Experimental Investigation on Hydrodynamic and Thermal Performance of a Gas-Liquid Thermosyphon Heat Exchanger in a Pilot Plant

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ABSTRACT: *Waste heat recovery is very important, because not only it reduces the expenditure of heat generation, but also it is of high priority in environmental consideration, such as reduction in greenhouse gases. One of the devices is used in waste heat recovery is heat pipe heat exchanger. An experimental research has been carried out to investigate the hydrodynamic and thermal performance of a gas- liquid thermosyphon heat exchanger "THE" in a pilot plant. The ε-NTU method has been used. The pressure drop has been calculated across tube bundle of the thermosyphon heat exchanger. It's module is composed of 6 "rows" and 15 "columns" copper pipes with aluminum plate fins with dimensions of 130 cm "height", 47 cm "width" and 20 cm "depth" . The tubes have been filled by water with filling ratio of 30 %, 50 % and 70 %. The density and thickness of fins are 300 fin/m and 0.4 mm, respectively. The configuration of tubes is in-line with 30 mm pitch. The results show that as the ratio of C_e/C_c raises the amount of heat transfer increases. The effectiveness of heat pipe heat exchanger remains constant as the temperature of hot stream rises, but the amount of heat transfer increases. Filling ratio in normal region (30-70 %) has no effects on experimental results. A new correlation for thermosyphon heat exchanger with individual finned tubes and in-line geometry has been proposed for calculating pressure drop across tube bank of a "THE". The error in pressure drop for 40 experimental points in the new correlation is less than 15 %. This indicates that the new correlation possesses an acceptable accuracy predicting pressure drop.*

KEY WORDS: *Thermosyphon heat exchanger "THE", ε-NTU method, in-line configuration, Pressure drop, Finned tube bundle.*

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1021-9986/08/3/115 12/$/3.20

INTRODUCTION

Heat recovery, one of the methods of energy conservation can be successfully implemented when the investment cost of additional equipment required is acceptably low. Thermosyphon based heat exchangers are very simple devices that can be used to heat transfer between two fluid phases. Features include no cross-contamination between streams, no moving parts, compactness and no need for any external power supply. Their heat transfer coefficient in the evaporator and condenser zones is 10^3-10^5 w/m^{2}k; heat pipe thermal resistance is 0.01-0.03 k/w, therefore leading to smaller area and mass of heat exchangers [1]. Appropriate performance of a heat pipe heat exchanger "HPHE" depends on many parameters, such as hot air mass flow rate, inlet air temperature, filling ratio and pressure drop across tube bank of heat pipes.

Since the mid-1970's, researches dealing with heat pipe heat exchangers have steadily increased in number. In this section, a brief review of some of the experimental and theoretical research conducted is presented. The ε-NTU model for gravity-assigned air to air heat pipe heat exchanger was applied by *Azad* and *Geoola* [2].

They developed a new correlation for condensing water vapor on vertical carbon-steel and determined that the performance of heat pipe heat exchanger is limited by the external thermal resistances in those cases where the thermosyphon are operating below the sonic limit. *Zhongliang Liu et al.* [3,4] have studied on heat transfer characteristics of "HPHE" with latent heat storage. They have reported a new thermal storage system and a heat pipe heat exchanger with latent heat storage. The new system may operate in three basic different operation modes, the charging only, the discharging only and the simultaneous charging/discharging modes.

Also they have studied the performance of the simultaneous charging/discharging operation modes of the heat pipe heat exchanger. *Shah* and *Giovannelli* [5] studied heat pipe heat exchanger performance from a comparative point of view. The performance of a single HPHE was modeled using both LMTD and the ε-NTU method. The thermal resistances were determined using many existing correlation and the final results were compared. Also they have presented a good correlation for predicting pressure drop across tube bundle of HPHE. *Tan* and *Liu* [6] have used the ε-NTU method to analyze an air to air heat pipe heat exchanger. They also have presented an equation to determine the optimum position separating a heat pipe into evaporator and condenser regions in a heat pipe heat exchanger was formulated by minimizing the total thermal resistance of the heat path. *Wadowski, T., et al.* [7], an experimental study of the performance of an air to air thermosyphon-based heat exchanger utilizing R-22 as the working fluid has been carried out to investigate its behavior under different operating conditions.

Yang et al. [8] have built a thermosyphon heat pipe heat exchanger for recovery of the gas heat emitting from automobile exhausts, and they have investigated thermal performance of "THE". *Noie* and *Majidian* [9] have built a "THE" for recovery the heat waste in hospital and laboratories. Also an experimental study of the performance of an air to air thermosyphon-based heat exchanger utilizing water as working fluid has been carried out to investigate its behavior under different operating conditions by *Noie* [10]. *Song Lin et al.* [11] have presented a design method by using CFD simulation of the dehumidification process with heat pipe heat exchanger. Their studies illustrate that the CFD modeling is able to predict the thermal performance of the dehumidification solution with HPHE. Detailed pressure drop analysis of various tubes and fin geometries has been presented by *Kays* and *London* [12], and *Rohsenow et al.* [13].

