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Experimental Performance of R134a Filled Thermosyphon Heat Pipe Heat Exchanger using Plain and Rifled Tubes

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ABSTRACT

Heat pipe technology becomes popular in waste heat recovery applications and in heating, ventilation and air conditioning (HVAC) systems in recent years, especially in increasing the dehumidification efficiency and cooling capacity of the cooling coil especially in warm-climate countries. An experimental study was carried out on air-to-air thermosyphon heat pipe heat exchanger (THPHE) filled with R134a as the working fluid and a fill ratio of 60% of the evaporator volume. Two configurations were tested; plain and grooved (rifled) inner surface THPHE. For each THPHE module, the lengths of the evaporator and condenser sections were 300 mm and the central adiabatic section was 100 mm. There were 6 rows of 48 copper tubes with 12 mm outside diameter. Aluminum wavy plate fins were fixed between the tubes to increase the heat transfer area. A test rig was set up to study the thermal performance of the THPHE, different sets of experiments were carried out by varying the heat load as well as the mass flow rate inlet to evaporator section of the heat exchanger; the two THPHEs were examined under low temperature (30-60^{o}C) operating conditions. Four evaporator section air face velocities namely, 1, 1.5, 2, and 2.5 m/s were tested while ambient air flowed through condenser section with air face velocity controlled at 1.5 m/s. The results shows that the THPHE effectiveness values are shown to vary with the evaporator inlet temperature and mass flow rates. Also, the inner grooved THPHE showed a significant effect on increasing the thermal performance of the heat exchanger as compared with the plain inner surface THPHE.

1. Introduction

Thermosyphons are two-phase heat transfer devices with extremely high effective thermal conductivity. The advantage of using the thermosyphon is that transfer high rates of heat across small temperature gradients. In addition, the simplicity of design, low manufacturing cost, low weight, low cost of maintenance, etc. makes this device more demanding. Thermosyphon heat transfer has certain operating and limiting mechanisms that need to be considered before further discussing thermosyphon technologies and applications.

Figure 1 illustrates a typical two-phase closed thermosyphon, which consists of a metal pipe with a fixed amount of working fluid sealed inside. During operation, heat is added through the bottom section (evaporator) and the working fluid becomes vapor. The vapor travels through the middle section (adiabatic section) to the top section (condenser) of the tube. In the condenser, the vapor releases the latent heat to the condenser wall and becomes liquid. In contrast to a heat pipe, which utilizes capillary forces for liquid return, the thermosyphon relies on gravitational or centrifugal force to return the condensed liquid to the evaporator [1]. This phase change cycle continues as long as there is heat (warm outside air) at the evaporator end of the heat pipe. This process occurs passively (no external electrical energy required).

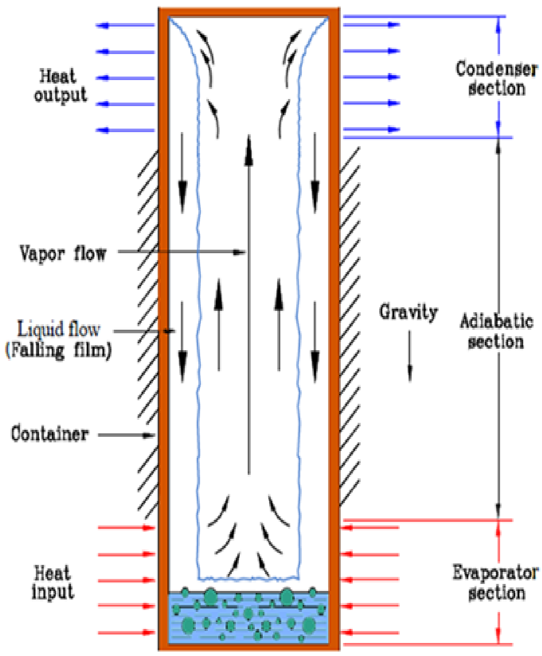

Figure 1. Gravity-assisted wickless heat pipe (two-phase closed thermosyphon).

Thermosyphon heat pipe heat exchangers (THPHE´s) are very effective heat exchangers utilized to transfer large

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amounts of heat through long distances without consuming any power for its operation. They are becoming popular in waste heat recovery and in air conditioning systems. One of the important applications of heat pipe heat exchangers is reducing energy consumption in heating, ventilation and air-conditioning (HVAC) systems in buildings, which leads to increase the dehumidification efficiency and cooling capacity of the cooling coil [2].

The most important factors affecting on thermal performance of a THPHE are:

- Velocity, relative humidity (RH) and dry-bulb temperature (DBT) of input air.
- Type and filling ratio (FR) of working fluid
- Number of rows and arrangement.
- Pipes material and inner surface configuration.

In this research, the thermal performance of two thermosyphon heat exchangers in a special test rig were investigated, those THPHEs differs in the pipe inner surface construction but both consist of 48 thermosyphons arranged in six rows. The variable parameters which were being altered are the air velocity and the inlet temperature (heat load) to the evaporator section.

2. Literature Review

In recent years, according to the global temperature increases, the energy consumption of most countries has been considerably increased. It can be anticipated that the energy consumption even has a faster growth in warm and tropical climate counties in comparison with the other counties. This is due to the fact that in the building industry of these countries, the AC systems are one of the major energy consumers, and usually the operating power cost of the AC systems accounts for more than 50% of the entire energy bills [3]. In order to save energy of the AC systems, Heat pipe heat exchanger is suggested as a new energy recovery technology.

However, there are many techniques to apply the new technologies of HPHE to reduce the energy consumption of air conditioning system. Wu et al. [4] applied the heat pipe heat exchanger to control the humidity in the air conditioning systems. Martinez [5] designed and experimental studied on a mixed energy recovery system, heat, heat pipes and indirect evaporative equipment for air conditioning system. Yau [6–8] studied experimentally the thermal performance of an inclined heat pipe heat exchanger operating in high humid tropical HVAC systems**.**

Moreover, the thermal performance of heat pipe heat exchangers has been studied experimentally and theoretically by many researchers. Noie and Majidian [9] designed, constructed and tested a heat pipe heat exchanger with a heat exchange potential of 0.8 W for heat recovery from a surgery room in a hospital. Their tests were under low temperature (15–55 ˚C) operating conditions and used methanol as the working fluid. El-Baky and Mohamed [10] designed, constructed and tested a heat pipe heat exchanger under relatively low temperature operating conditions of 15 ˚C to 35 ˚C. The results showed that the minimum heat transfer rate achieved by the heat pipe is well above the required heat transfer rate, and that the effectiveness of the heat pipe heat exchanger could be increased by increasing the number or rows of tubes in the direction of air flow.

Because of simplicity of thermosyphones, most of the researchers used a thermosyphone with plain inner surface to investigate the effect of different parameters on the HPHE performance. Noie [11] studied the thermal performance of an air-to-air thermosyphon heat exchanger consisted seamless pure copper plain inner surface pipes. The experimental results were compared with theoretical results. It was found that the minimum effectiveness of thermosyphone took place with equal mass flow rate in evaporator and condenser side.

Jouhara [12] investigated heat exchanger utilized nine plain inner surface thermosyphons in a modified inline configuration filled with water as the working fluid with finned evaporator and condenser sections. Different sets of experimental tests were carried out by varying the heat load as well as the inclination angle of the heat exchanger. Results show that significant energy savings can be achieved using thermosyphon based heat exchangers to transfer heat energy between two air streams at different temperatures.

An inner surface of a conventional plain and smooth inner surface heat pipe was initially used. However, as the investigation of thermodynamics advances, it is found that the heat transfer coefficient can be improved by forming a predetermined concave portion at the inner surface of a heat transfer pipe. Recently, the heat transfer pipe with grooved inner surface becomes the mainstream of the heat transfer pipes [13]. Hagens et al. [14] investigated an experimental and theoretical study of a heat pipe heat exchanger (as shown in figure 2), have utilized inner small spiral grooves surface thermosyphons to enhance the heat transfer in evaporation and condensation. The grooves are 0.2 mm wide and 0.2 mm deep each, separated 1 mm, under an angle of 25˚ with the vertical. They investigated the variation of air flow rate from 0.4 to 2.0 kg/s, the temperatures at the evaporator from 40 to 70 ˚C and the condenser section temperature from 20 to 50 ˚C. Their results demonstrated that a heat pipe heat exchanger is a good alternative for air–air exchangers typically in warm countries like Bahrain.

