International Journal of Engineering and Advanced Technology (IJEAT)
ISSN: 2249-8958 (Online), Volume-8 Issue-2S, December 2018

Experimental Investigation of Straight Shape Thermosyphon Filled with R410A Refrigerant

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Abstract*: *The thermal performances of a R410A filled thermosyphon subjected to low heat flux from 1882 W/m^2 to 4423 W/m^2 and evaporator wall temperatures between 20 °C and 40 °C with fill ratios 0.75 and 1.00 and at different inclinations from 45° to 90° were investigated. The axial temperature distribution of the thermosyphon was found to be uniform for all temperatures difference of evaporator at all power inputs. The performance of the thermosyphon which is determined from the heat transfer capability of the thermosyphon was found to be dependent of inclination angle and fill ratio. Experimental results show that heat transfer coefficient increases as the heat input increase while thermal resistance decreases exponentially with increasing input power. Increase in fill ratio and inclination angle at various heat input contributed to a better thermosyphon performance, at where heat transfer was highest at fill ratio 1.00 and inclination angle of 68°.

***Index Terms*: Heat transfer coefficient, Thermal resistance, Inclination angle, Filling ratio**

I. INTRODUCTION

Two-phase closed thermosyphon which are also known as heat pipes are highly effective heat transfer devices capable of transferring large amounts of heat effectively and efficiently. Fig. 1 shows an example of two-phase closed thermosyphon. It works by transferring latent heat of evaporation of the working fluid inside the system, which have been transport continuously by changing its phase from liquid to gas. As the amount of the heat absorbed increases, the vapor produced will be transport through the adiabatic section. Its low density causes it to flow upwards to the condenser end of the thermosyphon, at which it condenses back into liquid state by releasing the absorbed latent heat to a heat sink. As it condenses and the working fluid takes liquid state due to increasing in density, the liquid is drive back to the evaporator end by gravity. The cycle repeats continuously as long as heat is provided at the evaporator and removed at the condenser. There are many factors involved in the performance of a thermosyphon. These include the type of working fluid, input power, fill ratio (FR) and inclination angle.

Manuscript published on 30 December 2018.
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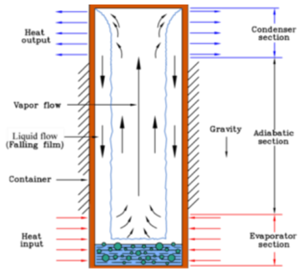

Fig. 1. Basic working principle of a thermosyphon [1]

II. LITERATURE REVIEW

Nguyen and Groll, [2] have investigated heat transfer limitations of two-phase thermosyphon by studying the maximum performance due to effects of liquid fill charge, inclination angle and operating temperature. The experiments carried out with copper tube length of 2.5 m and 20 mm outer diameter by using water as working fluid. Results showed that the fill charge exerts small influence on heat transfer and its influence is more noticeable for greater inclination angles. Thus, it was recorded that maximum heat transfer occurs for inclination angle between 40° to 60°.

Yong Joo Park, et al [3] have studied the heat transfer performance of a two-phase closed copper tube thermosyphon at heat input range of 50 W – 600 W and filling ratios of 10 % –70 % by using FC-72 (C_6F_{14}) as working fluid. For the relatively small fill ratio (< 20%) dry-out phenomena occurs and limits the performance at maximum heat flow rate of 100 W. Whereas, for the large fill ratio flooding limitation occurs and the maximum heat flow rate limited about 500 W –550 W.

Ong et al [4] investigated the thermal resistance of a thermosyphon filled with R410A refrigerant operating at evaporator temperature of 25 °C to 50 °C and power input from 40 W to 100W with fill ratios between 0.50 and 1.00 at different inclinations of 30°, 60° and 90°. From their study it was reported that thermal resistance of thermosyphon shows that fill ratio and inclination angle did not have significant effect on the thermal performance. However it was observed that inclination angle of 60° performed better compared to other inclination angles of 30° and 90°.In addition based on the comparisons made with other investigators using water, R410A and R134a it was generally concluded that thermal resistance increases exponentially with decreasing power. Fig. 2 shows their findings to compare other investigator.

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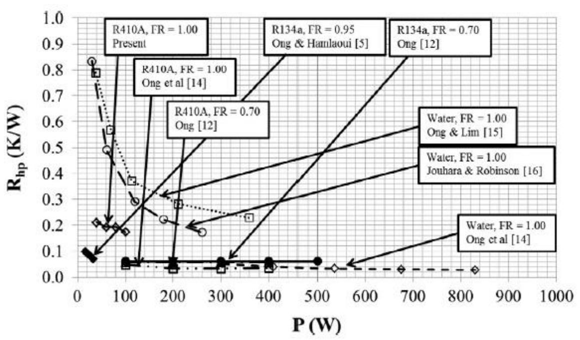

Fig. 2. Schematic of Experiment set up [4]

Asghar, et al [5] studied the gas/liquid flow by doing CFD simultaneous of the evaporation and condensation phenomena in a thermosyphon by applying the VOF technique. The copper tube thermosyphon constructed with a length of 100 cm and inner and outer diameters of 1.75 cm and 19 cm .The condenser and evaporator section consist length of 40 cm each and adiabatic section length of 20 cm. The experiment conducted by varying the heat flow rates of 350 W, 500 W and 700 W at three different fill ratios of 0.3, 0.5 and 0.8. Results showed that increase in inlet heat flow from 350 W to 500 W increases the thermosyphon's performance. However, applying higher energy to the evaporator decreases the performance. For each energy input to the evaporation section the optimum fill ratio is 0.5. Fig. 3 shows their findings.

