Experimental Study of Thermosiphon Heat Pipe Heat Exchanger Using Water as a Working Fluid

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Abstract:

The use of water as working fluid in thermosiphon heat pipe heat exchanger (THPHE) offers several distinct advantages over organic-based working fluid but research data appears to be still limited. In this study, the performance of water-filled heat pipe heat exchangers constructed of finned copper tubes was investigated in a test rig set up in the laboratory. Tests were carried out on heat pipe heat exchangers with two and four rows of copper tubes under different air flow rates through the condenser and evaporator, and at various evaporator inlet air temperatures of up to about 100 ^{o}C. The results show that the effectiveness of the heat pipe heat exchangers decreased with air flow rate; and when operating under different air flow rates through the condenser and evaporator, minimum effectiveness was found when the air flow rates were equal. It was found that the operation of the water-filled heat pipe heat exchanger required at least 20 K temperature difference between the air streams. With higher temperature difference, the effectiveness increased but the rate of increase tapered off appreciably beyond 45 K. for the range of operating conditions tested, the heat pipe heat exchanger produced marginally higher effectiveness than conventional cross-flow heat exchangers with two unmixed streams.

Keywords: Thermosiphon, Heat pipe, Heat exchanger.

1. Introduction:

A heat pipe heat exchanger (HPHE) is an efficient heat transfer device with good potential in process to process, process to comfort and comfort to comfort heat transfer applications to achieve energy saving is presented by [1] , [2]. By large, the majority of HPHE reported in the literature utilize organic-based working fluids. Such fluids can degrade over time, thus reducing the heat exchanger performance and operating life. Also, organic-based fluids are generally flammable and therefore any accidental leakage could pose substantial fire or explosion risks.

The use of water as working fluid in HPHE is attractive since it does not possess the above mentioned problems in addition to being cheap and readily available. The use of water-based HPHE in research or practical heat recovery applications has been reported by [3], [4], [5] but appears still limited. This study serves to provide further experimental findings on the water-based HPHE with slightly different construction features and in the lower temperature range of up to about 100 ^{o}C which is found in some exhaust conditions. The HPHE are used in heat recovery applications to cool the incoming fresh air in air conditioning applications. Two streams of fresh and return air have been connected with HPHE to investigate the thermal performance and effectiveness of heat recovery system was carried out by [6]. In this study, the temperature changes of fresh and return air are increased with the increase of inlet temperature of fresh air. The effectiveness and heat transfer for both evaporator and condenser sections are also increased when the inlet fresh air temperature is increased. The effect of mass flow rate ratio on effectiveness is positive for evaporator side and negative for condenser side.

[7] describes a theoretical analysis of a split heat pipe heat recovery system. The analysis is based on an effectiveness-NTU approach to deduce its heat transfer characteristics. In this study, the variation of overall effectiveness of heat recovery with the number of transfer units is presented. In the split heat pipe heat recovery, the evaporator is connected to the condenser through a piping system. The condenser section is located above the evaporator so that the

condensate is returned by gravity. Simple experiment for using heat pipe heat exchanger for heating automobiles using exhaust gas was carried out by [8]. It is obvious that the heat transferred by the HPHE increased with the rise of exhaust gas temperature. The effects of input heat transfer rate, the working fluid filling ratio and the evaporator length on the thermal performance of thermosiphon were investigated [9]. A HPHE was designed, constructed and tested under low temperature of 15–35°C.

10] set the variables to include overall heat transfer, effectiveness, pressure drop, and heat exchanger duty based on flow characteristics and thermosiphon configurations in heat exchangers. The effectiveness increases with increasing numbers of rows, but the effectiveness value will decrease as air mass flow value is increased. Increasing the mass flow value causes insufficient exposure to cold flow pipes. The results showed that there was not enough time for the thermal energy released by the pipe to be absorbed. [11] performed an investigation by testing an HPHE module within the air duct system. The heat pipe tubes were arranged in a staggered manner in several configuration variations. Their results showed that HPHEs can apparently reduce energy consumption in air conditioning systems.