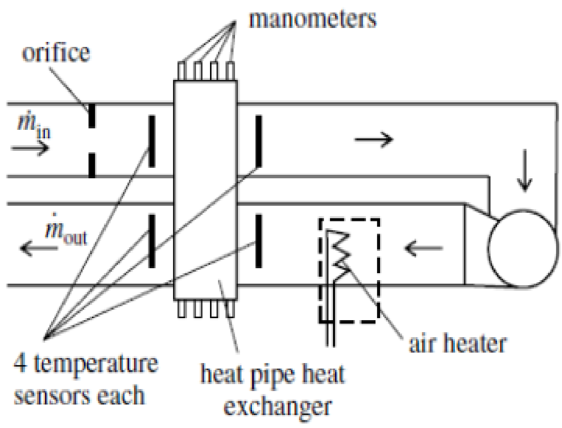


Figure 2. Schematic diagram of Hagens et al. [14] test rig.

3. Experimental set-up and procedures

Two heat pipe heat exchangers were designed and constructed with physical dimensions shown in Figure 3. Each of the THPHEs consists of a number of individual thermosyphons (plain and grooved inner surfaces pipes), R134a was used as the working fluid with a fill ratio of 60% of the evaporator section length. The first THPHE "which consists of plain inner surface" made of 48 seamless copper pipes with the specifications given in the Table 1.

The second type of THPHE contains grooved inner surface pipes (as recommended by SPC product limited) have the same dimensions in table 1 except the pipe inner diameter, figure 4A to 4C are schematic illustrations showing this thermpsyhon for heat pipe heat exchangers, Wherein figure 4A is a cross sectional view of the thermosyphon including a pipe axis line, figure 4B is a cross sectional view of the thermosyphon pipe cut along a vertical line to the pipe axis line, and figure 4C is an actual view of the rifled thermosyphons, grooved inner surface thermosyphon comprises a pipe body in which continuous spiral grooves and spiral fins are formed at an inner surface. Specifications of the grooved inner surface pipe can be presented as follows:

- The outside diameter (D) = 12 mm
- The thickness of the bottom wall (T_f) = 0.35 mm
- The groove depth (H) = 0.2 mm
- The groove angle (ridges angle) (α) = 50˚
- The helix angle, an angle with respect to the pipe axis line (torsion or rifle angle) (ß) = 18˚
- The inside diameter (d) at groove tip = 10.9 mm
- Number of grooves (no. of tooth N)= 60

Table 1. Specifications of Plain THPHE.

External diameter of pipes	12 mm
Internal diameter of pipes 14 mm	11.2 mm
Length of pipes	700 mm
Working fluid	R134a
Filling Ratio	60 %
Thickness of pipes	0.4 mm
Number of pipes	48
Pipe arrangement	Staggered
No. of Rows	6
No. of Pipes per Row	8
Tube/Row pitch	38.1/33 mm
Fin type	Aluminum
Fin thickness	0.15 mm
Fin spacing	12 fin/inch

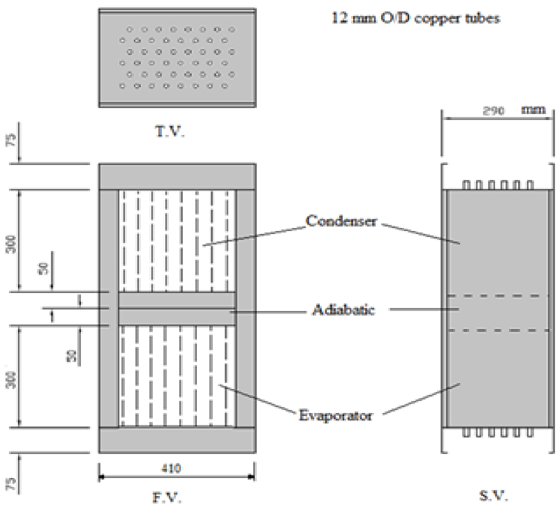


Figure 3. (a) Schematic diagram of the HPHE.

Figure 3. (b) Actual THPHEs used for investigation

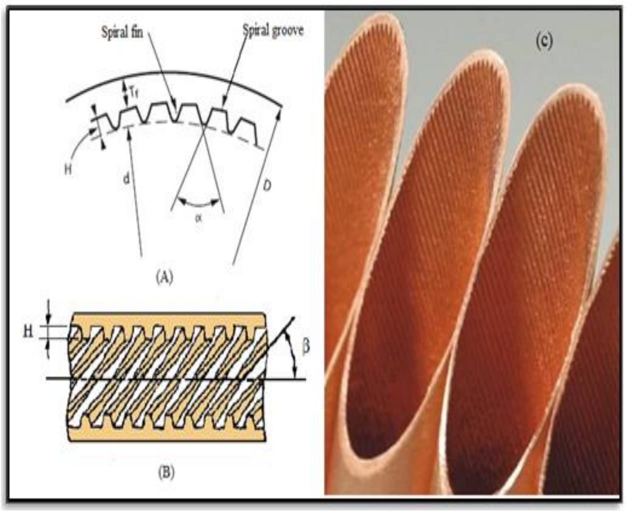


Figure 4. Thermosyphons with grooved inner surface

In order to investigate thermal performance of each THPHE, a special test rig was constructed and setup. The design of the test rig is based on the concept of using warm air to heat fresh air entering the building. With the introduction of the THPHE in the conventional cooling system pre-cooling or reheating can be achieved thereby providing substantial cost and energy savings. Figure 4 illustrates the schematic diagram of the test rig used in the investigation which composed of:

- A thermosyphon heat exchanger (THPHE)
- An axial flow fan
- A centrifugal fan and voltage transformer.
- An electrical heater (with total capacity of 9 kW)
- Temperature and relative humidity sensors.
- Data logger (Pico Technology PicoLog 1000)
- 30 cm square galvanized steel ducts; one for the hot air and the other for cool air.

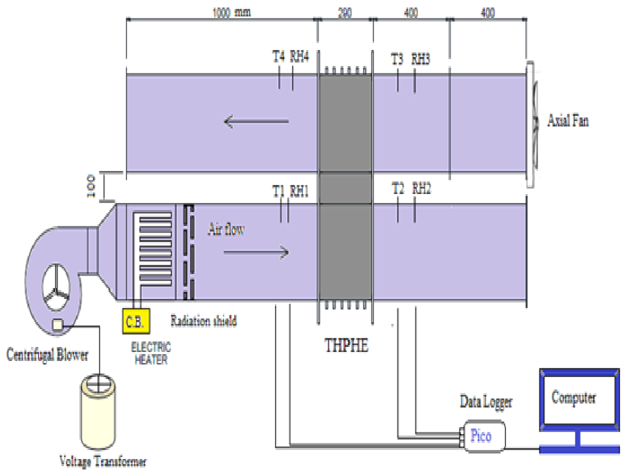


Figure 5. Schematic of the test rig

Hot air inter the evaporator section was supplied using 9 kW U-shaped powered electrical heaters cased by galvanize steel duct contains a terminal box, duct sensor goes downstream the ductwork and digital display. This heater is introduced to the test rig in order to control the input air dry bulb temperature (DBT) to the evaporator section of the THPHE. The 0.7 kW centrifugal blower (ebmpapst model D4E225-BC01-36) is used to control the air face velocity enters the evaporator section of the THPHE. This blower is speed controllable by varying the voltage across its windings which can be achieved by using a variable transformer (variac) allow the fan to generate a variety of air volume flow rates and face velocities across the THPHE evaporator section face. These methods are used to test the THPHE's effectiveness when evaporator section exposed to air flowing at a range of different temperatures and face velocities.

In order to provide a line of sight barrier between the heating elements and the measuring sensors plane, a radiation shield is placed between the electric heater and the heating elements. These shields prevent erroneous temperature and RH readings arising from exposure to the high temperature heating [15]. An axial fan installed on the other end of the ductwork with constant velocity to provide a constant air velocity enters the condenser section of the HPHE.

To ensure no mixing of air streams at the ends of the heat exchanger during the experimental procedure, flexible ducts are attached thereby ensuring the required inlet temperatures at each section. The air velocity in the hot and cool air streams of the two ductwork was measured using a hot wire anemometer type AM-4204with accuracy of 5%. During the experimental procedure, the air velocity was measured at the centre of the duct for each THPHE section. Figure 6 shows the image of the actual heat pipe heat exchanger test rig used for the experimental procedure.

