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Experimental Investigation on the Heat Transfer Coefficient of the Thermosyphon Cross Section Shape

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Abstract
Two phase closed thermosyphon is a good heat transfer device. A large heat is transferred from evaporator to condenser with relatively a small temperature difference. In the present work, the heat transfer performance of two phase closed thermosyphon is analyzed experimentally with different cross section shape for the thermosyphon tube. A copper thermosyphon has been constructed with three different cross section shape (circular, square and rectangular) having the same hydraulic diameter and length. Methanol is used as the working fluid. The temperature distribution across the thermosyphon outer surface was measured and recorded using thermocouples. The results showed that the heat transfer coefficient increases with the increase of input power, thermal resistance is indirectly proportional to the input power. The maximum heat transfer coefficient (1815 W/m2C) for square cross section at the input power (500 W).

I. Introduction

Charging and discharging of large quantities of heat into or from small transfer areas is the aim of many investigators in the field of heat transfer. This object can be achieved by absorbing or liberating of latent heat into or from some substances, which are called phase change process. Thermosyphons and heat pipes are useful applications of such processes. Thermosyphon heat transfer has certain operating and limiting mechanisms that need to be considered before further discussing thermosyphon applications and technologies.

Over the last few years many studies have been devoted to heat transfer devices based on a recirculating mass which may be accompanied by phase change such as, heat pipes and single-and two phase closed or open thermosyphon.

Due to high fuel prices, it has become necessary to investigate new methods for saving and more efficient use of energy. For this reason, in the last five decades there has been an important technological development in heat transfer equipment, to promote changes in configuration and applying heat transfer systems with high effectiveness. One example is the use of two-phase thermosyphons (Reay [1]; Azada et. al. [2]; Peterson [3]; Faghri [4]; Gershuni et. al. [5]; Noie [6]).

Thermosyphon is a device with high thermal conductivity that can transfer high quantities of heat. In its most simple form, a thermosyphon is a hollow evacuated metal pipe, charged by a pre determined amount of an appropriate working fluid. It can be divided into three main sections: evaporator, where the heat is delivered to the device, an adiabatic section (which may or may not exist) and a condenser, where the heat is released. The working fluid located in the evaporator, evaporates and go toward the condenser region, where it condenses, returning to the evaporator by means of gravity.

Grover [7] (Los Alamos Laboratory, USA) introduced the term heat pipe in 1964. The two-phase closed thermosyphon used in this study is essentially a gravity-assisted wickless heat pipe, which is very efficient for the transport of heat with a small temperature difference via the phase change of the working fluid. It consists of an evacuated-closed tube filled with a certain amount of a suitable pure working fluid. The simple design, operation principle, and the high heat transport capabilities of two-phase closed thermosyphons are the primary reasons for their wide use in many industrial and energy applications.

The thermosyphon is simpler in construction because that is no wick material,, smaller in thermal resistance, and wider in its operating limits than the wicked heat pipe.

It is well known that the high performance of the two-phase heat transfer system is attributed to the fact that it utilizes mainly the heat transfer characteristics of the phase change which are more efficient means of heat transfer than those of single phase. A two-phase open thermosyphon, which utilizes buoyancy force and phase change of liquid contained in a vertically long tube may be considered to be one of the convenient devices to achieve this, because no power supply or pumps are required to circulate the working fluid. Furthermore, the low cost and simplicity of the construction of such thermosyphon device are quite advantageous.

There are a number of engineering applications for thermosyphons such as, utilization of the geothermal energy in power generation, in snow melting in the northern cold districts, also in cooling of, gas turbine blades, electrical machine rotors, electronic components cooling, electric engine cooling, anti ice protection, transformers, nuclear reactors and internal combustion engines, and solar house heating or cooling.

The above mentioned engineering applications make thermosyphons as a wide research field, and a detailed knowledge of its heat transfer characteristics is required for design purposes.

II. Review of the previous work

In literature, single phase, and two phase heat transfer characteristics of open or closed tube thermosyphon were investigated experimentally and analytically.

Single-phase heat transfer characteristics of concentric tube open thermosyphon were investigated by Ski, Fukusako, and Koguchi [8]. The experiments were carried out with water and (R-11) refrigerant as working fluids. They concluded that the heat transfer coefficient of concentric tube thermosyphon increases by factors as large as 2 to 10 comparing with those of the single tube open thermosyphon.

Two phase thermosyphon working at low pressure with water as a working fluid was investigated by casarosa, et al. [9]. They found that the process of boiling is rather complicated, and at low pressure, boiling occurs with bubbles of large diameter. Large time intervals separate the nucleation of different bubbles, though such intervals decrease when the specific thermal flux rises. They also noted that, at low pressure (less than 0.15 bar), the overheating between wall and fluid is unaffected by variation in the specific thermal flux, the correlation which link the heat transfer coefficient with the working pressure and the specific thermal flux was derived.