2. EXPERIMENTAL SET-UP AND PROCEDURE

2.1. Construction of Thermosiphon Heat Pipe Exchanger

The experimental THPHE was designed in modular form. Each module was comprised of two rows of 12 mm nominal diameter copper tubes staggered and finned in a similar manner as the cooling coils in an air conditioning system. A header of 25 mm nominal diameter copper tubes was used to connect the ends of each row of tubes so that the working fluid was uniformly distributed inside the tubes; a smaller 6 mm diameter tube was brazed to one end of the header and installed with a vacuum valve for ease of charging and discharging the working fluid. The heat exchanger was divided equally into a condenser and an evaporator section, each with a face area of 0.147 m^2 and separated by an adiabatic section of 180 mm long.

Two modules were fabricated so that testing in two rows could be carried out suing one module of the heat exchanger and in four rows using two modules connected together. Figure 1 shows thermosiphon heat pipe heat exchanger (THPHE). And the specifications are summarized in table 1.

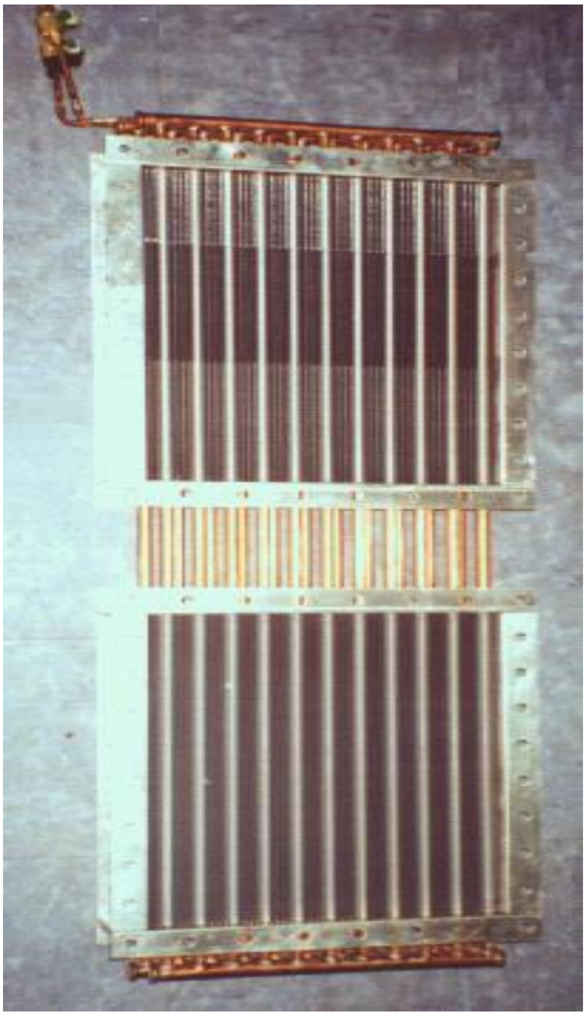


Figure 1 View of a two-row thermosiphon heat pipe heat exchanger module

Table3-2 Design specifications of the thermosiphon heat pipe heat exchanger

Item	Descriptions	
1. Casing	1020 mm (high)x 380 mm(wide)x 85 mm(deep)	
2. Thermosiphon Heath pipe	Copper tube	11 tubes in each row
	Overall length	1020 mm
	Evaporator length	420 mm
	Condenser length	420 mm
	Adiabatic length	180 mm
	Outside diameter	12.7 mm
	Inside diameter	11.8 mm
	Number of rows	2
3. Fin	Aluminum wavy plate	
	Density	315 fin per meter
	Length	320 mm
	Width	85 mm
	Thickness	0.15 mm
4. Tube arrangement	Staggered	
	Transverse pitch	27.5 mm
	Longitudinal pitch	31.75 mm
5. Working fluid	Distilled water	
	70% filling ratio	

2.2. Teat Rig and Instrumentation

The THPHE is constructed from mild steel and comprised a straight duct at the top and recalculating duct at the bottom for accommodating the condenser and evaporator sections of the THPHE respectively. A propeller fan was installed in each duct system and equipped with a speed controller to vary the flow rate. For the top duct, ambient air was drawn in through a bell mouth and passes straight through the condenser before being discharged to atmosphere. Foe the bottom duct, hot air was circulated in a closed loop through the evaporator. Six electrical finned heaters of 3.5 kW each were installed inside the duct and controlled by a temperature controller. Under this arrangement, the THPHE was tested for counter-flow performance for air face velocities at the condenser and evaporator of up to 1.5 m/s and inlet air temperatures at the evaporator of up to about 100 ^oC.