High precision, longterm stable temperature sensors IC´s (TSIC 501F TO92) and humidity sensors (Honeywell HIH-4021-004) were used in the heat exchanger to measure the inlet and outlet temperatures and relative humidity respectively. The air temperatures and relative humidity at the inlet and outlet for each section are measured 150 mm away from the heat pipe heat exchanger center. from these sensors, the temperature and relative humidity readings are obtained using a 16 Channel data logging device (Data logger-PicoLog 1000), which can be operated using LABTECH CONTROL Build Time software. A real-time vision screen was developed for the software which indicates the temperatures and relative humidity at various points in the test rig.

Figure 6. Actual test rig with all measuring instruments

4. Experimental calculations

A precise experimental calculation of the THPHE thermal performance requires accurate measurement of the temperatures of the air flow and temperatures at different sections of the heat exchanger, as well as the power transferred along the heat exchanger length.

Identifying the evaporator and condenser temperature is a usually uncomplicated task and can be done by averaging the temperature measurements along the respective sections periodically. The sensible energy added and removed at the evaporator and condenser respectively was determined by performing an energy balance across the sections such that;

$$\dot{Q}_{evap.} = \dot{m}_e . Cp_e (T_1 - T_2) \quad (1)$$

$$\dot{Q}_{cond.} = \dot{m}_c . Cp_c (T_3 - T_4) \quad (2)$$

Where:

T_1 = the evaporator section inlet air temperature = $T_{e,in}$.

T_2 = the evaporator section outlet air temperature = $T_{e,out}$.

T_3 = the condenser section inlet air temperature (room temperature) = $T_{c,in}$

T_4 = the condenser section outlet air temperature = $T_{c,out}$.

The mass flow rate at each section is calculated using the velocity measurement from the ducts and given as:

$$\dot{m}_e = \rho . u_e A_{cs} \quad (3)$$

$$\dot{m}_c = \rho . u_c A_{cs} \quad (4)$$

The effectiveness is judged to be the most relevant indicator in determining the thermal performance of a THPHE as a means of energy savings. The effectiveness for an HPHE is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate [16] such that:

$$\varepsilon = \frac{\dot{Q}_{act.}}{\dot{Q}_{max.}} \quad (5)$$

Where

$$\dot{Q}_{act.} = \dot{m}_e (h_1 - h_2) \quad (6)$$

and

$$\dot{Q}_{max.} = \dot{m}_{min} (h_1 - h_3) \quad (7)$$

From Eqs. (5) to (7), it follows that the total effectiveness refers to the total thermal performance of the THPHE depending on the sensible and latent energy transferred to and from the THPHE legs can be determined by:

$$\varepsilon_{tot.} = \frac{\dot{m}_e . (h_1 - h_2)}{\dot{m}_{min} . (h_1 - h_3)} \quad (8)$$

If the air mass flow rate is constant, $\dot{m}_c = \dot{m}_e$, therefore:

$$\varepsilon_{tot.} = \frac{(h_1 - h_2)}{(h_1 - h_3)} \quad (9)$$

For each of states (1), (2) and (3), the enthalpies are determinable from the fundamental psychrometric relationship [6]:

$$h = h_a + W . h_w \quad (10)$$

h_a: is the specific enthalpy of dry air component [KJ/kg air]

W: is the humidity ratio [kg water/kg air], and

h_w: is the specific enthalpy of the water vapor[KJ/kg water].

Equation (10) may be approximated by:

$$h = Cp . T + W . h_w \quad (11)$$

Where:

Cp: is the specific heat of the dry air [KJ/kg K],

T is the dry-bulb temperature [ºC] and

h_w is the specific enthalpy of water vapor saturated at dry-bulb temperature [KJ/kg water].

Eq. (11) is good accurate in these analyses, the reason can be explained as the enthalpy of superheated water vapor at low water vapor pressures (as it exists in under-saturated air) is little different from the enthalpy of saturated water vapor (hg) at the same temperature.

Hence, an approximation used to find the enthalpies as a function of the temperature:

$$h_w = 2501 + 1.84\, T \quad (12)$$

Taking Cp at a constant value equal to 1.007 kJ/kg K and using Eq. (12), the Eq. (11) may be rewritten as:

$$h = 1.005\, T + W(2501 + 1.84\, T) \quad (13)$$

Equation (5) is used to evaluate the second type of effectiveness which is called the sensible energy effectiveness ($\varepsilon_{sen.}$) by substituting dry-bulb temperature (DBT) instead of specific enthalpy, therfore:

$$\varepsilon_{sen.} = \frac{\dot{m}_e.(T_1 - T_2)}{\dot{m}_{min}.(T_1 - T_3)} \quad (14)$$

If the air mass flow rate of the evaporator and condenser sections is constant then:

$$\varepsilon_{sen.} = \frac{(T_1 - T_2)}{(T_1 - T_3)} \quad (15)$$

The following assumptions were considered for determining the sensible and total effectiveness;

1. The test rig has a steady flow and the air is well mixed at each measuring state so that all measured thermodynamic properties are appearance of each air state.
2. The THPHE is operating under steady state conditions for supply of warm evaporator section air and fresh condenser air during each experiment.
3. No external energy is supplied into or lost from the THPHE to the surrounding.

5. Results

Many factors affect the thermal performance of heat pipe heat exchangers. In this study, the operating parameters investigated are:

A. Controlled parameters

- Working fluid: R134a
- Filling ratio: 60%
- Inlet cold air temperature into the condenser section: $T_{c,in}$ = 22-23 ˚C
- Inlet cold air velocity into the condenser section: $\nu_{c,in}$ = 1.5 m/s
- Material of tubes and fins: Copper and aluminum

B. Variable parameters

- Inner surface configuration of thermosyphons (plain and rifled)
- Evaporator section air Face velocity values 1 - 2.5 m/s with 0.5 increments (typically face velocities used in AC systems).
- Evaporator section inlet DBT values start from 30˚C to 60 ˚C with 5˚C increments "to approach the warm climates in Iraq at summer for most cities [17]" this is done by varying the electric heater power between 0.5 to 9 KW.

An ambient temperature in the laboratory facility and constant velocity were used as inlet conditions to the THPHE condenser section. Therefore, there were a total of 56 (2×4×7) experimental runs. The measurements are performed at steady state condition, and it typically took 15 min to reach steady state. Measurements were done at each condition during 12 min to check steady state condition and to guarantee proper averaging. Figure (7) shows a typical example of the temperatures reading across the THPHE during a measurement, it can be seen that the variations is very small and not exceed 0.1.

The following sub-sections describe the temperature variation across the two THPHE and the effect of the evaporator inlet DBT and face velocity on the heat flow rates from the THPHE sections; also the thermal performance results as well as the comparison of the effectiveness are discussed later in sub-sections

5.1. Effect of evaporator inlet DBT and face velocity

The effect of inlet DBT and face velocity to the evaporator section on the evaporator outlet DBT and temperature difference for the two THPHEs are shown in Figure 6 a-d. According to figure 6a and 6b, it can be seen that, under the same inlet conditions, the evaporator outlet DBT of rifled THPHE was lower than that of the plain THPHE, also for each THPHE it is clear that evaporator outlet temperatures increases at increasing the face velocity despite the fact that the increases take the same curvature for all velocities which have been selected. Figure 6c and 6d approves what was found in figures 6a and 6b, were the temperature difference ΔT_{ein} increases atincreasing the evaporator inlet temperature, also it is clear that the temperature different across the evaporator section of the grooved inner surface THPHE are larger than that of the plain THPHE.

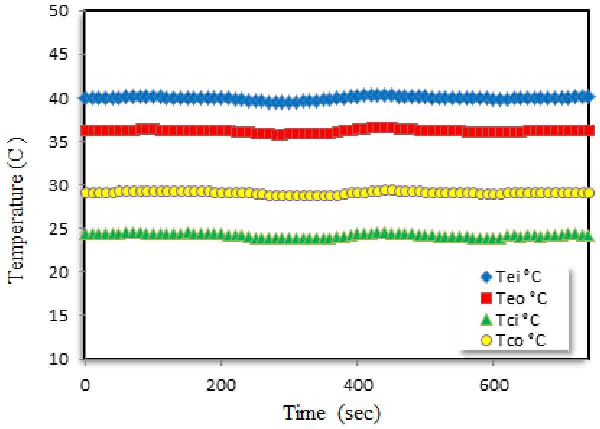


Figure 7. Typical reading of air temperatures at steady state

Figure 7 (a-d) shows the effect of evaporator face velocity on the evaporator outlet temperature and temp difference across the evaporator section of both THPHE types. It can be seen that evaporator face velocity have a significant effect on the evaporator outlet DBT were the evaporator temperature difference increases with the decreases of the evaporator face velocity (especially when the ratio of heat capacity of the evaporator section to the condenser section less than one), so that the temperature difference across the evaporator section of both THPHEs have the greatest values when the flow rate ratio (m_e/m_c) be less than one and decreases when increasing the evaporator face velocity.

