Inclination and Fill Ratio Effects on Water Filled Two-Phase Closed Thermosyphon

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ABSTRACT

An experimental investigation was conducted to determine the effects of inclination and fill ratio on the performance of a water-filled two-phase closed thermosyphon at low evaporator temperatures below 65°C. The evaporator section was heated by resistance band heaters. The condenser section consisted of a concentric pipe water-cooled jacket. Experiments were carried out with fill ratios between 0.25 and 1.0 and at angles of inclination between 30 – 90 degrees from the horizontal. Evaporator power input was varied from 304 – 830 W. In order to maintain uniform cooling at the condenser section, coolant water mass flow rate was kept as high as possible. The effects of evaporator power input, inclination and fill ratio were determined. The performance of the thermosyphon was found to be independent of fill ratio and inclination within the limits of the experimental investigation. The mean evaporating and condensing heat transfer coefficients and the relationship between thermosyphon heat transfer and operating temperature difference between evaporator and condenser were determined.

Keywords: *Thermosyphon, evaporator power input effect, inclination effect, fill ratio effect, evaporating and condensing heat transfer coefficients, overall thermal resistance.*

1. INTRODUCTION

Heat pipe heat exchangers (HPHEs) are very effective heat exchangers employed to transfer large amounts of heat through long distances without parasitic power for its operation. They are becoming popular in waste heat recovery and in heating, ventilation and air conditioning (HVAC) systems to increase the dehumidification efficiency and cooling capacity of the cooling coil. In a compact air handling unit (AHU), the HPHE has to fit within the existing system ducting. The HPHE has to be fitted nearly horizontally in-line before and after the cooling coil of the AHU. Wickless heat pipes (thermosyphons) are employed because of their simplicity and low manufacturing cost. However, they can only operate with the condenser section higher than the evaporator section to allow for the condensate to return to the evaporator. Hence the performance of the HPHE could be affected by the inclination of the unit. As a direct consequence of the tilt, the upper level of the fill liquid could flow into the adiabatic section at high fill ratios. Hence fill ratio could affect its performance.

Hahne and Gross [1] showed that an R115 filled thermosyphon performed best when it is tilted at an angle of around 50 degrees to the vertical. The amount of liquid in the thermosyphon was such that the evaporator was always filled with liquid. Negishi and Sawada [2] investigated the interactive influence of the inclination angle (-10 to 90 degrees from the horizontal) and fill ratio (5-100%) of an inclined two-phase closed thermosyphon without an adiabatic section. Heating and cooling were performed by water jackets surounding the evaporator (up to 85°C) and condenser (25°C) sections. They showed that for high performance it is necessary to have a fill ratio of between 25 – 60% for water and between 40 – 75% for ethanol and tilt angles between 20 – 40 degrees for water and more than 5 degrees for ethanol. They determined that the overall heat transfer coefficient of the water thermosyphon was between $2.4 \times 10^3 – 3.0 \times 10^3$ W/m^2 K and for the ethanol thermosyphon, between $0.9 \times 10^3 – 1.1 \times 10^3$ W/m^2 K. Shiraishi et al. [3] investigated the influence of aspect ratio, fill charge, fill liquid and operating pressure of six inclined two-phase closed thrmosyphons and determined a new dimensionless parameter called a modified Kutateladze number to correlate the data of the maximum critical heat transfer rate based on the critical heat transfer rate at the vertical position. Terdtoon et al. [4] concluded that the performance of R-22, ethanol and water-filled thermosyphons depended upon aspect ratio and inclination angles. For better performance, angles should lie in the range from 40 – 70^o from the horizontal and the critical apect ratio is 10. Payakaruk et al. [5] considered the effects of Bond number, Froude number, Weber number and Kutateladze number on the performance of an inclined thermosyphon with R22, R134a, ethanol and water as the working fluids. Experiments were conducted with fill ratios between 0.5 - 1.0, aspect ratios between 5 – 40, vapor temperatures between 0 – 30^{o}C and inclination angles from 0 – 90 degrees. Their results showed that performance was dependent upon type of fluid but not filling ratio. Nitipong et al. [6] determined the performance limit of an inclined two-phase closed thermosyphon filled with ethanol, R113 and R123 and showed that Bond number and aspect ratio had no effect on the performance. Shalaby et al. [7] found that for an R-22 filled thermosyphon at low heat transfer rate between 100 – 300W, filling ratio between 30 – 100% and inclination angles between 22.5 – 90 degrees, the optimum filling ratio was 50% and best inclination was 30^o. Nguyen-Chi and Groll [8] investigated the grooved thermosyphon with various fill ratios and inclinations and showed that operating temperature also affected its performance. Gurses et al. [9] showed that water-filled brass-wicked heat pipes are strongly dependent on inclination and heat source temperatures with inclination angles between 45 – 90 degrees being most effective. Beckert and Herwig [10] showed that a wicked heat pipe heat exchanger filled with R22 could operate in a nearly horizontal position. Said and Akash [11] showed that a wicked water filled heat pipe performed better than one with no wick. Loh et al. [12] showed that wick structure affects the performance of heat pipes.