Air temperatures at the inlet and outlet of the evaporator and condenser were measured by an array of six type-T thermocouples distributed uniformly cross each section. The readings were recorded automatically by a data logging system connected to a personal computer. The air flow rate in each duct was determined by measuring the pressure differential across a flow nozzle of 100 mm throat diameter mounted inside the duct. Figure 2 shows schematic of experimental set-up.

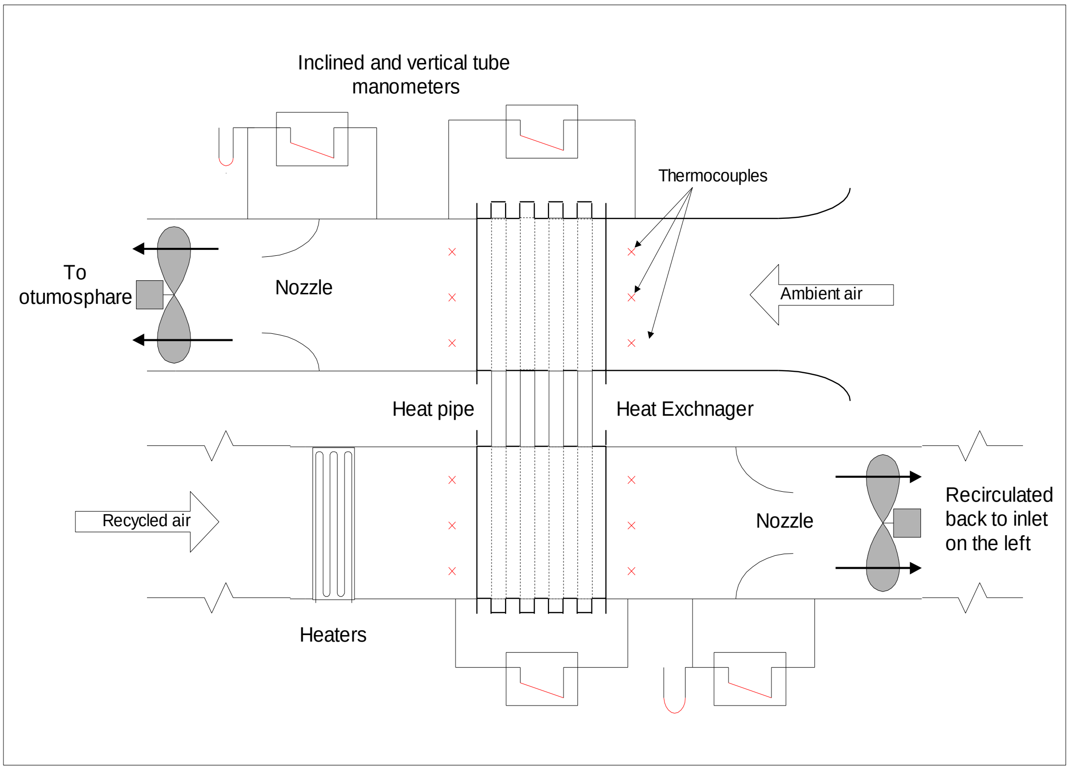


Figure 2 Schematic diagram of experimental set-up of THPHE.

2.3. Processing of experimental results

Form the experimental data obtained, the mean inlet and outlet temperatures of the evaporator and condenser were computed by averaging the readings from the thermocouples. Next, the mean air temperatures at the evaporator and condenser were calculated by taking the averages of their inlet and outlet air temperatures. Air properties were evaluated at the mean air temperatures and the air flow rates through the evaporator and condenser computed from the pressure differential across the flow nozzle (ASHRAE standard 41.2.1987) [12].

To obtain the heat transfer rates in the THPHE, the heat capacity rates C_c and C_e of the condenser and evaporator were calculated by the expressions

$$C_c = \rho_c V_c C_{pc} \tag{1}$$

$$C_e = \rho_e V_e C_{pe} \tag{2}$$

Where V is the volumetric air flow rate, ρ the air density, c_p the specific heat of air and the subscripts c and e refer to the condenser and evaporator were evaluated by the following equation:

$$Q_c = C_c (T_{co} - T_{ci}) \tag{3}$$

$$Q_e = C_e (T_{ei} - T_{eo}) \tag{4}$$

Where T_{ci} and T_{co} are the mean inlet and outlet air temperatures at the condenser respectively, and T_{ei} and T_{eo} are the mean inlet and outlet air temperatures at the evaporator respectively.

