Penerbit Akademia Baru

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences

Journal homepage: www.akademiabaru.com/arfmts.html

ISSN: 2289-7879

Two-phase Thermosyphon Filled with R410A Refrigerant Operating at Low Evaporator Temperature

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ARTICLE INFO

Article history:
Received 9 October 2018
Received in revised form 4 January 2019
Accepted 1 February 2019
Available online 5 February 2019

ABSTRACT

The thermal performances of a R410A filled thermosyphon subjected to low heat flux from 1882 W/m2 to 4423 W/m2 and evaporator wall temperatures between 20 °C and 50 °C with fill ratios 1.00 and 0.75 and at different inclinations from 45°, 68° and 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°. In addition experimental and application measured readings have a percentage error less than 10 %.

Keywords:
Heat transfer coefficient, thermal resistance, inclination angle two phase thermosyphon, dry-out limitation, filling ratio

1. 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. Figure 1 shows an example of two-phase closed thermosiphon [1]. It works by transferring latent heat of evaporation of the working fluid inside the system, which have been transported 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

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many factors involved in the performance of a thermosyphon. These include the type of working fluid, input power, fill ratio (FR) and inclination angle.

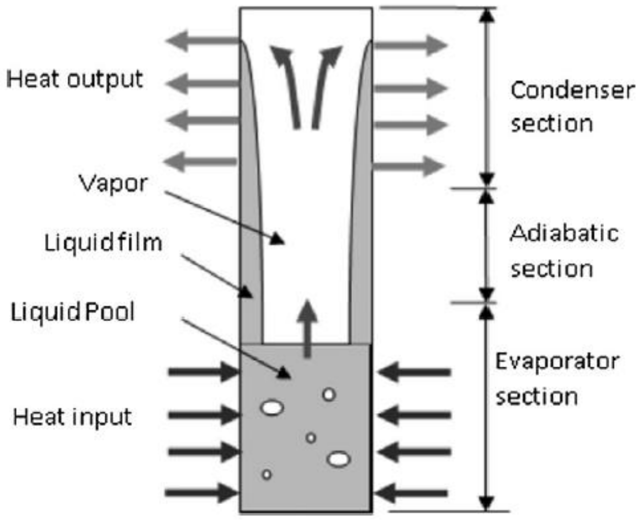

Fig. 1. Comp basic working principle of a thermosyphon [1]

Nguyen and Groll [2] 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. 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 *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.

Ong *et al.*, [4] investigated the thermal resistance of a thermosyphon filled with R410A refrigerant operating with fill ratios between 0.50 and 1.00 at different inclinations of 30°, 60° and 90°. From their study it was reported 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°.

Asghar *et al.*, [5] studied the gas/liquid flow by doing CFD simulations on thermosyphon by applying the VOF technique. Results showed that the optimum fill ratio is 0.5 and thermosyphon performs the best at heat input from 350 W to 500 W.

Akash [6] reviewed the performance of the two-phase thermosyphon and multiple factors which affecting it. It was 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°.

Anjankar and Yarasu [7] experimentally analyzed effects of different condenser length of thermosyphon. Result shows that the thermal performance at heat input 500 W and flow rate 0.0027 kg/s with condenser length of 450 mm are highest and concluded that condenser length should be 1.5 times to that of evaporator length to obtain better thermal performance.

Davoud *et al.*, [8] investigated the unsteady experimental and numerical analysis of a two-phase closed thermosyphon at filling ratios of 16%, 35% and 135%. 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.

Mihir *et al.*, [9] studied on a trans-critical R744 based summer air-conditioning unit and its impact of refrigerant charge on system performance. 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.

Thanaphol and Naris [10] studied the effects of bending and tilting using a flexible hose thermosyphon by using R-134a as working fluid. Result show that the higher the pressure drop, the higher the evaporation temperature. This proves that even a small change in structure can vary the performance of the thermosyphons.

