Thermal Performance of a Heat Pipe with Sintered Powder Metal Wick Using Ethanol and Water as Working Fluids

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Received on : 2/1/2011 **Accepted on : 29/3/2011**

Abstract.

An experimental study is carried out to compare the thermal performance of a sintered powder metal wick heat pipe. Pure water and absolute ethanol are used as two different working fluids. The pipe is made of copper with 300 mm length, 14 mm diameter, and 1.0 mm wall thickness. The wick is made of copper powder. All the experiments are accomplished and the heat pipe is at the horizontal position ($\theta=0^o$). The heat flux changed within the range (2.8 -13.13) kW/m^2, while all other conditions remained constant.

The results show that the thermal performance of the heat pipe is better when water is the working fluid, where the operating temperature and the thermal resistance of the heat pipe are lower when the water is the working fluid.

Keywords: Heat pipe, sintered powder wick, thermal performance, working fluids, water, ethanol.

1. Introduction.

The heat pipe is a 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].

Guoping et al. [3], have experimentally studied the heat transfer characterization of a pulsating heat pipes. They used two heat pipes with different working fluids of HFC-134a and butane, the heat pipes were made of aluminum plate with the same width of 50 mm and thickness of 1.9 mm. The effects of heat flux, heat pipe orientation, length of cooling section, heating surface area, cooling mode and the working fluid were examined. High thermal performance of 0.05^o C/W was demonstrated for the heat pipe using HFC-134a.

Thermal performance of a thermosyphon heat pipe using ethanol-water with various parameters such as the mixture content, the pipe diameter, and the working temperature has been studied by Kiatsiriroat et al. [4].

An experimental study to determine the performance of silicon micro heat pipes (MHPs) filled with different fluids was conducted by Pandraud et al. [5]. The results show that the use of (MHPs) with low liquid charges can increase the effective thermal conductivity of the array. Chandratilleke et al. [6] investigated four different working fluids in an attempt to develop loop heat pipes that can work in the cryogenic temperature range of 4 to 77 K.

Kaya and Hoang [7] used primary and secondary wick inside the evaporator to ensure that the liquid remains in the wick all the time.

A copper heat pipe sintered with a copper powder wick was investigated theoretically and experimentally by Fadhil [8]. The experimental side include manufacturing and testing three heat pipes sintered with different thick wicks; 1.0, 1.5, 2.0 mm in addition to a fourth wickless pipe (thermosyphon). The results show that the optimum amount of the liquid charge for heat pipe with sintered copper powder wick is about twice the amount theoretically estimated. Also, the heat pipe thermal performance decreased when the pipe was operating against gravity. The increase in wick thickness led to an increase in heat pipe operating temperature.

Zhang [9] performed experimental studies on a pulsating heat pipe. The capillary tube used in this study has an inside diameter of 1.18 mm and a wall thickness of 0.41 mm. The experiments were conducted under pure natural convection, for heating powers from 5 to 60 W. Three working fluids –FC-72, ethanol, and deionized water were used. He concluded that the thermal oscillation amplitude is much smaller for FC-72 due to its lower surface tension, than for ethanol and water, while the oscillation cycle period for FC-72 is shorter than for the other two fluids. Also, he suggested that FC-72 may be used for low heat flux situations, due to its lower minimum heating power.

The effect of evaporator section lengths and working fluids on operational limit of closed loop oscillating heat pipes with check valves with R-123, ethanol, and water used as the working fluids was investigated by Meena et al. [10]. They concluded that, when the evaporator lengths increased from 5 cm to 10 and 15 cm the critical heat transfer flux decreased, also when the working fluids changed from R-123 to ethanol and water the critical heat flux decreased.

The aim of the present experimental study is to investigate the thermal performance of a copper heat pipe sintered with porous media wick when the ethanol represents a working fluid, and compare this performance with that when the pure water represents a working fluid. The present study differs from most previous studies in that it uses a heat pipe with sintered powder metal wick.