In theory, Q_c and Q_e are equal in an adiabatic system by energy conservation. In practice, noticeable differences can result from different heat losses and air leakage in the condenser and evaporator. Hence, the heat transfer rate for the THPHE was calculated by taking the mean as follow:

$$Q = \frac{Q_e + Q_c}{2} \tag{5}$$

The performance of THPHE may be quantified by its effectiveness ε, defined as the ratio of the actual heat transfer Q to the maximum possible amount of heat $Q_{\max}$ that could be transferred in an infinitely long heat exchanger. Therefore,

$$\varepsilon = \frac{Q}{Q_{\max}} \tag{6}$$

And $Q_{\max}$ was determined by the air stream with the smaller heat capacity rate $C_{\min}$ as follows:

$$Q_{\max} = C_{\min}(T_{ei} - T_{ci}) \tag{7}$$

The overall heat transfer coefficient U of the THPHE was calculated by

$$U = \frac{Q}{A\Delta T_{lmtd}} \tag{8}$$

Where A is the total heat transfer surface area of the THPHE and ΔT_{lmtd} is the log-mean temperature difference defined as follows:

$$\Delta T_{lmtd} = \frac{(T_{ei} - T_{co}) - (T_{eo} - T_{ci})}{\ln\left(\frac{T_{ei} - T_{co}}{T_{eo} - T_{ci}}\right)} \tag{9}$$

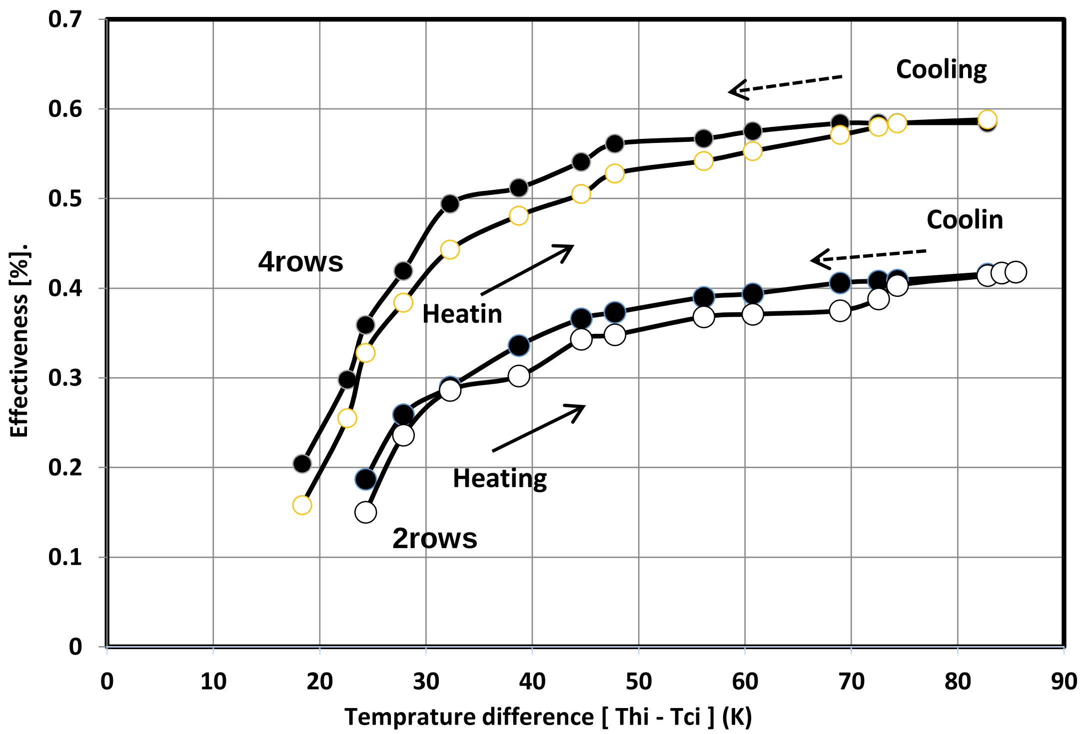
Finally, the number of heat transfer unit NTU of the HPHE was calculated by the expression.

$$NTU = \frac{UA}{C_{\min}} \tag{10}$$

2.4. Results and Discussion

2.5. Hysteresis Effect.

Tests were carried out by increasing the inlet air temperature of the evaporator in several heating steps followed by decreasing the temperature in several cooling steps. The results obtained for two different THPHE are shown in figure 3. The only difference between the THPHEs was the depth, which was represented by the number of heat pipe heat rows. For each THPHE a substantial difference between the effectiveness values during heat up and



cooling down stages was observed. In the 2 row deep thermosiphon heat pipe heat exchanger the effectiveness became independent of the thermal history and also of the temperature difference when the temperature difference was greater than 60 C^0. For the 4 row deep THPHEs similar relationship was observed if the temperature difference was somewhere around 70 C^0.