Hamidreza *et al.*, [11] 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 other researcher's studies. Results show that increase in fill ratio at various heat input can contribute to a better performance which was parallel with other researchers findings, but it was observed that there was a limit for each system which causes it not to perform further if exceeded.

Engin [12] investigated the thermal performance of a two-phase closed thermosyphon using different working fluids at different flow rates, heat input and inclination angles. Results show that at all inclination angles 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 the inclination angle and heat inputs have varies effects on the efficiency of the thermosyphon operating with different working fluids.

Kyung and In [13] compared the thermal performance between annular and concentric thermosyphons based on changes of the entrainment limit at different fill ratios. Their findings shown that concentric thermosyphon has enhanced the entrainment limit far better than an annular thermosyphon as the fill ratio increases. This was 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.

Penglei *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 thermosyphon. The results show that the performance of the ''ideal cycle" is the upper limit of the performances of all real cycles.

Ahmadou *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 where the system manages to operate at minimum temperature and result smaller thermal resistance.

Although there have been a lot of works conducted on thermosyphon performance not much is available on comparison between experimental data and field test unit (Application). Most of the comparison studies which been done are more focused based on parameters of the thermosyphon and numerical analysis comparison using computational fluid dynamics (CFD) simulations. Thus, the main objective of this investigation is to gain an understanding of the heat transfer performance comparing both experimental data and field test data. In this article will show how the experimental performance related to the actual field test performance.

2. Methodology

The thermosyphon consists of a close copper tube that has an outer and inner diameter of 0.0127 m and 0.0105 m respectively with a total length of 0.52 m. Figure 2 shows the experiment set up and

Figure 3 shows the cut-away section. Total of eight thermocouples placed on the tube to measure the temperature of evaporator, adiabatic and condenser sections and two other used to measure the inlet and outlet temperatures of the cooling air. 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. Power (P) to evaporator was set using an electrical regulator with accuracy of ±0.0625 W and measured using laboratory standard voltmeter and ammeter.

