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Experimental Study of the Thermal Characteristics for a Thermosyphon Pipe with Finned Condenser

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Abstract

The thermosyphon pipes are two-phase heat transfer devices with extremely high effective thermal conductivity. In this work, the thermal characteristics of a thermosyphon pipe with finned condenser (free convection condenser) is studied experimentally. The thermosyphon pipe is manufactured from copper. Distilled deionized water is used as a working fluid, the pipe is charged at filling ratio equal to 50% of the evaporator volume. Annular fins manufactured from aluminum are installed on the outer surface of the condenser section. The thermosyphon pipe is tested experimentally at different input power (2 W, 5 W, 10W, 15 W, 20 W, 25 W, and 40 W) and different inclination angle (30^o, 60^o, and 90^o from horizontal). The results show that the increase in the input power leads to increase in the thermosyphon pipe operating temperature, while the thermal resistance decreases with increasing the input power. Also the results showed the thermal performance of the pipe is improved when the pipe is positioned at inclination angle of 30^o comparing with the inclination angles 60^o, and 90^o.

Keywords: Thermosyphon, Thermal resistance, Thermal performance, Finned condenser

1.Introduction

Thermosyphon pipes are two-phase heat transfer devices with extremely high effective thermal conductivity. The advantage of using the thermosyphon is its need to small area and temperature difference. In addition, to simplicity of design, high rate of heat transfer, one way heat transfer (thermal diode), low cost, low weight, low cost of maintenance **[1,2]**. All thermosyphons have three sections including: evaporator, adiabatic and condenser, and the condenser always placed above the evaporator. In a thermosyphon, the heat is inputted through the evaporator section where a liquid pool exists, turning the working fluid into vapor. The vapor rises and passes through the adiabatic sectiontoward the condenser. In the condenser the vapor condenses and gives up its latent heat. Then, the condensate returns to the evaporator due to its gravity **[3]**. Regarding to their high efficiency, reliability and cost effectiveness, thermosyphons have been used in many different applications. These applications include preservation of permafrost, de-icing of roadways, turbine blade cooling, and applications such as in heat exchangers, solar systems**,** humidity control, reactors and food industry **[4]**. A numerous experimental investigations were carried out to understand the thermosyphons characteristics and the effect of various parameters on their performance. **Negishi and Sawada [5]** investigated experimentally the thermal performance of an inclined two-phase closed thermosyphon. They use water and ethanol as working fluids. Their conclusions are that the highest heat transfer rate is obtained when the filling ratio (ratio of volume of working fluid to volume of evaporator section) is between 25% and 60% for water and between 40% and 75% for ethanol, and the inclination angle is between 20° and 40° for water, and more than 5° for ethanol. **Wang and Ma [6]** studied condensation heat transfer inside vertical and inclined thermosyphon. Their conclusions were that the inclination angle has a notable influence on the condensation coefficient, and the optimum inclination angle varies with liquid filling from 20- 50 degrees. **Payakaruk et al. [7]** investigated the heat transfer characteristics of copper thermosyphon with inner diameters of 7.5, 11.1 and 25.4 cm. Water, ethanol, R-22, R- 123, and R-134a are used as working fluids. They concluded that the optimum inclination angle for water is between 40° and 70°. **Abou-Ziyan et al. [8]** investigated the thermal performance of two phase closed thermosyphon under stationary and vibratory conditions with water and R134a as a working fluid. They carry out the experiments for filling ratio of range (40% to 80%). The thermosyphon is tested for various adiabatic lengths of (275,325 and 350mm), vibration frequency (0.0-4.33Hz) and input heat flux (160-2800 kW/m^2). They concluded that the adiabatic length of 350mm and liquid filling ratio of 50%