Lee and Mital [10], studied the heat transfer performance of a two-phase closed thermosyphon. The effect of working fluid quantity, heated length to cooled length ratio, mean operating pressure, heat flux, and working fluid, on the performance of the thermosyphon was investigated. They found that: the heat transfer coefficient is not sensitive to the quantity of the working fluid above a certain amount of the working fluid, decreasing the heated length to cooled length ratio increases heat transfer coefficient, the heat transfer coefficient is very sensitive to the operating pressure and rapidly increases with increasing pressure, and a theoretical analysis for the maximum heat transfer rate predicts the trends closely and the agreement with experimental results for water and refrigerant (R-11) is good.

Negishi and Sawada [11] made an experimental study on the heat transfer performance of an inclined two-phase closed thermosyphon. They used water and ethanol as working fluids. The highest heat transfer rate was obtained when the filling ratio (ratio of volume of working fluid to volume of evaporator section) was between 25% and 60% for water and between 40% and 75% for ethanol. The inclination angle was between 20° and 40° for water, and more than 5° for ethanol.

Park, Kang and Kim [12] studied the heat transfer characteristics in thermosyphon depending on the amount of working fluid and when the operation limits occur. The two-phase element was made of copper and as working fluid FC-72 (C6 F14) was used. The thermosyphon was subjected to a heat supply in the range of 50-600 W and with FR 10-70%. For the heat transfer coefficients in the condenser and in the evaporator, the authors used the theory of Nusselt and Roshenow respectively. They found that the operation limits manifest in different forms depending on the filling ratios of the fluid. For small filling ratio (FR = 10%) the drying limit occurs in the evaporator, while for high filling ratio (FR = 50%) the flooding limit appears. In the first case, evaporator temperature increases from the evaporator bottom; in the second case the evaporator temperature increases at the top of the evaporator. These conclusions were made by observing the temperature distribution along the thermosyphon.

Zuo and Faghri [13] conducted an analytical and experimental research on the thermodynamic behavior of the working fluid in a thermosyphon and a heat pipe, using a temperature-entropy diagram. The authors divide the thermodynamic processes into two categories: 1) heat transfer by conduction through the tube wall and 2) heat and mass transfer, by convection inside the two-phase thermosyphon.

Noie [6] presented in his work an experimental study of a thermosyphon of (980 mm length and 25 mm internal diameter) made of smooth copper tube, with distilled water as a working fluid. The goal of the study was model when the liquid film and liquid pool are continuous, based on the empirical heat transfer correlations from his experimental results. It can only determine the temperature distribution at the inside wall of thermosyphon, and cannot confirm the location of liquid film and liquid pool inside the thermosyphon.

Karthikeyan, Vaidyanathan and Sivaraman [14] investigated experimentally the thermal performance of two phase closed thermosyphon with various working fluid. They found that the maximum thermal performance than compared to thermosyphon charged with distilled water.

Anjankar and Yarasu [15] made Experimental study of Condenser Length Effect on the Performance of Thermosyphon. They used the highly efficient thermal transport process of evaporation and condensation to maximize the thermal conductance between a heat source and a heat sink and they also studied the new design and thermal performance of thermosyphon.

Experimental investigations of thermosyphons show that the heat transfer coefficient depends mainly upon the following parameters: heat flux, operating pressure of the working fluid, properties of the working fluid, the void ratio and thermosyphon tube geometry.

Therefore, It is purpose of the current study to investigate heat input, condenser coolant flow rate and different cross section shape on the temperature distribution on the surface and the heat transfer coefficient.

III. The Experimental Setup and Instrumentation:

An experiment investigation of the characteristics of heat transfer for thermosyphon with different cross section shape was carried out.

This section deals with two main parts; the first part is description of setup and the second part is the instrumentation and measuring devices used in collecting the experimental data.

- Experimental Setup Description:

The experimental setup is shown schematically in Fig. (1). The main components of the setup are as follows: (thermosyphon tube, electric heater, variac, cooling water tank, flow meter, vacuum pump, thermocouples, digital clamp meter and data logger). A closed copper tube with total length of 540 mm and three different cross sections has same hydraulic diameter 16 mm (circular, rectangular and square cross sections) as shown in Fig. (2). The evaporation section length 180 mm and energy was transferred to it by an electrical heater with a predefined power input. The heat input to the evaporator was set using an electrical energy regulator (Variac). The condenser section length 240 mm that surrounding the upper region of the thermosyphon, and cooling water was introduced to it in upward direction. The flow rates of the cold water were controlled by a valve measured by a flow meter with an accuracy of $\pm 0.3\%$ of full scale. A length of 120 mm is adiabatic section between both of evaporator and condenser sections. Ten thermocouples were placed on the thermosyphon tube to measure the temperature of evaporator, adiabatic and condenser section surfaces, two thermocouples were used to measure the inlet and outlet temperatures of the cooling water as shown in Fig. (2). The thermocouples used in the present experiment had a measurement error of ± 0.5 C.