2. Experimental Apparatus and Procedure.

The schematic diagrams of the experimental setup and the arrangement of thermocouples for wall temperature measurement are shown in **Figs. (1)** and **(2)** respectively. The container of the heat pipe was made of a copper pipe of 16 mm OD, 14 mm ID and its length was 300 mm. A copper powder was selected to form the wick structure. The average particle diameter of the powder used to make the wick structure was 0.03 mm.

Sintering was used to produce the wick structure. The simplest way of making wicks by this method is to sinter the powder in the pipe that will form the final heat pipe. In the case of copper powder used in the present work, a stainless steel mandrel is satisfactory as the copper will not bond to stainless steel and, thus, the bar can easily be removed after sintering [1]. The pipe was fitted with the mandrel and a collar at one end. The powder was then poured in from the other end. No attempt was made to compact the powder apart from tapping the pipe to make sure that there were no gross cavities left. When the pipe was full, the other collar was put in place and pushed up against the powder. The complete assembly was then sintered by heating in an environment which was vacuumed from oxygen at 850^{o}C for half hour. After the pipe was cooled and removed from the furnace, the mandrel was removed and the pipe, without the mandrel was then resintered to complete the sintering process. After this process the pipe was ready for use. The wick porosity was measured experimentally according to the B328-73 Designation (Reapproved 1986) and it was found tube about 0.32. **Fig. (3)** shows a cross section of a completed pipe. When the fabrication of the wick structure was finished, the heat pipe was cleaned by flushing with R-11, also was taken to make sure that the end caps were very clean, prior to the welding of the end caps. The end caps were welded by using the Tungsten inert gas welded (TIG), to minimize the probability of recontamination of the cleaned parts.

Prior to changing, a heat pipe must be evacuated to remove materials that may subsequently appear unwanted non-condensable, or that chemically react with the working fluid forming undesirable corrosion products. When the evacuation process was completed, the heat pipe became ready for charging. The amount of working fluid was charged to the heat pipe and then the charging valve was closed. For measurement of the pipe-wall temperature, 12-copper-constanton thermocouples of 1mm OD were arranged at the locations shown in **Fig.(2)**. The vapour temperature inside the pipe was measured by two thermocouples inserted inside two copper probes. The probe tips were all sealed by welding. The condenser cooling water temperature was measured by two thermocouples placed at the center core of the condenser jacket water input and output tubes. The outer surface temperature of the heat pipe insulating materials was measured by three thermocouples. Each thermocouple bead was placed in a very thin pocket made at the outer surface of the insulation material.

The heat loss from the outer surface of the insulation during test was calculated and it's found to be less than 20% of the heat input.

A resistance heating wire with electrical insulation sleeve was used to input the heat to the heat pipe evaporator. The electric power from a constant-voltage regulator was controlled by variable transformer in order to supply a uniform heat input to the heating wire. The input heat flux was measured by an ammeter and voltmeter. A thermally controlled water jacket was used to cool the condenser. The jacket was of a 30.0 mm diameter. The condenser section was insulated by 10.0 mm thick armaflex insulation. Also, the evaporator and adiabatic sections were wrapped by 25.0 mm thick glass wool and then covered by 1.0 mm carton layer with a white paint. The specifications of the heat pipe used in this work are given in **Table (1).**

In this study the heat pipe was fixed at the horizontal position ($\theta=0^o$). The thermal load was adjusted from 2.8 to 13.13 kW/m^2. At any one power level setting, the power was set and then the heat pipe allowed to reach steady state condition. After wards, the power input, the condenser water flow rate and the temperature of the various locations were all recorded. The same procedure was repeated at each power input until the maximum power input was reached. Two types of working fluid are used in this study, one of them is the pure water and the other is the ethanol.

3. Results and Discussions.

The results of the thermal performance can be described from the evaporator temperature, condenser temperature, operating temperature, and the input heat flux, or in terms of the thermal resistance.