From the literature survey it could be seen that there is an inter-relationship between fill ratio, inclination angle, aspect ratio and operating temperature for the optimum performance of the thermosyphon. Further, the evaporating and condensing heat transfer coefficients in a single pipe thermosyphon are required in order to predict the performance of an array of thermosyphons connected together to form a heat pipe heat exchanger.

2. OBJECTIVE

The objectives of the present experimental investigation are to determine the effects of evaporator input, inclination and fill ratio on the performance of a water filled two-phase closed thermosyphon operating at low temperatures and to obtain the evaporating and condensing heat transfer coefficients in order to present a theoretical model for a heat pipe heat exchanger for air conditioning application.

3. EXPERIMENTAL INVESTIGATION

Dimensions of the 38 mm O/D x 32 mm I/D copper pipe thermosyphon are shown in Fig. 1. The experimental set-up is shown in Fig. 2. Evaporator, adiabatic and condenser sections were 405, 110 and 327 mm long, respectively. Electrical band heaters wrapped around the evaporator section provided up to 1000W of electrical heating via a transformer. Power to the band heaters was measured using a voltmeter and ammeter. Coolant water was circulated in the 71 mm I/D concentric copper water jacket condenser of the thermosyphon using a water pump. Temperatures were measured with Type-T (Cu-con) thermocouples (accuracy $\pm$ 0.5^{o}C). The thermocouples were tied onto the pipe wall by wire to measure surface temperatures. The condenser wall thermocouples were insulated from the coolant water stream using water proof tape to isolate the cooling effects of the coolant water on the readings. Saturation temperature was measured with another thermocouple inserted into the thermosyphon. Two thermocouples located at the inlet and outlet of the jacket measured coolant temperatures. All temperatures were logged on a data logger and displayed on a PC. The thermosyphon was also fitted with a pressure gauge to measure the saturation pressure inside during operation. Water flow rate was measured with a float-in-glass type flow meter with an instrument quoted accuracy of $\pm$ 2%. In order to obtain as near a uniform cooling in the condenser section as possible a high water flow rate of 0.0883 kg/s was employed. As a consequence of the high coolant flowrate employed, the temperature rise from water inlet to outlet was less than 1^{o}C. By assuming that the thermosyphon was perfectly insulated and heat loss from the thermosyphon to the ambient was neglected, heat transfer from evaporator to condenser section (P) was thus equal to the power input.

The thermosyphon was first charged with water and then fully vacuumed down to 0.001 mm Hg vacuum pressure. Experiments were conducted with fill ratios (defined as the ratio of the volume of fill liquid to the evaporator section volume) between 0.25 and 1.0, inclination angles from 30 to 90 degrees from the horizontal and at power inputs varying from 304 – 830 W. There was no

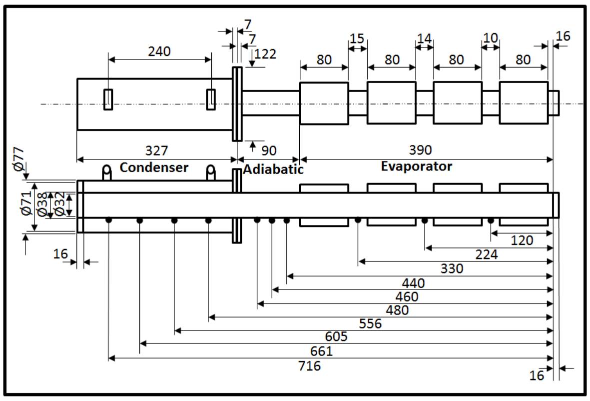

Fig. 1. Details of thermosyphon.