- Instrumentation and Measuring Devices:

The setup was instrumented to perform the following measurements (temperature measurements, flow rate measurements, voltage and current measurements). All temperature measurements within the setup were measured by using copper-constantan thermocouples.

The local temperature along the outer surface of the thermosyphon tube, inlet and outlet of cooling water were measured by ten thermocouples. All the wires of the ten thermocouples were the same length, a grounded sheathed extension copper leads of the same length were also used. Therefore the all slopes (C/mV) were almost the same for all thermocouples.

The flow rate of the circulating cooling water is measured by a flow meter with an accuracy of $\pm 0.3\%$ of full scale.

The power supplied to the electrical heater is determined monitoring the supplied voltage and current using DT266C 3 1/2 DIGITAL CLAMP METER with voltage accuracy is ± 0.8% and the current accuracy of ± 3%.

IV. Experimental Work analysis:

The experiments are performed under different thermosyphon cross section shape to determine its effect on the heat transfer coefficient. The experimental program consists of different test runs while varying one of the following parameters:

1. Thermosyphon cross section shape (circular, rectangular and square cross sections).
2. Power applied to the evaporator heater, ranging from 200 W to 500 W.
3. Cooling water rates from 0.8 lit/min to 1.5 lit/min.

- Mathematical Equations Used:

- Hydraulic diameter (D_h):

$$D_h = \frac{4A}{P} \qquad (mm) \tag{1}$$

For circular cross section:

$$D_h = \frac{4 \times \pi \times r^2}{2 \times \pi \times r} = 2r = 16 \quad mm \tag{2}$$

For rectangular cross section:

$$D_h = \frac{4 \times 24 \times 12}{2 \times (24 + 12)} = 16 \quad mm \tag{3}$$

For square cross section:

$$D_h = \frac{4 \times 16 \times 16}{4 \times 16} = 16 \quad mm \tag{4}$$

- Hydraulic area (A_h):

$$A_h = \pi \times D_h \times L \qquad (m^2) \tag{5}$$

- Input heat transfer rate (Q_{in}):

Where: Q_{in} input heat energy to evaporator

$$Q_{in} = V \times I \quad (W) \tag{6}$$

- Output heat transfer rate (Q_{out}):

Where: Qout is outlet heat energy from condenser which can be calculated from cooling water mass flow rate, inlet and outlet cooling water temperature according to the following equation:

$$Q_{out} = \dot{m} \times c_p \times (T_o - T_i) \quad (W) \qquad (7)$$

- Average heat transfer rate ($Q_{av.}$):

Where: $Q_{av.}$ is heat transfer rate which calculated from the following equation:

$$Q_{av.} = \frac{Q_{in} + Q_{out}}{2} \quad (W) \qquad (8)$$

- Thermal resistance ($R_{th.}$):

$$R_{th.} = \frac{(T_e - T_v)}{Q_{av.}} \quad (^oC/W) \qquad (9)$$

- Heat transfer coefficient (h):

$$h_{\exp.} = \frac{Q_{av.}}{A_h (T_e - T_v)} \quad (W/m^2C) \qquad (10)$$

- Heat Transfer Limitations:

The maximum rate of thermosyphon heat transfer is limited due to various parameters. These limiting parameters as the following:

1. Dry-out Limit: The dry-out limit occurs at the bottom of the evaporator in the liquid falling film mode. This limit prevails for very small liquid fill charges and relatively small radial evaporator heat fluxes.
2. Boiling Limit: The boiling limitation occurs at the large liquid fill ratio and high radial heat fluxes. Under this limitation, at the critical heat flux, vapor bubbles coalesce near the pipe wall prohibiting the contact of working liquid to wall surface, resulting in the rapid increase in evaporator wall temperature.
3. Counter Current Flow (Flooding)Limit: The counter current flow limitation is one of the most important and common limitations found in closed two-phase thermosyphon with large liquid fill ratio, large axial heat fluxes and small radial heat fluxes. The vapor shear prevents the condensate from returning to the evaporator and leads to a flooding.

V. Results and Discussion:

- Temperature distribution along thermosyphon tube:

The experimental results indicated that the temperature was nearly uniform and fell slightly toward the adiabatic section because the energy losses may be occurs in adiabatic section. The temperature reached at the condenser section bottom part the minimum value (inlet of cooling water) and then increased at the condenser section bottom part (outlet of cooling water).

