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Experimental Study of Effect the Load Applied and Length of Adiabatic Section on the Performance of the Wicked Horizontal Heat Pipe.

Amer I. Ismael And Majid H. Majeed
Middle Technical University, Technical Engineering Collage - Baghdad

Abstract: *An experimental study is carried out to evaluate the thermal performance of wicked horizontal heat pipe (WHHP).R134-a is used as working fluid with mass 123 gram. The WHHP was made of three parts condenser, evaporator and adiabatic section. Condenser and evaporator sections were made of copper with 100 mm length, 35 mm outer diameter,24 mm inner diameter and 5.5 mm thickness wall. Adiabatic section was made of UPVC with 100 mm, 43.5 mm outer diameter, 24 mm inner diameter and 9.75 mm wall thickness.The evaporator section of the heat pipe was surrounded by heater coil representing the heat source. The condenser was jacketed with plastic cylinder to accommodate the cooling water flow. The entire WHHP was insulated. WHHPconsists of a copper and UPVC lined with a six-layer stainless steel screen mesh wick with size 100.All the experiments are accomplished and the heat pipe is at the horizontal position (θ=0o). The flow rate of cooling water changed within the range (1, 1.5 and 2) L/M, and heat power applied with range (10, 20, 30 and 40) watt, while all other conditions remained constant. The results show that the thermal resistance decreases when increase the power applied to evaporator while the overall heat coefficient of a heat pipe increased, also the deference temperature between average of evaporator and condenser was increases and the power transferred too.*

Keywords: *Heat pipe, screen mesh wick, thermal performance, working fluids, R134-a.*

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I. Introduction

The heat pipe is a passive device of very high thermal conductance. The amount of heat that can be transported as a latent heat of vaporization is usually several times larger than that which can be transported as sensible heat in a conventional convective system Dunn and Chi [1, 2]. Majeed and Skheel. [3],An experimental study has been done on heat pipe 1000mm length and 25.4mm diameter. Evaporator section is formed by adopting a coil heater over 240mm from one side of heat pipe length. The condenser section is formed by adopting a water jacket over 300mm from the other end of heat pipe length. R134_a was used as working fluid. Two type of screen meshes used as wick structure with six layers. The supplied power over evaporator (20, 30, 40 and 60) W. it wasfound that the thermal conductivity has no effect on the WHHP performance. But size of screen meshinfluenced on the performance of WHHP and rises the temperature differences.

Kempers et al. [4].An experimental studyhas been performed to evaluate the effect of the number of mesh layers and mass of working fluid on the heat transfer performance of copper –water heat pipes with screen mesh wicks. It was found that this non-linearity in the thermal resistance is larger for wicks with fewer mesh layers. There is a small growth in thermal resistance of the WHHP when the thickness of the wick is enlarged, but this is significant smaller than that predicted by models based on conduction heat transfer across the wick.

The heat pipe operates on a closed two phase cycle. Figure (1) shows a schematic of a typical heat pipe set up. Inside the container there is a liquid under its own pressure that enters the pores of the capillary material, wetting all internal surfaces. When the heat is supplied to the evaporator, this equilibrium breaks down, as the working fluid evaporates, causing the liquid at that point to boil and enter a vapor state. The vapor, which then has a higher pressure in the evaporator than that in condenser, vapor moves inside the sealed container to a colder section where it condenses. Thus, the vapor gives up the latent heat of vaporization. In other words,evaporation and condensation result in transfer heat from the input to the output end of the heat pipe. A considerable experimental and theoretical works have been done on the application and design modification for improving heat pipe performance.

Shwin Chung et al. [5] in their work presented visualization of the evaporation/boiling process and thermal measurements of operating horizontal transparent heat pipes. Yahya [6] carried out an experimental study concerned with the development of an indirect type of solar cooker using a heat pipe for transporting energy from the focal spot. Based on the results of the heat pipe, a complete solar cooking system was developed and tested to find its effectiveness. The present work investigates experimentally heat transfer characteristics performance of HP. Tests for different orientations, different working fluid charges and a range of heat loads were carried out. One of the most important design parameters involved in different applications of twophase flow heat pipes, is the heat transfer coefficient. This parameter was calculated from the experimental results of this work and compared with well-known theoretical and empirical correlations of Rohsenow and Imura [7, 8,9] and Bejan [10].