Fig.(4) shows a comparison of the average evaporator temperature($T_{e.a}$) for the tested heat pipe with two working fluids under the same conditions. It is obvious that the average evaporator temperatures when the ethanol and the water are the working fluids are near together at the smaller heat flux (<11.0 kW/m^2). With such high heat flux (> 11.0 kW/m^2) a clear difference can be seen between the two curves, i.e. dry out was observed for the ethanol. This may be attributed to the difference between the boiling temperatures for the two fluids.

The effect of input heat flux on the average condenser temperature ($T_{c.a}$) is shown in **Fig.(5).** At the low heat flux, as seen from this figure, the ($T_{c.a}$) for the ethanol is slightly higher then that for the water. However, at the higher heat flux, the ($T_{c.a}$) for the ethanol is higher than that for the water. This is considered to be due to the influence of specific heat, where the specific heat of the water twice than of the ethanol.

The effects of input heat flux on the operating temperature are shown in **Fig.(6)** . The operating temperature means the average vapour temperature at the core of the heat pipe. The test results indicate that the operating temperatures of the ethanol are higher than that of the water for any input heat flux.

Fig.(7) represents the thermal resistance of the heat pipe with sintered powder metal wick, as a function of the input heat flux for the two working fluids at the horizontal orientation ($\theta=0$). The overall thermal resistance can be defined from the heat flux (q), the average evaporator temperature ($T_{e.a}$) and the average condenser temperature ($T_{c.a}$) Stephane et al[11]:

$$R_{th} = \frac{T_{e.a} - T_{c.a}}{q} \quad (1)$$

For both working fluids, the overall thermal resistance decreases when increasing the input heat flux. The thermal resistance of the heat pipe with ethanol working fluid is higher than that with water working fluid. The thermal resistance of the heat pipe was affected by the latent heat of vaporization, where the working fluid with a higher latent heat of vaporization exhibits the lower thermal resistance.

4. Conclusion.

The main findings of the present study can be summarized as follows;

1. The average evaporator temperature of the heat pipes higher when ethanol is the working fluid.
2. The average condenser temperature of the heat pipe is higher when ethanol is the working fluid.
3. The operating temperature of the heat pipe is higher when ethanol is the working fluid in comparison with water.
4. The thermal performance of the heat pipe is better when water is used as the working fluid in comparison with ethanol. In other words, when both are available for use in the same conditions, water is preferable.

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5. Nomenclature.

La	adiabatic section length (m)
Lc	condenser length (m)
Le	evaporator length (m)
q	input heat flux (kW/m^2)
R_{th}	thermal resistance ($m^2.C/kW$)
$T_{c.a}$	average condenser temperature (^oC)
$T_{e.a}$	average evaporator temperature (^oC)
T_o	operating temperature (^oC)
θ	angle of orientation (^{o})
δ_w	wick thickness (m)