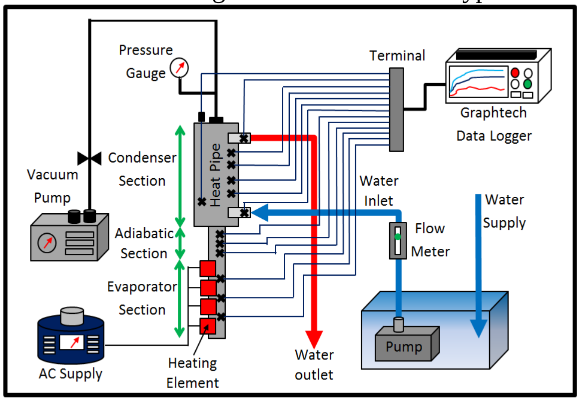

Fig. 2. Experimental set-up.

control over the coolant inlet temperature which varied between 28^o to 29^{o}C. Temperatures were logged every minute. Readings showed that even after about an hour, temperatures still fluctuated about $\pm$0.8^{o}C. Pressure fluctuations ($\pm$ 0.02 kPa at low fill ratio to $\pm$ 0.01 kPa at high fill ratio) were observed. Due to the pressure fluctuations, the difference between the measured saturation temperature and the value from the pressure-temperature chart amounted to about 2^{o}C. Each run was repeated three times in order to check for repeatability or consistency of results. It was found that the temperatures obtained were repeatable to within 2^{o}C. Average values were then calculated from the results of three runs.

4. RESULTS

4.1 Effect of evaporator power input (Fig. 3)

Typical axial temperature distributions at a fill ratio of 0.5 and 90 degree inclination are shown in Fig. 3 at various power inputs. Wall and saturation temperatures increased with evaporator power input as expected. Generally, the wall temperature was found to be highest near the lower end of the evaporator section. The temperature then decreased towards the adiabatic section. The condenser wall temperature varied about 1^o – 2^{o}C along the length of the section. Coolant water temperature rise from inlet to outlet was less than 1^{o}C.

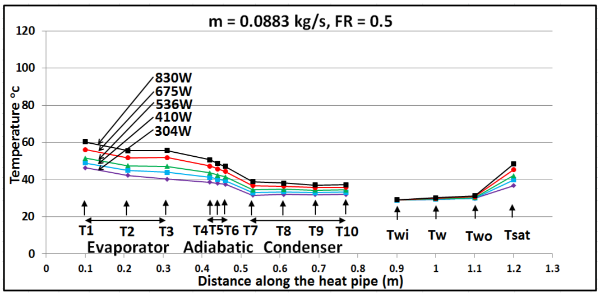

Fig.3.Axial temperature distribution (FR=0.5, 90^o).

4.2 Effect of inclination (Fig. 4)

The effect of inclination is shown by the typical result of Fig. 4 at a fill ratio of 0.5 and various inclinations. At a given evaporator heat input (P) the difference in operating temperature difference (T_e – T_c) was about 2-3^{o}C. Since the results are consistent to within 2^{o}C, it is concluded that within the experimental conditions, inclination had no distinct or noticeable effect on the performance.

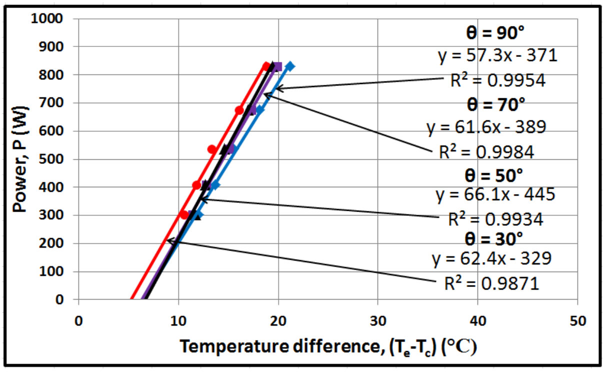

Fig. 4. Effect of inclination (FR=0.50).

