Study of Thermal Performance of Heat Pipe with Different Coated Wick Layers.

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دراسة الأداء الحرارى لأنبوب حراري ذوفتيل مطلى بعدد مختلف من الطبقات

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الملخص:

تم فى هذا البحث دراسة تأثير بعض المتغيرات على الأداء الحرارى لأنبوب حرارى مثل كمية الحرارة المضافة ونسبة ملءالمبخر بالمياه النقية وعدد طبقات الطلاء لفتيل مستخدم داخل الأنبوب الحرارى,حيث ان طبقات الطلاء المستخدمة عبارةعن شريحة رفيعة من البوليمرات يتم طلاؤها على الفتيل على هيئة طبقات مختلفة من الدرجة الاولى والثانية والثالثة يتراوح سمكها بين 20 الى 100 ميكرومتر ويحتوى على مكونات كيميائية مختلفة يستخدم لتغيير خصائص سطح الفتيل.

وقد أوضحت النتائج أن المقاومة الحرارية الكلية للأنبوب الحرارىتقل مع زيادة نسبة الملء حتى تصل لنسبة تتراوح من 0.5 إلى 0.6 ثم تزيد بعد ذلك مع زيادة نسبة ملء المبخر بالمياه النقية وكذلك تقل المقاومة الحرارية الكلية مع زيادة عدد طبقات الدهان على سطح الفتيل كما أن المقاومة الحرارية الكلية تقل مع زيادة كمية الحرارة المضافة .

Abstract

An experimental investigation is carried out to study the effect of operating parameters such as filling ratio, input heat, number of wick layers and thin coated wick on the thermal performance of a heat pipe .A film coating is a thin polymer- based coat applied to wick .the thickness of such a coating is usually between 20-100 micro meter .film coating formulations usually contain the following components: polymer, plasticizer, colourant, opacifier, and solvent. Coated material is a covering that is applied to the surface of an object ,the purpose of applying the coating may be decorative ,functional, or both ,The coating itself may be an allover coating, completely covering the substance, Functional coatings may be applied to change the surface properties of the wick, demineralized water used as working fluid. The total thermal resistance decreases with increasing filling ratio until filling ratio around between 0.5 and 0.6 then increases with increasing filling ratio, The total thermal resistance decreasing with increasing the number of coated layers of wick ,The total thermal resistance decreases with increasing heat input .

Keywords:

Heat pipe, Coated wick, UN coated wick, Thermal resistance, Thermal performance.

1. Introduction

Heat pipes were developed especially for space applications during the early 60´ by the NASA. One main problem in space applications was to transport the temperature from the inside to the outside, because the heat conduction in a vacuum is very limited. Hence there was a necessity to develop a fast and effective way to transport heat, without having the effect of gravity force. The idea behind is to create a flow field which transports heat energy from one spot to another by means of convection, because convective heat transfer is much faster than heat transfer due to conduction .Nowadays heat pipes are used in several applications where one has limited space and the necessity of a high heat flux. It is still in use in space applications, but it is also used in heat transfer systems, cooling of computers, cell phones and cooling of solar collectors.Especially for micro applications there are micro heat pipes developed as for cooling the kernel of a cell phone down Due to limited space in personal computers and the growing computational power it was necessary to find a new way to cool the processors down. By means of a heat pipe it is possible to connect the processor

cooling unit to a bigger cooling unit fixed at the outside to cart of the energy. A heat pipe is a device that efficiently transports thermal energy from its one point to the other. It utilizes the latent heat of the vaporized working fluid instead of the sensible heat. As a result, the effective thermal conductivity may be several orders of magnitudes higher than that of the good solid conductors a heat pipe consists of a sealed container, a wick structure, a small amount of working fluid that is just sufficient to saturate the wick and it is in equilibrium with its own vapor. The operating pressure inside the heat pipe is the vapor pressure of its working fluid. The length of the heat pipe can be divided into three parts viz. evaporator section, adiabatic section and condenser section. In a standard heat pipe, the inside of the container is lined with a wicking material. Space for the vapor travel is provided inside the container. studied that Novel wick with two distinguished characteristic pore sizes were successfully developed. **Chien-chin yeh., [1],** said that because the evaporative heat transfer of a wick structure in aloop heat pipe is exceedingly sensitive to the internal volume fractions of liquid and vapor phases, the purpose of this study was to investigate the evaporative heat transfer of various bi porous wick parameters by controlling the particle size of pore former, the pore former content, and the sintering temperature. A statistical experiment was carried out to analyze the evaporative heat transfer of the biporous wicks and to understand the effects of the barameters more effectively. The statistical analysis indicated a clear and strong relationship between the effect of the pore former content and the evaporative heat transfer of a biporous wick. This is because the pore former content had a great influence on the probability of large interconnecting pores and an extended surface area for liquid film evaporation in a biporous wick. Experimental results also showed that, at the sink temperature of 10 ºC and the allowable evaporator