Table (1) Heat pipe specifications

δ_w (m)	Porosity	L_e=L_c (m)	L_a (m)	Charge (g)
0.001	0.32	0.075	0.15	8.0

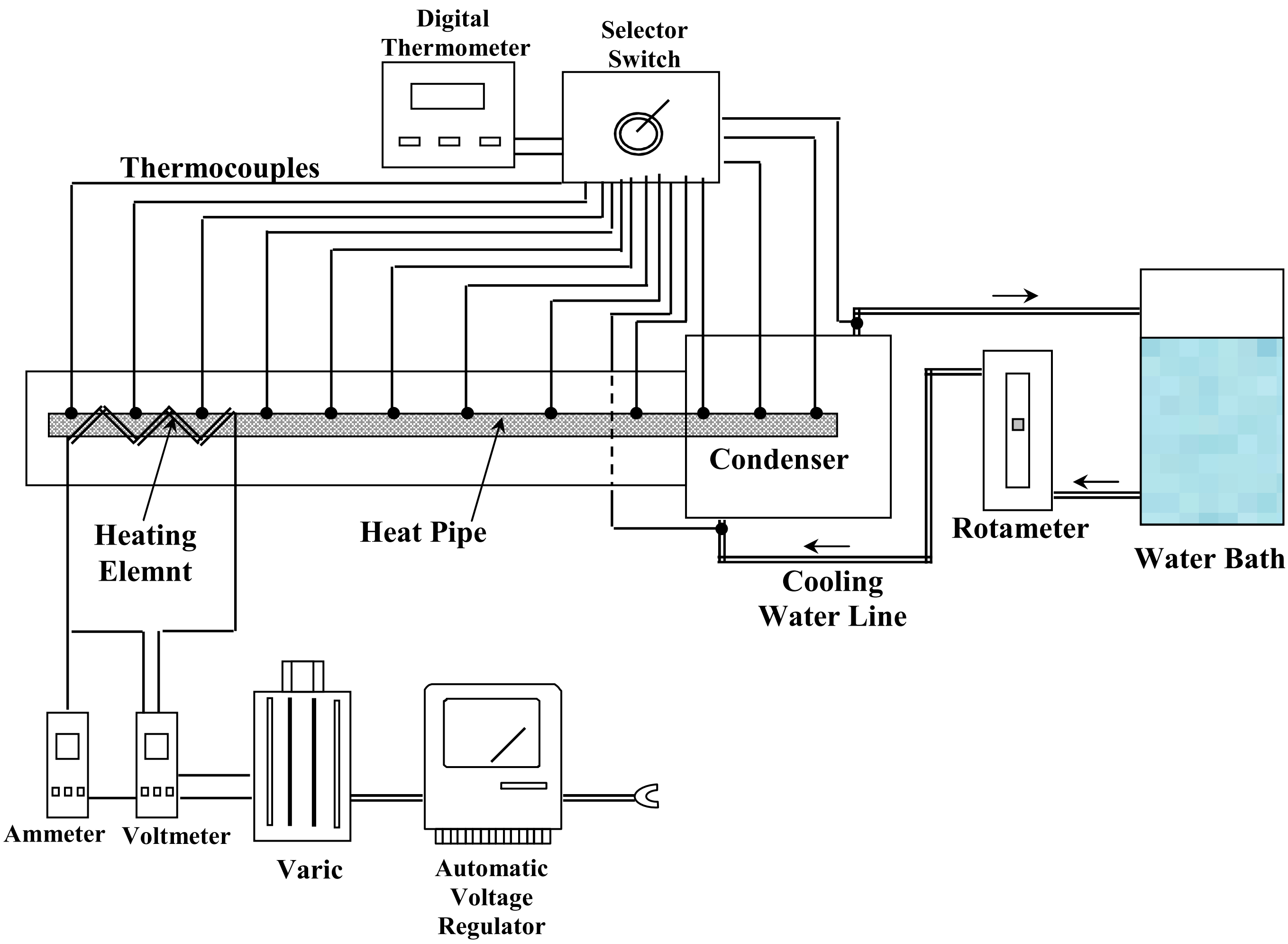

Figure (1): Schematic diagram of the experimental rig**.**

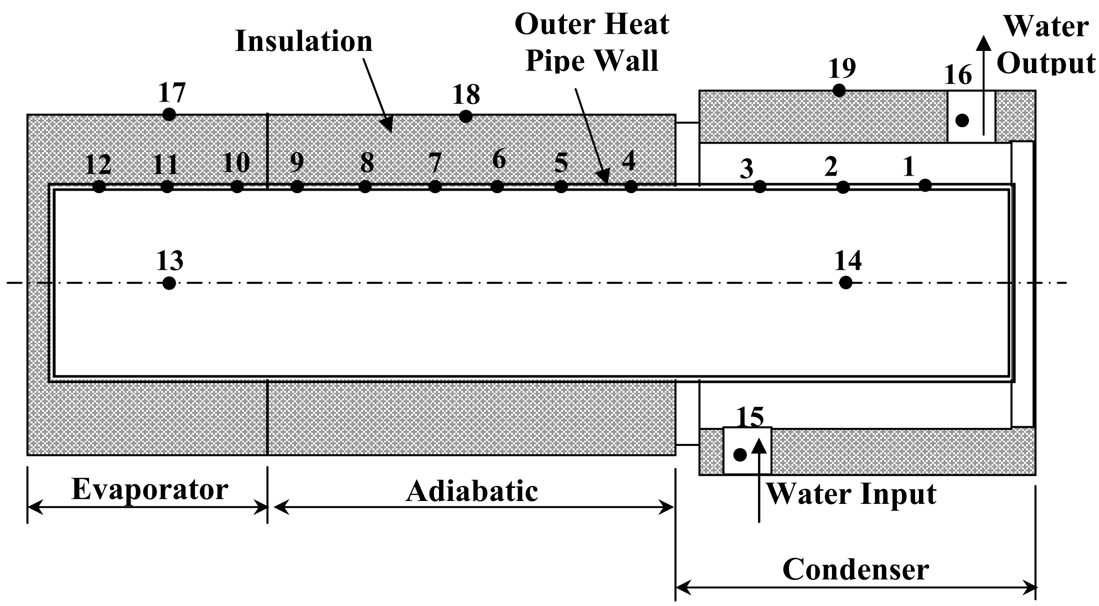