Experimental rig was designed and constructed to study the important parameters such as the heat load, the working fluid charge and the angle of inclination. Calibrated thermocouples were inserted into the heat pipe outside surface, along and around it, to measure and to show the temperature distribution profile. Heptane was used as a working fluid. Results show that the maximum conductivity obtained was more than one thousand times that of stainless steel solid bar which is the material of the heat pipe container. Also, results of the experimental work show a good agreement with that obtained from theoretical and empirical correlations derived by other researchers especially when the power input is lower than (1000)W where after a dry out condition can be clearly seen.Ahmad and Rajab [11].

An experimental test rig was designed and manufactured to investigate the performance of a heat pipe(HP). The heat pipe consists of a stainless steel pipe lined with athree-layer stainless steel mesh wick. The evaporator section of the heat pipe was surrounded by three heaters representing the heat source. The condenser was jacketed with galvanized cylinder to accommodate the cooling water flow. The entire HP was insulated.Different affecting parameters were investigated experimentally in this study including thepower input the filling charge of the working fluid(water) represented by a volumetric ratiowith respect to evaporator volume and the inclination angle with a horizontal line. All testswere carried out at a pressure around the atmospheric pressure during steady stateconditions. The experimental results showed that the conductivity was about (2060) timesthat of the solid piece of the stainless steel (the material of the HP). Ahmad and Rajab [12].

The effects of different refrigerants on heat transfer performance of pulsating heat pipe (PHP) are investigated experimentally. The working temperature of pulsating heat pipe is kept in the range of 20 C°–50 C° . The startup time of the pulsating heat pipe with refrigerants can be shorter than 4 min, when heating power is in the range of 10W–100W. The startup time decreases with heating power. Thermal resistances of PHP with filling ratio 20.55% were obviously larger than those with other filling ratios. Thermal resistance of the PHP with R134a is much smaller than that with R404A and R600a. It indicates that the heat transfer ability of R134a is better. In addition, a correlation to predict thermal resistance of PHP with refrigerants was suggested. Xingyu and Li [13]

Several heat pipes were designed and manufactured to study the effect of the working fluids, container materials, and the wick structures on the heat transfer mechanism of the heat pipes. Also, the effect of the number of wick layers on the effective thermal conductivity and the heat transfer characteristics of the heat pipes have been investigated. It was found that the flow behavior of the working fluid depends on the wicking structures and the number of wick layers. The heat transfer characteristics and the effective thermal conductivity are related directly to the flow behavior. Increasing the number of wick layers (up to 16 layers) increases the heat flux with smaller temperature differences. Abo El-Nasr and El-Haggar [14].

A. Faghri,[15].they was investigate the use of heat pipes for thermal management due to rising heat flux requests and thermal limits in many industrial functions. The operation of heat pipes is illustrated both by the effective thermal resistance and the largest heat transport capacity. The largest heat transport in mean temperature applications is limited by the capillary force that can be produced by the wick structure.

Zheng, et, al. [16]. Experiments were carry out to calculate the condensation heat transfer on models having several surface treatments at multi temperatures, a typical condensation temperature for power plant air-cooled condensers. The coating types calculated involved sintered powder porous surfaces. A bare uncoated surface was worked as a standard. The experimental results showed porous coated surface presented a surprisingly high heat transfer coefficient for film wise condensation mode.

II. Experimental Setup And Methods

2.1 Experimental Model and Setup

The heat pipe for the test was manufactured using a copper tube with a 30 mm outer diameter and 3.1 mm thickness for Evaporator and Condenser section and using a Polyvinyl chloride (PVC) tube with a 33 mm outer diameter and 4.6 mm thickness for adiabatic section. As a capillary wick structure, The total length of the heat pipe was 300 mm, in which the evaporator, adiabatic section, and condenser was 100 mm for each one . The dimensions of the heat pipe were as shown in Fig. 1.