- Effect of the input power:

Fig. (3) to Fig. (5) shows thermosyphon tube distance versus its surface temperature along the thermosyphon tube at three different rates of cooling water and input heat transfer and thermosyphon cross section shapes. Data were recorded when the system steady state was reached. When the input power is small (200 W), the evaporator surface temperature is almost equal as shown in Fig. (5). Then the evaporator top part temperature gradually rise with rise of input power (500W) and with decrease of cooling water rate (0.8 lit/min) to becomes higher than the bottom part of the evaporator in order to increase the amount of bubbles at the to, as shown in Fig. (3).On the other hand, and generally increases the evaporator surface temperature with the increase of input power, due to the weakness of the effect of thermal resistance, as is the amount of change in the of the evaporator surface temperature (6-9 C) is influenced by the condensate liquid which returns to the evaporator. When input power increases the heat transfer coefficient increases as a result of the occurrence of bubble boiling. As well as the figures show the temperature distribution along the surface of the condenser was lower value and this decline as a result of the thermal resistance of the expected thermal conductivity on the surface of the tube length and cause of both boiling and condensation of the working fluid. The difference of evaporator, adiabatic and condenser parts are due to increasing of cooling water flow rates.

- Effect of the heat transfer coefficient:

The nature of the surface and the geometric shape of the evaporator and condenser section a major impact on the process of heat transfer between the surface and fluid operation as well as the boiling process within the evaporator. Thermal conductivity of the tube and the interactions between the surface of the tube on the one hand, liquid and vapor on the other hand as adhesion and portability wetness has a significant effect on the process of heat transfer. Since the heat transfer coefficient in the evaporator increases with the decrease in contact angle vapor bubbles with the heating surface and this is due to the increase in contact between the working liquid and the surface of the evaporator, which is working to reduce the thermal resistance, and the opposite in the condenser. Figs.(6,7) show that the best value obtained for the heat transfer coefficient for the thermosyphon square shape, followed by thermosyphon rectangle shape and then the circular shape because of the increased thermal resistance where the heat transfer coefficient is inversely proportional to the thermal resistance.

- Effect of the cooling water rate:

Fig. (8) shows the cooling water flow rate which uses to cool the condenser versus a thermosyphon thermal resistance, Noting that the relation between the cooling water flow rate and both of input power and thermosyphon cross section shape is linear relationship. The increase in the cooling water flow rate reduces the thermosyphon surface temperature about(6-9 C) and that because of the thermal resistance, and note the thermal resistance almost constant at each thermosyphon cross section shape and not directly dependent on the cooling water flow rate. And that the thermal resistance of the square thermosyphon is the lower of the other cross section, and thus got the best performance of the heat transfer.

VI. Conclusion

The effect of various input power, cooling water flow rate and different thermosyphon cross section shape on the thermosyphon performance were investigated in this work. The following results were obtained:

- Evaporator surface temperature almost equal at lower input power, but start gradually rise with the increase of input power.
- Heat transfer coefficient increases with the input power.
- Experimentally, maximum heat transfer coefficient in this work is (1815 W/m2C) for square cross section at input power (500 W).
- Thermal resistance is indirectly proportional to the input power, i.e., thermal resistance decrease with the increase of input power.
- Thermal resistance almost constant for all cross section shape and does not depend on cooling water flow rate.

Nomenclature:

Symbol	Definition	Unit
A	Area	(m2)
cp	Specific heat at constant pressure	(j/kg °C)
Dh	Hydraulic diameter	(m)
h	Heat transfer coefficient	(W/ m2 °C)
L	Length of tube	(mm)
m	Cooling water flow rate	(lit/ min)
p	Circumference	(m)
Q	Heat transfer rate (input power)	(W)
q	Heat flux	(W/m2)
R	Heat resistance	(°C /W)
T	Temperature	(°C)

Subscripts

av.	Average
c	Condenser
e	Evaporator
exp.	Experimental
v	Vapor
in	Input
out	Output
th.	Thermal