Figure 3 Effectiveness hysteresis in the two and four-rows THPHE.

2.6. Change in effectiveness under equal air flow rates through the condenser and evaporator

Tests were carried out under different air flow rates between 0.07 and 0.18m^3/s, and at evaporator inlet air temperature between 60 and 80 ^{o}C. In each test, the flow rates through the condenser and evaporator were kept constant and equal. Figure 4, shows that the effectiveness for both the two-row and four-row THPHE dropped gradually with increasing flow rate. The decrease in effectiveness with flow rate under constant inlet temperatures at the condenser and evaporator is to be expected due to the lesser increase in heat transfer rate in comparison with the increase in flow rate as may be clearly seen from equations (6) and (7).

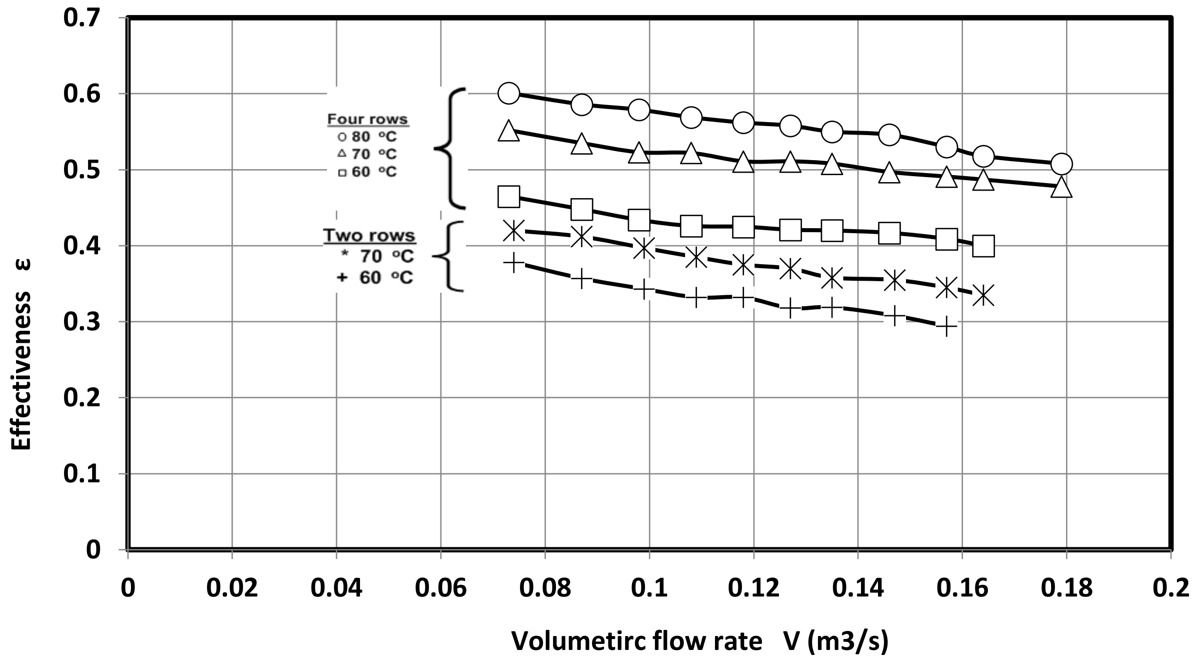


Figure 4 Effectiveness as a function of air flow rate (equal both condenser and evaporator) at different evaporator inlet air temperatures.