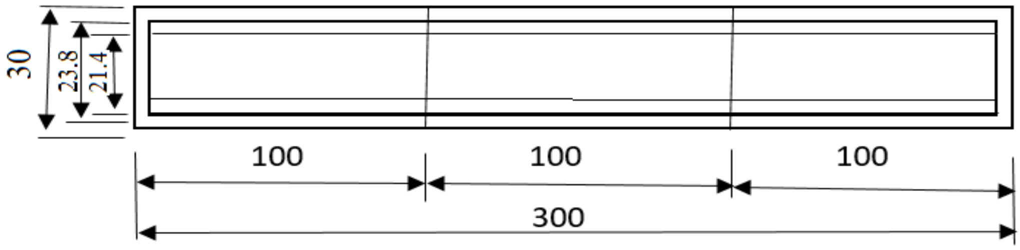


Fig. 1. Dimensions of the considered heat pipe (in mm)

Wick structure size 100- screen mesh stainless steel with six layers were inserted and pressed onto the inner wall surface using a coiled spring. R134-a was used as a working fluid. Its charge ratio,Φ, was 600% based on the wick space volume of the evaporator section. One meter of an electric resistance heater coil with maximum power 1000 Watt, voltage (220-230) V, wrapped helically around outer diameter of evaporator section, as shown in Fig. 2 . The surrounding were insulated by multi layers of Mineral wool with 110 mm outer diameter.

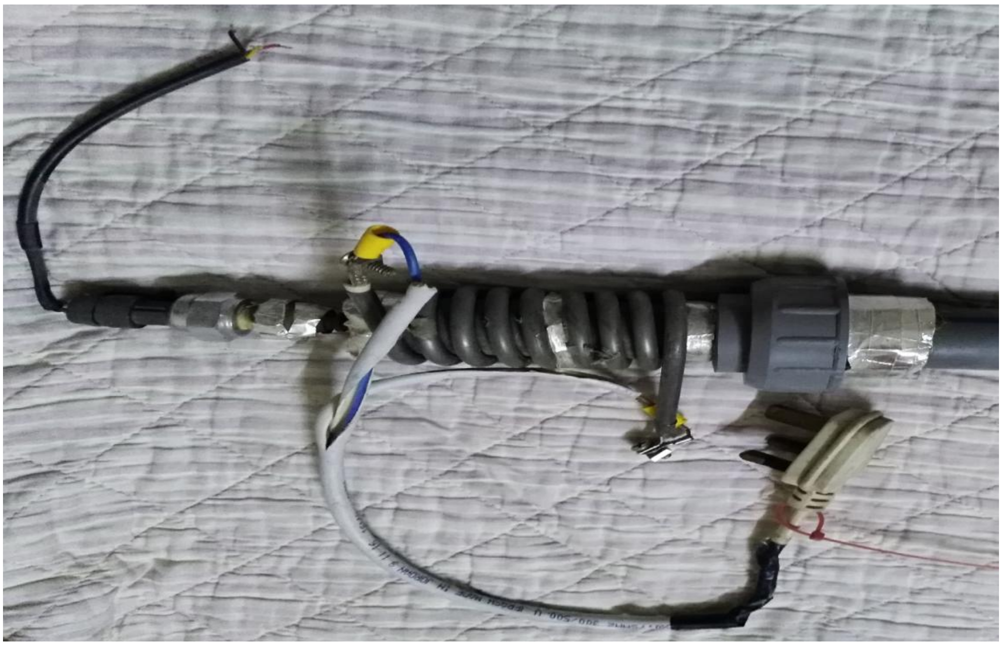


Fig.2. Electric resistance heater coil wrapped helically around outer diameter of evaporator section

To measure the temperatures at 9 locations along the heat pipe, LM35 – temperature sensors were attached, as shown in Fig. 3. The temperature sensors were calibrated for a measurement accuracy of 0.01 C°. as in its data sheet.