Figure 8 (a-h) shows the effect of evaporator air velocity on the temperatures across the heat pipe heat exchanger for a range of evaporator inlet DBT. The resulting data show that the heat exchanger provides optimal evaporator section performance at 1 m/s evaporator face velocity for all evaporator inlet temperatures. This finding is to be expected due to the reduction of the evaporator flow rate less than the flow rate of the condenser section($\dot{m}_e/\dot{m}_c$), By observing the temperature variations across the condenser section it can be shown that the heat pipe heat exchanger exhibits a nearly isothermal behavior, which is a appropriate indicator of correct heat pipe operation.

5.2. Effect of energy added and removed at the condenser and evaporator sections

In order to check the accuracy of the recorded data, a sample of energy balance ratio (EBR) was indicated in Table 2, EBR is defined as the energy removed from the air passing through the evaporator section divided by energy transferred to the condenser section. It can be seen from Table 2 that EBR values for the tests examined at equal flow rates have EBR values approximately should be equal to unity (see Figure 11). Two possible reasons explain the EBR deviation from unity at equal flow rates, the first is possible heat transfer to the ambient air because of no satisfactory insulation and the second reason is no uniform temperature profile in measuring points because of the not well-mixed air. It was obvious that not well-mixed air at measuring points caused deviation of the EBR from unity.

Table 2. Energy balance ratio (EBR) of plain inner surface THPHE

$$EBR = \frac{\dot{m}_e\,(T_1 - T_2)}{\dot{m}_c\,(T_4 - T_3)}$$

Evaporator inlet DBT (˚C)	Evaporator face velocity (m/s)			
	1	1.5	2	2.5
	EBR			
30	1.23112	1.06086	1.23266	1.14599
35	1.1903	1.09508	1.22548	1.06287
40	1.17483	1.12663	1.13943	1.15082
45	1.1637	1.05534	1.18145	1.17421
50	1.10924	1.0699	1.19664	1.20309
55	1.12369	1.02346	1.18382	1.21883
60	1.13241	1.0373	1.17657	1.25733

Table 3. Uncertainty for energy balance ratio (EBR) and sensible effectiveness

Effectiveness	EBR	Ɛ max. error %	Ɛ max. absolute error	EBR max. error %	EBR max. absolute error
0.39167	1.23112	±4.61647	±0.01808	±11.5665	±0.1424
0.41872	1.1903	±2.57678	±0.01079	±8.65671	±0.10304
0.45652	1.17483	±1.7626	±0.00805	±7.83569	±0.09206
0.47025	1.1637	±1.3449	±0.00632	±7.52557	±0.08757
0.475	1.10924	±1.14276	±0.00543	±7.38734	±0.08194
0.48049	1.12369	±0.95787	±0.0046	±7.30181	±0.08205
0.48816	1.13241	±0.83887	±0.0041	±7.25406	±0.08215

From the above table, it can be said that the maximum experimental uncertainty associated with the obtained effectiveness and EBRvalues, is around 10%, which is anacceptable uncertainty value in engineering applications.

Figures 12 (a-h) shows the comparison of the heat flow rates at the evaporator and condenser sections for each THPHE. It can be seen that the heat flow rate of the evaporator is about 17%, 19%, 6% and 16% larger than the heat flow rate of the condenser at the evaporator velocities 2.5 m/s, 2 m/s, 1.5 m/s and 1 m/s respectively for plain inner surface THPHE tests and 40%, 34%, 7% and 35% larger than the heat flow rate of the condenser for grooved inner surface THPHE at the same above evaporator velocities. This is due to the fact that higher evaporator mass flow rate than the condenser leads to higher evaporator heat flow rate and at lower evaporator mass flow rate leads to higher the temperature difference across the evaporator section.

5.3. Effectiveness results

The results of sensible and total effectiveness for plain THPHE and rifled THPHE are plotted against the evaporator section inlet air DBT in figure 13 (a-f) at different evaporator flow rates. From the figures, it can be observed that for each type of THPHE, effectiveness generally increases slightly with increasing temperature at constant evaporator velocity.

Furthermore, ε_{sen} .and ε_{tot} values are increases rapidly with the increasing flow rate ratio ($\dot{m}_e/\dot{m}_c$) higher than one or decreasing flow rate ratio lower than one. From the figures, it can be observed that for each type of THPHE, sensible and total effectiveness generally increases with increasing temperature and increasing or decreasing flow rate ratio. Comparison between the plain and grooved inner surface THPHE under the same evaporator face velocity shows that the grooved inner THPHE has significantly higher effectiveness.

It may be observed that the effectiveness of the grooved THPHE is about 10 to 35% higher than plain inner surface THPHE at all temperature difference and flow rate ratio, this improvement in performance is due to the enhancement of the internal surface area contributed to the helical ridges inside the tube which reduced the thickness of the liquid film between the bubbles and the tube wall in the evaporator section and leads to decrease the thermal resistance. Furthermore, the tube side evaporative heat transfer coefficient is improved contributing to increased internal surface area.

In figure 14, for the two THPHE, effectiveness becomes minimum when the flow rates of the evaporator and condenser air steams are identical, suggesting that the flow rate ratio of 1 should be avoided in THPHE operation.

6. Error analysis

One method of error analysis that can be used for engineering applications is the root-sum-square (RSS) method which used for the Thermal Environmental Engineering laboratory report data analysis [18]. An error analysis was conducted for some representative experimental runs. The uncertainties of energy balance ratio (EBR) and sensible effectiveness were calculated by using RSS method. The main source of uncertainty came from the temperature measurements. The uncertainties associated with the temperature readings are given in data sheet to be ±0.1 ˚C. The equations of uncertainties associated with the calculated effectiveness values (S_ε) can be calculated from:

$$\frac{S_\varepsilon}{\varepsilon} = \sqrt[2]{\left(\frac{S_{\Delta T_\varepsilon}}{T_1 - T_2}\right)^2 + \left(\frac{S_{\Delta T_{max}}}{T_1 - T_3}\right)^2} \quad (16)$$

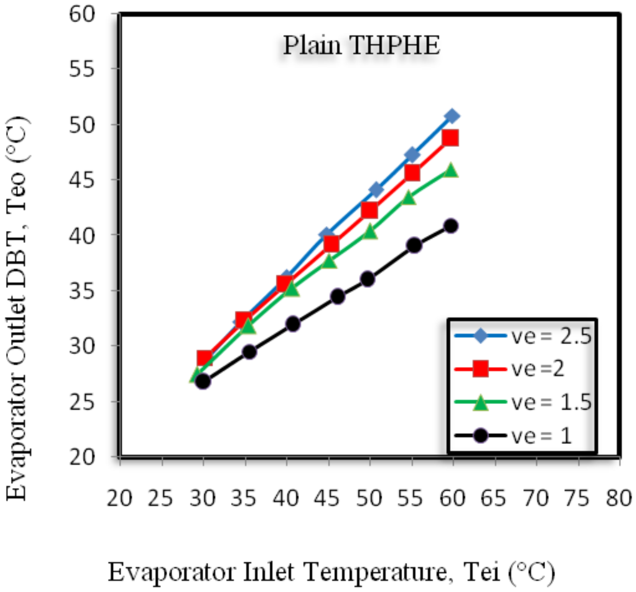
Where;

$S_{\Delta T_\varepsilon}$ and $S_{\Delta T_{max}}$ are the errors associated with ($T_1 - T_2$) and (T1 – T3) respectively, so that;

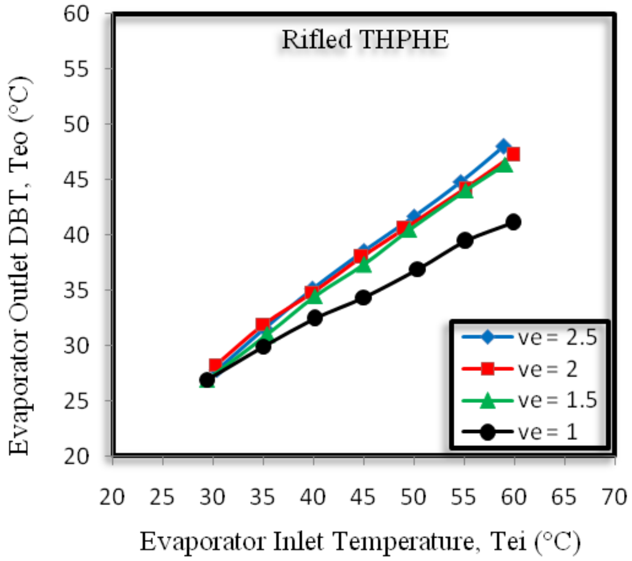
$$S_{\Delta T_\varepsilon} = \sqrt[2]{\left(S_{T_1}\right)^2 + \left(S_{T_2}\right)^2} \quad (17)$$

$$S_{\Delta T_{max}} = \sqrt[2]{\left(S_{T_1}\right)^2 + \left(S_{T_3}\right)^2} \quad (18)$$