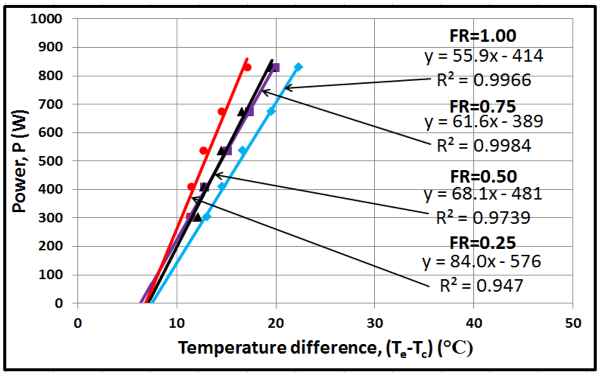

Fig. 5. Effect of fill ratio (70 degrees).

4.3 Effect of fill ratio (Fig. 5)

The effect of fill ratio is shown by the typical result of Fig. 5 at an inclination of 70 degrees and various fill ratios. Differences of about 3-4^{o}C were observed in the operating temperature difference at the higher heat input and about 1-2^{o}C at the lower heat input. Again, it could be concluded that because the differences were within the experimental consistency, fill ratio had no distinct or noticeable effect effect on the performance.

4.4 Condensing and evaporating heat transfer coefficients (Figs. 6 - 8).

Local evaporating and condensing heat transfer coefficients were calculated from the wall and saturation temperatures at each thermocouple location along the thermosyphon, viz.,

$$h_{evap} = \left(A_e \left[\frac{(T_e - T_{sat})}{P} - \frac{\ln(d_o / d_i)}{2\pi \, k_{wall} \, L_e} \right] \right)^{-1} \quad (1)$$

$$h_{cond} = \left(A_c \left[\frac{(T_{sat} - T_c)}{P} - \frac{\ln(d_o / d_i)}{2\pi \, k_{wall} \, L_c} \right] \right)^{-1} \quad (2)$$

Typical results are shown in Fig. 6 for 0.5 fill ratio and 90 degrees inclination. Mean condensing and evaporating heat transfer coefficients were then obtained by averaging the local values.

Linear regression lines are shown in the mean evaporating (h_{evap}) and condensing (h_{cond}) heat transfer coefficients results of Figs. 7 and 8. Regression coefficients greater than 0.9 were obtained for the mean evaporating coefficient and about 0.8 for the mean condensing coefficients. Within the limits of the experimental investigation where temperature fluctuations and small temperature differences were encountered, it could be concluded that the heat transfer coefficients were independent of angle of inclination but depended to a slight extent on the fill ratio. The average linear relationships are proposed to withion $\pm$ 20%.

$$h_{evap} = 1.742\, P + 791 \quad (3)$$

$$h_{cond} = 1.195\, P + 1822 \quad (4)$$

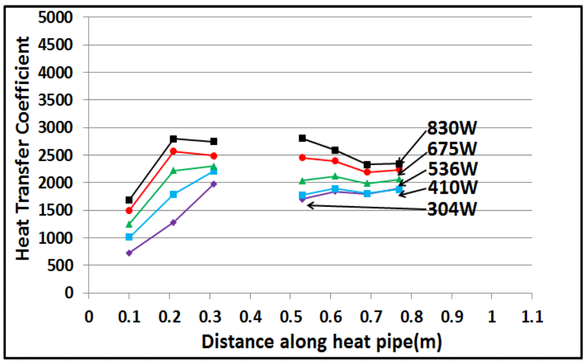

Fig. 6. Local evaporating and condensing heat transfer coefficients (FR = 0.5, 90 degrees).