temperature of 85 ºC, the evaporative heat transfer coefficient of the bi porous wick, which reached a maximum value of 64,000 W/m2 K, was approximately six times higher than that of the mono porous wick. **Ji Li G.P, [2],** studied that A practical quasi three-dimensional numerical model is developed to investigate the heat and mass transfer in a square flat evaporator of a loop heat pipe with a fully saturated wicking structure. The conjugate heat transfer problem is coupled with a detailed mass transfer in the wick structure, and incorporated with the phase change occurring at the liquid–vapor interface. The three-dimensional governing equations for the heat and mass transfer (continuity, Darcy and energy) are developed, with specific attention given to the wick region. By comparing the results of the numerical simulations and the experimental tests, the local heat transfer mechanisms are revealed, through the obtained temperature distribution and the further derived evaporation rates along the liquid–vapor interface. The results indicate that the model developed herein can provide an insight in understanding the thermal characteristics heat pipes during steady-state operation, especially at low heat loads. **Fang-Chou Lin., [3],** a mathematical model of evaporative heat transfer in a loop heat pipe was developed and compared with experiments. The steady-state thermal performance was predicted for different sintered nickel wicks, including mono porous and bi disperse structures. The effect of wick pore size distribution on heat transfer was taken into consideration. The wick in the evaporator was assumed to possess three regions during vaporization from an applied heat load: a vapor blanket, a two-phase region, and a saturated liquid region. The evaporator wall temperature and the total thermal resistance at different heat loads were predicted using ammonia as the working fluid. The predictions showed distinct heat transfer characteristics and higher performance for the bi disperse

wick in contrast with mono porous wick. A bi disperse wick was able to decrease the thickness of the vapor blanket region, which presents a thermal resistance and causes lower heat transfer capacity of the evaporator. Additionally, a validation test presented good agreement with the experiments. **Brusly solomonA.,[4],** An experimental investigation is carried out to study the thermal performance of a heat pipe operated with nanoparticle coated wick. Screen type wicks (100 mesh/inch) with and without deposition of nanoparticles are used in this study. Copper particles with average particle size of 80e90 nm are coated over the surface of the screen mesh. A simple immersion technique followed by drying is used to coat the wick with nanoparticles. The performance of the heat pipe is investigated at three different heat inputs.The thermal resistance and heat transfer coefficient in the evaporator of the heat pipe operated with coated wick are lower and higher respectively than that of conventional one whereas the same are opposite in the condenser. The total resistance of heat pipe operated with coated wick is lower than that of conventional one and it decreases with increasing heat input. At the evaporator section, 40 percent thermal resistance reduction and 40 percent heat transfer coefficient enhancement are observed. It is also found that the decrement in total resistance is 19%, 15%, and 14% at 100,150 and 200 W respectively .The aim of the present work is to study the effect of operating parameters such as filling ratio (FR), Input heat (Qin) ,number of Wick layer and coated on the total thermal resistance of the loop heat pipe.**Li Zhang.,[5],** The operating characteristics of a copper–water loop heat pipe (LHP), including start-up property, heat-transfer capacity, and heat resistance, under four different charge ratios (52.2–78.3%) were experimentally investigated. The LHP had a cylindrical evaporator (22 mm diameter and 80 mm long) with a sintered copper

wick, and the clearance between the evaporator envelope and the wick surface was eliminated to attain a low heat contact resistance. The porosity of the wick was 44.54%, and the effective pore radius was 19.11 m.

The experiment showed that higher inventory is an effective method in keeping thewick saturated within the low heat-load range and in supplying adequate flow of the subcooled liquid from the condenser to cool the water in the compensation chamber within the medium heat-load range. Thus, fast start-up of 260 s (heat load: 30 W; heat flux: 8.5 _ 103 W/m2), maximum heat-transfer capacity of 500W (under allowable evaporator wall temperature of 85 ºC), and minimum LHP thermal resistance of 0.075 C/W (under 500W heat load) were realized by the LHP charged with an optimal 69.6% volume ratio.

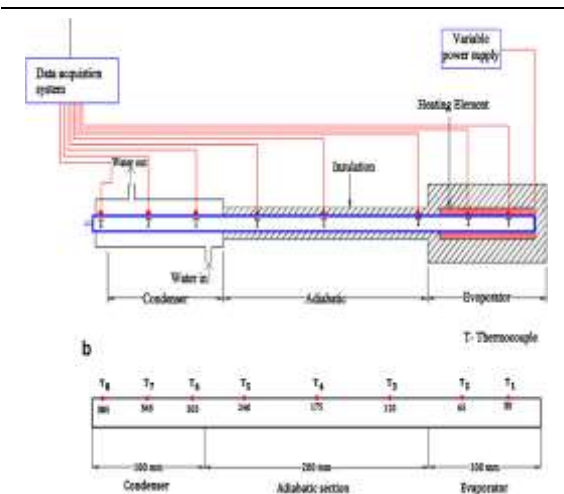## 2. Experimental setup and operation

Measurement scheme of heat pipe and Arrangement of thermocouples on the heat pipe are shown in Fig(1).

The heat pipes tested here were constructed using copper tubing with an outer diameter of 25.4 mm and an inner diameter of 23.4 mm as shown in Fig (2) ,the heat pipes were 360 mm long, and the two ends were closed with 3.2 mm thick copper end caps that were silver soldered in place.