provide the highest heat flux. **Park et al. [9]** investigated the heat transfer characteristics of a copper two phase closed thermosyphon with Copper Ferrous as working fluid in the range of 50 – 600W heat input and 10 – 70% fill charge ratio. They concluded that the heat transfer coefficient of the evaporator section increases with the increase of the power. The effect of the filling ratio is negligible for both the smooth and grooved surface. On the other hand, the heat transfer coefficient showed some enhancement with increase in the filling ratio in the evaporator section by the expanded working fluid pool. **Samuel Luna Abreu and Sergio Colle [10]** worked an experimental study of two-phase closed thermosyphon (TPCT) for compact solar domestic hot-water systems. The effects of cooling temperature, slope, fill ratio, and evaporator length are determined for different heat fluxes. They concluded that an increase in the cooling temperature decreases the overall thermal resistance. For the smallest slope, better results are achieved; however a dry-out limitation can occur, although it was not observed during the experiments. The thermal resistance decreases for the lowest fill ratio, but it is necessary to take care about the dry-out limitation too. Changing the evaporator length will also change the power input, but the lowest thermal resistances were achieved for the shortest TPCT evaporator. **Ong and Haider E. Alahi [11]** investigate performance of an R134a filled thermosyphon. They carried out the experiments to study the effects of temperature difference between bath and condenser section, fill ratio and coolant mass flow rate, which was water. The thermosyphon is of a copper material with inside and outside diameter of 25.5mm and 28.2mm respectively. The overall length of thermosyphon is 780mm (300mm-evaporator length, 300mm-condenser length). They concluded that the heat flux transferred increased with increasing coolant mass flow rate. **Abreu and Colle [12]** investigated experimentally the analysis of the thermal behavior of two-phase closed thermosyphons with an unusual geometry characterized by a semicircular condenser and a straight evaporator. All the tests are done in an experimental indoor setup that uses electrical skin heaters to simulate the solar radiation. Different evaporator length, fill ratios of working fluid, cooling temperatures and slopes of the evaporator are tested for different heat fluxes. The analysis of the transient results and the steady state performance are used in order to provide information for the design of a compact solar domestic hot-water system. **Noie et al. [13]** conduct experimental investigation of boiling and condensation heat transfer of a two phase closed thermosyphon. They concluded that the heat flux of the thermosyphon is nearly 250 times that of a copper rod with the same dimensions. Maximum heat transfer rates for each aspect ratio occur at different filling ratios evaluate, where for an aspect ratio of 9.8 the maximum heat transfer rate occurs when the filling ratio is 60%, while for an aspect ratio of 11.8 the highest value occurs at filling ratio of 30%.

2. Data reduction

The dimensions of the thermosyphon pipe are selected to fit the application desired to be used in the pipe, which represent cooling water reservoir in summer season. The researcher takes into account that the removal of heat in the condenser section are by free convection. On these bases, fins installed are made from aluminum on the condenser surface for calculated distance, so that transmission of heat from the condenser to the outside air can be obtained. Here are some theoretical calculations regarding the selection of the dimensions of the pipe and the number of fins that must be installed on the surface of the condenser.

2.1 Estimating the optimum space between the fins

To determine the number of the fins fitted to the condenser surface, or the ideal distance between each fin and its neighboring, the relationship of Bar-Cohen and Rohsenow [**14]** has been used, where:

$$S_{opt} = 2.71\left(\frac{R_a}{S^3 L}\right)^{-0.25} \ldots\ldots\ldots\ldots\ldots..(1)$$

where:

R_a : Rayleigh number.

S: The space between the fins.

L : Length of the fin.

2.2 Estimating the convection heat transfer coefficient

The convection heat transfer coefficient is calculated in the following manner:

$$h = \frac{\overline{Nu}\,K}{S_{opt}} \ldots\ldots\ldots\ldots\ldots\ldots.(2)$$

where:

h : Convection heat transfer coefficient.

K: Thermal conductivity of air.

Nu: Nusselt number.

The Nusselt number computed from Bar-Cohen and Rohsenow for symmetric isothermal plate **[14].**

$$\overline{Nu} = \left[\frac{576}{\left[\frac{R_a S_{opt}}{L_c}\right]^2} + \frac{2.87}{\left[\frac{R_a S_{opt}}{L_c}\right]^{0.5}}\right]^{-0.5} \ldots\ldots\ldots\ldots\ldots.(3)$$

where:

Lc: correct length.

The correct length computed from the following equation **[15].**

$$L_c = L + \left(\frac{A_{cs}}{P}\right) \dots\dots\dots\dots (4)$$

where:
A_{cs}: Cross sectional area of the fin.
P: Perimeter of the fin at the tip.
The Rayleigh number computed from the following correlation **[14].**

$$R_a = \frac{g\beta(T_s - T_\infty)S^3}{\alpha v} \quad \dots\dots\dots (5)$$

Where:

g : Gravitational acceleration.

β : Volumetric thermal expansion coefficient.

α: Thermal diffusivity.

v: Kinematic viscosity.

T_s: Temperature of fin surface.

T_∞ : Ambient temperature.

