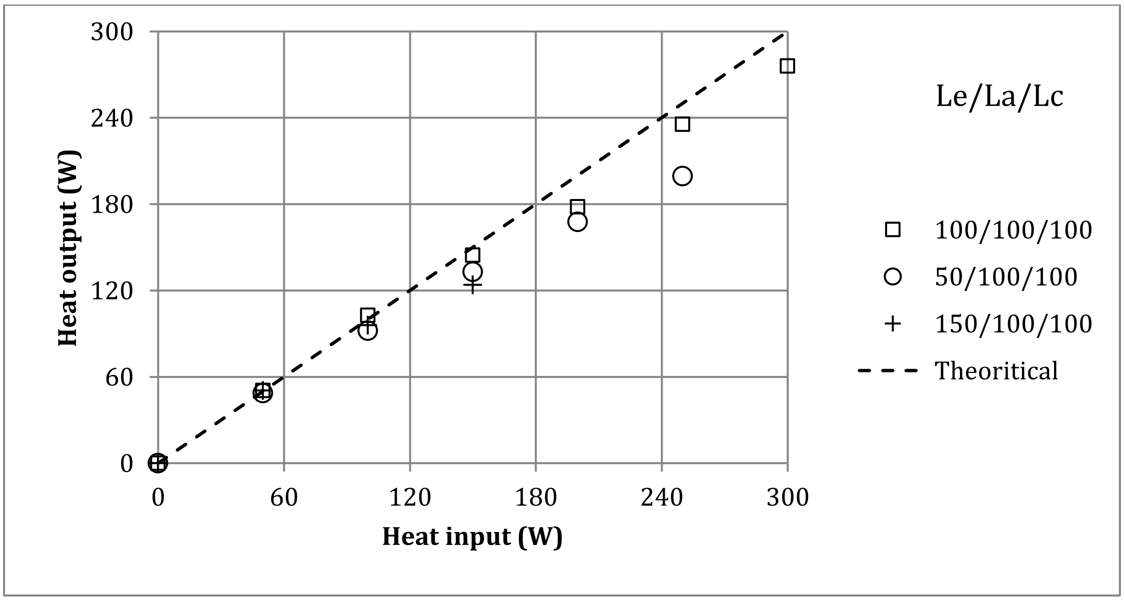

Fig. 6 Comparison between heat input at the evaporator and heat output at the condenser of heat pipes charged with de-ionic water

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Fig. 6 shows the heat output measured from the temperature difference in the cooling water at the condenser side. The heat input is measured at the evaporator side.

In the steady state condition, the measured heat output at the condenser side should be equal to the heat input at the evaporator side. A straight line is drawn at 20 % limit in the fig.6. It is seen that the variation between the expected and measured heat outputs is less than 20%. This variation is mainly due to the heat loss from the electric heater to atmosphere at the evaporator and additionally from the adiabatic section to atmosphere.

3.2 Thermal resistance of heat pipe charged with methanol

Fig. 7 shows the total thermal resistance of the heat pipe charged with methanol for various lengths of evaporator, condenser and adiabatic region. The thermal resistance and heat transport limit, respectively, decreases and increases with increasing evaporator length. It is interesting to note that the heat pipe having evaporator longer than condenser and adiabatic section shows the best performance in methanol whereas the same show worst performance in DI water.