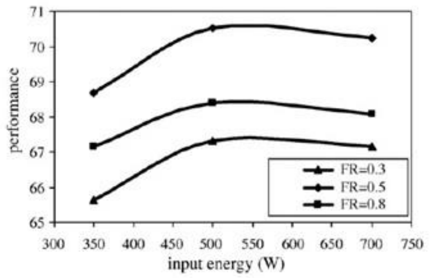

Fig. 3. The thermosyphon performances at various energy inputs and fill ratios [3].

Hamidreza et al [5] studied the thermal characteristics of a closed thermosyphon by varying the working fluid and fill ratio at different heat inputs and the results were compared with numerical results from other studies. Based on their study it was found that the optimum filled thermosyphon has the shortest response time and the lowest thermal resistance. The overfilled thermosyphon posed a slightly slower thermal response and greater thermal resistance compared to the optimal condition. Based on comparison of result made, it was noted that there are small differences in values obtain by the researcher. This is highly suspected due to differences of the possible heat loss from the evaporator end cap of the experiments. In conclusion it could be observed that increase in fill ratio at various heat input can contribute to a better performance, but it is clear that there is a limit for each system which leads the system not to perform further if exceeded.

Akash [6] reviewed the performance of the two-phase thermosyphon and multiple factors which affecting it. Refrigerant show better performance at lower temperature ranges and it is necessary to use refrigerant of lower GWP due to its effects in global warming. It was also observed that working fluid completes the cycle with the aid of gravity effect, thus thermosyphon cannot work in horizontal position and heat transfer performance is better between angles 50° to 90°.

Amatachaya and Srimuang, [7] compared the heat transfer characteristics of a FTPCT and a CTPCT. Both copper tube thermosyphons in the study have and inner diameter of 32 mm and length of 980 mm. The different type thermosyphons compared by varying the heat input, filling ratios and aspect ratios by using distilled water as working fluid and it is reported that the heat transfer coefficient of the CTPCT and FTPCT increased with decrease in filling ratios. Based on their finding it shows that thermosyphon structure has significant effect on overall heat transfer coefficient.

P.G.Anjankar and R.B.Yarasu [8] studied the experimental analysis effects of condenser length of 350 mm, 400 mm and 450 mm on the performance of vertical thermosyphon with different flow rates to condenser and heat input to evaporator. The result shows that the thermal performance of thermosyphon at heat input 500 W and flow rate 0.0027 kg/s with condenser length of 450 mm is higher and it was concluded that condenser length should be 1.5 times to that of evaporator length to obtain better thermal performance.

Davoud Jafari et al [9] investigated the unsteady experimental and numerical analysis of a two-phase closed thermosyphon at filling ratios of 16 %, 35 % and 135 %. The experiments performed to determine the heat transfer rate and overall thermal resistance at range of 30 W – 700 W by using heat pipe length and diameter of 500 mm and 35 mm. Result show that the thermosyphons performance better at filling ratio 35%. But at this filling ratio there are high risk of thermosyphon undergo dry-out effect.

Engin Gedik [10] investigated the thermal performance of a two-phase closed thermosyphon at different operating conditions. The experiment is conducted by using water, ethanol, and ethylene glycol as the working fluids and to operate at different inclination angles of 30°, 60°, and 90°, heat inputs of 200 W, 400 W, and 600 W, and flow rates of cooling water 10 L/h, 20 L/h, and 30 L/h. The results show that water, ethylene and ethanol were the best working fluid at the heat inputs and flow rate of (200 W /10L/h), (400 W /20L/h) and (600 W / 30L/h) . In conclusion it was found that the inclination angle and heat inputs have varied effects on the efficiency of the TPCT with different working fluid.

Thanaphol and Naris [11] studied the effects of bending and tilting using a flexible hose thermosyphon with 12.7 mm inner diameter, 8.5 mm outer diameter and 700 mm length at the adiabatic section by using R-134a as working fluid. Based on their study it was found that bending the pipe had a significant effect on two-phase flow pattern and pressure drop in the pipe. The higher the pressure drop, the higher the evaporation temperature. This proves that even a small change in its structure can vary the performance of the thermosyphons. Kyung Mo and In Cheol [12] compared the thermal performance between annular and concentric thermosyphons which have an outer diameter 25.4 mm and 22 mm inner diameter with length of 1000 mm at different fill ratios.