2.3 Estimating heat dispassion from the condenser

The total heat dissipation from the thermosyphon condenser part estimate by the flowing equation

$$Q_{total} = Q_{fins} + Q_{pipe} \quad \dots\dots\dots\dots (6)$$

where:

Q_{fins}: Heat dissipation by the fins.

Q_{pipe} : Heat dissipation from thermosyphon wall between the fins.

The amount of heat dissipation from fins computed by the following equation **[16].**

$$Q_{fin} = \eta_{fin}\, hA_{fin}(T_s - T_\infty) \quad \dots\dots (7)$$

where:

η_{fin} : fin efficiency.

A_{fin} : Cross sectional area of fin.

While the amount of heat dissipation from thermosyphon wall between the fins computed from the following equation

$$Q_{pipe} = hA(T_w - T_\infty) \dots\dots\dots\dots\dots\dots (8)$$

where:

A : Surface area of the pipe condenser wall between the fins.

T_w : Temperature of the surface condenser wall.

3. Experimental Work

In this work, the thermal performance of a thermosyphon pipe with finned condenser (free convection condenser) is studied experimentally. The thermosyphon pipe is manufactured from copper, with length 1.0 m. The inner and outer diameters are (49 mm and 53 mm) respectively. Thermosyphon pipe consist of three sections: evaporator, adiabatic, and condenser. The evaporator length is 0.45 m, and the adiabatic section length is 0.1 m, and the condenser length is 0.45 m . The fins installed on the surface of the condenser;The thickness of the fin is 0.1cm and the distance between each fin and its neighboring is 0.9cm, and the outer diameter of the fin is 15.9cm . The number of fins are 45. The pipe dimensions are taken to employ it to dissipate the heat from water tanks in summer season. Distilled deionized water is used as a working fluid, the pipe is charged at filling ratio equal to 50% of the evaporator volume. Annular fins manufactured from aluminum are installed on the outer surface of the condenser section.

The experiments are carried out on a two phase closed thermosyphon pipe as illustrate photography in figure **(1.1)** and schematicly in figure **(1.2)** to investigate the effect of two parameters, input heat flux and inclination angle , on the performance of two phase closed thermosyphon. The evaporator of the thermosyphon is heated electrically with an electrical heater of length of 200 cm and power of 2000 Watt. The heater is fixed tightly around the outer surface of the evaporator as a spiral to insure distribute the heat worthy on the evaporator. The heat input to the evaporator is set using an electrical energy regulator (Variac). To prevent the heat losses, the entire length of evaporator and adiabatic of the thermosyphon pipe is insulated with 5cm thick of fiber glass insulation. The condenser part for the pipe is cooled by natural convection. The wall temperature of the thermosyphon is measured using eight calibrated thermocouples type (K) (Ni Cr/Ni Al) with measurement error of ±0.5 °C. Each thermocouple welded together and calibrated and inserted in (1 mm) grooves, machined in the outer surface of the pipes, and fixed by an epoxy steel in positions. All thermocouples are connected to data logger type BTM-4208 with twelve channel. Two digital Ammeter and Voltmeter are used to determine the voltage and the current used by the heating element. Voltage is measured directly across the output from an electrical energy regulator (Variac) by a digital voltmeter type INDO that has a resolution of 1V for the used range, and with accuracy was ± 0.5% of reading. The current is measured by a digital ammeter type DP3-72A, with range AC 0~10 amp. With accuracy of ±1 % of reading. The thermosyphon pipe is investigated in three angles 90°, 60°, and 30°, and at various input power namly 2, 5, 10, 15, 20, 25, 40 Watt. At any power adopted the thermosyphon pipe allowed to reach to the steady state condition before taking the readings of temperature and input power. The time required for each experiment is approximately from one to three hours to complete.

Figure (1.1): Photograph of the test rig.

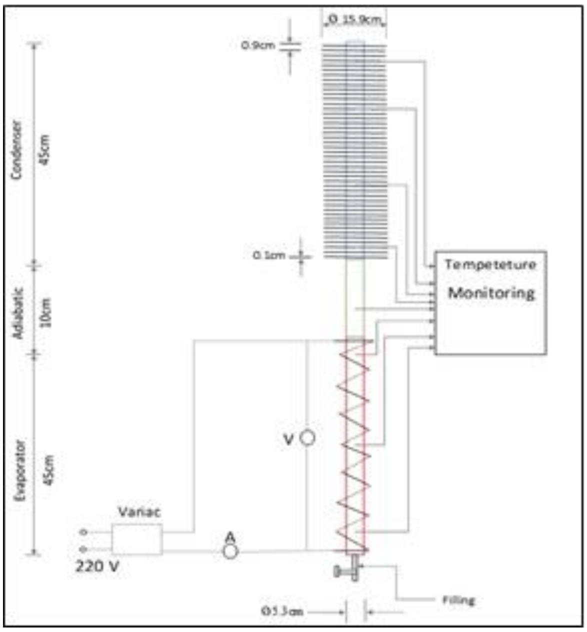

Figure (1.2): Schematic diagram of experimental apparatus.