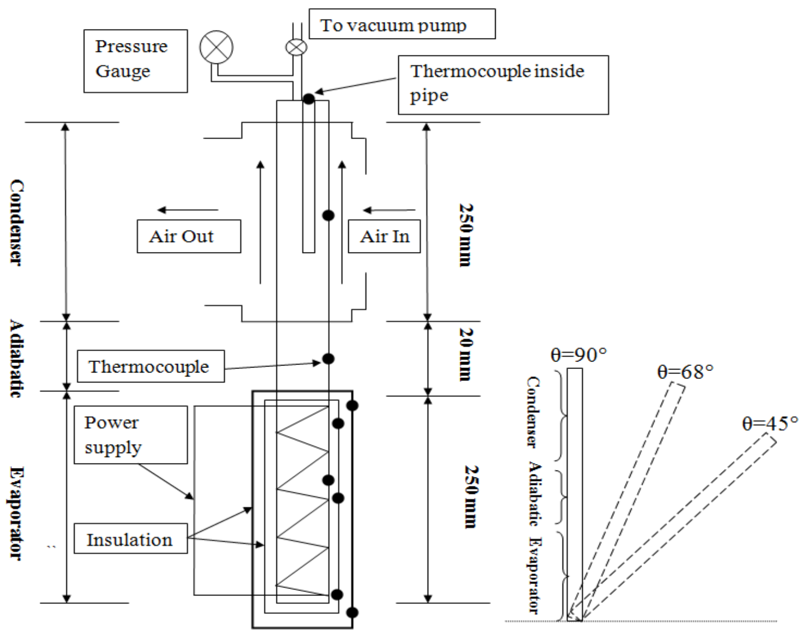

Fig. 2. Schematic of experiment set up of straight shape two-phase thermosyphon

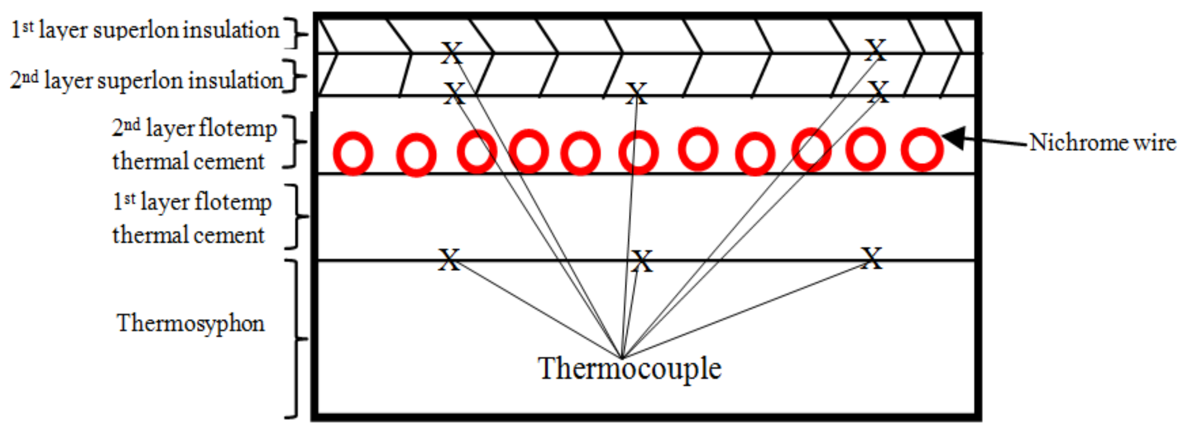

Fig. 3. Vertical cut-away section view for straight shape two-phase thermosyphon at evaporator

The experiment carried out with a straight shape thermosyphon filled with R410A. Input power range was varied from 20 W to 50 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 45°, 68° and 90° 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 °C. Average values were then calculated from the two runs and utilized to plot the results.

2.1 Application Details

The field study was has been carried out by using a 6 horse power floor standing unit (Figure 4). The unit consists of two different heat exchangers which are a coiling coil (evaporator) and heat pipe system (evaporator and condenser). The evaporator and condenser side individually have a heat transfer area of 12.21 m^2. The copper tube has an outer and inner diameter of 0.007 m and 0.00678 m with a total length of 0.747 m. There are total of 20 thermocouples been installed, 10 per each section for the evaporator and condenser side to measure the inlet and outlet side temperature of the cooling coil and heat pipe system. 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 application test carried out by operating the unit with R410A at different fill ratios which are 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 measured condenser out temperature values were compared to the experimental values. Each run was repeated twice and measured were taken at steady state in order to check for repeatability. Average values were then calculated from the two runs and utilized to plot the results.

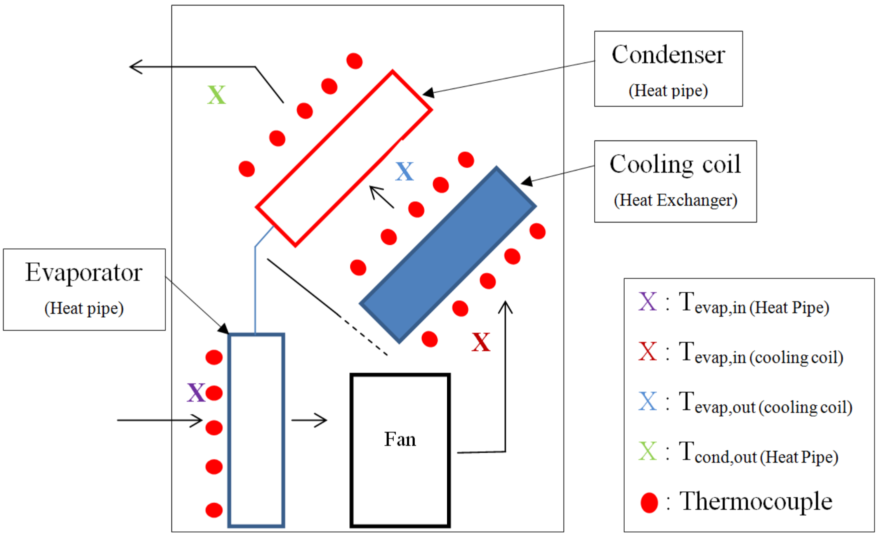

Fig. 4. Schematic of application set up

3. Results

3.1 Heat Transfer Coefficient

The results from Figures 5 and 6 show that heat transfer coefficient of evaporator has an increasing trend as the heat flux increases. Based on Figures 5 and 6, 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^2K 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. 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.