Type of structure wick used un coated,1st coated,2nd coated, and 3rd coated is shown in Fig (3). The effect of fluid loading on the performance of the heat pipes was tested using heat pipe that each had a three-layer screen mesh wicks. The range of fluid in the heat pipes was determined by initially estimating the volume of fluid required to saturate the wick. A porosity model typically used for wire screen meshes was used to calculate the porosity of the wick. This was then multiplied by the bulk volume of the wick structure. Tests were performed here for

heat pipe that included filling ratio 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9, 1. The amount of water required to saturate the wick. In this case, each heat pipe was charged with the estimated amount of water required to fully saturate the wick. The heat pipes tested here were characterized using the experimental facility . Heat was applied to the heat pipe in a 150 mm long section using a 300 W electrical band heater that was rapped around a 25.4 mm diameter brass annular block. Thermal paste was used to ensure a good thermal contact between the heat pipe and the block. The condenser section of the heat pipe was cooled using a 150 mm long acrylic water jacket with an inner diameter of 23.4 mm. Chilled water supplied from a dedicated chiller entered and exited the water jacket through 12.5 mm diameter ports positioned 50 mm from the ends of the condenser section. The inlet and outlet temperatures of the cooling water to the condenser section were measured using thermometers with an uncertainty of 0.01 ºC, and the flow rate was measured using flow meter with an uncertainty of 0.19% of flow rate. The 150 mm section between the evaporator and condenser sections was insulated using several layers of ceramic insulation to create an adiabatic section. The surfaces of the heat pipes tested here were instrumented with eight T-type thermocouples mounted along the wall to measure the axial surface temperature distribution in the evaporator and condenser sections. Thus, it was thought the effect on the local temperature distribution inside the pipe was negligible. The lower thermal conductivity around the thermocouple junction would result in a slightly longer time response in the measurement, but this was not of interest in the present steady-state results. The uncertainty of the thermocouples was 0.5 ºC. The signals from the thermocouples, flow meter were logged continuously using a data acquisition system. The heat pipes were tested by setting the chiller to hold

the average condenser wall temperature at 25 ºC.

Fig(1)Measurement scheme of heat pipe and Arrangement of thermocouples on the heat pipe
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Fig(2) photo of expererimental set up

Fig (2) type of wick

Fig(2) photo of types of wick

3. Results and discussion.

The experimental results are investigated, the effects of operating parameters on convective heat transfer

coefficient and on nusselt number are presented .the experiments on heat pipe performance are carried out for several values of heat input rates up to 200 W. pure water is the working fluid .the amount of the fluid is charged to give the filling ratio of (0.2,0.4,0.5,0.6,0.7,0.8,0.9,and 1.0) respectively and the wick is changed by 1st coated, 2nd coated ,and 3rd coated from apainting available to 400˚c .finally the obtained experimentally results are compared and discussed.Experiments were performed on the considered loop heat pipe using pure water as working fluid .the effect of heat input (Q),and filling ratio,(FR) and coated, uncoated wick on its performance are investigated. The values of local surface temperatures along all sections of the heat pipe were measured (0≤X≤360 mm), where X is measured from the beginning of the evaporator section. (0≤X≤120 mm) represents the evaporatorsection, (120<X≤240mm) represents the adiabatic section ,and (240<X≤360 mm) represents the condenser section.

3.1 Results of pure water as working fluid, coated and uncoated wick heat pipe.

Fig (4 to 7) illustrate the surface temperature along the heat pipe for different heat input rates (50,100,150,and 200W) at fixed FR=0.5.from all figures one can show that the surface temperature take the higher value at the evaporator section(0≤X≤120 mm)and take the fixed value at the adiabatic section(120<X≤240 mm) and the temperature decreases inside the condenser (240<X≤360 mm).it is observed that the surface temperature decreases with increasing the distance from the evaporator ,for any heat input rate .Comparing between Fig(4) and Fig(7)the effect of the coated layers is positive ,one can observed that the temperature of the wall decreases for both sections evaporator and condenser for 3rd coated layer wick than for un coated wick.

Fig (8) illustrates the variation of the total average thermal resistance for heat pipe with filling ratio FR =0.5 and for different heat inputs for the un coated ,1st coated,2nd coated, and 3rd coated wick heat pipe, it can be noticed the total thermal resistance decreases with increasing heat input and increasing number of layers of coated wick.

Fig (9,10,11,12) illustrate variation of the temperature difference (Te-Tc) of the heat pipe with filling ratio and for different heat inputs for the un coated wick, 1st layer coated wick, 2nd layer coated wick, 3rd layercoated wick respectively. it is observed that the temperature difference (Te-Tc) decreases with increasing heat input and increasing number of layers of coated wick till filling ratio FR = 0.5 then increases for high filling ratio

Fig (13,14,15,16) illustrate variation of the total thermal resistance of the heat pipe with filling ratio and for different heat inputs for the uncoated wick, 1st layer coated wick, 2nd layer coated wick,3rd layer coated wick respectively. It is observed that the total thermal resistance decreases with increasing filling ratio and increasing number of layers of coated wick till filling ratio FR = 0.5 then increases for high filling ratio then decreases at

The optained data of temperature and power are used to calculate the overall heat transfer coefficient and Nusselt number. At steady state condition,the produced heat generated by electrical heating is given by the equation:

$$Q = I\,V \qquad (1)$$

Where Q is the input heat rate (W), V is the applied voltage across the heat element terminal (V), and I is the electric current (Amp).