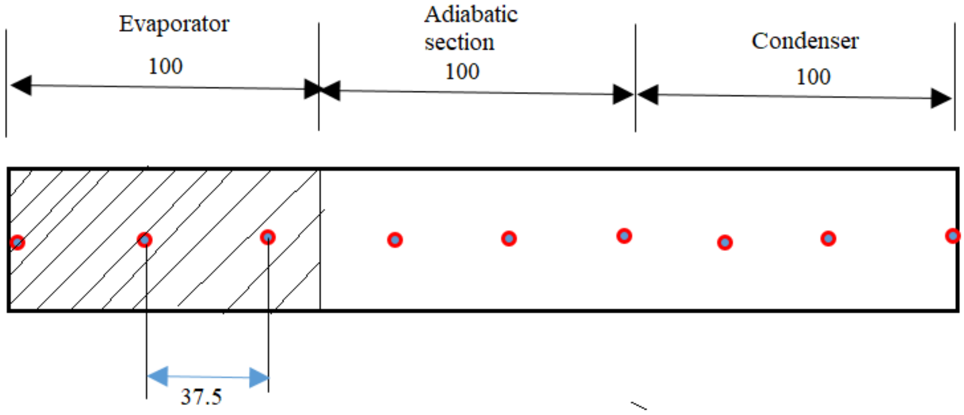
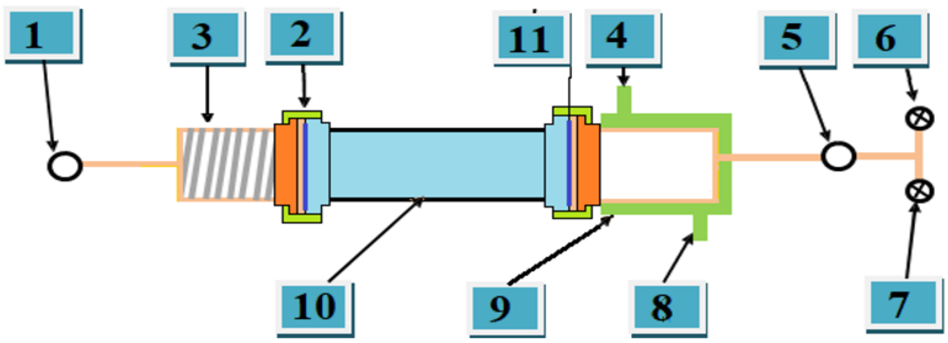


Fig. 3. Nine sensors along the outer surface of heat pipe

The performance of the heat pipe with uniform heat flux conditions was tested to identify the reference characteristics of the heat pipes. The Voltage Variac was regulated the input power to heater for control the power applied on a heat pipe by range (10, 20, 30, 40 watt). Fig. 4 is a schematic of the experimental setup. A cooling jacket was coupled with the condenser section through which coolant (water) from an isothermal bath passed.



1-	*Pressure Gauge*	2-	*Flange*	3-	*Electrical heater*
4-	*Water outlet*	5-	*Pressure Gauge*	6-	*Charging valve*
7-	*Vacuum valve*	8-	*Water inlet*	9-	*water jacket*
10-	*Heat pipe shell*	11-	*O ring*		

Fig. 4 is a schematic of the experimental setup for Heat pipe

The flow rate of the coolant was measured by Arduino flow meter sensor of which the accuracy was ±10% of the full scale. The flow rate of the coolant was controlled by valve with range (1, 1.5, 2 liters per minute) during the experiment. The temperatures at the inlet and outlet of the cooling jacket were measured using LM53 Arduino temperature sensor of which accuracy ±1⁄4 C° to estimate the effective heat transport to the condenser section of the heat pipe.Temperatures distribution was measured by put 9 sensors along the outer surface of heat pipe as shown in Fig. 4. Temperature readings from the sensors were viewed and managed by computer.The (WHHP) was in capillary force -assisted mode with a horizontal to extend the thermal input range to several tens of watts.

2.2 Performance Parameters and Experimental Methods

As primary performance parameters, the thermal resistance,R_{th}, and the effective thermal conductance, k_{eff} were computed based on the gotten data to denote the thermal characteristics of the (WHHP) as a function of thermal load and effective thermal load. They are expressed as follows.

$$R_{th} = \frac{T_{evp} - T_{cond}}{Q} \quad (1)$$

$$k_{eff} = \frac{Q.L_{tr}}{A_c(T_{evp} - T_{cond})} \quad (2)$$

Where T_{evp} *and* T_{cond} represent the average temperatures of the evaporator and condenser surface, respectively, L_{tr} is the effective heat transport length between the centers of the evaporator and condenser, A_c is the cross sectional area of the (WHHP), and Q denotes transported thermal load.
Q_e Measured at the condenser by the following equation.

$$Q_e = \dot{m}_{coolant} .C.(T_{out} - T_{in}) \quad (3)$$

Where $\dot{m}_{coolant}$ are the mass flow rate and the C specific heat of the coolant flowing through the cooling jacket, respectively, and T_{out} *and* T_{in} are the outlet and inlet temperatures of the coolant, respectively. Temperature deference dT represented to deference between the temperature average of condenser andtemperature average of evaporator where it calculate by the equation 4.