Figure (2): Schematic of thermocouples location.

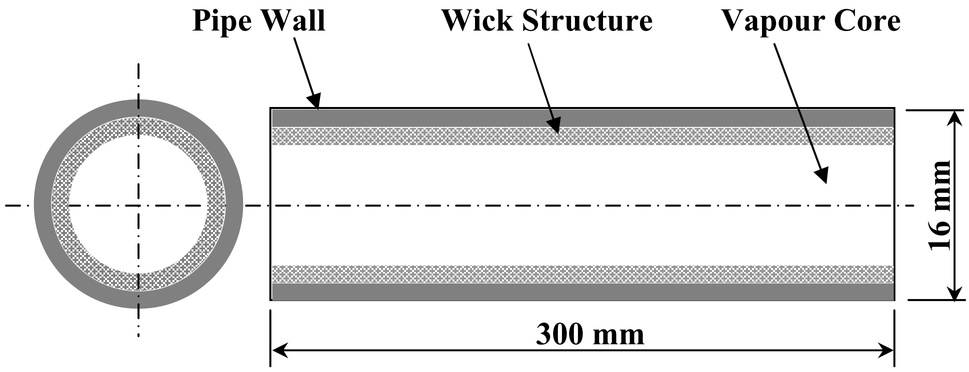

Figure (3): Heat pipe cross section.

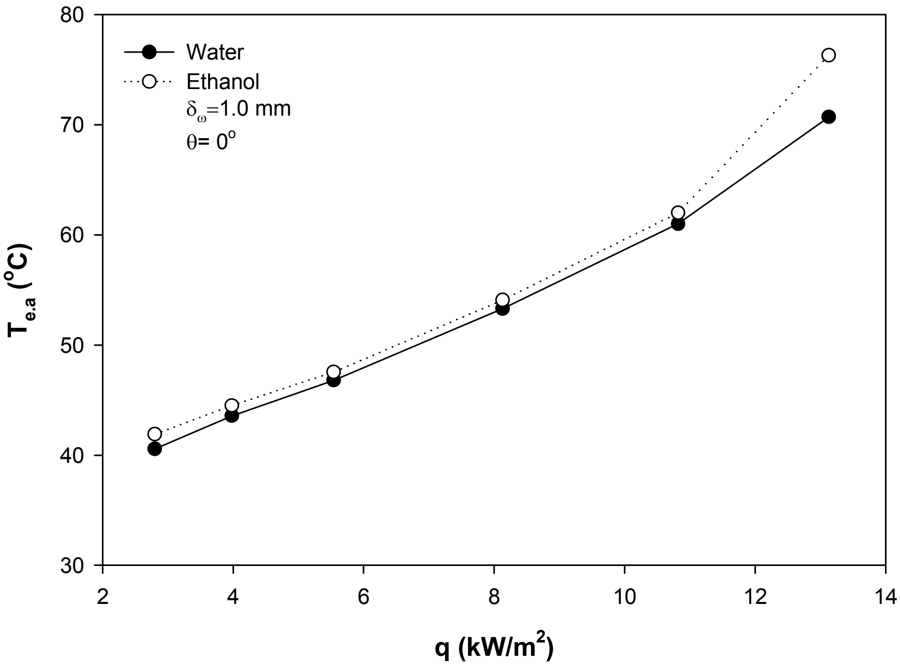

Figure (4): Average evaporator temperature Vs. input heat flux.