The heat flux can be calculated by the following:

$$q'' = Q/A_e \qquad (2)$$

Where q" is the heat flux density (W/m²), and A_e is the area of the evaporator surface attached to the electric heater (m²).

Thermocouples are used to measure the temperature for the evaporator, adiabatic section and condenser. One can calculate the average temperature to the evaporator and the condenser by the following relation.

$$\overline{Te}=\{(\sum Te_i)/Nte\} \ i=1 \ to \ i=N \quad (3)$$
$$\overline{Tc}=\{(\sum Tc_i)/Ntc\} \ i=1 \ to \ i=N \quad (4)$$

Where $\overline{Te}$ and $\overline{Tc}$ are the average of the evaporator temperature and the average of the condenser temperature. Te and Tc are the evaporator temperature and the condenser temperature. Nte and Ntc are the number of thermocouples on the evaporator and the number of thermocouples on the condenser.

The average overall heat transfer coefficient is calculated by the following equation:

$$U=q"/ (\overline{Te} - \overline{Tc}) \quad (5)$$

Where, U is the overall heat transfer coefficient, $W/m^2.k$.

The average Nusselt number is calculated by the following equation:

$$\overline{Nu}= (U*L_1/K_{ef}) \quad (6)$$

Where $\overline{Nu}, L_1$ and K_{ef} are the average Nusselt number, the characteristics length(m) and the effective thermal conductivity, W/m.k.

The total thermal resistance of heat pipe can be expressed as:

$$R= (\overline{Te} - \overline{Tc})/Q \quad (7)$$

4.   Comparison with available literature.

Fig (17) shows a comparison between the present experiment results with those reported by BRUSLY SOLOMON,[17]., in case of using pure water as working fluid and for the un coated wick heat pipe .one can observed that the present experiment results for the used working fluid and wick have the same trend as those reported byBRUSLY SOLOMON,[17]., the difference between both results when using pure water may be un certainty in measurements.

5. Reduction in total thermal resistance

The reduction in total thermal resistance of heat pipe charged with the wick as shown in Fig (18)

$$RR= (R \ uncoated-R \ coated)/ \ R \ uncoated \quad (8)$$

The following table illustrates the reduction in total thermal resistance of heat pipe charged with the number of coated layers of the wick.

Fig (19, 20, and 21) illustrate variation of reduction ratio (RR) in total thermal resistance of the heat pipe with filling ratio and for different heat inputs for the1st layer coated wick, 2nd layer coated wick and 3rdlayercoated wick respectively.one can observed that the reduction in total thermal resistance increases with increasing filling ratio and increasing number of layers of coated wick till filling ratio FR = 0.5 then increases for high filling ratio then decreases at FR=1

6. Conclusions

An experimental investigation is carried out to study the thermal performance of a heat pipe operated with a different coated wick. Conclusions may be drawn from the experimental results as follows

1- Increaseing the input heat increases the wall temperature and decreases the total thermal resistance.
2- The optimum filling ratio is found to be FR around 0.5 for the uncoated wick.
3- Improvement in thermal performance and reduction in thermal resistance higher usingcoated wick than uncoated wick.
4- Improvement in thermal performance and reduction in thermal resistance with incrasing no. of layers of coated wick.

7. Nomenclature

Ae   area of evaporator surface, m^2
FR   filling ratio (volume of working fluid to the evaporator volume)

Uoverall heat transfer coefficient, W/m² K

I   Electric current, Amp

R   thermal resistance of heat pipe, C/W

Kef   effective thermal conductivity, W/m K

N   number of thermocouples

Q   heat transfer rate, W

T   temperature, °C

ΔT   temperature difference between evaporator and condencer, °C

q   heat flux, W/m2

V   applied voltage, V

RR   reduction in total thermal resistance of heat pipe

A. Subscripts

h,   hydraulic

w,   wall

l,   liquid (cooling water)

hp, heat pipe

B. Dimensionless numbers

Nu   Nusselt number, (Nu=hd/k)

C. Abbreviation

CPL   Capillary pumped loops

LHP   Loop heat pipe

MHP   Micro heat pipe

8. Refrences.

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[4]   Brusly Solomon A., Ramachandran K., and Aillai B.C. "Thermal performance of a heat pipe with nanoparticles coated wick" Applied Thermal Engineering vol 36 (2012) pp106-112.