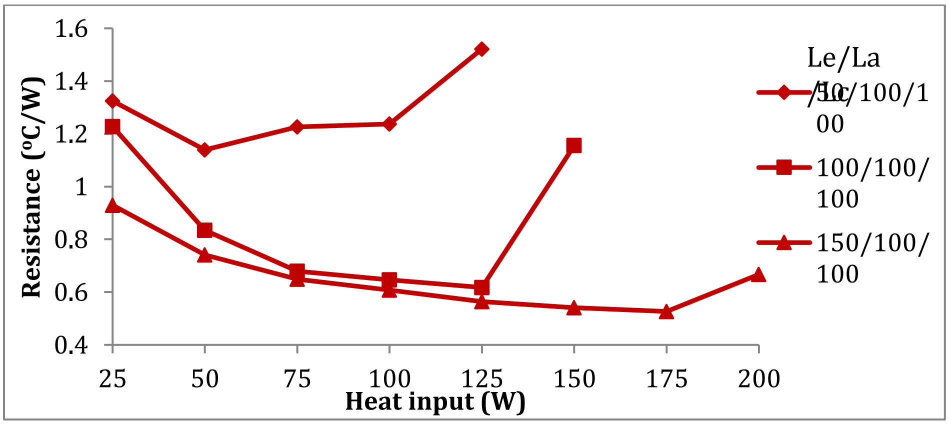

Fig.7 Total thermal resistance of heat pipes charged with methanol

Since methanol has lower latent heat of evaporation, the possible dry out due to the boiling limitation in the heat pipe filled with DI water at the evaporator length of 0.15 m is not occurred in the heat pipe filled with methanol and it may occur if the evaporator length is increased above 0.15m. Both shorter condenser and adiabatic section enhance the heat transport limit. The heat transport limit of the heat pipes filled with DI water is higher than that of heat pipes filled with methanol. The main reason for the same is water has higher latent heat of evaporation. Though thermal characteristics of the heat pipes filled with DI water and methanol are dissimilar, the difference between thermal resistances of the same is not significant.

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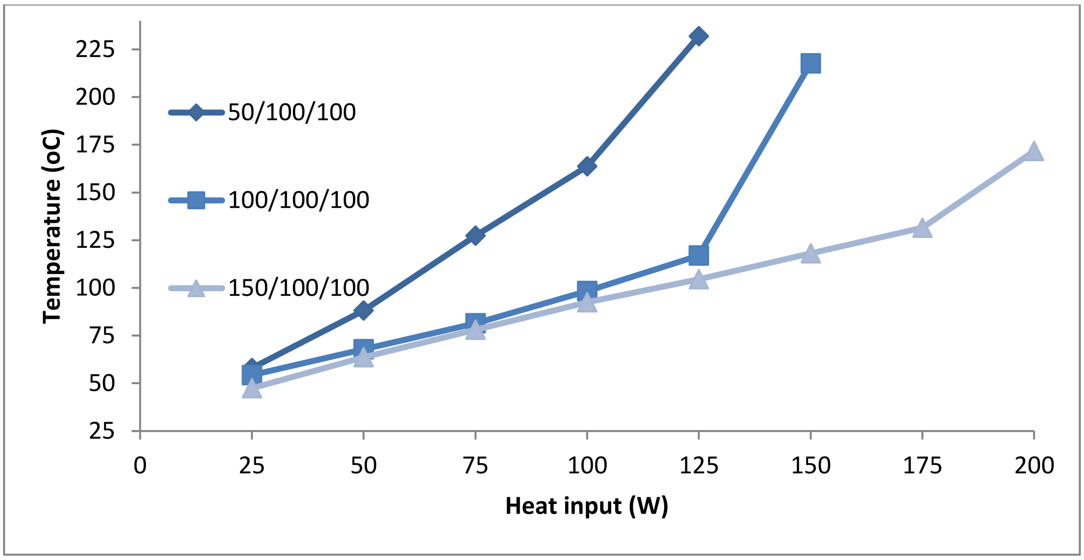

Fig. 8 Evaporator temperature of heat pipes charged with methanol

The effect of heat input on temperature of the evaporator is shown in Fig. 8. The temperature of the evaporator increases linearly with heat input. A sharp increase in temperature is due to the dry out. The variation in the temperature of the evaporator of various sized heat pipes increase with rising heat input. Almost similar trend is observed in the heat pipes filled with DI water. The difference in the temperature of the evaporator of the heat pipes filled with DI water and methanol is not noteworthy.