The uncertainties related to the energy balance ratio (EBR) can be found from:



(8-a)



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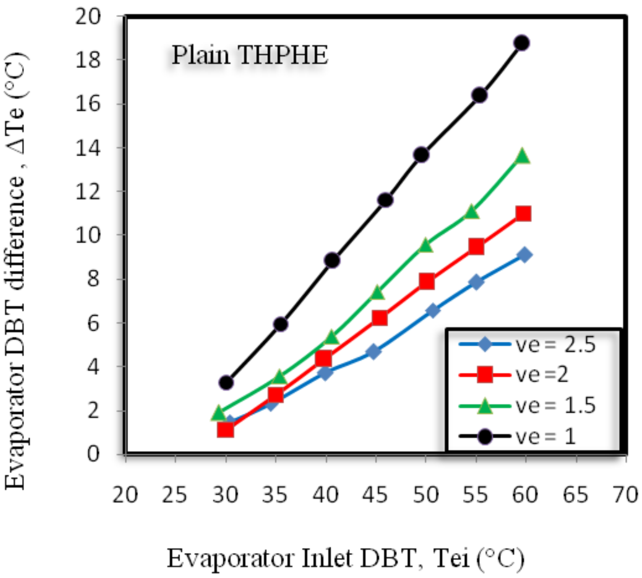
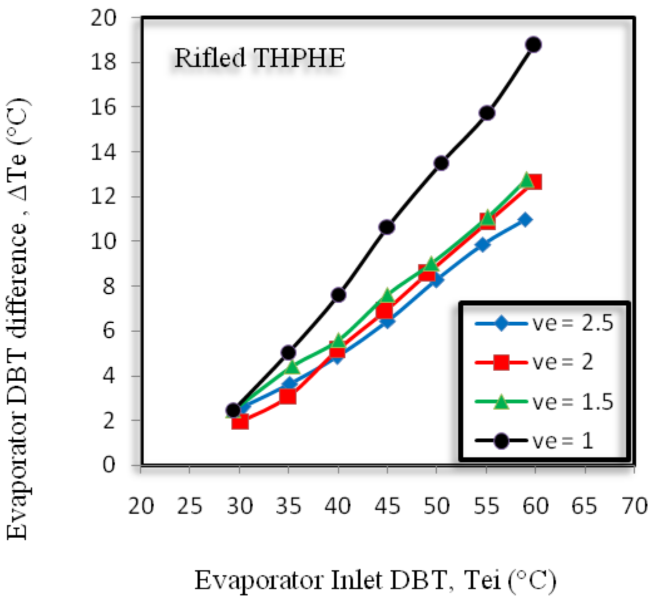
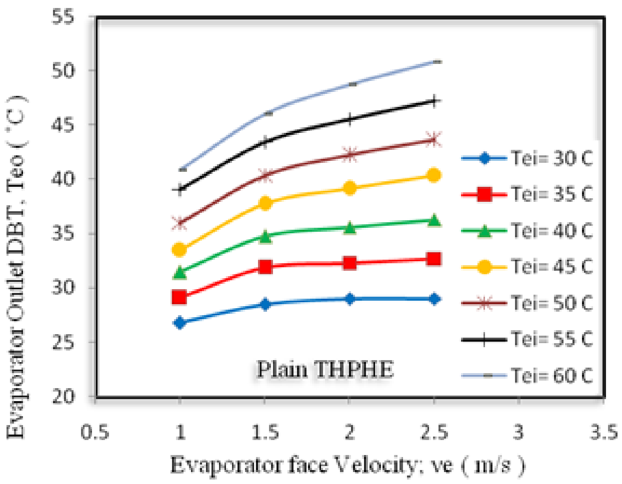


(8-c)

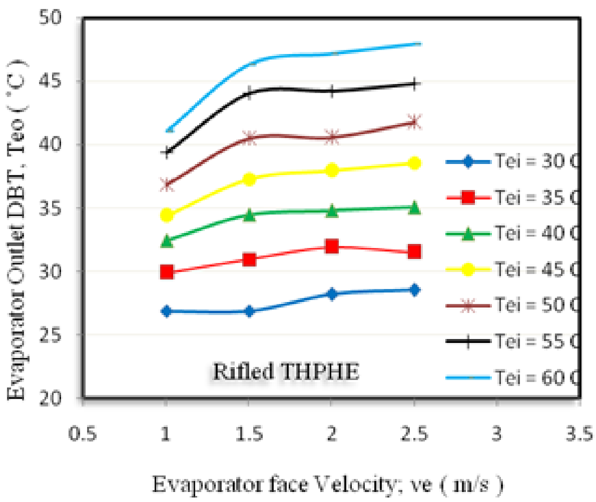


(8-d)

Figure 8 (a-d). Effect of evaporator inlet DBT on the evaporator outlet DBT.



(9-a)



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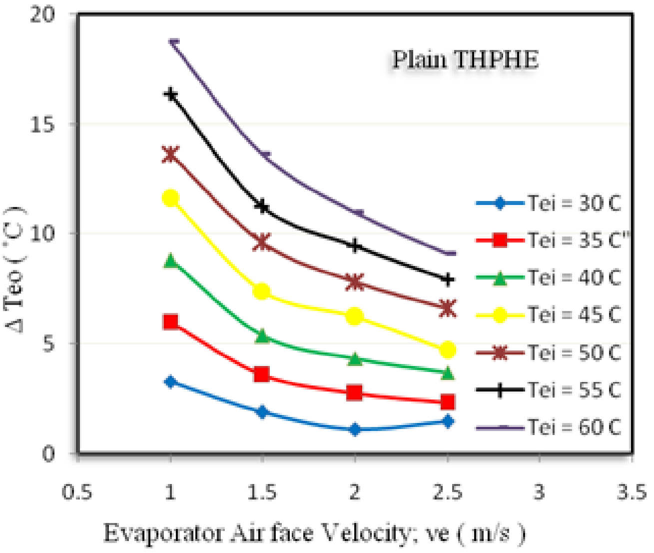

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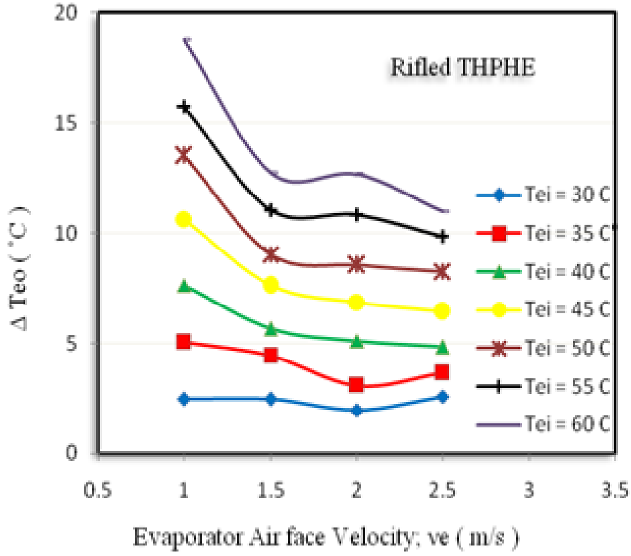
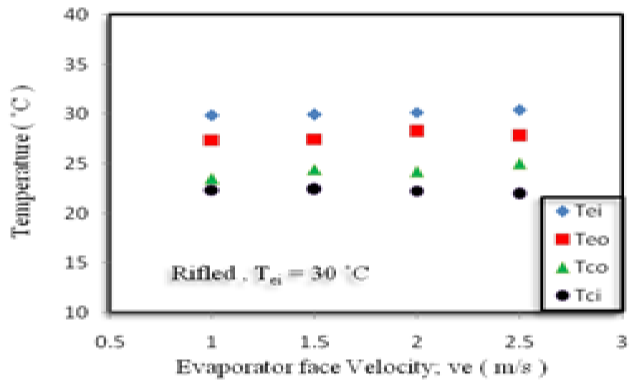
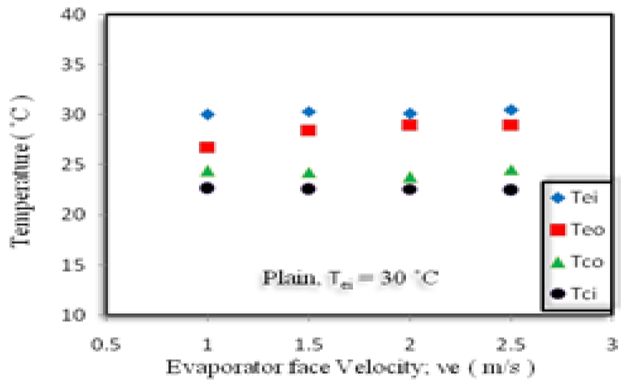