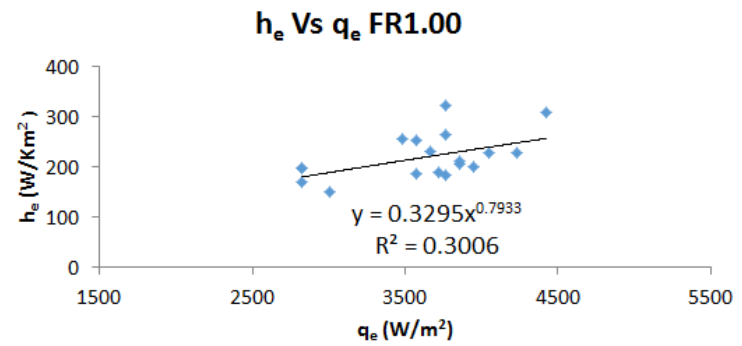

Fig. 5. Evaporator heat transfer coefficient (W/m^{2}K) vs heat flux (W/m^2) at filling ratio 1.00

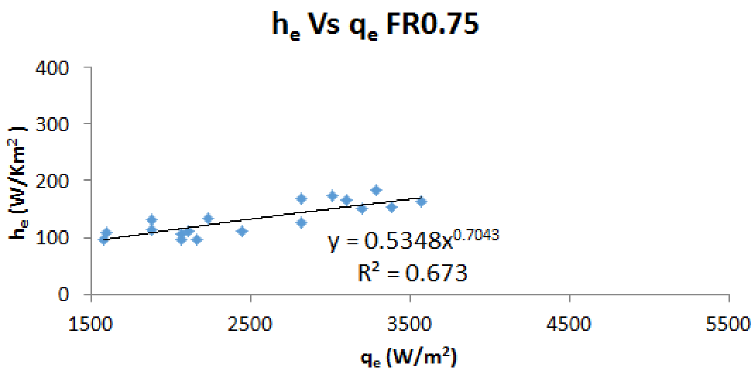

Fig. 6. Evaporator heat transfer coefficient (W/m^{2}K) vs heat flux (W/m^2) at filling ratio 0.75

Figures 7 and 8 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 has 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.

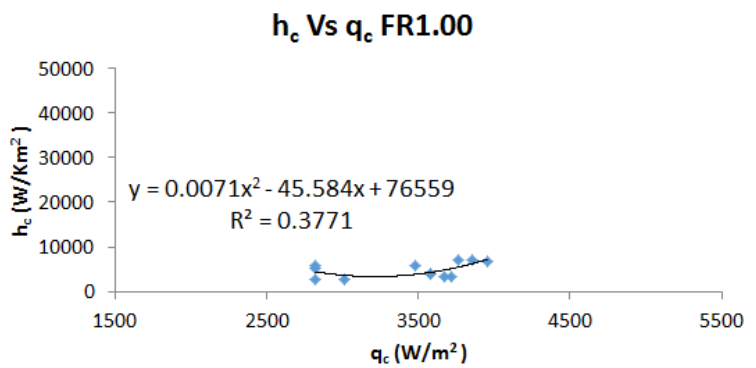

Fig. 7. Condenser heat transfer coefficient (W/m^{2}K) vs heat flux (W/m^2) at filling ratio 1.00