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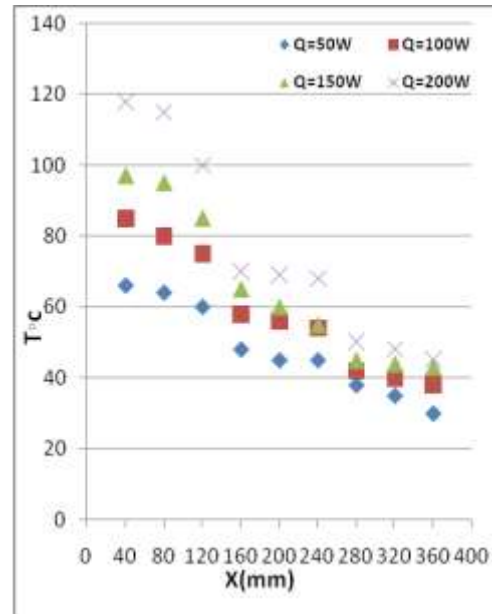
Fig(4) Effect of input heat on heat pipe wall temperature distribution for FR = 0.5 and uncoated wick.
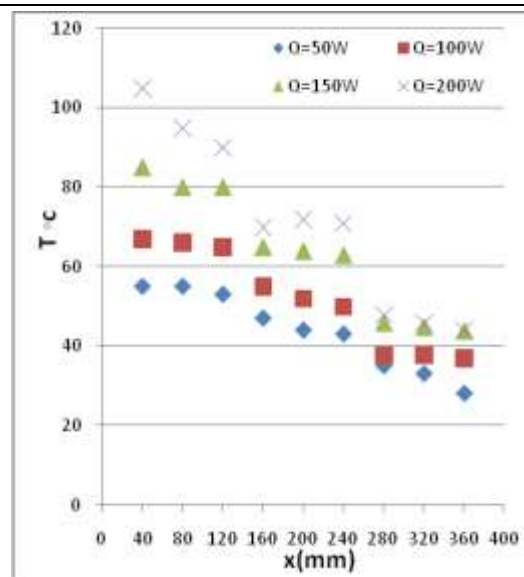
Fig (5) Effect of input heat on heat pipe wall temperature distribution for FR =0.5 and 1st coated wick.
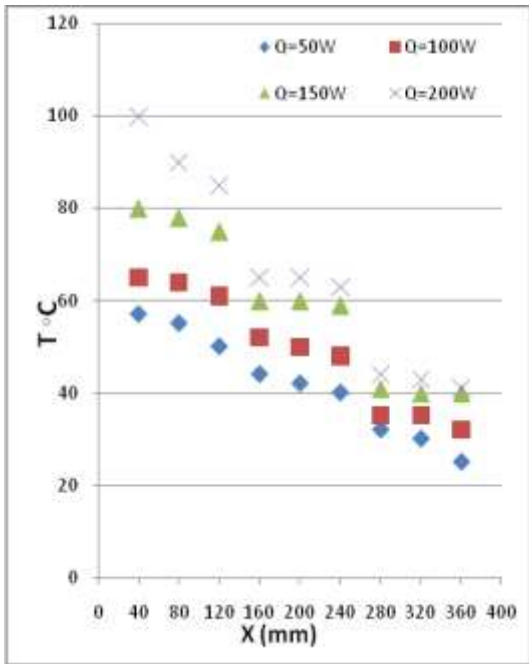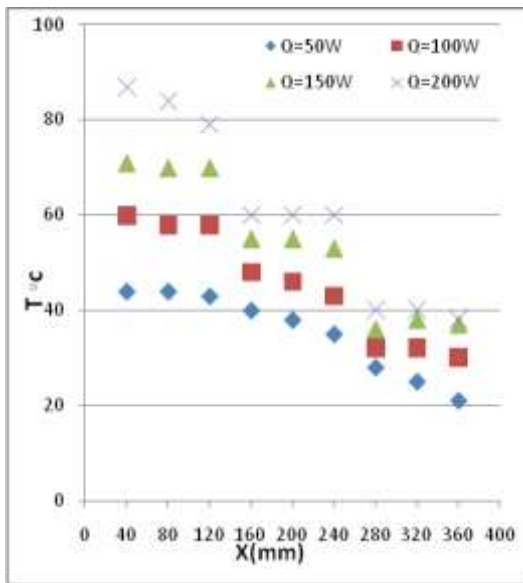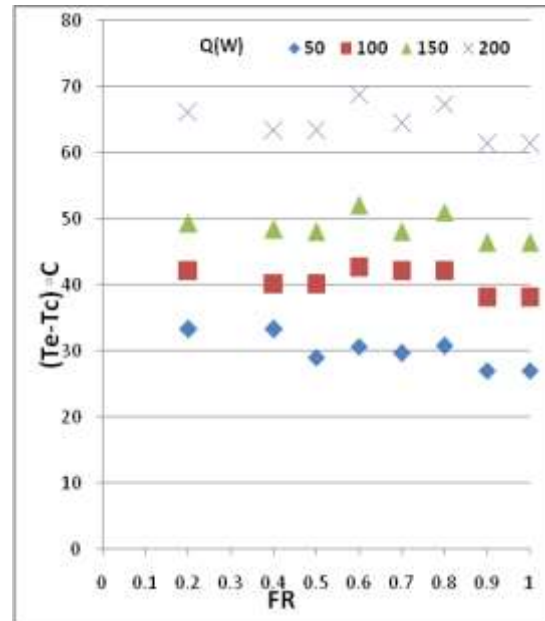
Fig (6) Effect of input heat on heat pipe wall temperature distribution for FR =0.5 and 2nd coated wick
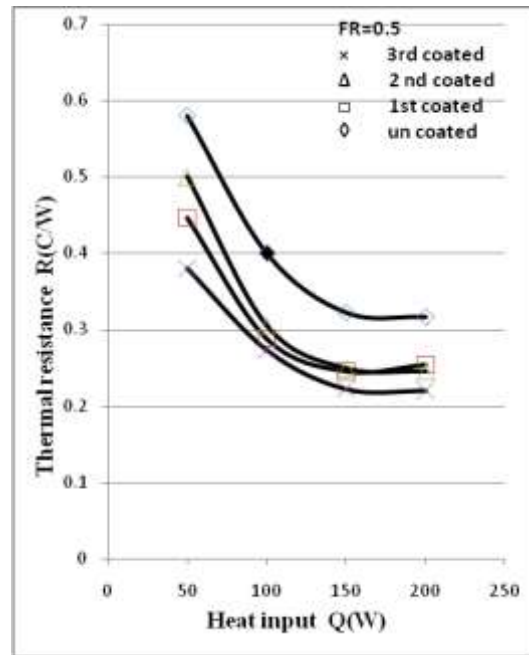
Fig (8) effect of input heat on heat thermal resistance for FR = 0.5 and no. of layers of coated wick