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The thermosyphons performance was studied based on changes of the entrainment limit at different fill ratio. Based on their findings it was shown that concentric thermosyphon has enhanced the entrainment limit far better than an annular thermosyphon as the fill ratio increases. This is due to the reduction of the cross-sectional area for vapor flow which results in increase of the shear at the vapor–liquid interface causing such enhancement.

Mihir et al [13] studied on a trans-critical R744 based summer air-conditioning unit and its impact of refrigerant charge on system performance. The experiment was carried to validate the theoretical model by investigating the effects of refrigerant charge, ambient temperature, and gas cooler face velocity on the performance of the system. Results show that for each 10 K increase in the ambient temperature, the system COP decreases by about 24 %. COP decreases significantly if the charge varies more than ±18% from the optimum value.

Penglei Zhang et al [14] investigated performance evaluation index for two-phase thermosyphon loop used in air condition systems. The study carried out by comparing the performance of the ideal and real TPL. The results show that the performance of the ''ideal cycle'' is the upper limit of the performances of all real cycles. From the perspective of refrigerant distribution, including under-charged refrigerant, over-charged refrigerant, insufficient height difference, and considerable flow resistance; and the ''ideal cycle'' can be approached, only when the filling ratio (100 %) and the height difference (1.2 m) is optimal, and the flow resistance is negligible simultaneously.

Ahmadou Samba et al [15] studied the two-phase thermosphon loop for cooling outdoor telecommunication equipment. The investigation was carried out by analyzing transient and steady states analysis of the thermosyphon loop efficiency, the thermal resistance and heat losses by convection in the walls of the cabinet as a function of heat load by using n-pentane as working fluid at different fill ratios. The result shows that the optimal filling ratio is about 9.2 %. At the optimal fill ratio the system manage to operate at minimum temperature and result smaller thermal resistance.

III. EXPERIMENTAL PROCEDURE

Fig. 4 shows the experiment set up. The thermosyphon consists of a close copper tube that has an outer and inner diameter of 0.0127 m and 0.0102 m respectively with a total length of 0.52 m. All temperatures were measured using thermocouples type-T with an accuracy of ±0.5 °C connected to a Yokogawa data logger and logged every minute. The condenser section is constructed from a cylindrical shell with a length of 0.25 m that surrounded the upper region of the thermosyphon and cooling air introduced to it in upward direction. The condenser section was keeping cool by using cold air supply through a ducting system at different air flow speed which is low, medium and high. The evaporator section has a length of 0.25 m constructed by having two layers of sauereisen flotemp thermal cement which was cast to eliminate all air pockets that acts as a thermal resistance between the heater and thermosyphon. The thermal cement has thermal conductive and electrical insulative properties. Thermal cement has a high thermal conductivity feature of 0.79 W/mK, for even heat distribution. The first layer casting consists of three thermocouples molded together at the evaporator section and the second layer casting consists of nichrome wire which been molded together and the whole part was insulated all around with two layers of superlon (insulation) with thickness of 50 mm each to prevent heat from escaping to obtain accurate readings. The nichrome wire used in experiment has a resistivity of 1.009 ± 0.05μΩ.m. Fig. 5 shows the cut-away section.

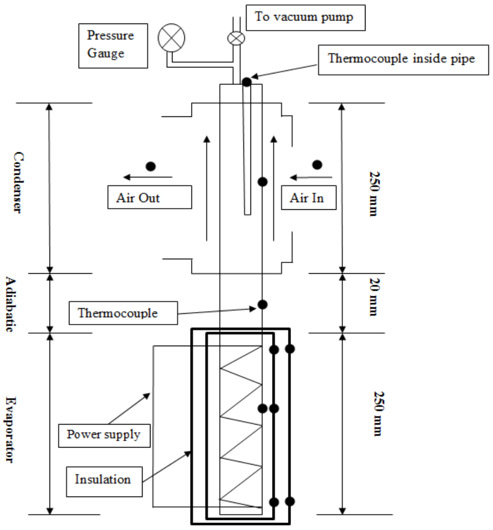

Fig. 4. Schematic of experiment set up of straight shape two-phase thermosyphon

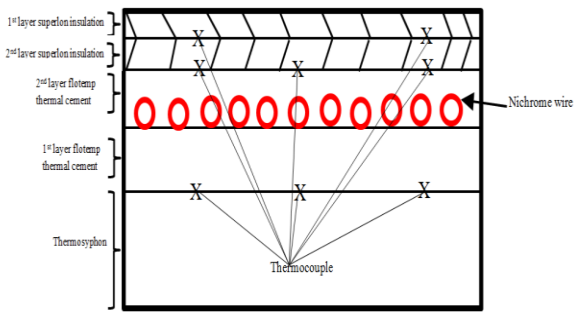

Fig. 5. Cut-away section for straight shape two-phase thermosyphon at evaporator.