(9-d)

Figure 9 (a-d). Effect of evaporator air face velocity on the evaporator outlet temperature



(10-a)



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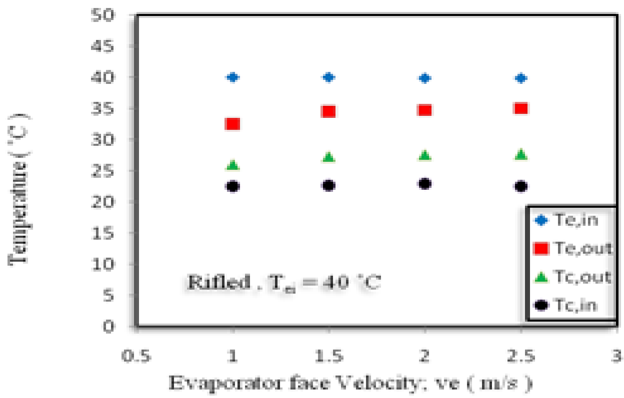

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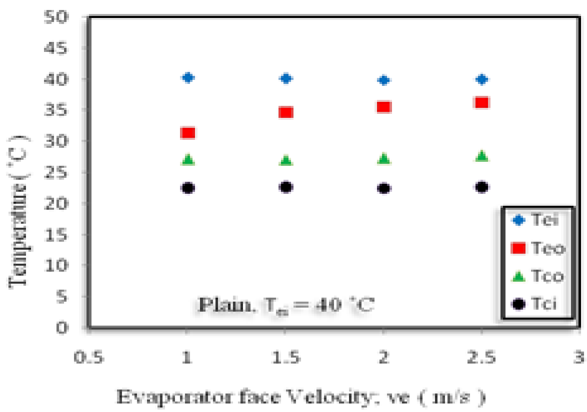
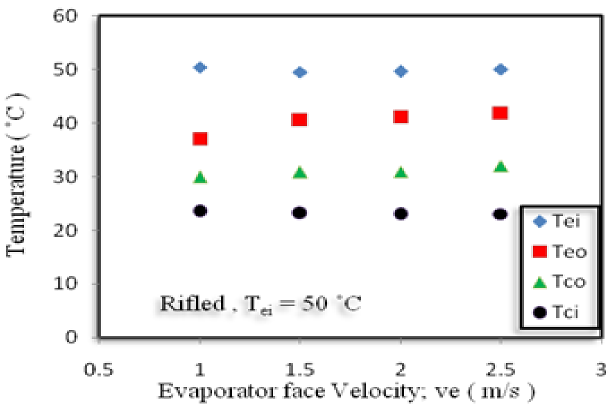
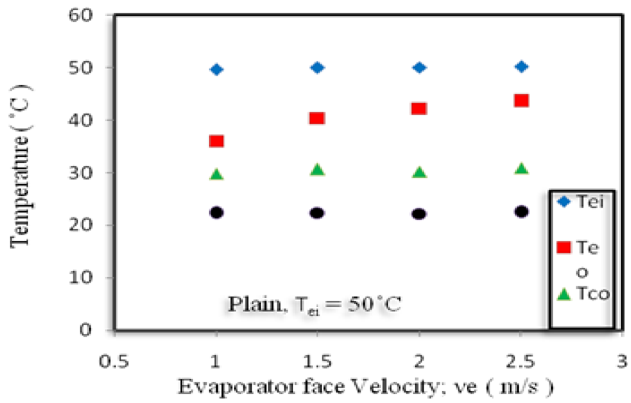


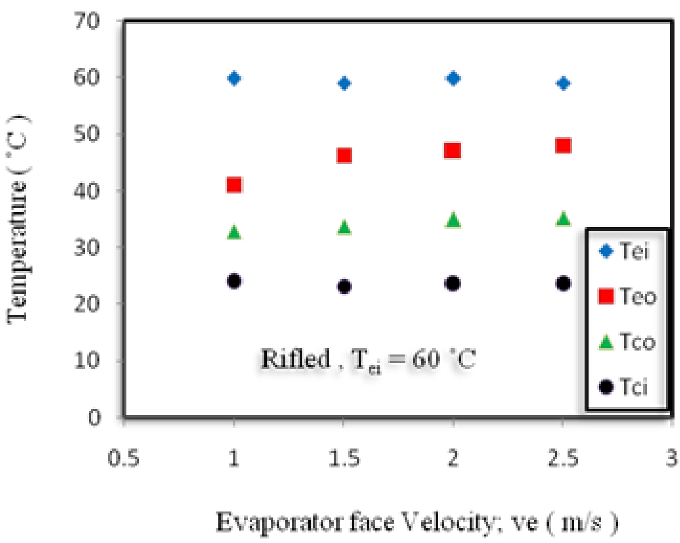
(10-d)



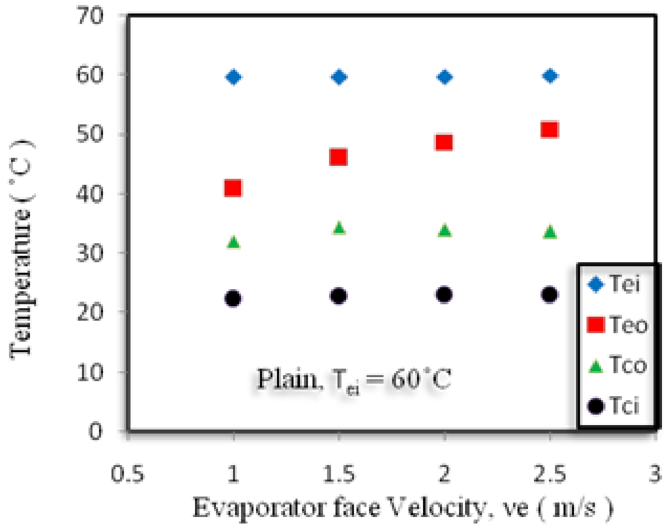
(10-e)



(10-f)



(10-g)



(10-h)

Figure 10(a-h). Effect of evaporator air face velocity on Temperatures across THPHE

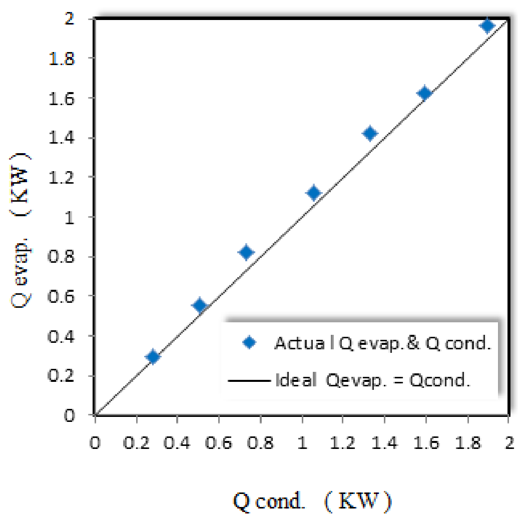


Figure 11. Comparison of heat flow rates at evaporator and condenser section of the THPHE.

$$\frac{S_{EBR}}{EBR} = \sqrt[2]{\left(\frac{S_{Q_{evap.}}}{Q_{evap.}}\right)^2 + \left(\frac{S_{Q_{cond.}}}{Q_{cond.}}\right)^2} \quad (19)$$

Where $S_{Qevap.}$ and $S_{Qcond.}$ are the error associated with the evaporator and condenser flow rates respectively, as for the air velocity, it was measured using a hot wire probe (anemometerAM-4204) which gave velocity measurements with uncertainty of ±5% rdg., so that :

$$\frac{S_{Q_{evap.}}}{Q_{evap.}} = \sqrt[2]{\left(\frac{S_{u_E}}{u_E}\right)^2 + \left(\frac{S_{\Delta T_E}}{\Delta T_E}\right)^2} \quad (20)$$

$$\frac{S_{Q_{cond.}}}{Q_{cond.}} = \sqrt[2]{\left(\frac{S_{u_C}}{u_C}\right)^2 + \left(\frac{S_{\Delta T_C}}{\Delta T_C}\right)^2} \quad (21)$$

Table (3) shows a sample of the uncertainties for energy balance ratio and sensible effectiveness for iplain THPHE in some representative experimental runs.