2.7. Change in effectiveness under different air flow rates through the condenser and evaporator

In practical heat recovery applications, the air flow rates through the condenser and evaporator need not be equal. Tests under unequal air flow rates were carried out with the condenser inlet air at ambient temperature and the evaporator inlet air temperature varying between 60 and 80 ^{o}C. The result of effectiveness versus the volumetric air flow ratio V_e/V_c is shown in figure 5 for the two-row and four-row THPHE. It has been observed that for $V_e > V_c$ the effectiveness increased with increasing flow rate ratio while for $V_e < V_c$ the effectiveness decreased with flow rate ratio. The curves converged to minimum effectiveness as V_e/V_c approached unity. The tendency towards minimum effectiveness when the flow rates became equal may be explained by expressing the effectiveness in the following form:-

$$\varepsilon = \frac{\rho_c c_{pc}\left(T_{co} - T_{ci}\right)}{\frac{V_{\min}}{V_c}\rho_e c_{pe}\left(T_{ei} - T_{ci}\right)} \qquad (11)$$

For $V_e > V_c$, V_{min} is equal to V_c and therefore the effectiveness increases with T_{co} which rises with V_e as results of better heat transfer. For $V_e < V_c$, V_{min} is equal to V_e and therefore the effectiveness decreases as the ratio V_e/V_c in the denominator increases. However, the rate of decreases is limited by a simultaneous increase in T_{co} due to better heat transfer with increasing V_e. Therefore the curves to the left of unity flow rate ratio tend to have smaller slopes. The above trends also hold true if the heat capacity rates are used instead of volumetric flow rates since the product of density and specific heat of air do not vary significantly within the temperature range of the tests.

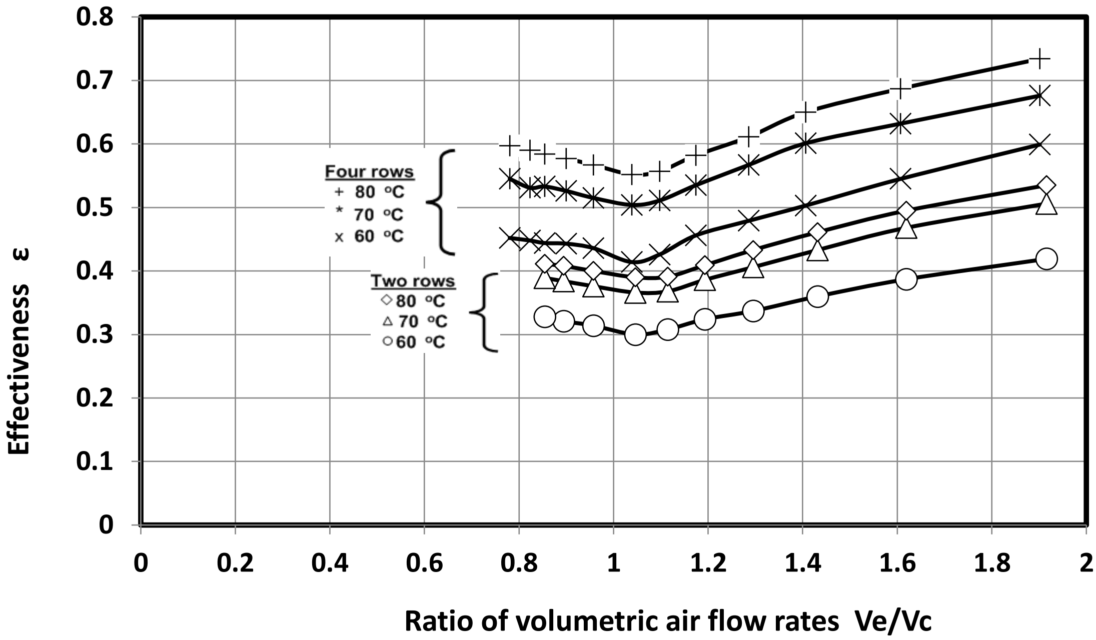


Figure 5 change in effectiveness with ratio of air flow rates through the evaporator and condenser and evaporator inlet air temperature

2.8. Change in effectiveness due to temperature different between the air streams

Tests were carried out with different evaporator inlet air temperature between 45 and 105 ºC and keeping the incoming air to the condenser at ambient temperature. The effect of temperature difference between the air streams on the effectiveness is shown in figure 6. It may be seen that effectiveness increased with higher temperature difference and flow rate ratio. The rate of increase in effectiveness tapered off appreciably beyond a temperature difference of about 45 K. Also, the THPHE was able to operate only if the temperature difference between the air streams was above 20 K.