4. Uncertainty analysis

The main source of uncertainty for the calculated R (effective overall thermal resistance) came from the temperature readings, which were measured using K-type thermocouples and power reading. According to Taylor **[17]**, the propagation of uncertainties associated with the calculated R (u_R) can be estimated from the equation:

$$\frac{u_R}{R} = \sqrt{\left(\frac{u_{\Delta Te-c}}{\Delta Te-c}\right)^2 + \left(\frac{u_{Qin}}{Qin}\right)^2} \; \dots\dots\dots\dots\dots\dots (9)$$

where:

$$u_{\Delta Te-c} = \sqrt{(u_{Te-av})^2 + (u_{Tc-av})^2}$$

The uncertainty associated with $\Delta Te - c$.

u_{Qin}: The uncertainty associated with the reading of the energy throughput Qin.

here :ΔTe_c = Te-av – Tc-av .

Te-av = (Te1 + Te2 +Te3)/3 and Tc-av = $\sum_{j=1}^{4} T_{cj}/4$ are the average wall temperatures in the evaporator and condenser sections, respectively. By calculating u_R, for the entire experimental range, the maximum uncertainty associated with the resulting R values was found to be around 2.1%, which is an acceptable value in engineering applications.

5. Results and Discussion

Figure **(5.1)** shows the temperatures distribution along the outer surface of the thermosyphon pipe. The input power of the thermosyphon pipe are varying from (2W to 40W) with three different inclination angles (30°, 60° and 90°). it's clear that the wall temperatures are increasing with the increase of the input power. As the heat input increases, the temperature distribution becomes everywhere decreasing toward the top of the pool. This occurs due to the transition from the geyser boiling to the nucleate figure **(5.2)** shows the relationship between the thermal resistance and the overall heat transfer coefficient with input heat flux for thermosyphon pipe with angle 90° (vertical position), where the performance of a thermosyphon can be characterized by the thermal resistance R and overall heat transfer coefficient U. Where the rate of heat transfer to the evaporator and the heat sink from the condenser, and inversely proportional to the equivalent thermal resistance to heat transfer between the two regions **[18]**. The thermal resistance R_{total} can be defined as follows **[19]**:

$$R_{total} = \frac{T_{e.m} - T_{c.m}}{Q_{in}} \; \dots\dots\dots\dots\dots (10)$$

where; Q_{in} rate of input heat. $T_{e.m}$ and $T_{c.m}$ are the mean evaporator temperature and the mean condenser temperature respectively.

Different heating power inputs have been used to investigate the performance of a two-phase closed thermosyphon. The behavior of the thermal resistance began to decrease with the increasing of the input heat flux even reaching the lower value of thermal resistance at the highest energy input for each angle.

The overall heat transfer coefficient U can be defined as follows **[15]**:

$$UA_{cs} = \frac{1}{R_{total}} \dots\dots\dots\dots\dots\dots\dots (11)$$

where; A_{cs} is a cross sectional area.

The behavior of the overall heat transfer coefficient is true, due to its inversely proportional with thermal resistance. Where the thermosyphon with higher input heat flux is more effective in the overall heat transfer coefficient compared to lower range of input heat flux **[19]**.

Figures **(5.3)** and **(5.4)** show the wall temperatures distribution for the thermosyphon pipe at difference inclination angles (30°, 60° and 90°). It can be seen that the maximum performance of the thermosyphon pipe is at angle 30°. This is due to the free convection which can be enhanced when the position of the pipe move from vertical to horizontal position then the fins reache from vertical channel case which lead to easy movment for boundary layer. The mean evaporator temperature at angle 90° is higher than 60° and 30°. At angle 30° the free convection from condenser is best than the other two angles, also the mean condenser temperatures has the same behavior of the mean evaporator temperature, where the maximum mean condenser temperature is at angle 90°. Also the mean condenser temperatures are increasing with increasing of heat input.