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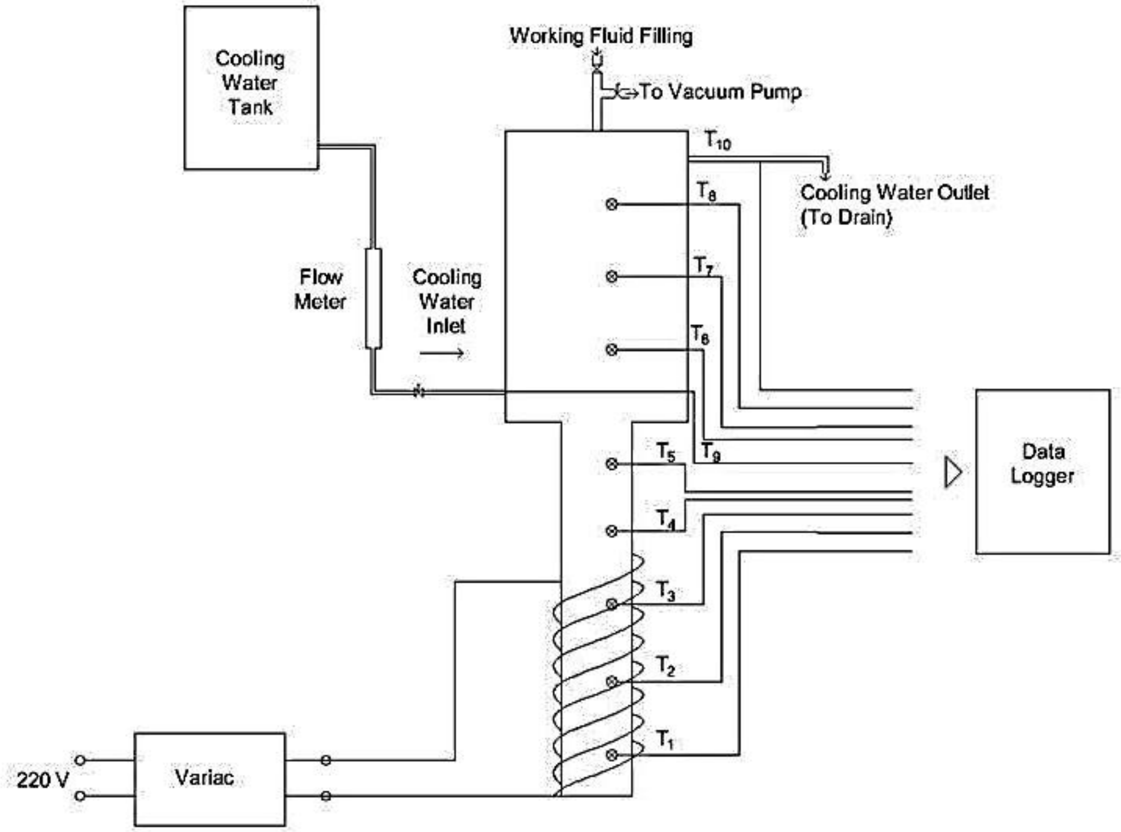


Fig. (1) Schematic