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dT= $T_{evp} - T_{cond}$ (4)

Overall Heat Transfer Coefficient (Uo) calculated from the equation 5.

$$U_o = \frac{\frac{Q_{eva} - Q_{cond}}{2}}{A_{ave}\,(T_{eva} - T_{cond})} \quad (5)$$

Where $A_{ave}\,(m^2)$ is surface area of evaporator section .

III. Results And Discussion

1. **3.1 General discussion**

The amount of heat load applied on evaporator is an important parameter that greatly influences on the performance of (WHHP). This study presents four range heat power values applied (10, 20, 30 and 40) watt to estimate that effect on performanceof (WHHP). Thermal resistance calculated by Eq. (1). Compared to the thermal load inputs to the evaporator Q_{in} and flow rate of coolant water liters per minute, the R_{th} values were decrement 33 % for the higher Q_{in} (10-20 W) with 1L/M but decrement was 37% for the Q_{in} (10-20W) with 1.5L/M, while the decrement was 50% for the Q_{in} (10-20W) with 2 L/M. The difference between Q_{in} and Q_e can be regarded as the heat loss from the experimental operation between the evaporator and condenser of the heat pipe when a steady state was realized. As a result, the readings might have been taken before a complete steady state was achieved. In this respect, the analyses for the heat inputs ($Q_{in} \leq 40$ W) can be regarded as those for quasi-steady states, which maintained nearly equilibrium condition for at least half an hour within 0.5 K bounds in all temperature locations in theheat pipe. In addition, and condenser, the results were analyzed in terms of Q_e, which is denoted as 'thermal load' hereinafter.

3.2 Effectpower input on the thermal resistance

Figs.5.represent the thermal resistance and the effective thermal conductance of the heat pipe with 600 % fluid charge, against the thermal load, Q_e.

For these experiments, the range of heat input Q_{in} in the evaporator was from 10W to 40W by 10 W increments. The thermal resistance for a uniform heat flux was plotted as a reference case (denoted as uniform Q in thegraphs). exhibited polynomial with third order decrease in thermal load, as expected, with a minimum of 1 K/W at Q = 40 Wand flow rate 1 L/M for coolant water . While it was maximum at 2.7 k/W with Q =10 W with flow rate 2 L/M for coolant water.

Thermal loads and flow rates of coolant water are adjusted as described in follow. Where the thermal resistance decrease in percentage as follow :-

1- the decrement of thermal resistance was 33% With 1 liter per minute of coolant water with rate of thermal load (10 – 20) watt, but it was 14% for (20-30) watt, while 9% for (30-40) watt
2- the decrement of thermal resistance was 37% With 1.5 liter per minute of coolant water with rate of thermal load (10 – 20) watt, but it was 20% for (20-30) watt, while 23% for (30-40) watt
3- The decrement of thermal resistance was 50% With 2 liter per minute of coolant water with rate of thermal load (10 – 20) watt, but it was 23% for (20-30) watt, while 2% for (30-40) watt.