The effect of the inclination angle on the thermal resistance is shown in figure **(5.5)** the minimum values of thermal resistance are at 30°, this means the maximum thermal performance of thermosyphon pipe. The behavior of the thermal resistance also begins to decrease with increasing of the input power even reaching lower value of thermal resistance at highest input power. The reason of low heat thermal resistance with the increase in input heat flux is due to the enhancement of boiling techniques. The nucleate boiling activity improves in the evaporator section with increase in power input there by reducing the thermal resistance of the evaporator section hence reducing the thermal resistance of the thermosyphon. The improvement in boiling activity with increase in heat input, enhanced the performance of thermosyphon **[18]**.The overall heat transfer coefficient is influenced by the inclination angle, where figure **(5.6)** shows the behavior for the thermosyphon at angle (30°, 60° and 90°). The maximum value at angle 30°,since minimum value of thermal resistance at this angle.

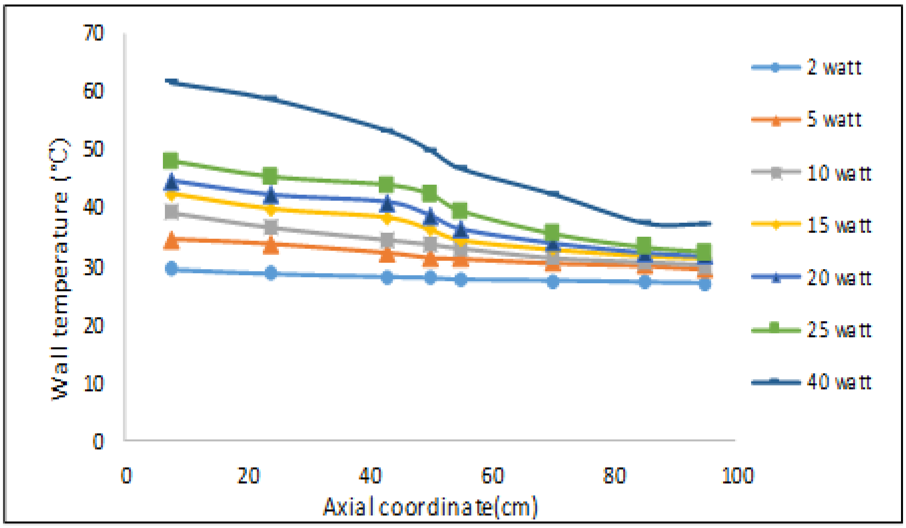

Figure 5.1: Temperature distribution along outside wall surface at angle 30° **.**

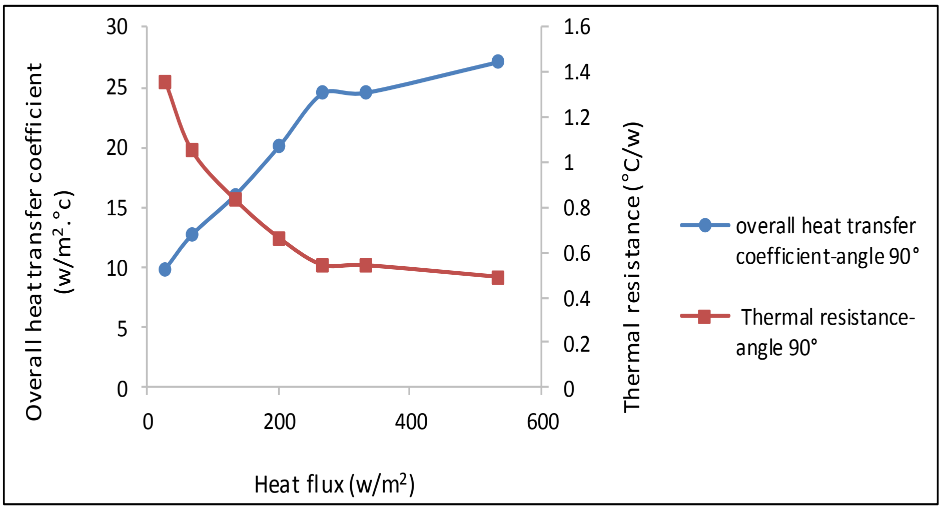

Figure 5.2: Variation of the thermal resistance and overall heat transfer coefficient vs. heat flux.