Fig (7) Effect of input heat on heat pipe wall temperature distribution for FR around 0.5 and 3rd coated wick

Fig (9) effect of filling ratio on uncoated wick heat pipe temperature difference for different heat input
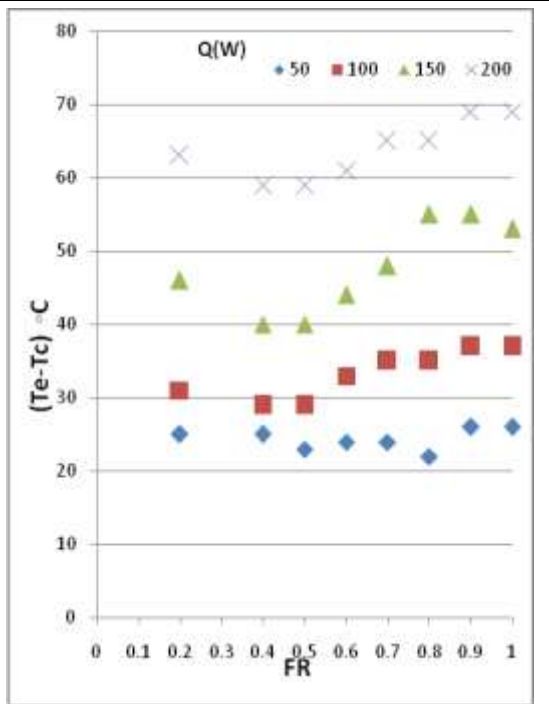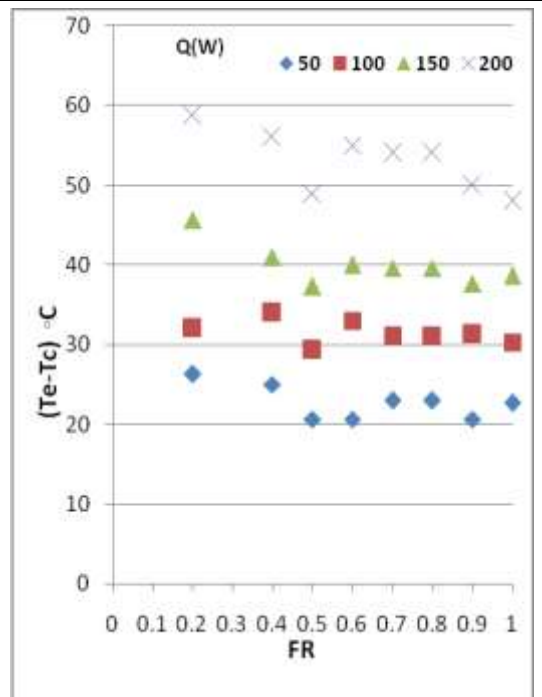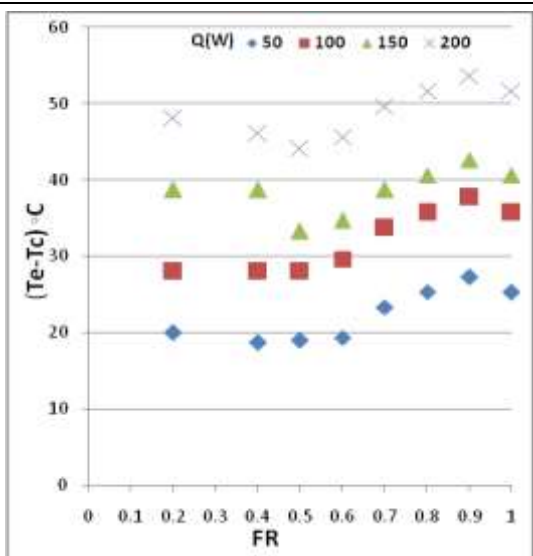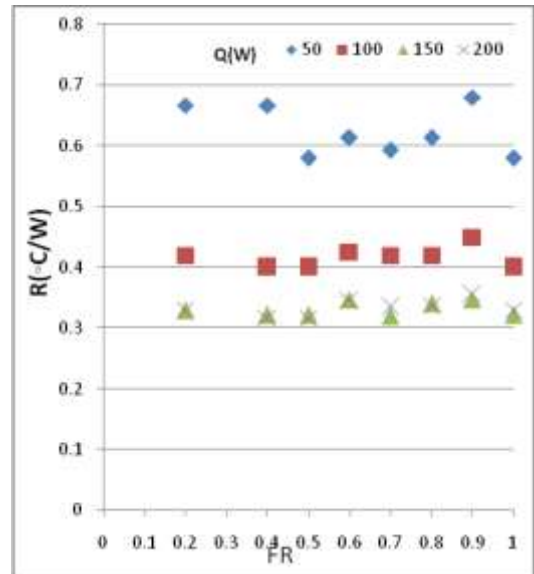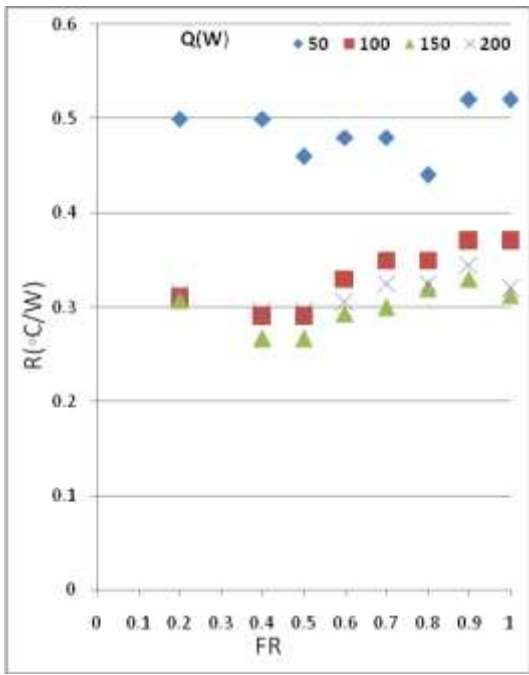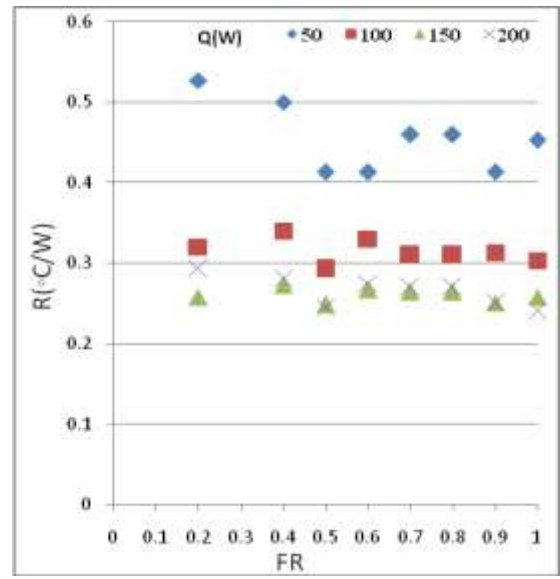
Fig (10) effect of filling ratio on 1st layer coated wick heat pipe temperature difference for different heat input