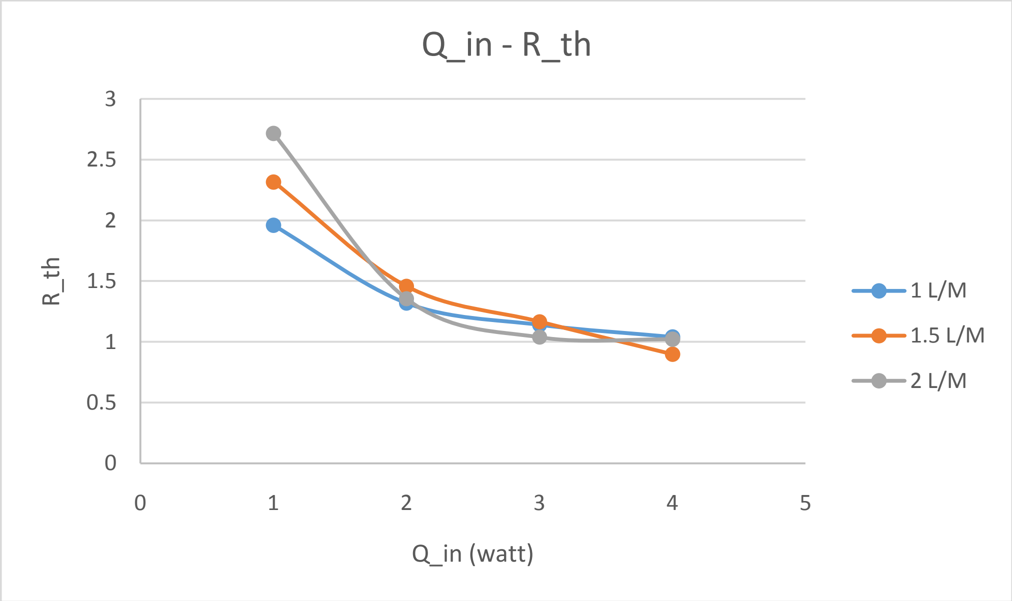

Fig.5. Thermal resistance profile at multi thermal loads

3.3 Effect power input on the temperature deference between Evaporator and Condenser (dT)

Fig.6. represent the input heat power and the temperature deference between evaporator and condenser of the heat pipe (dT).

For these experiments, the range of heat input Q_{in} in the evaporator was from 10 W to 40 W by 10 W increments. The temperature deference for a uniform heat flux was plotted as a reference case (denoted as uniform *Q* in the graphs). exhibited linear relation in thermal load, as expected, with a minimum of 17.24 C at Q = 10 W and flow rate 1 L/M for coolant water, While it was maximum at 39.8 C° with Q =40 W. the temperature deference between evaporator and condenser was increments with respect to heat power applied in percentage as follow :-

1- The temperature deference between evaporator and condenser was increments with linear relation $y = 7.561x + 9.575$ with respect to heat power applied with percent 131% at range input heat power from 10 watt to 40 watt for 1 L/M of coolant water.
2- the temperature deference between evaporator and condenser was increments with polynomial relation $y = -0.7425x^2 + 8.5535x + 14.327$ by maximum value 36.5 C° and minimum value 22.3 C° with respect to heat power applied an increase68% at range input heat power from 10 watt to 40 watt for 1.5 L/M of coolant water.
3- the temperature deference between evaporator and condenser was increments with polynomial relation $y = 1.275x^2 - 0.765x + 20.975$ by maximum value 38.5 C° and minimum value 21.3 C° with respect to heat power applied an increase 81% at range input heat power from 10 watt to 40 watt for 2 L/M of coolant water.

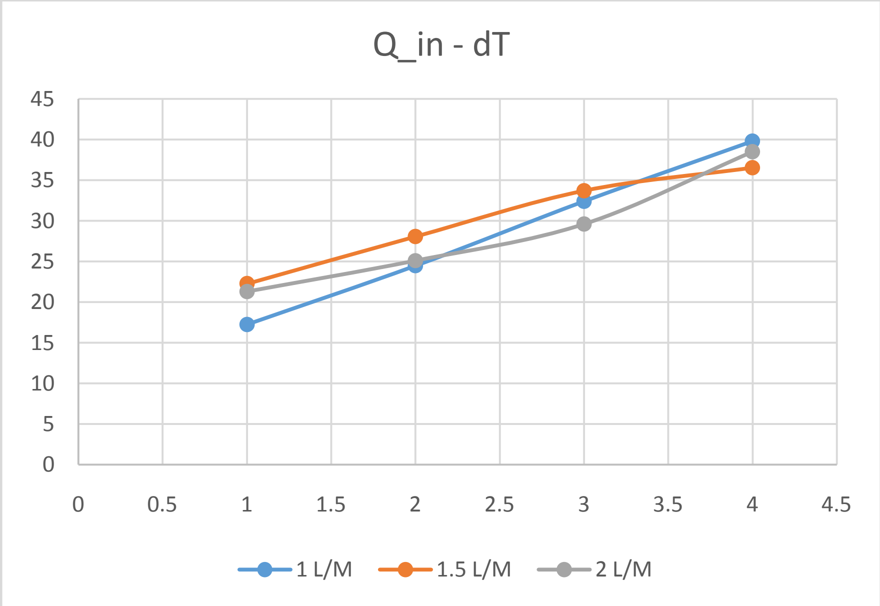

Fig.6. Temperature deference profile at multi thermal loads

3.3 Effect power input on the overall heat transfer coefficient U_o

Figs.7. represent the input heat power and the overall heat transfer coefficient U_o. For these experiments, the range of heat input Q_{in} in the evaporator was from 10 W to 40 W by 10 W increments. The overall heat transfer coefficient for a uniform heat flux was plotted as a reference case (denoted as uniform *Q* in the graphs). exhibited linear relation in thermal load, as expected, with a minimum of 17.24 C° at Q = 10 W and flow rate 1 L/M for coolant water, While it was maximum at 39.8 C° with Q =40 W. the temperature deference between evaporator and condenser was increments with respect to heat power applied in percentage as follow :-