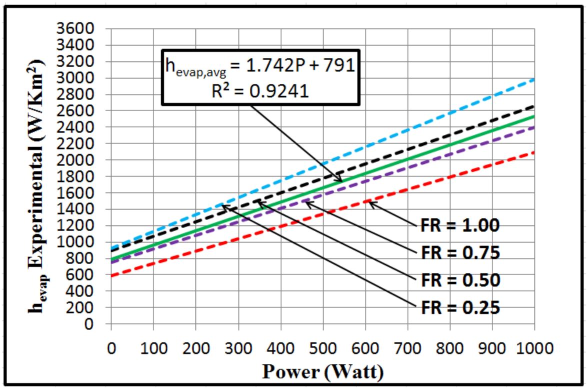

Fig. 7. Mean evaporating heat transfer coefficient.

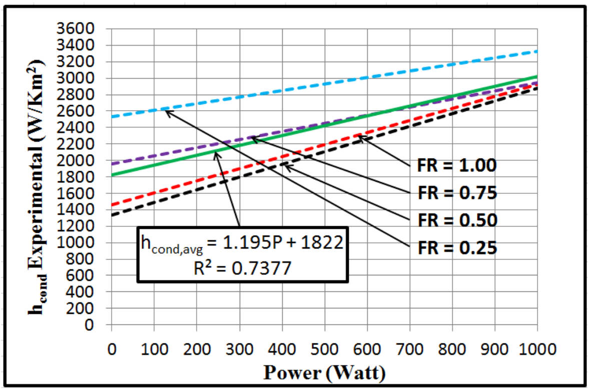

Fig. 8. Mean condensing heat transfer coefficient.

4.5 Overall heat transfer coefficient (Fig. 9)

The overall performance of the thermosyphon could be represented by the heat transfer vs operating temperature difference equation

$$P = \frac{(T_e - T_c)}{R_o} \quad (5)$$

Figure 9 shows the heat transfer rate plotted against the temperature difference. The results show that the heat transfer rate varied very nearly linearly with the temperature difference and that the thermosyphon required at least about 5 - 8°C temperature difference for it to function. A mean regression line indicated by

$$P = 57.7\,(T_e - T_c) - 306 \quad (6)$$

is shown in the graph.The results also showed that the performance was independent of angle of inclination. At a given power input, operating temperature difference at fill ratios between 1.0 and 0.25 from the mean was about 3-4°C at high input power and about 2-3°C at low input power.

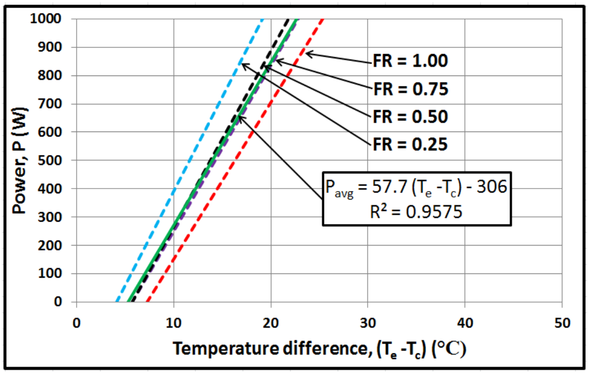

Fig. 9. Performance of thermosyphon.

5. CONCLUSIONS

The effects of fill ratio and inclination on the thermal performance of a water filled two-phase closed thermosyphon was investigated at fill ratios between 0.25 and 1.0, angles of inclination between 30 – 90 degrees from the horizontal and evaporator power input from 304 – 830 W. Within the limits of the experimental investigation, it could be concluded that the performance was independent of fill ratio and inclination. The mean evaporating and condensing heat transfer coefficients and the relationship between thermosyphon heat transfer and operating temperature difference between evaporator and condenser were determined.

NOMENCLATURE

A_e evaporator wall surface area [m^2]
A_c condenser wall surface area [m^2]
d_i pipe inner diameter [m]
d_o pipe outer diameter [m]
FR Fill ratio
h_{evap} evaporating htc [W/m^2 K]
h_{cond} condensing htc [W/m^2 K]
k_{wall} wall thermal conductivity [W/m K]
L_e evaporator section length [m]
L_c condenser section length [m]
P power input at evaporator [W]
R_o overall thermal resistance [K/W]
T_c condenser temperature [K]
T_e evaporator temperature [K]
T_{sat} saturated temperature [K]

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