diagram of an Experimental Setup

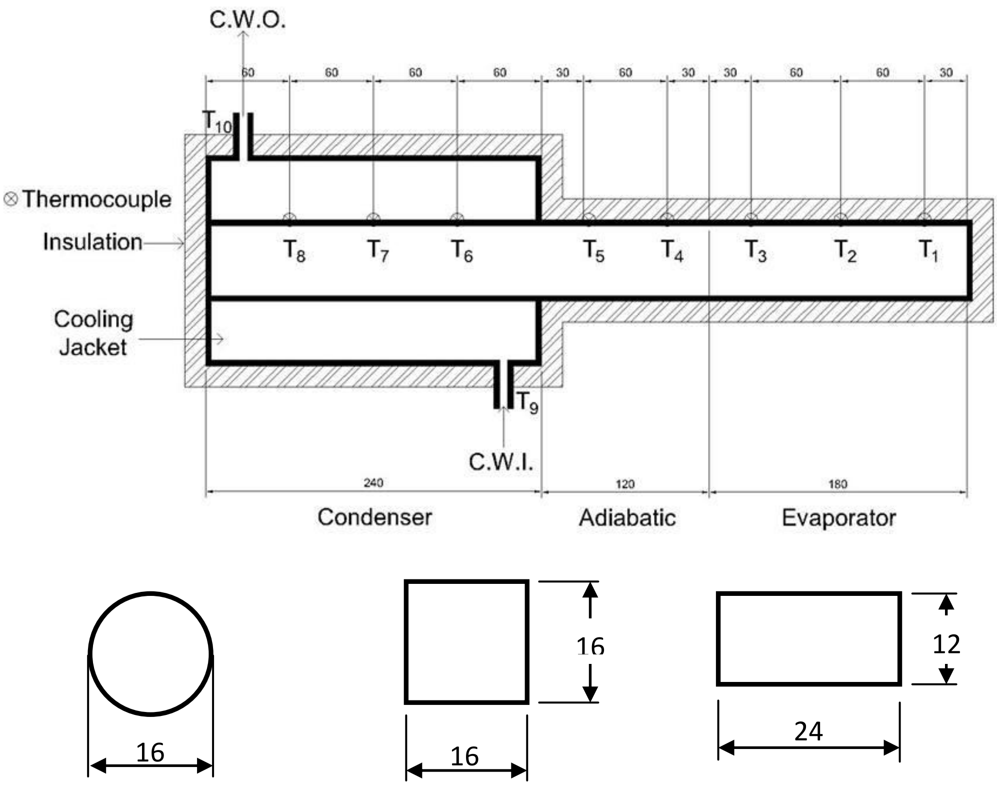
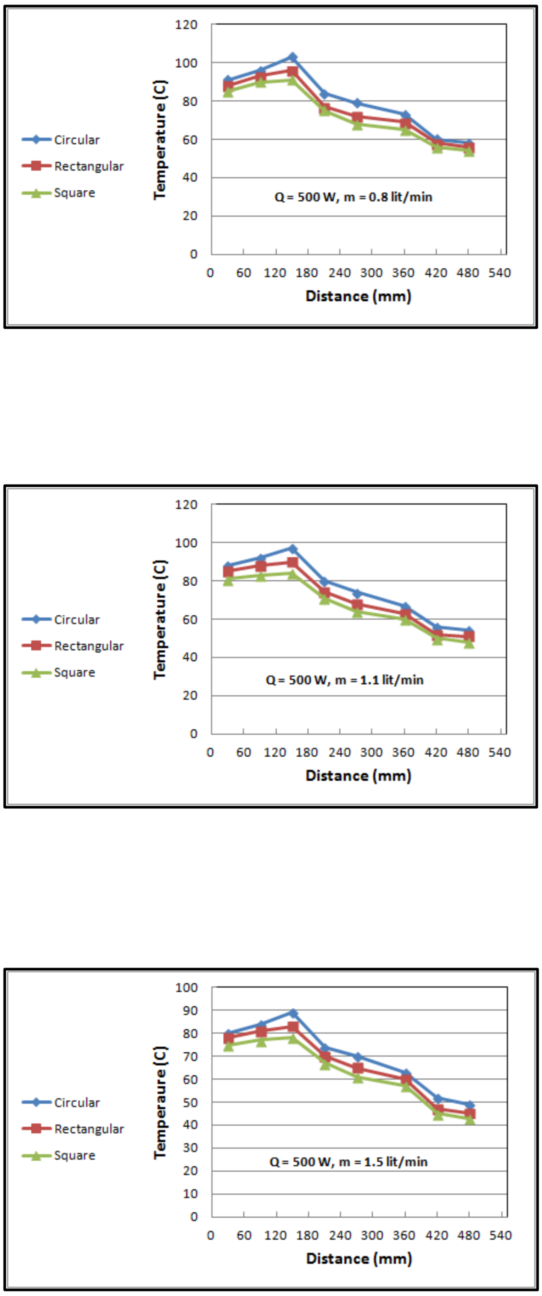