Power (P) to evaporator was set using an electrical regulator with accuracy of ±0.0625 W and measured using laboratory standard voltmeter and ammeter. Total of eight thermocouples placed on the tube to measure the temperature of evaporator, adiabatic and condenser sections (T_1 till T_8). Two other thermocouples used to measure the inlet and outlet temperatures of the cooling air (T_9 and T_{10}). Details of thermocouples are as shown in Fig. 6.

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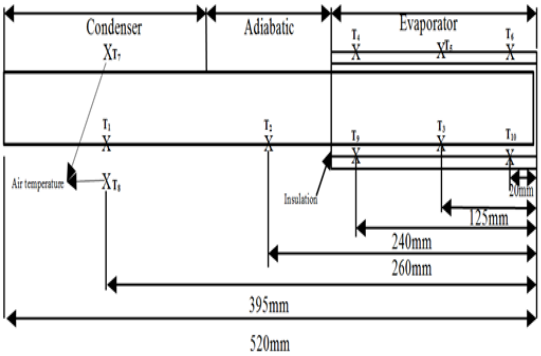

Fig. 6. Thermocouple details

The experiment carried out with a straight shape thermosyphon filled with R410A. Input power ranges were varied from 20 W to 30 W at heat flux from 1882 W/m^2 to 4423 W/m^2 . Fill ratios kept at 1.00 and 0.75. Fill ratio (FR) is defined as the ratio of the volume of the fill liquid to the evaporator section volume. The thermosyphons were tested inclined at angles from 45º, 68 º and 90 º to the horizontal. Each run was repeated twice and measurements were taken at steady state in order to check for repeatability. It was found that the temperatures obtained were repeatable to within ±1 ºC. Average values were then calculated from the two runs and utilized to plot the results.

A. Error analysis

h_e is used to determine the heat transfer coefficient of the evaporator. The following Eqn. (1) is used to determine the total error of the measured data.

$$h_e = \left(A_e \left(\frac{T_{e,wo} - T_{sat}}{P} - \frac{\ln \frac{d_o}{d_i}}{2\pi k_{wall} L_e} \right) \right)^{-1} \quad (1)$$

Total error at surface area (A_e) can be calculated from Eqn. 2. At where d is the diameter of the tube and h is the height of the tube with uncertainty of ± 0.1 mm and ±0.2 mm. At d = 12.7 mm and h= 260 mm.

$$Error, A_e = 2\pi\left(\frac{d}{2}\right)^2 + 2\pi\left(\frac{d}{2}\right)h \quad (2)$$
$$= \left(\left(\frac{0.1mm}{12.7mm}\times 2\right) + \left(\frac{0.1mm}{12.7mm}\right) + \left(\frac{0.2mm}{260mm}\right)\right)\times 100\%$$
$$= 2.43\%$$

The thermocouples have an uncertainty of ±0.5°C.The error at evaporator wall temperature ($T_{e,wo}$) can be calculated from Eqn. 3. At Average $T_{e,wo}$ = 32.67 °C.

$$Error, T_{e,wo} = \left(\frac{0.5°C}{T_{e,wo}}\right)\times 100\% \quad (3)$$
$$= \left(\frac{0.5°C}{32.67°C}\right)\times 100\%$$
$$= 1.53\%$$

The error at saturated temperature (T_{sat}) can be calculated from Eqn. 4. At Average T_{sat} = 14.84 °C

$$Error, T_{sat} = \left(\frac{0.5°C}{T_{sat}}\right)\times 100\% \quad (4)$$
$$= \left(\frac{0.5°C}{14.84°C}\right)\times 100\%$$
$$= 3.36\%$$

The measuring tape has an uncertainty of ±0.2mm. The error at evaporator length (L_e) can be calculated from Eqn. 5. At L_e= 260 mm.

$$Error, L_e = \left(\frac{0.2mm}{L_e}\right)\times 100\% \quad (5)$$
$$= \left(\frac{0.2mm}{260mm}\right)\times 100\%$$
$$= 0.08\%$$

The power regulator has an uncertainty of ±0.0652 W. The error at input power (P) can be calculated from Eqn. 6. At average P = 32.60 W.

$$Error, P = \left(\frac{0.0652W}{P}\right)\times 100\% \quad (6)$$
$$= \left(\frac{0.0652W}{32.60W}\right)\times 100\%$$
$$= 0.2\%$$

Sum of total error:
= 2.43 %+1.53 %+3.36 %+0.2 %+2(0.79 %)+0.08 %
= 9.18 %

The sums of total errors obtain from Eqn. 1 is 9.18%, thus this indicates the overall experimental measured values has an error of less than 10%. The evaporator and condenser have the same surface area and power input thus it is assumed that total error of h_c is equivalent to h_e.

IV. EXPERIMENTAL RESULTS

B. Heat Transfer Coefficient

Figures 7 through 17 are plotted based on a steady state operation. The results show that the largest heat transfer coefficient recorded is 323.9 W/m^{2}K at heat flux of 3763.9 W/m^2 whereas the smallest heat transfer coefficient rate recorded is 152.2 W/m^{2}K at heat flux of 3011.2 W/m^2.