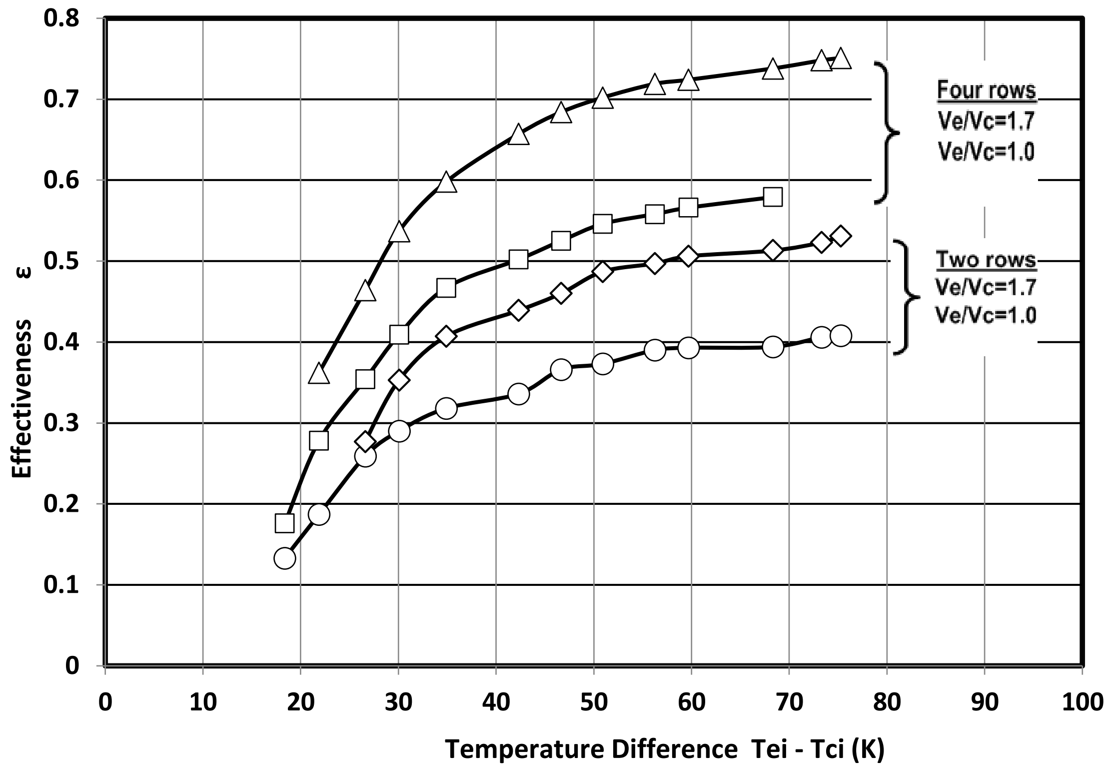


Figure 6 Change in effectiveness with temperature difference between the inlet air to the evaporator and condenser.

2.9. Effectiveness-NTU of thermosiphon heat pipe heat exchanger:

The performance data of the HPHE is plotted in terms of effectiveness versus NTU in figure 7 for different evaporator inlet temperatures. The result suggests the existence of a good correlation between effectiveness and NTU as in conventional heat exchangers. Figure 8 shows the comparison between the data from the THPHE and the effectiveness-NTU curves of cross-flow heat exchanger with both unmixed streams extracted from reference [13] and the symbol represented our experimental data. It appears that the THPHE had marginally higher effectiveness than the cross-flow heat exchanger for the range of NTU tested.