Fig. (2) Schematic diagram of a two phase thermosyphon details



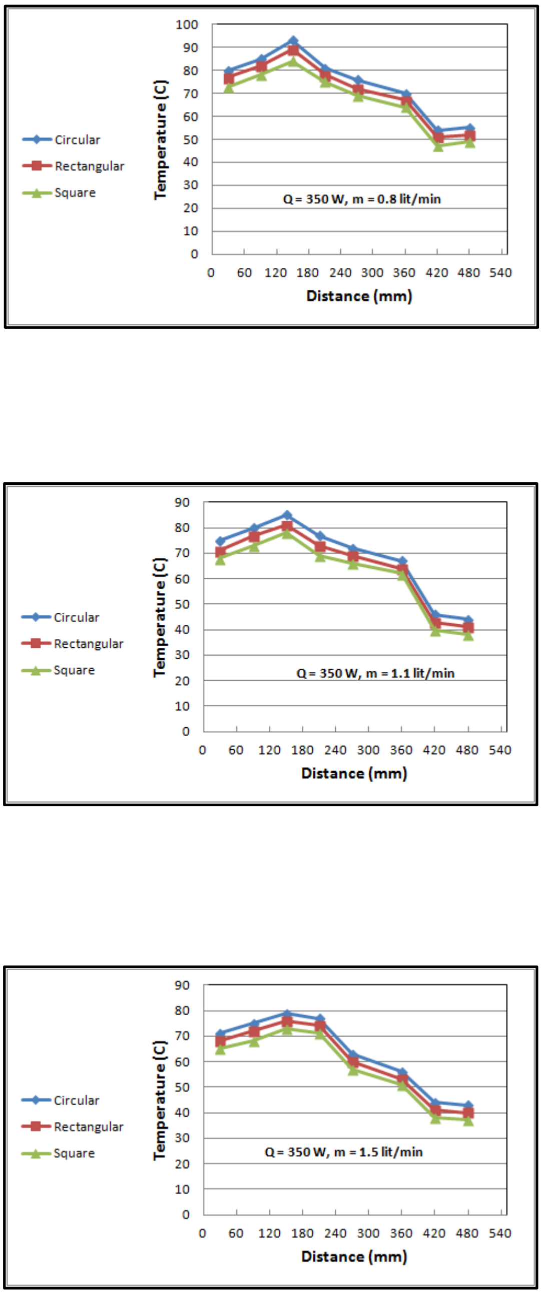
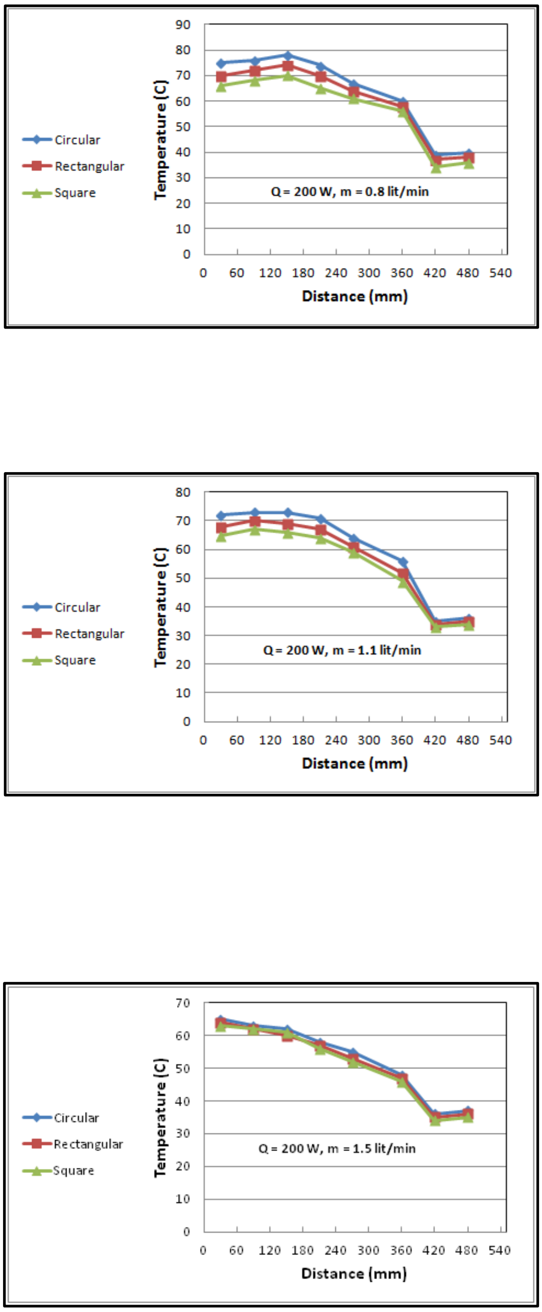
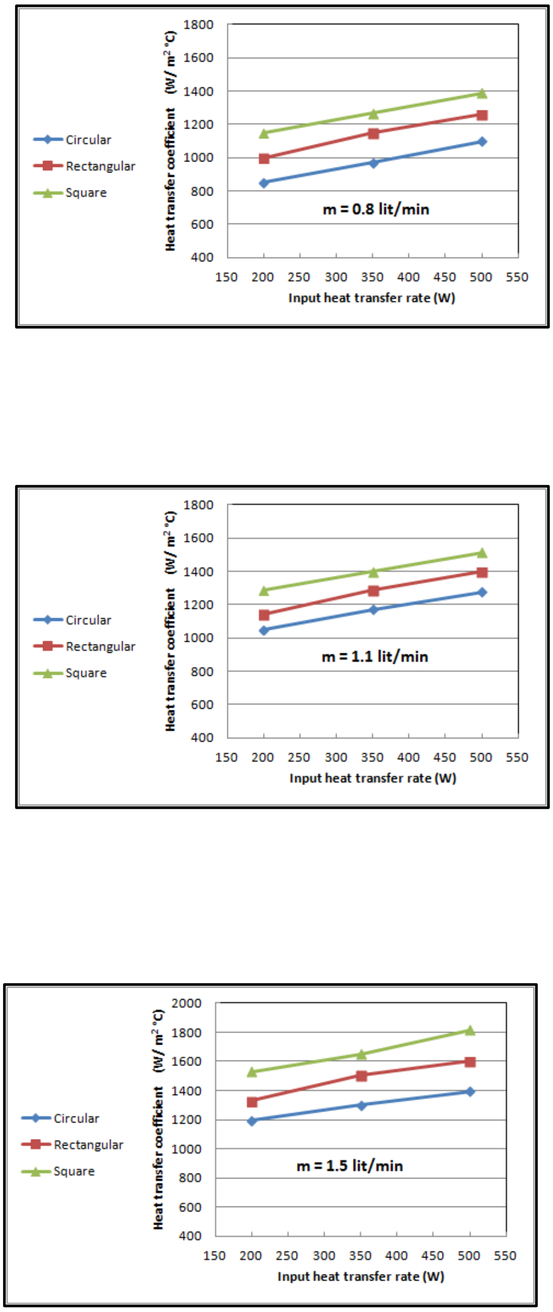


Fig. (3) Temperature distributions along the outside wall surface of thermosyphon at input power (500 W) and three cooling water flow rates and different cross section

Fig. (4) Temperature distributions along the outside wall surface of thermosyphon at input power (350W) and three cooling water flow rates and different cross section