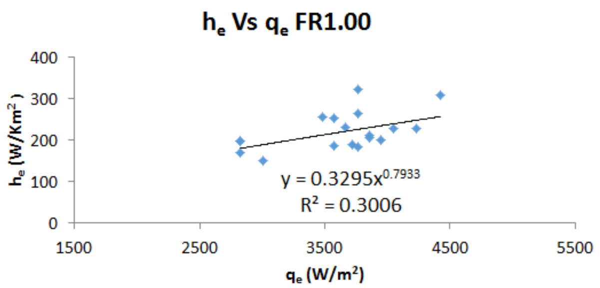

Fig. 7. Evaporator heat transfer coefficient (W/m^{2}K) vs. heat flux (W/m^2) at filling ratio 1.00

Fig. 8 shows that the largest heat transfer coefficient rate recorded is 183.4 W/m^{2}K at heat flux of 3293.5 W/m^2 whereas the smallest heat transfer coefficient values recorded is 95.1 W/m^{2}K at heat flux of 2164.3 W/m^2.

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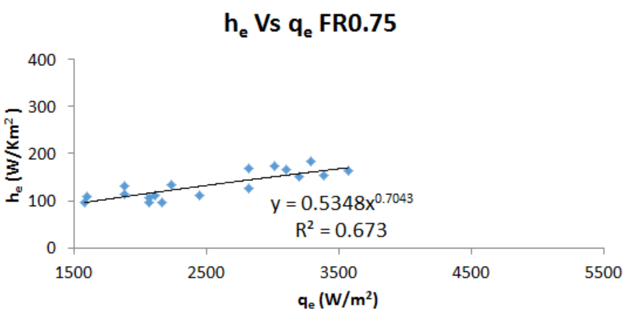

Fig. 8. Evaporator heat transfer coefficient (W/m^{2}K) vs. heat flux (W/m^2) at filling ratio 0.75

Overall the results from Fig. 7 and 8 shows that heat transfer coefficient of evaporator have an increasing trend as the heat flux increases. Based on Fig. 7 and 8, filling ratio 1.00 performs better compared to 0.75. This is because fill ratio 1.00 exhibits the largest heat transfer coefficient at same input power. For instance at heat flux value 2000 W/m^2 the heat transfer coefficient values at fill ratio 1.00 and 0.75 are 136.9 and 113 W/m^{2}K respectively. Thus this shows that straight pipe with R410A of fill ratio 1.00 exhibits better heat transfer properties compared to fill ratio 0.75 with same refrigerant. At fill ratio 1.00, the evaporator section has a larger volume of liquid compare to fill ratio 0.75 thus allowing more heat is to be absorbed by the liquid refrigerant enabling it to transfer more heat energy (W) resulting in better overall heat transfer.

Fig. 9 Shows that the largest heat transfer coefficient rate recorded is 7194.3 W/m^{2}K at heat flux of 3858.1 W/m^2 whereas the smallest heat transfer coefficient rate recorded is 2590 W/m^{2}K at heat flux of 2823 W/m^{2}K.

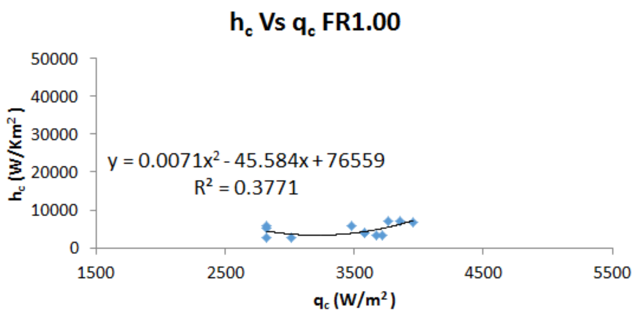

Fig. 9. Condenser heat transfer coefficient (W/m^{2}K) vs. heat flux (W/m^2) at filling ratio 1.00

Fig. 10 shows that the largest heat transfer coefficient rate recorded is 9735.9 W/m^{2}K at heat flux of 2822 W/m^2 whereas the smallest heat transfer coefficient rate recorded is 1608.9 W/m^{2}K at heat flux of 1599.7 W/m^{2}K.

Overall, Figs. 9 and 10 show that heat transfer coefficient of condenser has an increasing trend as the heat flux increases. Based on the result as shown, performance is better with filling ratio 1.00 compared to 0.75. This is because fill ratio 1.00 exhibits the largest condenser heat transfer coefficient at same input power. For example at heat flux value 2000 W/m^2 the condenser heat transfer coefficient values at fill ratio 1.00 and 0.75 are 13791 and 3242.5 W/m^{2}K respectively. Thus this shows that straight pipe with R410A of fill ratio 1.00 exhibits better heat transfer properties compared to fill ratio 0.75 with same refrigerant. Similar to evaporator side condenser side also been influenced by the working fluid volume thus at higher fill ratio it enables heat absorption to occur inside the system thus resulting better heat transfer.