7. Conclusion

An experimental comparison between plain and grooved inner surface THPHE constructed of copper tubes and aluminium wavy plate fins and using R134a as the working fluid have been studied under different inlet air temperature and flow rate through the evaporator section and constant conditions inlet to the condenser section.

The experimental results show that significant energy savings can be achieved using thermosyphon heat pipe heat exchangers to transfer heat energy between two air streams at different temperatures and mass flow rates. The performance testing rig is cost effective means of testing without the need for full scale testing of a HVAC system.. It was observed that, for all cases studied, the sensible and total effectiveness increases with increase in evaporator inlet temperature and unequal ratio of flow rate between the two air streams which has major effect on the THPHE effectiveness. Also, grooved inner surface THPHE produce higher effectiveness than plain inner surface THPHE.

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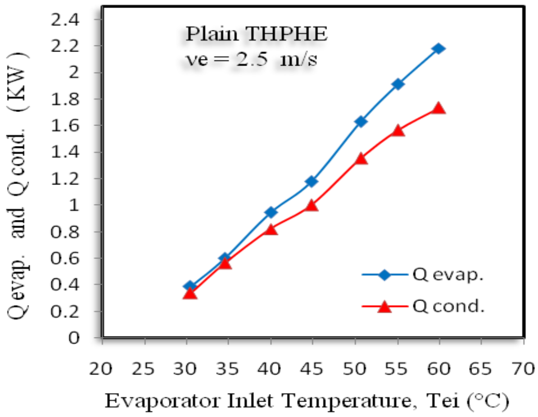
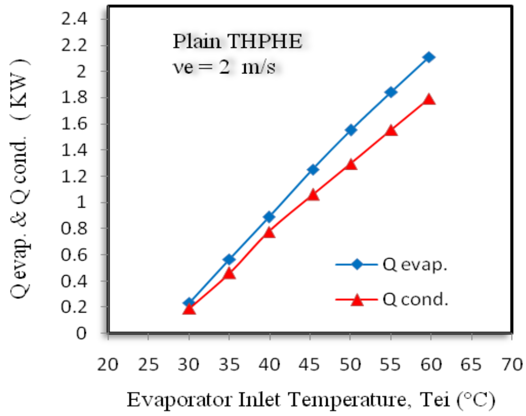
Nomenclature

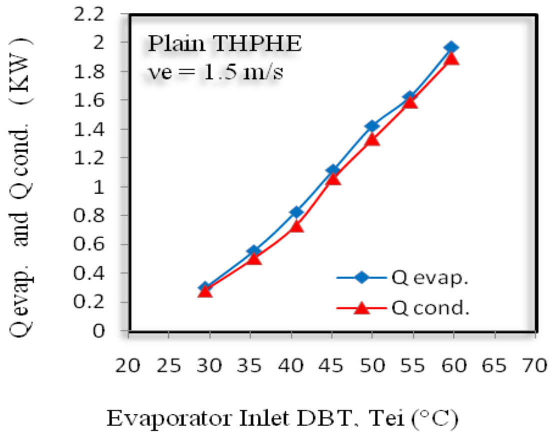
A	Heat transfer area (m^2)
Cp	Specific heat of the ambient air (J /kg K)
D	Diameter (mm)
DBT	Dry bulb temperature (˚C)
EBR	Energy balance ratio
HPHXE	Heat pipe heat exchanger
h	Specific enthalpy (k J / k g)
$\dot{m}$	Mass flow rate (kg/s)
Q	Heat transfer rate (KW)
T	Temperature (˚C)
ΔT	Temperature difference (C)
RSS	root-sum-square method
S	Uncertainty of measurements



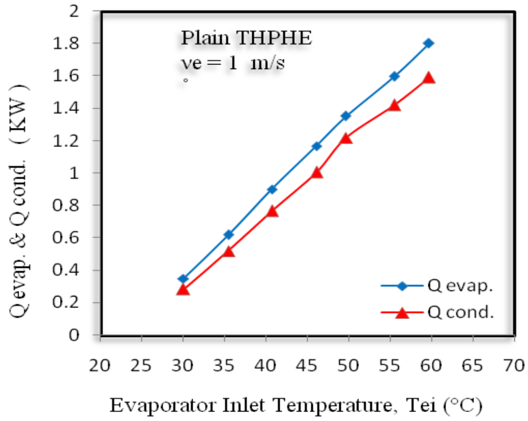
(12-a)



(12-b)



(12-c)



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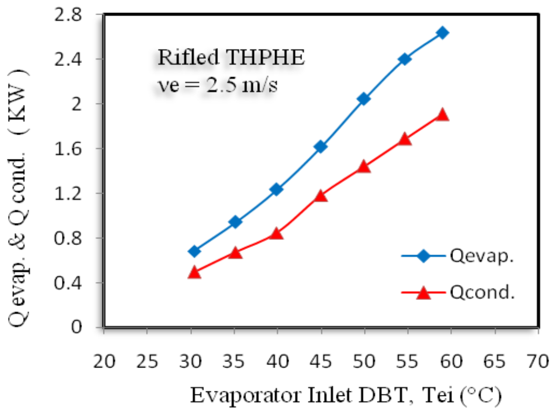
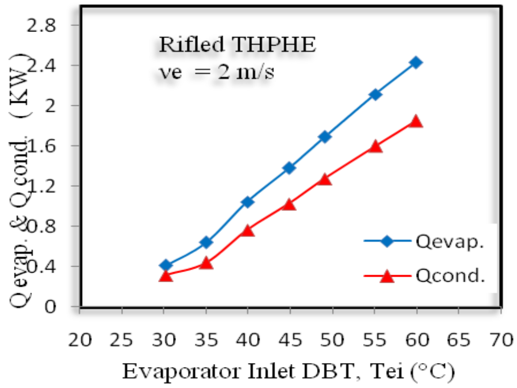
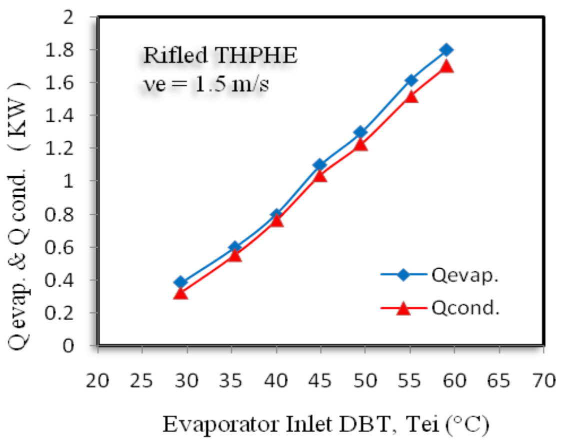


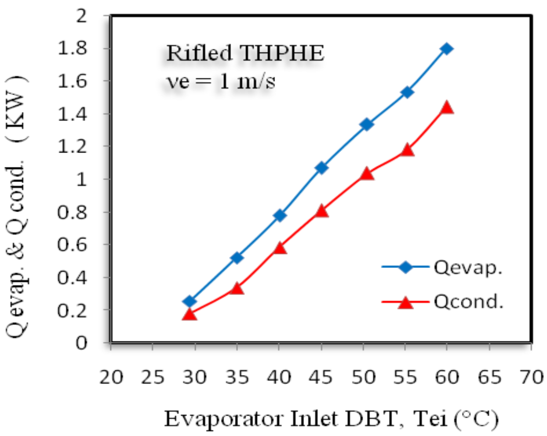
(12-e)



(12-f)