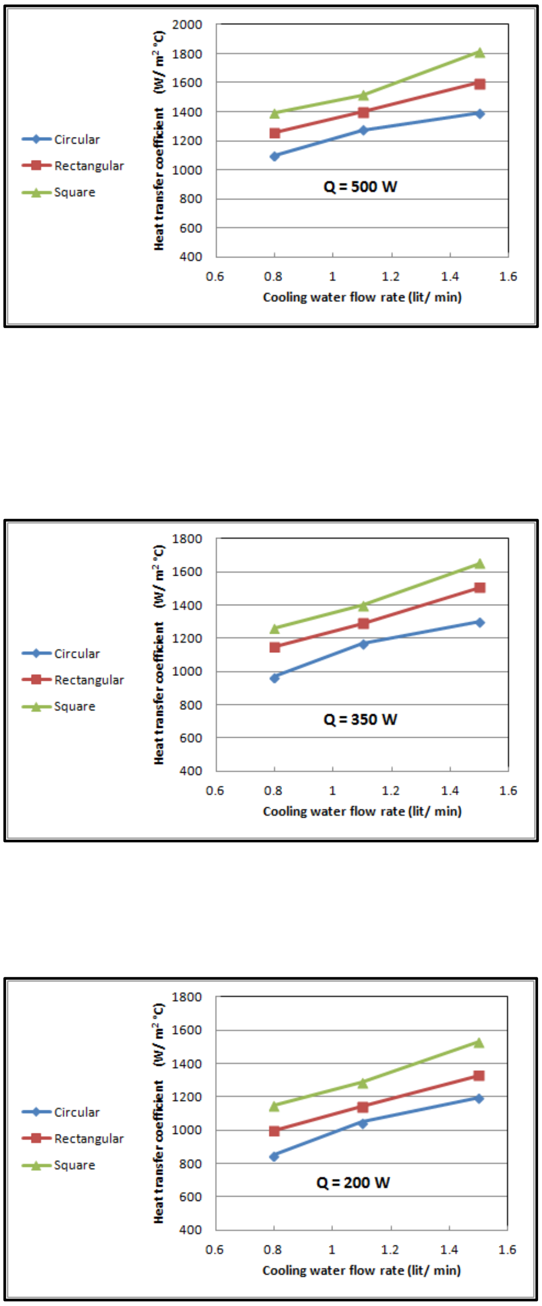




W) and three cooling water flow rates and different cross section

Fig. (6) Variation of heat transfer coefficient with input power at different cooling water flow rates

Fig. (5) Temperature distributions along the outside wall surface of thermosyphon at input power (200



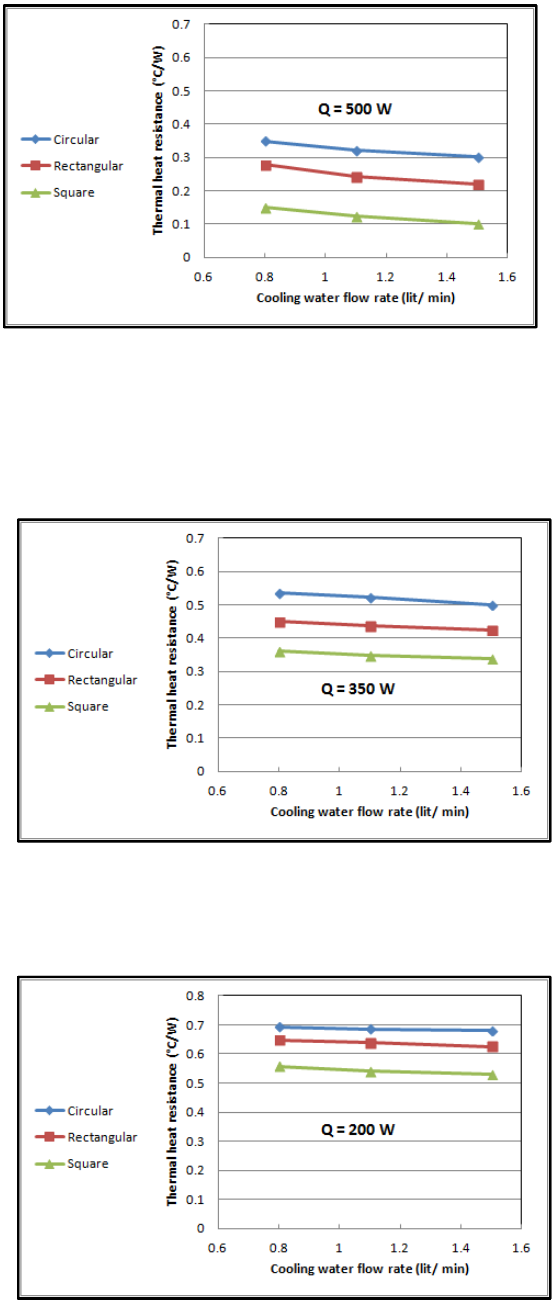


Fig. (8) Variation of thermal resistance with cooling water flow rates at different input power

Fig. (7) Variation of heat transfer coefficient with cooling water flow rates at different input power