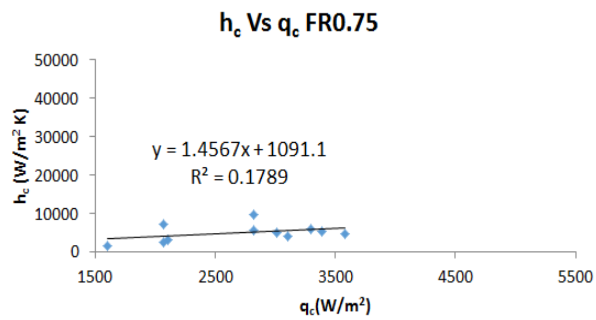

Fig. 10. Condenser heat transfer coefficient (W/m^{2}K) vs. heat flux (W/m^2) at filling ratio 0.75

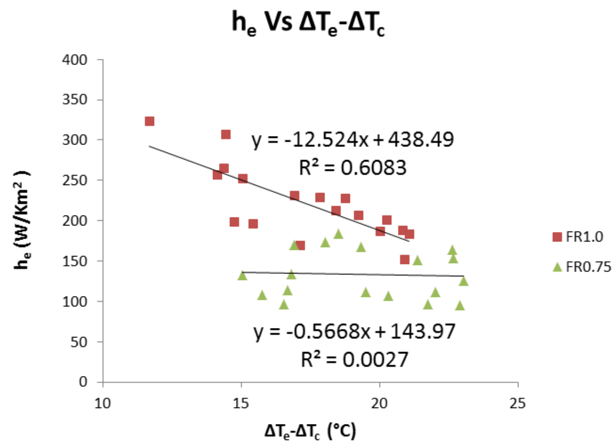

Fig. 11. Evaporator heat transfer coefficient (W/m^{2}K) vs. change of temperature (°C) between evaporator and condenser

Fig. 11 results show that the both experimented fill ratios display a decreasing trend as the change of temperature between evaporator and condenser side increases. The heat transfer coefficient value becomes smaller as the change of temperature between evaporator and condenser side increases. It could be seen that certain recorded results were not uniform. Example of non-uniform value of temperature difference for fill ratio 1.00 and 0.75 at heat flux 323.93 W/m^2 and 96.23 W/m^2 were 11.68 °C and 16.53 °C respectively, thus a mean temperature difference (ΔT) was obtained by taking the difference between the average evaporator (T_e) and condenser temperature (T_c). The axial wall temperature distribution of the thermosyphon at filling ratio 1.00 seems to be better compared to filling ratios 0.75 based on the plotted values.

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At fill ratio 1.00 the thermosyphon has more working liquid in the evaporator section thus the system can have even more heat exchange and contributes to a better overall temperature distribution. Based on the Fig. 11, filling ratio 1.00 performs better compared to 0.75 as it has the smallest change in temperature between evaporator and condenser section that indicates better overall heat transfer occurrence. Fill ratios 0.75 exhibits largest change of temperature different between evaporator and condenser compared to fill ratio 1.00 indicates less heat transfer occurred at larger power input. Such occurrence is due to the system could not perform the fullest for the amount of heat input (power), thus resulting lesser overall heat transfer hence causing thermosyphon to be less efficient.

C. Change of Temperature at Different Inclination Angle

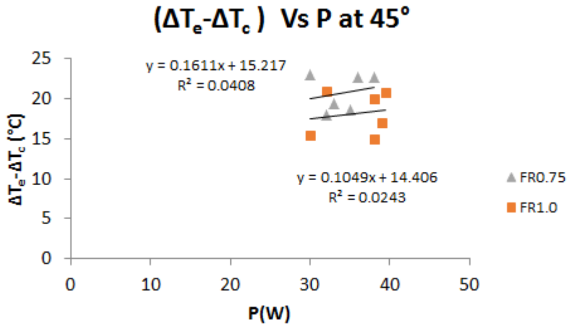

Fig.12. Temperature (°C) vs. Power (W) change in evaporator and condenser at an inclination angle of 45°

Fig. 12 shows temperature change between evaporator and condenser at an inclination angle of 45º. The results show that the entire fill ratio displaying an increasing trend as the change of temperature between evaporator and condenser side increases. The largest change of temperature between T_e and T_c recorded for fill ratio 1.00 and 0.75 is 20.8 °C and 23.0 °C respectively. Based on the result it shows that fill ratio 1.00 performed better compared to other fill ratio at same inclination angle due to smaller change in temperature between evaporator and condenser section that indicates better overall heat transfer occurrence. Hence fill ratio 1.00 required smaller input powers to achieve the targeted T_e and T_c values.