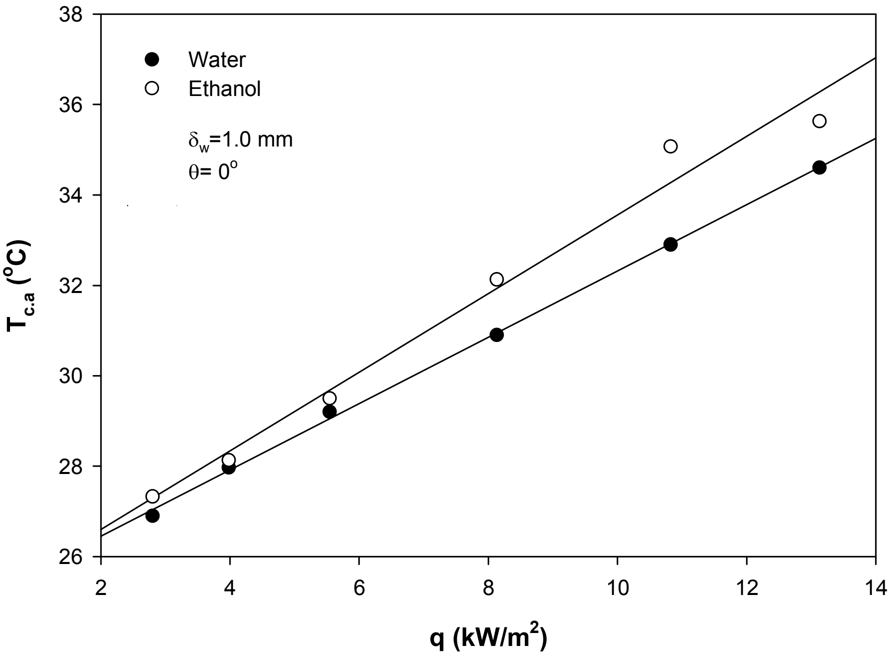

Figure (5): Average condenser temperature Vs. input heat flux.