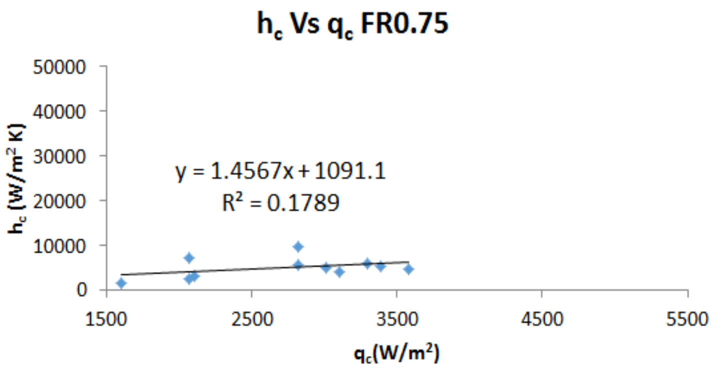

Fig. 8. Condenser heat transfer coefficient (W/m^{2}K) vs heat flux (W/m^2) at filling ratio 0.75

3.2 Overall Thermal Resistance

Figures 9, 10 and 11 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 vapour 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.

3.3 Application Results

The changes in temperature profile of a R410A filled system are plotted in Figure 12 and 13.Based on the plot the average change in temperature between the precooling and reheating are 2.85 °C and 2.68 °C for FR 1.00 and 1.96 °C and 1.55 °C for FR 0.75. Thus this indicates that system efficiency is 94% and 79%.