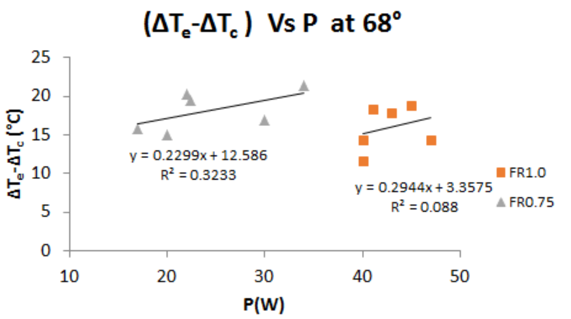

Fig.13. Temperature (°C) vs. Power change in evaporator and condenser at an inclination angle of 68°

Fig. 13 shows temperature change between evaporator and condenser at an inclination angle of 68º. The results show that the entire fill ratio displays an increasing trend as the change of temperature between evaporator and condenser side increases. The largest change of temperature between T_e and T_c recorded for fill ratio 1.00 and 0.75 is 18.8 °C and 21.4 °C respectively. Based on the result it shows that fill ratio 1.00 performed better compared to other fill ratio at same inclination angle due to smallest change in temperature between evaporator and condenser section that indicates better overall heat transfer occurrence. Thus it indicates that fill ratio 1.00 required smaller input powers to achieve the targeted T_e and T_c values.

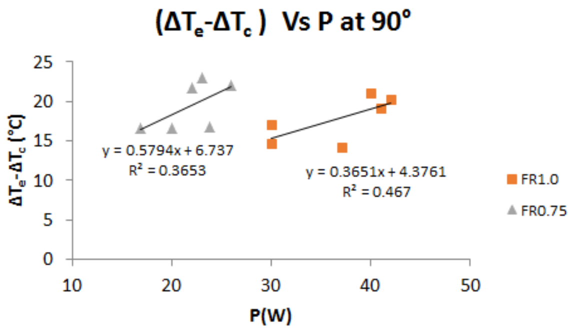

Fig.14. Temperature (°C) vs. Power change in evaporator and condenser at an inclination angle of 90°

Fig. 14 shows temperature change between evaporator and condenser at inclination angle of 90º. The results show that the entire fill ratio displays an increasing trend as the change of temperature between evaporator and condenser side increases. The largest change of temperature between T_e and T_c recorded for fill ratio 1.00 and 0.75 is 21.1 °C and 22.2 °C. Based on the result it shows that fill ratio 1.00 performed better compared to other fill ratio at same inclination angle due to smallest change in temperature between evaporator and condenser section that indicates better overall heat transfer occurrence. Thus, fill ratio 1.00 requires smaller input powers to achieve the targeted T_e and T_c values.

Taking into account all the observations fill ratio1.00 has performed better at all inclination angles compared to 0.75, especially at inclination angle of 68° which has the best performance. This is because at inclination angle of 68° the working fluid which changes phase from liquid to vapor, condenses back to the evaporator side faster at larger drag force with aid of gravity compare to other inclination angles hence resulting better overall performance. This results are similar to findings of Ong et al [4] who observed that inclination angle of 60° had smaller change in temperature between evaporator and condenser compare to other inclination angles of 30° and 90°. Thus this shows that thermosyphon performed better in between horizontal and vertical position. Moreover similar findings could be observed from findings of Nguyen and Groll [2] who reported that maximum heat transfer occurs for inclination angle between 40° to 60°.

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Thus this indicates different inclination angle also play a role in thermosyphons behavior when compared with each other at different filling ratios. The force of gravity acting on the condensate liquid refrigerant which assists the liquid back to the evaporator section will be opposed by the force created by the vaporizing refrigerant and drag force created by the condensate liquid itself. Thus at inclined position the vapor and drag force exerted on the condensate liquid becomes less dominated which makes gravity to be more dominated and fastens the flow of liquid refrigerant back to the evaporator section. Nevertheless Asghar, Yong Joo Park and Ong team [3,4, 5] has reported that there is operating limit for every system which leads to drop in performance if surpassed.

The change of thermal resistance by varying input power for different fill ratios at inclination angle of 45° are plotted in Fig. 15 The smallest thermal resistance recorded for fill ratio 1.00 and 0.75 is 0.50 °C/W and 0.67 °C/W respectively. From the results it shows that thermal resistance decreases with increase of input power.