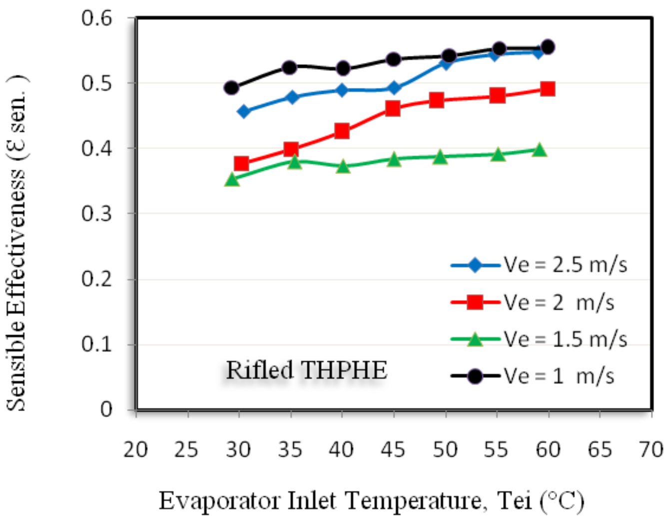


(12-g)

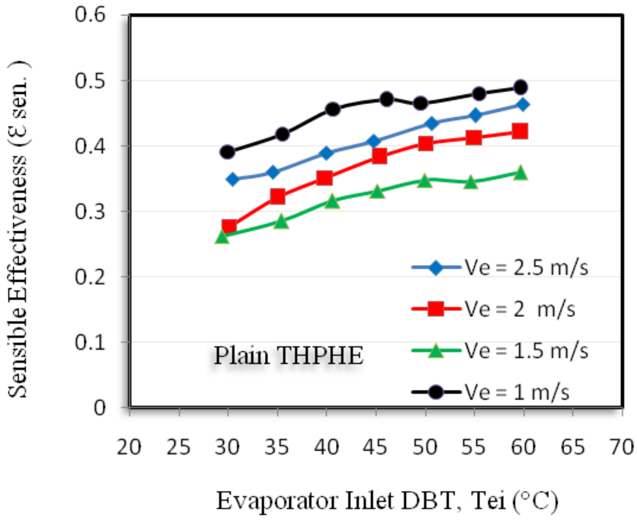


(12-h)

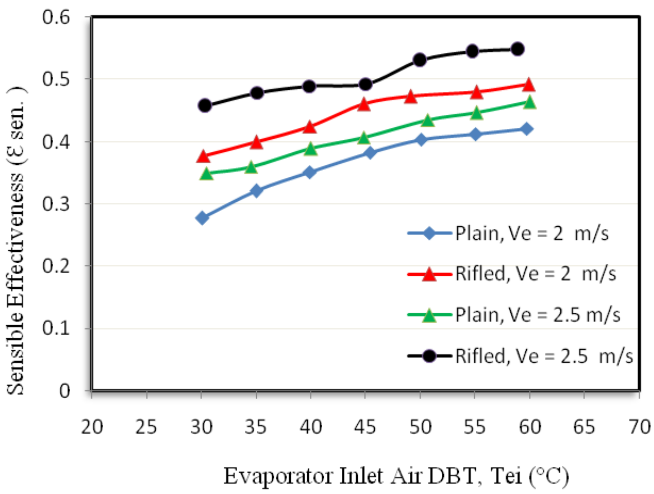
Figure 12 (a-h). Effect of evaporator and condenser heat added and removed



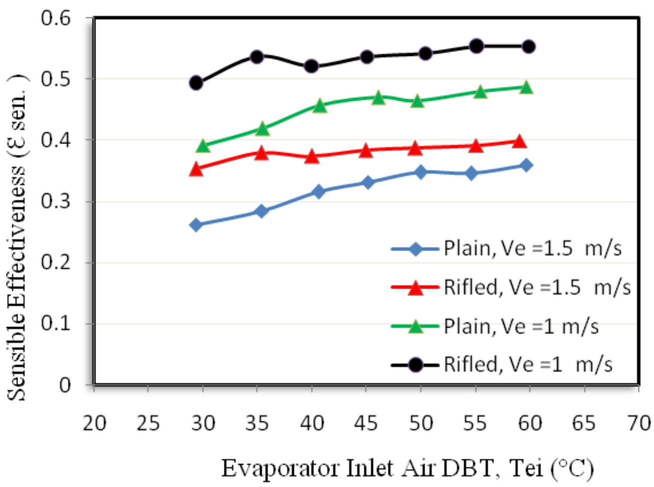
(13-a)



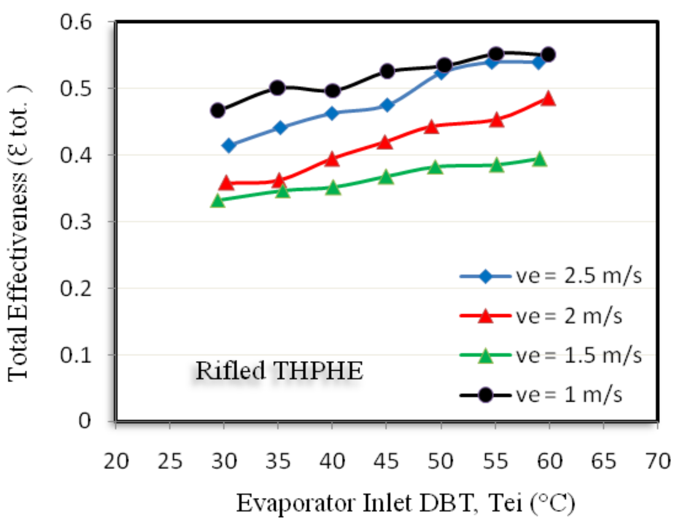
(13-b)



(13-c)



(13-d)



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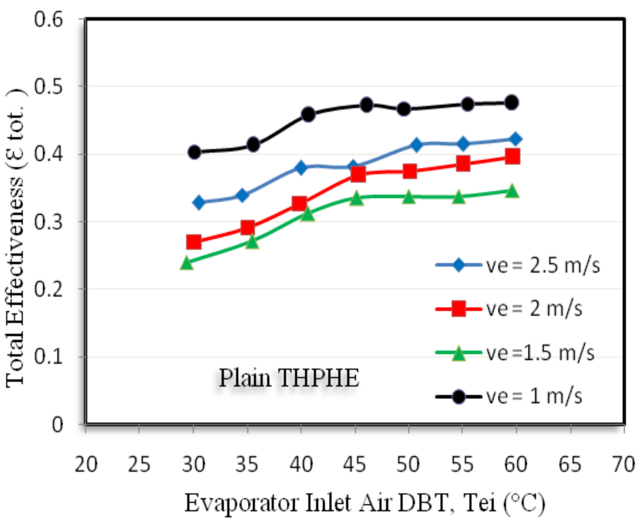
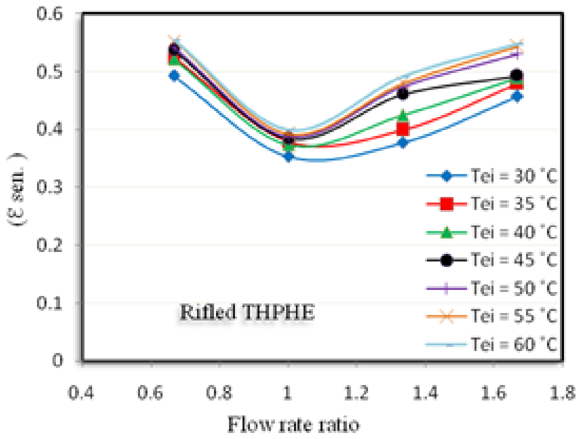
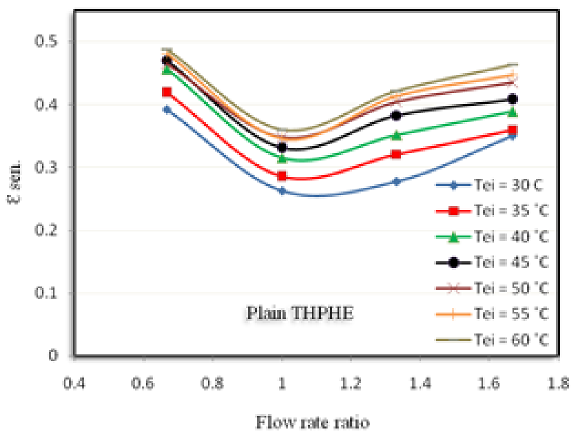


(13-f)

Figure 13 (a-f). Effect of Evaporator inlet DBT and face velocity on THPHE thermal performance



(14-a)



(14-b)

Figure 14. Change in effectiveness with air flow rate ratio

Greek symbols

Ɛ	Effectiveness

Subscripts

Act.	Actual
e, evap.	Evaporator
c, cond.	Condenser
i	Inlet
o	outlet
sen.	Sensible
tot.	Total

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