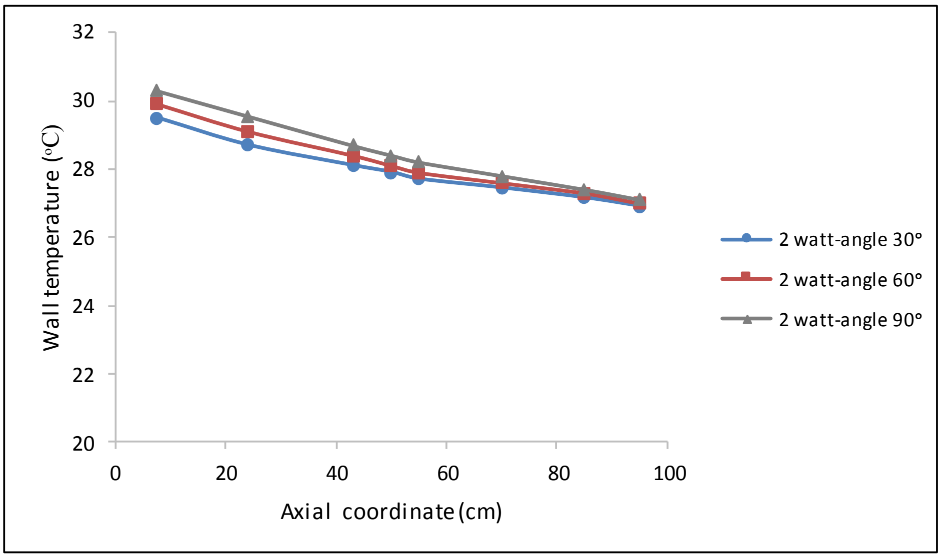

Figure (5.3): Wall temperature distribution along outside wall surface vs. axial coordinate**.**

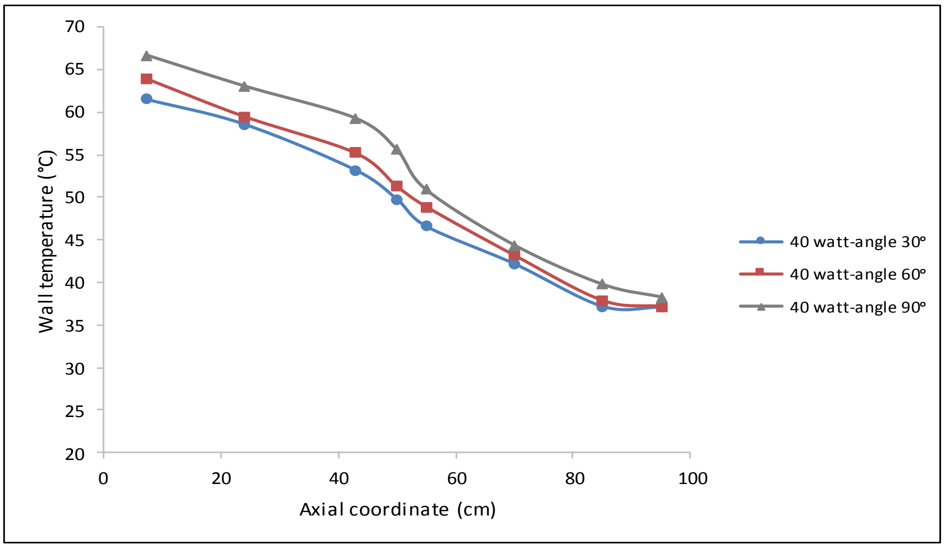

Figure (5.4): Wall temperature distribution along outside wall surface vs. axial coordinate**.**