D. Overall Thermal Resistance

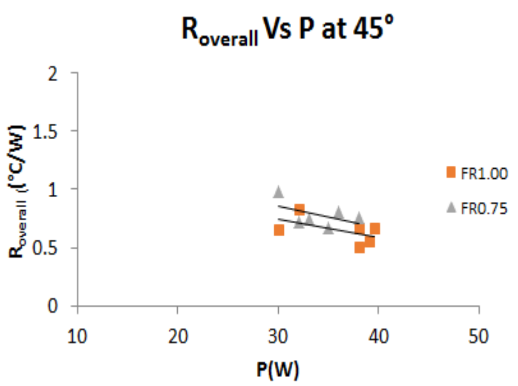

Fig.15. Overall thermal resistance (°C/W) vs. input power (W) at 45°

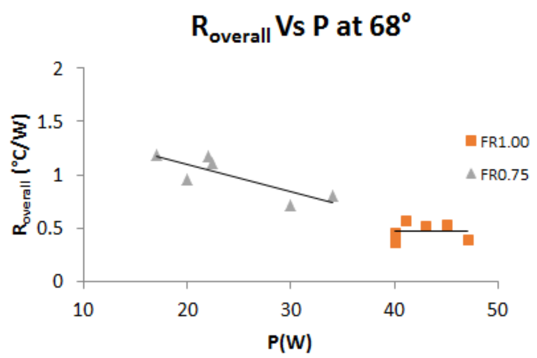

Fig.16. Overall thermal resistance (°C/W) vs. input power (W) at 68°

The change of thermal resistance by varying input power for different fill ratios at an inclination angle of 68° are plotted in Fig. 16. The smallest thermal resistance recorded for fill ratio 1.00 and 0.75 is 0.37 °C/W and 0.72 °C/W respectively. From the results it shows that thermal resistance decreases with increase of input power.

The change of thermal resistance by varying input power for different fill ratios at an inclination angle of 90° are plotted in Fig. 17. The smallest thermal resistance recorded for fill ratio 1.00 and 0.75 is 0.49 °C/W and 0.90 °C/W respectively. The results show that thermal resistance decreases with increase of input power.

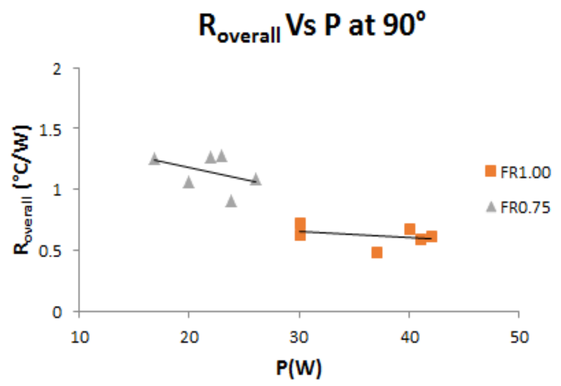

Fig. 17. Overall thermal resistance (°C/W) vs. input power (W) at 90°

Overall Figs. 15 through 17 results show that thermal resistance has a decreasing trend as the input power increases. These results are similar to findings of Ong et al [4] who reported the thermal resistance decreases as input power increases. Based on the result it shows that fill ratio 1.00 performed better compared to fill ratio 0.75 at all inclination angles where it has the smallest thermal resistance values. Especially at inclination angle of 68° which has the smallest thermal resistance values compare to other angles. This is because at inclination angle of 68° the force of gravity acting on the condensate liquid refrigerant becomes more dominant than of vapor and drag force is exerted on the liquid and quickens the flow of liquid refrigerant back to the evaporator section thus resulting better heat transfer and making the system more efficient.

V. CONCLUSION

The thermal performance of a R410A refrigerant filled thermosyphon subjected to low heat flux from 1882 W/m^2 to 4423 W/m^2 and evaporator wall temperatures between 20 °C and 50 °C was investigated. The effects of input power, fill ratio and inclination angle on the heat transfer coefficient and thermal resistance were determined. The following were concluded from the study:

1. Heat transfer coefficient increases as the heat flux increases, thus causes higher heat transfer occurrence in the overall system. The largest heat transfer coefficient value of h_e and h_c recorded is 323.9 W/m^{2}K and 9735.9 W/m^{2}K respectively. However increase of heat transfer coefficient does not solely depend on heat input where there are other contributing factors toward a more effective and efficient thermal performance.
2. Increase in fill ratio at various heat input can contribute to a better thermosyphon performance. However based on findings from other researchers such as Asghar, Yong Joo Park team findings [3,5] it was reported that there is a limit for each system which causes the system not to perform further if it exceeds the limit.

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3. Heat transfer coefficient value becomes larger as the difference of temperature between evaporator and condenser side decreases. Thus this shows that uniform temperature distribution between evaporator and condenser contributes to better heat transfer.
4. Other than influence of filling ratio towards heat transfer, inclination angle also contributes as one of the factors for a greater thermosyphon system. It's a fact that the working principle of thermosyphon completes due to the effect of gravity. Therefore, thermosyphon is unable to perform in horizontal position and heat transfer performance is best at an inclination angles of 68°.Which indicates better working fluid circulation hence resulting better heat transfer.
5. Thermal resistance decreases exponentially with increasing input power, thus resulting better heat transfer and making the system more efficient and effective. However based on findings of Yong Joo Park and Davoud Jafari [3,9] dry out effect could easily occur in the case of high input power at low fill ratio, as it will cause an increase of evaporator side temperature due to reduction in contact surface with working fluid. Thus indicating that each system has its own operating limit and its performance will drop once it's exceeded.

ACKNOWLEDGMENT

The authors would like to acknowledge the RIGS Project of International Islamic University Malaysia and Daikin Malaysia Sdn Bhd for the support to carry out this work.

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