1- The overall heat transfer coefficient was increments with logarithmical relation $y = 32.244\ln(x) + 51.155$ with respect to heat power applied with percent 131% at range input heat power from 10 watt to 40 watt for 1 L/M of coolant water.
2- The overall heat transfer coefficient was increments with polynomial third order relation $y = 2.7775x^3 - 20.777x^2 + 68.063x - 7.4359$ by maximum value 110.1 C° and minimum value 42.6 C with respect to heat power applied an increase 158% at range input heat power from 10 watt to 40 watt for 1.5 L/M of coolant water.
3- The overall heat transfer coefficient was increments with polynomial third order relation $y = -1.0539x^3 - 0.8363x^2 + 46.451x - 8.2074$ by maximum value 97 C° and minimum value 36 C° with respect to heat power applied an increase 169% at range input heat power from 10 watt to 40 watt for 2 L/M of coolant water.

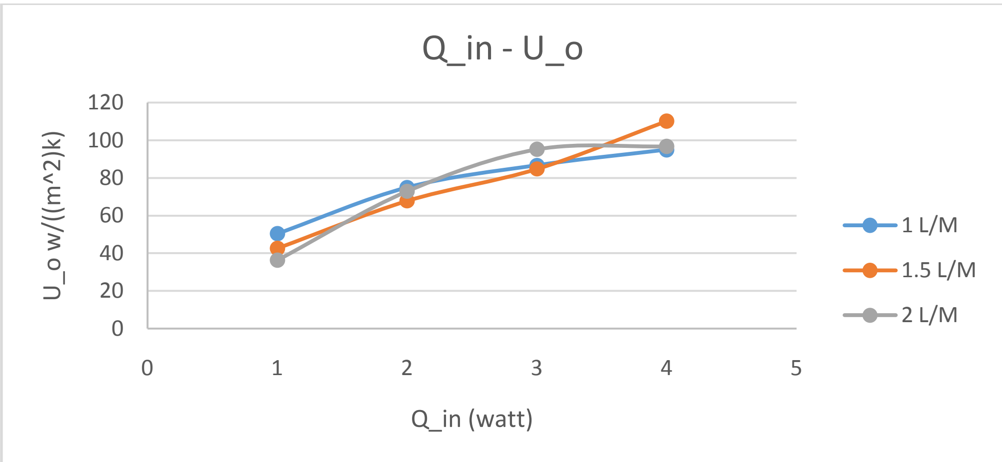

Fig.7. Overall heat transfer coefficient profile at multi thermal load

IV. Conclusions

Quantitative as well as qualitative differences in the performance of a cylindrical heat pipe with uniformheat source was acquired experimentally in terms of thermal resistance, overall heat transfer coefficient, and operating temperature. The performance differed with the flow rate of the coolant water fluid charge and the configuration of the uniform thermal loads. For a 1L/M flow rate of coolant water, noticeable thermal resistance degradation for the uniform heat source was observed compared to a 1.5 L/M flow rate of coolant water case. It was presumed that the rate of coolant water was at an optimum flow rate amount for the operation temperature in 1.5 L/M case. The behavior of the heat pipe with a 1L/M exhibited a little difference from that with a 2 L/M. It was presumed that a 2 L/M was a little more than an optimum flow rate coolant water for the heat pipe dimensions. Performance increases was observed for uniform heat source for a thermal load lower than that of the minimum thermal resistance. The best performance was observed for the a 1.5 L/M flow rate of coolant water having increasing magnitude toward the evaporator exit with a 40 watt heat load, and the effective thermal conductance exhibited approximately 12.9 % higher than a 1 L/M flow rate of coolant water with a 40 watt heat load. The results herein can be employed to estimate the performance deviation of a heat pipe due to thermal load distribution and flow rate coolant water. In the future, further studies are required to provide experimental data from extended cases as well as the physical reasoning for the unexpected deviations in performance behavior. Extended work is also desired for better understanding of heat pipe performance depending on the operating temperature, by varying ratio of evaporator and condenser length or adiabatic section length while maintaining a specified operating temperature. Furthermore, a theoretical model is desired that can reasonably predict the thermal performance of a heat pipe with multiple shape of cross sectional.

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