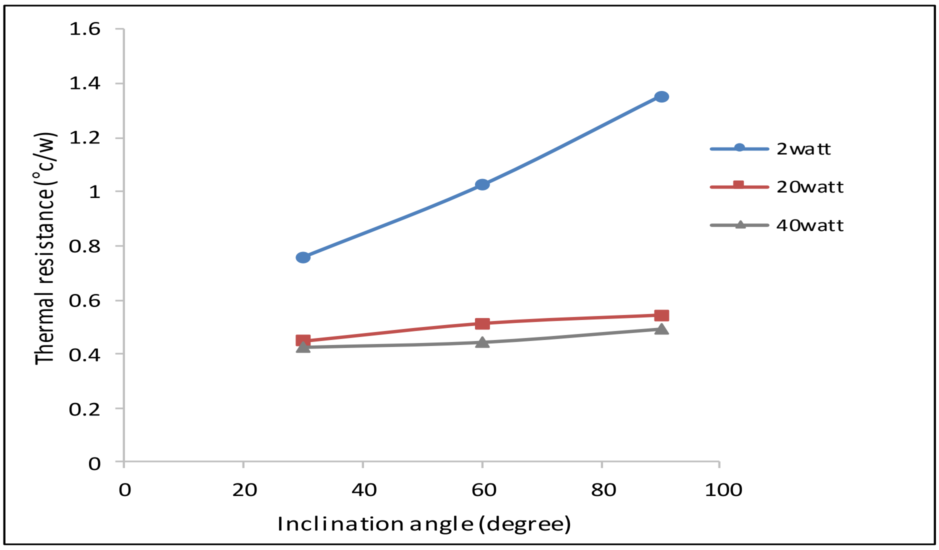

Figure 5.5: Variation of the thermal resistance v s. inclination angle.

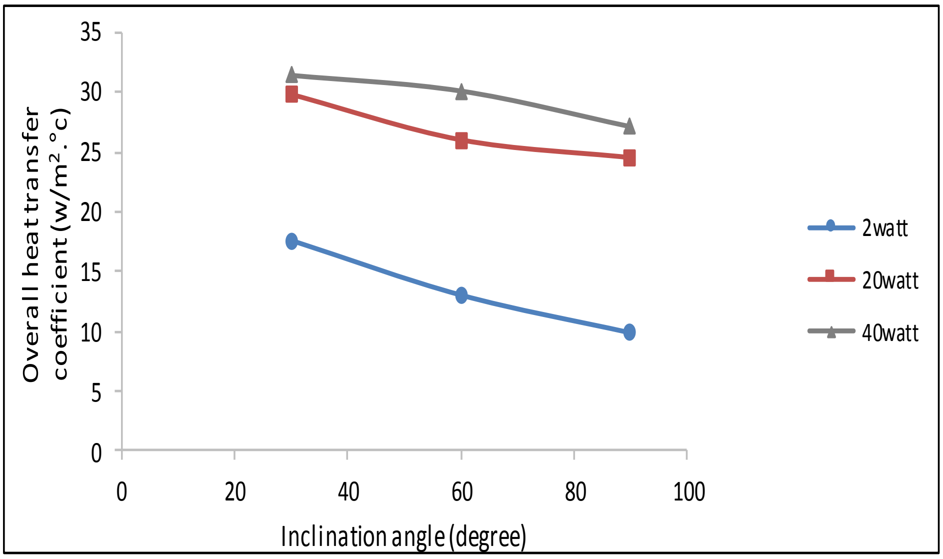

Figure 5.6: Variation of the overall heat transfer coefficient vs. inclination angle.

6. Conclusions

In the present study, experiments performed on a two phase closed thermosyphon had the inner and outer diameter with 4.9cm and 5.3cm respectively. The experiments carried out in order to investigate the effect of input heat at 2 W, 5 W, 10W, 15 W, 20 W, 25 W, 40 W and inclination angle at 30°, 60°, 90°. The working fluid used was deionized distilled water with filling ratio 50% from evaporator volume. The experimental results show that:

- ➢ The temperature distribution of the two phase closed thermosyphon is nearly uniformed in evaporator section and fell slightly toward the adiabatic section. The temperature reaches minimum value at the top end of the condenser section.
- ➢ The incline position at 30° of thermosyphon has higher performance than other positions 60° and 90°.
- ➢ Lower thermal resistance was 0.4247 ˚C/W found at 40watt heat input, consequently it has highest performance.
- ➢ The overall heat transfer coefficient is increased with increasing input heat flux.

7. References

[1] Faghri, A., Heat pipe science and technology, Taylor & Francis, Washington DC, USA, p. 341 (1995).

[2] Peterson, G.P., An introduction to heat pipes, modeling, testing and applications, John Wiley & Sons Inc., New York, USA, p.1 (1994).

[3] Asghar Alizadehdakhel, Masoud Rahimi, Ammar Abdulaziz Alsairafi." CFD modeling of flow and heat transfer in a thermosyphon". International Communications in Heat and Mass Transfer 37 312–318, (2010).

[4] Dussadee, N., Punsaensri, T., Kiatsiriroat, T., "Temperature control of paddy bulk storage with aeration.thermosyphon heat pipe", Energy Conversion and anagement, 48, 138 (2007) .

[5] Negishi, K. & Sawada, T. Heat transfer performance of an ITPCT. Int. J. Heat Mass Transfer, Vol. 26, No. 8, pp. 1207-1213, (1983).