The thermal resistance of evaporator and condenser of heat pipes charged with methanol are shown in Fig. 9 (a) for the heat pipe having total length of 0.25 m and 0.35 m. Similar to the heat pipes charged with DI water, heat pipes charged with methanol show higher thermal resistance in the evaporator than in the condenser. In the heat pipe having total length of 0.25m, a shorter evaporator compared to condenser and adiabatic section produces highest evaporator and lowest condenser resistances similar to results of the heat pipe charged with DI water. Unlike to the results of the heat pipe charged DI water, heat pipe having total length of 0.35 m, a shorter evaporator compared to condenser and adiabatic section reduces evaporator resistance significantly whereas the same produces almost same condenser resistance. Further studies are needed to explain these effects.

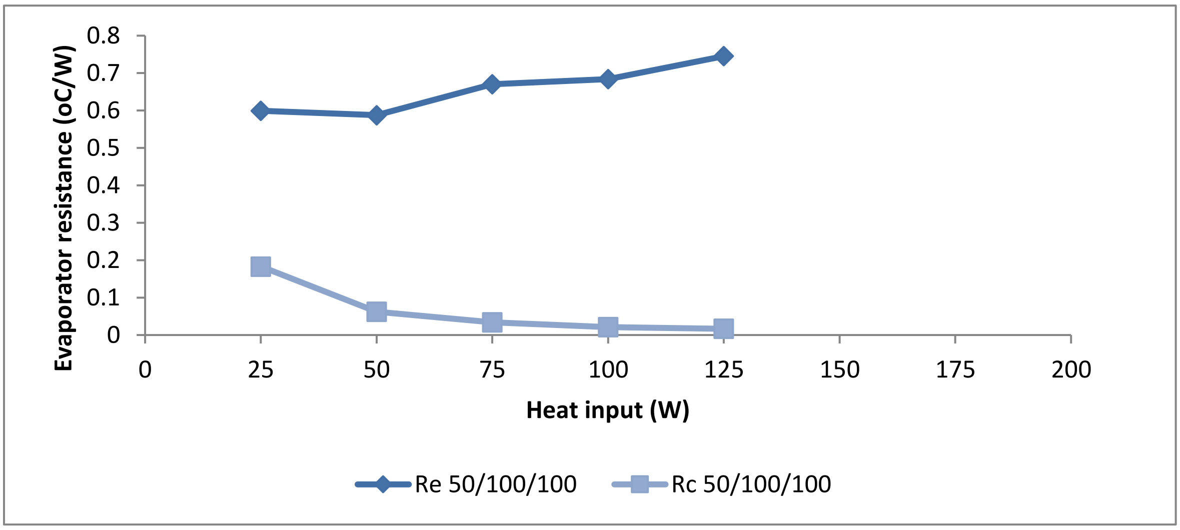

Fig. 9 Evaporator and condenser resistances of heat pipes charged with methanol of heat pipe with evaporator length 0.25m

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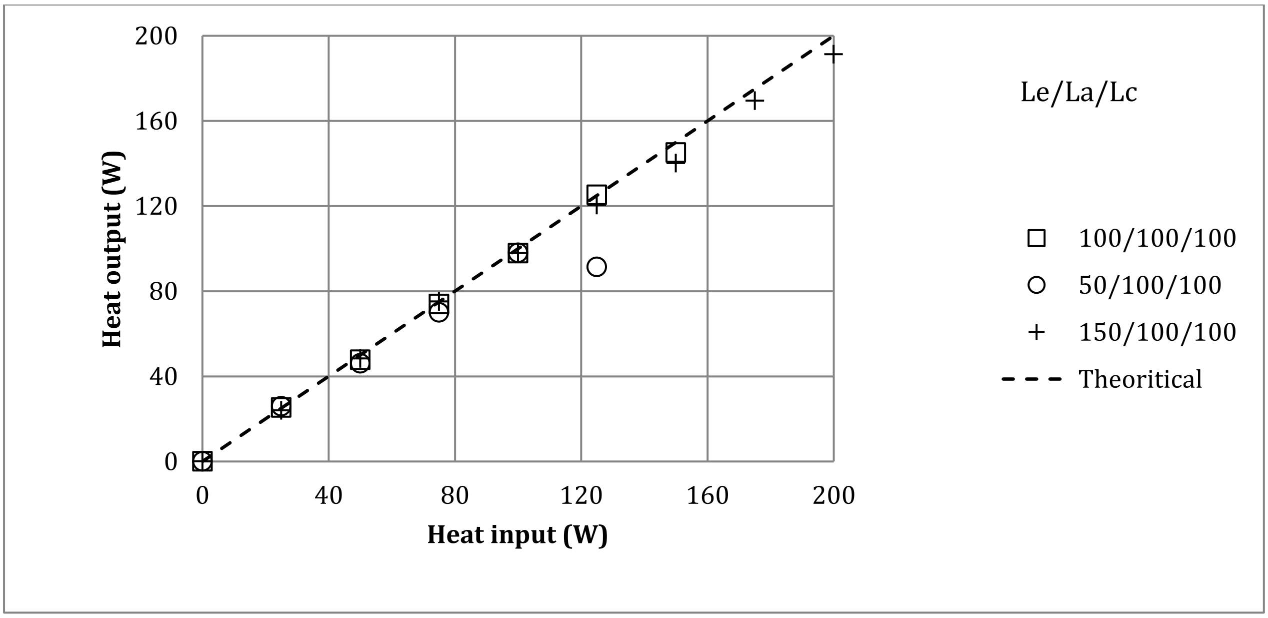