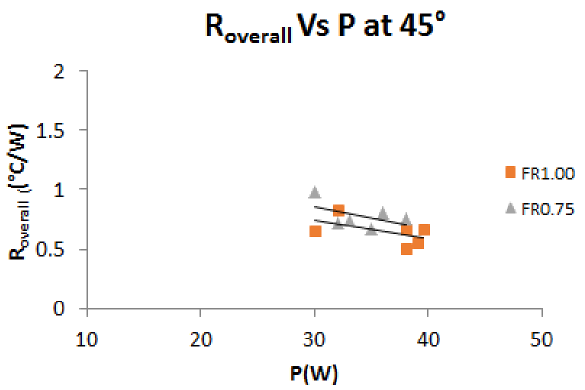

Fig. 9. Overall thermal resistance (°C/W) vs input power (W) at 45°

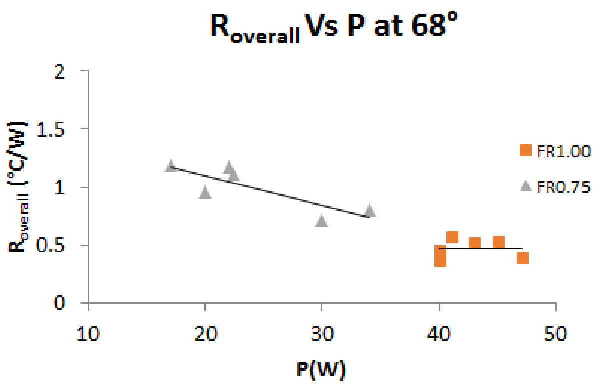

Fig. 10. Overall thermal resistance (°C/W) vs input power (W) at 68°

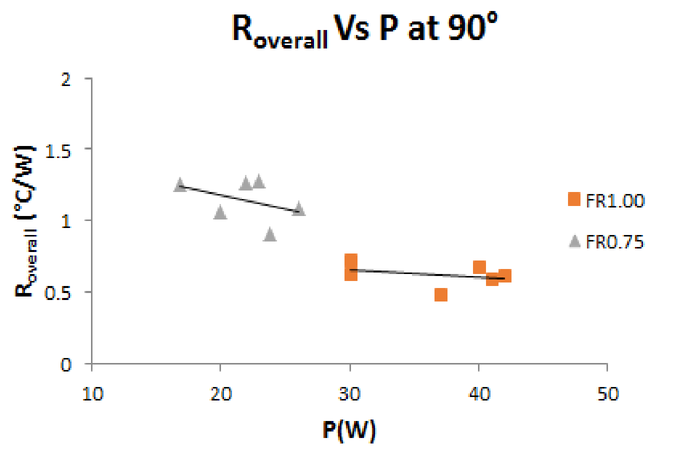

Fig. 11. Overall thermal resistance (°C/W) vs input power (W) at 90°

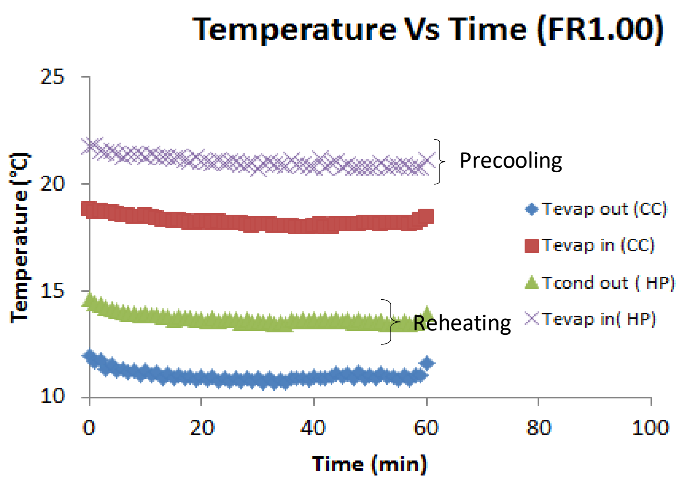

Fig. 12. Temperature (°C) vs Time (min) at FR 1.00

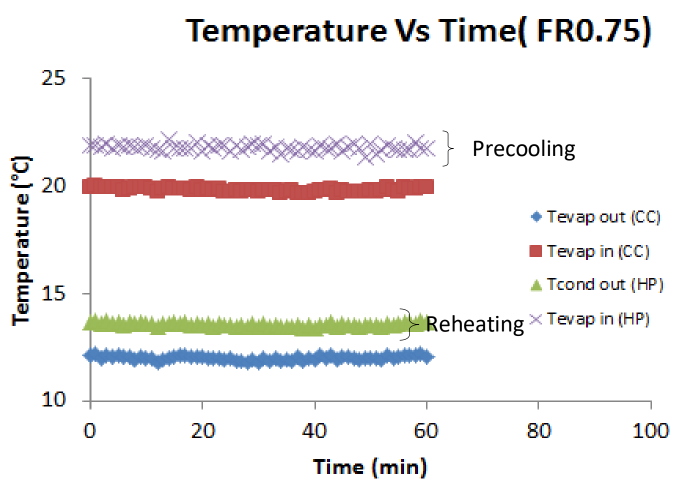

Fig. 13. Temperature (°C) vs Time (min) at FR 0.75

The calculation below shows the percentage error difference between the measured condenser out temperature from the field test unit with calculated condenser out temperature based on collinear equations of h_e and h_c in Figures 7 through 10 that obtained from measured experimental data for fill ratios 1.00 and 0.75.

At fill ratio 1.00 average q_e / q_c obtain from field test unit is 221.04 W/m^2. Find h_e from Equation in Figure 7.

$$h_e = 0.3295({q_e}^{7933}),$$

At q_e=221.04 W/m^2

$$h_e = 22.86\ W/m^2\,K$$

At $T_{e,AVG}$=23.35°C,$T_{c,AVG}$=14.36°C,P=2699.52 W and A_e= 12.21 m^2

$$T_{sat,AVG} = T_{e/c,AVG} - P\left(\frac{ln\frac{d_o}{d_i}}{2\pi K_{wall} L_{e/c}}\right) - \frac{P}{h_{e/c}\,A_{e/c}} \tag{1}$$

$$T_{sat,AVG} = 14.03\ °C$$

Find h_c from equation in Figure 9.

$$h_c = 0.0071({q_c}^2) - 45.584(q_c) + 76559$$

$$h_c = 66835.88\ W/m^2\,K$$

Find $T_{c,newAVG}$ value from Equation 1, at P=2699.52 W, d_o = 0.007 m, d_i = 0.00678 m L_c= 0.747 m and A_c= 12.21 m^2

$$T_{c,new\ AVG} = 14.11\ °C$$

$$Coeffcient\ Factor = \frac{T_{c,out\ new\ AVG} - T_{c,in}}{T_{c,new\ Avg} - T_{c,in}} \tag{2}$$