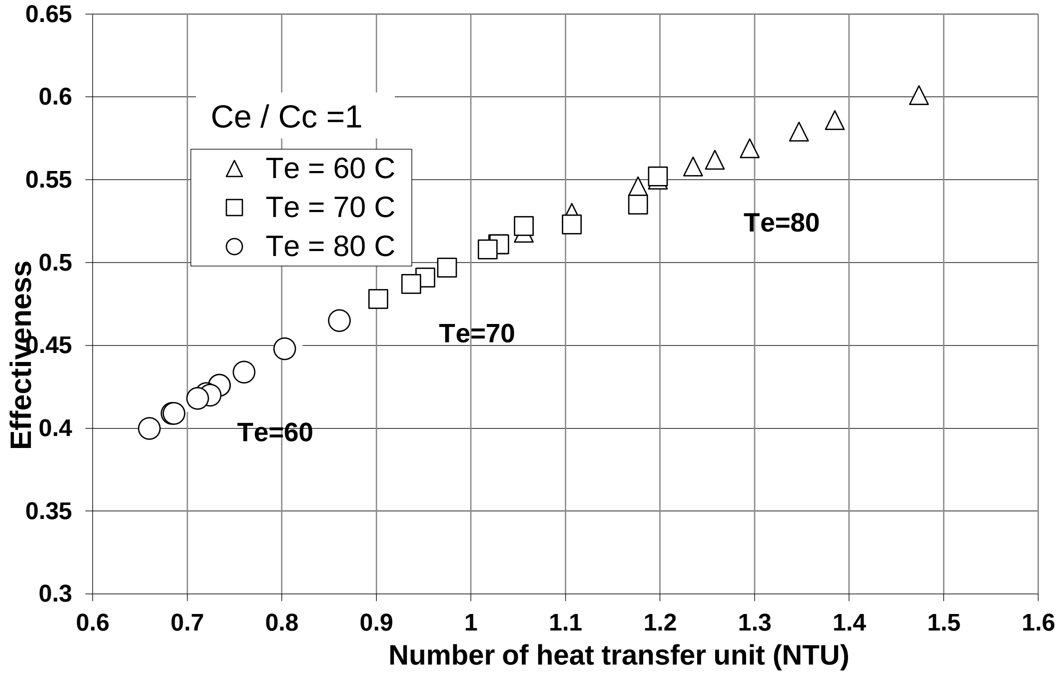


Figure 7 Effectiveness as a function of number of heat transfer unit (equal air flow rates in condenser and evaporator)

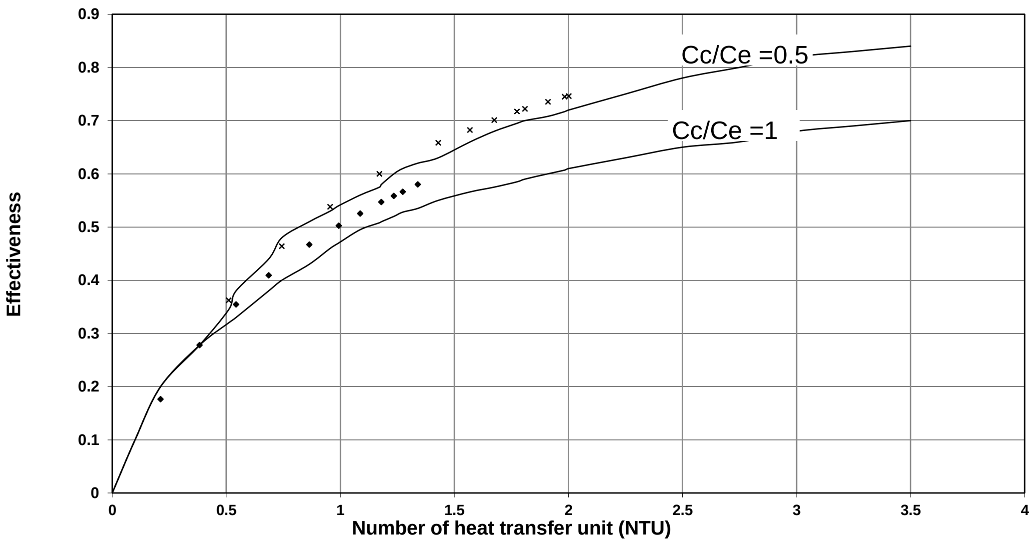


Figure 8 comparison between data of the THPHE and the effectiveness-NTU curves of a cross-flow heat exchanger with both unmixed streams.

Conclusions:

Thermosiphon heat pipe heat exchanger of finned copper tubes construction and using water as the working fluid have been tested at evaporator inlet air temperature of up to about 100 ^{o}C and under various air flow rates through the condenser and evaporator. The effectiveness of the THPHE was found to decrease gradually with air flow rate. Under different air flow rates through the condenser and evaporator, the effectiveness of the THPHE tended towards a minimum value when equal flow rates were approached. Operation of the water-filled THPHE required at least 20 K temperature difference between the air streams. With higher temperature difference, the effectiveness increased but the rate of increase tapered off beyond 45 K, finally, the effectiveness of the THPHE was found to be marginally higher than the cross-flow heat exchanger with unmixed air streams.

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