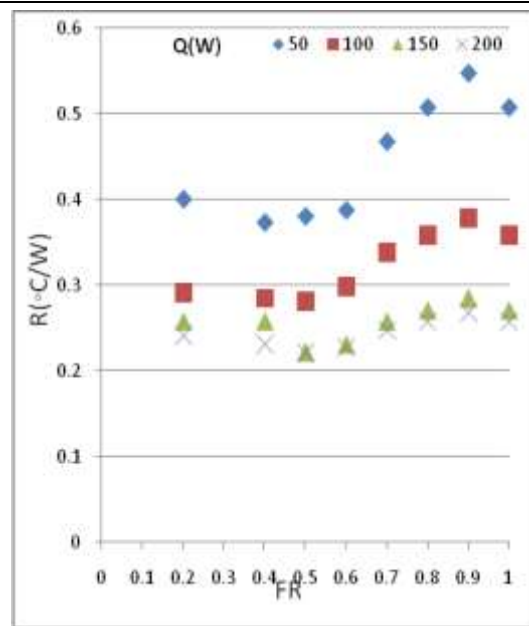*Fig (12) effect of filling ratio on 3rd layer coated wick heat pipe temperature difference for different heat input*

Fig (11) effect of filling ratio on 2nd layer coated wick heat pipe temperature difference for different heat input

Fig (13) effect of filling ratio on uncoated wick heat pipe total thermal resistance for different heat input

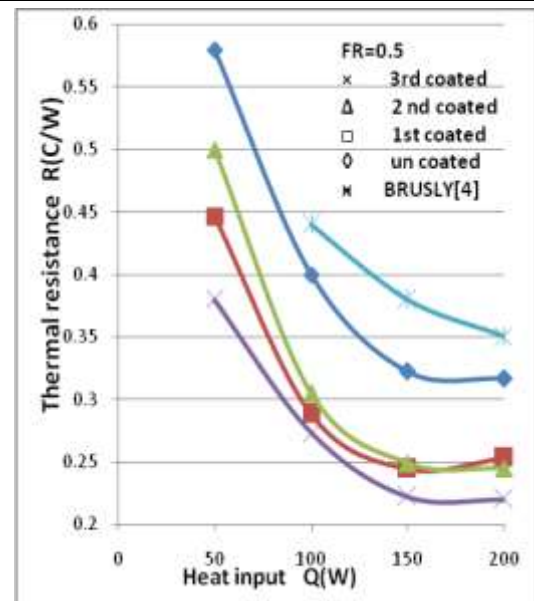**Fig (14) effect of filling ratio on 1st coated wick heat pipe total thermal resistance for different heat input**