Find $T_{c,out,newAVG}$ value from Equation 2, at coefficient factor is 0.8

$$T_{c\ out,new\ AVG} = 13.47\ °C$$

At $T_{C\ out,\ AVG}$ from field test 13.69 °C

$$Error, \% = \frac{T_{c,outAVG} - T_{C,out\ new\ AVG}}{T_{c,outAVG}} \times 100 \tag{3}$$

$$Error\ , \% = \frac{13.69 - 13.47}{13.69} \times 100\%$$

$$Error\ , \% = 1.61\ \%$$

At fill ratio 0.75 average q_e / q_c obtain from field test unit is 152.21 W/m^2.

Find h_e from Equation in Figure 8.

$h_e = 0.5348(q_e{}^{0.7043})$, At q_e=152.21 W/m^2

$h_e = 18.42\ W/m^2\,K$

At $T_{e,AVG}$=23.34°C,$T_{c,AVG}$=13.94°C, P=2699.52 W and A_e= 12.21 m^2

$$T_{sat,AVG} = T_{e/c,AVG} - P\left(\frac{ln\frac{d_o}{d_i}}{2\pi K_{wall}L_{e/c}}\right) - \frac{P}{h_{e/c}\,A_{e/c}} \tag{4}$$

$T_{sat,AVG} = 15.03\ °C$

Find h_c from equation in Figure 10.

$h_c = 1.4567(q_c) + 1091.1$

$h_c = 1312.82\ W/m^2\,K$

Find $T_{c,newAVG}$ value from Equation 1, at P=2699.52 W, d_o = 0.007 m, d_i = 0.00678 m L_c= 0.747 m and A_c= 12.21 m^2

$T_{c,new\ AVG} = 15.16\ °C$

$$Coeffcient\ Factor = \frac{T_{c,out\ new\ AVG} - T_{c,in}}{T_{c,new\ Avg} - T_{c,in}} \tag{5}$$

Find $T_{c,out,newAVG}$ value from Equation 2, at coefficient factor is 0.8

$T_{c\ out,new\ AVG} = 14.53\ °C$

At $T_{C\ out,\ AVG}$ from field test 13.55 °C

$$Error, \% = \frac{T_{c,outAVG} - T_{C,out\ new\ AVG}}{T_{c,outAVG}} \times 100 \tag{6}$$

$$Error\ , \% = \frac{14.53 - 13.55}{13.55} \times 100\%$$

$Error\ , \% = 7.23\ \%$

4. Conclusions

Straight pipe filled with R410A of fill ratio 1.00 exhibits better heat transfer properties compared to fill ratio 0.75. Thus this shows that increase in fill ratio at various heat input can contribute to a better thermosyphon performance. However based on findings from other researchers such as Hamidreza *et al.*, [11] team it was reported that there is a limit for each system which will causes the system not to perform further if it exceeds the limit.

Thermal resistance decreases exponentially with increasing input power, thus resulting better heat transfer and making the system more efficient and effective. Heat transfer performance is best at inclination angles of 68° which has the smallest thermal resistance. However based on findings of Yong *et al.*, [3] and Davoud *et al.*, [8] team dry out effect could easily occur in the case of high input power at low fill ratio. Thus indicating that each system has minimum operating limit and its performance will be affected once it goes below the limit.

Based on experimentally plotted h_e and h_c equations, total percentage error obtained comparing both theoretical and field study values are 1.61% and 7.32% for fill ratio 1.00 and 0.75. Thus this indicates the experimental and application measured readings have a difference of less than 10%.

Acknowledgement
The authors would like to acknowledge the RIGS Project ID: RIGS16-090-0254 of International Islamic University Malaysia and Daikin Malaysia Sdn Bhd for the support to carry out this work.

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