Fig. 10 Comparison between heat input at the evaporator and heat out put at the condenser of heat pipes charged with methanol

The variation between the expected and measured heat outputs in the heat pipes charged with methanol (Fig. 10) is also less than 20% as seen in Fig. 6.

Conclusion

The performance of the heat pipe is studied for various lengths of evaporator, condenser and adiabatic sections. The effects of length of evaporator, condenser and adiabatic sections on thermal resistances of heat pipe as well as thermal resistance of evaporator and condenser of the heat pipe are clarified. Also, the effects of heat input on thermal resistance and heat transport limit of the heat pipe as well as temperature of the evaporator are investigated. The following conclusions are arrived from this study.

1. The heat pipe having equal length of evaporator and condenser as well as longer condenser than evaporator shows better performance in the case of water is used as working fluid whereas the same is achieved in heat pipe having evaporator longer than condenser and adiabatic section when methanol is used as working fluid.
2. Thermal resistance of the heat pipes charged with both DI water and methanol decrease with increasing heat input if there is no dry out.
3. The variation in temperature of the evaporator among the heat pipes with various lengths of evaporator is stronger at higher heat input both in DI water and methanol.
4. Heat transport limit is higher in the heat pipes charged with water than in the same charged with methanol since latent heat of evaporation of methanol is lower than water.
5. Selection of suitable length of the evaporator is necessary to achieve lower thermal resistance and higher heat transport limit of the heat pipes.

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Nomenclature

R thermal resistance
Q heat input
T temperature
L length

Subscripts

a adiabatic
a,c interface between adiabatic to condenser section
c condenser
c,end condenser end
e evaporator
e,end evaporator end
e,a interface between evaporator and adiabatic section
in input
t transported
T total

Figure captions

Fig. 1 Experimental set up

Fig. 2 (a) Heat pipes with various evaporator lengths and (b) heat pipes with various condenser lengths

Fig. 3 Total thermal resistance of heat pipes charged with De-ionized water

Fig. 4 Evaporator temperature of heat pipe charged with DI-water

Fig. 5 Evaporator and condenser resistances of heat pipes charged with de-ionized water at different lengths of the heat pipe (a) 0.25 m and (b) 0.35 m

Fig. 6 Comparison between heat input at the evaporator and heat output at the condenser of heat pipes charged with de-ionic water

Fig .7 Total thermal resistances of heat pipes charged with methanol

Fig. 8 Evaporator temperature of heat pipes charged with methanol

Fig. 9 Evaporator and condenser resistances of heat pipes charged with methanol at different lengths of heat pipe (a) 0.25 m and (b) 0.35 m

Fig. 10 Comparison between heat input at the evaporator and heat output at the condenser of heat pipes charged with methanol