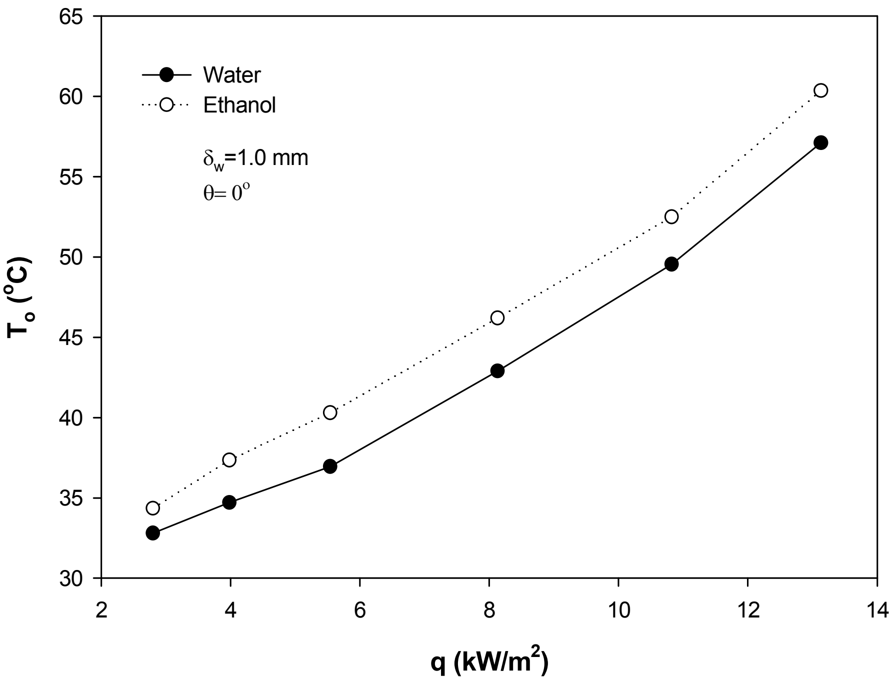

Figure (6): Operating temperature as a function of input heat flux.

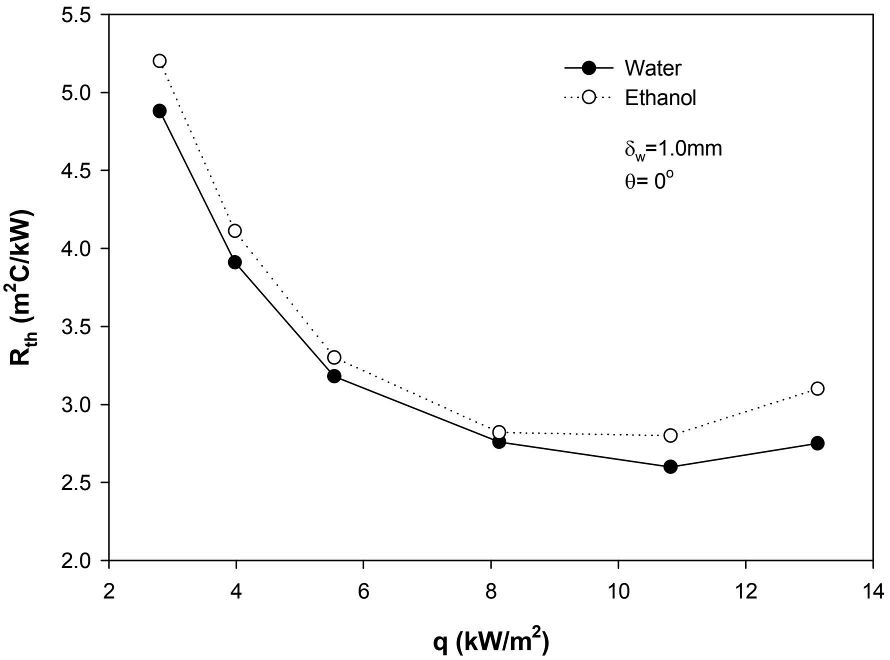

Figure (7): Thermal resistance as a function of input heat flux.

الأداء الحراري لأنبوب حراري مبطن بفتيل من مسحوق معدني عندما يكون الايثانول أو الماء مائعا التشغيل.

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الخلاصة.

لقد أجريت دراسة تجريبية على أنبوب حراري مبطن بفتيل مسامي لمقارنة الأداء الحراري للأنبوب عند استخدام كل من الماء النقي والايثانول المطلق كمائع تشغيل. الأنبوب من النحاس وهو بطول (300 mm) وقطر داخلي (14mm) وسمك جدار الأنبوب (1.0 mm)، أما الفتيل فهو من مسحوق النحاس.

أجريت جميع التجارب والأنبوب الحراري بوضع أفقي ($\theta=0^o$). تم تغيير الفيض الحراري ضمن مدى تراوح بين $(2.8 - 13.13)kW/m^2$ وتثبيت الظروف الأخرى كافة.

أظهرت النتائج أن الأداء الحراري للأنبوب يكون أفضل عند استخدام الماء مائعا للتشغيل، حيث لوحظ أن درجة حرارة الاشتغال وكذلك المقاومة الحرارية للأنبوب تكونان أوطأ عندما يكون الماء مائعا للتشغيل.

الكلمات الدالة: أنبوب حراري، فتيل مسامي مبطن، أداء حراري، موائع تشغيل، ماء، أيثانول.