Fig (16) effect of filling ratio on 3rd coated wick heat pipe total thermal resistance for different heat input

Fig (15) effect of filling ratio on 2nd coated wick heat pipe total thermal resistance for different heat input

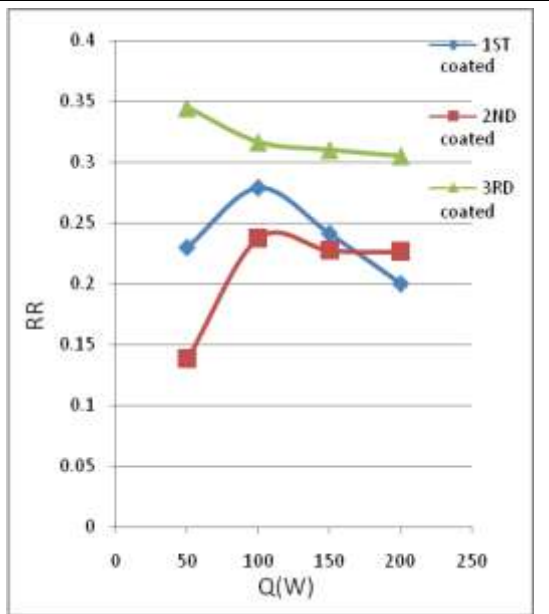**Fig (17) comparison between total resistance at different heat inputs for present experimental results and previous work**

Fig (18) reduction in total thermal resistance of heat pipe for 1st, 2nd, 3rd coated wick with heat input
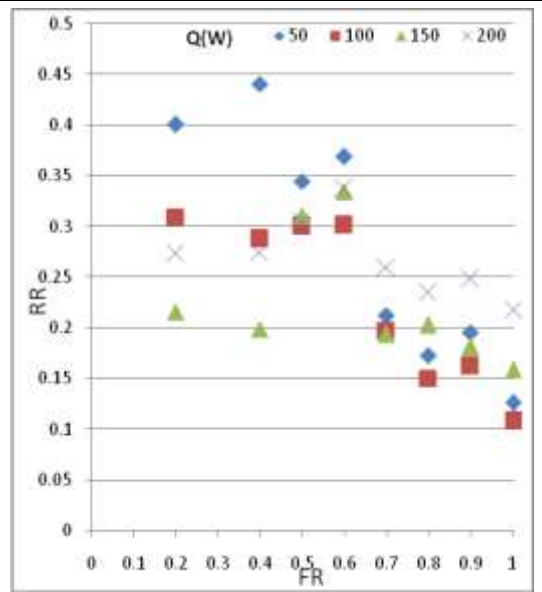

Fig (20) effect of filling ratio on 2nd coated wick heat pipe reduction ratio in total thermal resistance for different heat input
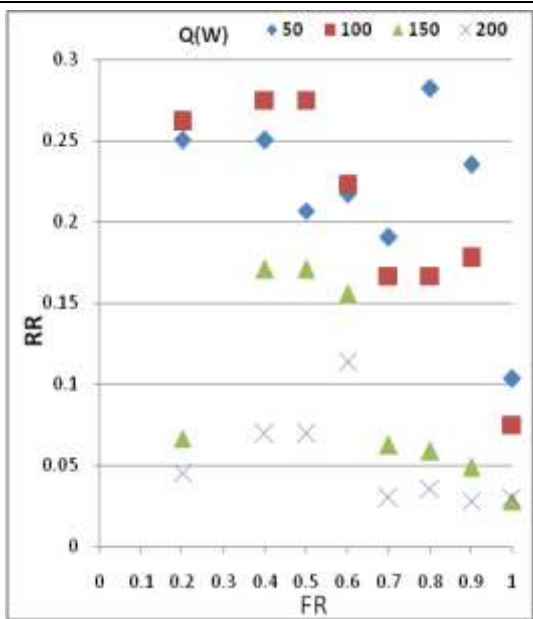

Fig (19) effect of filling ratio on 1st coated wick heat pipe reduction ratio in total thermal resistance for different heat input
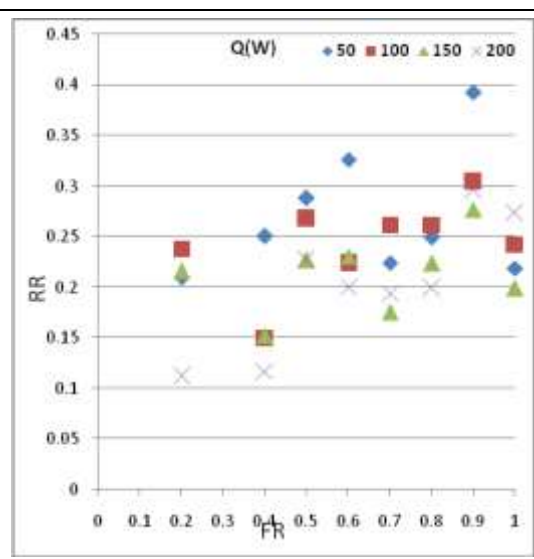

Fig (21)effect of filling ratio on 3rd coated wick heat pipe reduction ratio in total thermal resistance for different heat input