[6] Wang, J. C. Y. & Ma, Y. (1991). Condensation heat transfer inside vertical and inclined thermosyphons. Journal of Heat Transfer, Vol. 113, pp. 777-780.

[7] Payakaruk, T., Terdtoon, P. & Ritthidech, S. Correlations to predict heat transfer characteristics of an inclined closed two-phase thermosyphon at normal operating conditions. Applied Thermal Engineering, Vol. 20, pp. 781-790, (2000).

[8] H.Z. Abou-Ziyan, A. Helali, M. Fatouh and M.M. Abo El-Nasr,"Performance of stationary and vibrated thermosyphon working withwater and R134a". Applied thermal engineering, Vol. 21, pp813-830, (2001)

[9] Yong Joo Park, Hwan Kook Kang, Chul Ju Kim. Heat transfer characteristics of a two phase closed thermosyphon to the fill charge ratio. International Journal of Heat and Mass Transfer. 45: 4655 – 4661, (2002).

[10] Samuel Luna Abreu and Sergio Colle," An experimental study of two-phase closed thermosyphons for compact solar domestic hot-water systems" available online at www.sciencedirect.com, Solar Energy 76 ,141–145, (2004).

[11] K.S. Ong and Md. Haider-E-Alahi, ―Performance of an R-134afilled thermosyphon‖. Applied thermal engineering, Vol. 23, pp 2373-2381, (2003).

[12] Abreu, S. L., and Colle, S., "An experimental study of two-phase closed thermosyphons for compact solar domestic hot-water systems", Solar Energy, 76, 141, (2004).

[13] S. H. Noie, M. H. Kalaei, M. Khoshnoodi," Experimental investigation of boiling and condensation heat transfer of a two phase closed thermosyphon" International Journal of Engineering Vol. 18, No. 1, 100-101, (February 2005).

[14] Incropera, Dewitt, Bergman and Lavine. Text book "Fundamentals of Heat ad mass transfer". Sixth edition, (2002).

[15] Yunus A. Çengel. Text book "Heat transfer ". Second edition, (July 2002).

[16] Rajput. Text book "Heat and mass transfer". Third edition.

[17] J.R. Taylor, An Introduction to Error Analysis: The Study of Uncertainties in Physical Measurements, University Science Books, Sausalito, 1997.

[18] Bandar Fadhl, Luiz C. Wrobel, Hussam Jouhara." Numerical modelling of the temperature distribution in a two-phase closed thermosyphon". School of Engineering and Design, Brunel University, Kingston Lane, Uxbridge, Middlesex UB8 3PH, UK. Applied Thermal Engineering 60 ,122-31, (2013).

[19] Nikhil E. Chaudhari, Nishtha Vijra, T. P. Singh." Computational fluid dynamics analysis of two-phase thermosyphon". International Journal of Engineering and Technology (IJET), Vol 5 No 5, ISSN: 0975-4024, (Oct-Nov 2013).

دراسة تجريبية من الخصائص الحرارية لأنابيب الثرموسيفون مع مكثف مزعنف

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الخلاصة

ان أنبوب الثرموسيفون هو من أدوات نقل الحرارة ثنائية الطور ويعتبر ناقل حراري كفوء جداً.في هذا البحث تم دراسة الاداء الحراري لانبوب ثرموسيفون ذي مكثف مزعنف (مكثف يبرد بالحمل الحر) تجريبياً. تم تصنيع أنبوب الثرموسيفون من مادة النحاس .أستخدم الماء المقطر كمائع تشغيل، حيث تم شحن الانبوب بكمية تعادل 50% من حجم المبخر. وقد ثبتت زعانف حلقية مصنوعة من الالمنيوم على طول السطح الخارجي للمكثف. تم أختبار أنبوب الثرموسيفون تجريبياً عند قيم مختلفة للقدرة الداخلة (2 واط و 5 واط و 10 واط و 15 واط و 20 واط و 25 واط و 40 واط) و زوايا ميل مختلفة (30° و 60° و 90° من الوضع الافقي). أظهرت النتائج أن زيادة القدرة الداخلة تؤدي الى زيادة درجة حرارة اشتغال الانبوب، بينما تنخفض المقاومة الحرارية مع زيادة القدرة الداخلة. وهذا يعني تحسن الاداء الحراري للانبوب مع زيادة القدرة الداخلة. أيضاً بينت النتائج أن الاداء الحراري للانبوب يكون افضل عند وضع الانبوب بزاوية ميل 30° عن الافق مقارنة مع زوايا ميل 60° و 90°.