In this research, we have investigated pressure drop across tube bundle and thermal performance of thermosyphon heat exchanger, experimentally and theoretically.

THEORY

Thermal analysis

The analysis of the heat transfer aspects of heat pipe heat exchanger is based on the heat transfer rate equation obtained by an energy balance of the heat exchanger:

$$Q = U \cdot S(T_h - T_c) \quad (1)$$

There are two main approaches for designing of a HPHE:

1) The Log-mean temperature difference model (LMTD).

2) The effectiveness-number of transfer units model (ε-NTU) [14, 15].

ε-NTU Method

The ε-NTU method is based on the heat exchanger effectiveness, ε, which is defined as the ratio of the actual heat transfer of a heat exchanger to the one that would have occurred in a heat exchanger with infinite surface. The exit temperature of the low-temperature fluid would equal the inlet temperature of the high-temperature fluid. Therefore, the effectiveness can be defined as [16]:

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_h\left(T_{h,in} - T_{h,out}\right)}{C_{min}\left(T_{h,in} - T_{h,out}\right)} = \frac{C_c\left(T_{c,out} - T_{c,in}\right)}{C_{min}\left(T_{h,in} - T_{c,in}\right)} \tag{2}$$

Applying conservation of energy, the general exponential function for a counter-flow heat exchanger is [17]:

$$\varepsilon = \frac{1 - \exp\left[\frac{U_t S_t}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}{1 - \frac{C_{min}}{C_{max}}\exp\left[\frac{U_t S_t}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]} \tag{3}$$

The ratio of $U_t S_t / C_{min}$ is defined as the number of transfer units:

$$NTU = \frac{U_t S_t}{C_{min}} \tag{4}$$

The C_{min} and C_{max} are the minimum heat capacity and the maximum heat capacity of fluid through the thermosyphon heat exchanger.

$$C_{min} = \left(\dot{m}C_p\right)_{min} \ , \ C_{max} = \left(\dot{m}C_p\right)_{max} \tag{5}$$

And the C_e and C_c are the heat capacities of the fluid streams in evaporator and condenser sections of the heat pipe heat exchanger, respectively.

$$C_e = \left(\dot{m}C_p\right)_e \ , \ C_c = \left(\dot{m}C_p\right)_c \tag{6}$$

$$\frac{C_e}{C_c} = \frac{\left(\dot{m}C_p\right)_e}{\left(\dot{m}C_p\right)_c} \tag{7}$$

The heat capacity ratio of high- and low-temperature fluid streams (C_e/C_c) is being used to investigate thermal performance of "THE".

Therefore, the effectiveness can be obtained by the following correlations:

$$\varepsilon = \frac{T_{c,put} - T_{c,in}}{T_{h,in} - T_{c,in}} \ , \text{ if } \quad C_e < C_c \tag{8}$$

$$\varepsilon = \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} \ , \text{ if } \quad C_e > C_c \tag{9}$$

Due to phase change, the maximum heat capacity is several orders of magnitude larger than the minimum heat capacity, $(C_{min}/C_{max} \approx 0)$.

The effectiveness will be expressed as:

$$\varepsilon = 1 - \exp\left(-NTU\right) \tag{10}$$

The effectiveness of evaporator and condenser sections of the heat pipe heat exchanger can be defined as:

$$\varepsilon_e = 1 - \exp\left(-NTU_e\right) \tag{11}$$

$$\varepsilon_c = 1 - \exp\left(-NTU_c\right) \tag{12}$$

Where ($NTU_e = U_e.S_e/C_e$) and ($NTU_c = U_c.S_c/C_e$).

These correlations have been defined for a single row of pipes, while the effectiveness of a heat pipe heat exchanger with *n* rows of pipes is as follows:

$$\varepsilon_{e_n} = 1 - \left(1 - \varepsilon_{e_1}\right)^n \tag{13}$$

$$\varepsilon_{c_n} = 1 - \left(1 - \varepsilon_{c_1}\right)^n \tag{14}$$

At least overall effectiveness of heat pipe heat exchanger is obtained by the following correlations [2]:

$$\varepsilon_o = \frac{1}{\left(1/\varepsilon_{c_n}\right) + \left(\frac{C_c}{C_e}\Big/\varepsilon_{e_n}\right)} \quad \text{if} \quad C_e > C_c \tag{15}$$

$$\varepsilon_o = \frac{1}{\left(1/\varepsilon_{e_n}\right) + \left(\frac{C_e}{C_c}\Big/\varepsilon_{c_n}\right)} \quad \text{if} \quad C_e < C_c \tag{16}$$

Determination of the overall heat transfer coefficient

To determine the overall heat transfer coefficient, the heat transfer can be modeled as a thermal resistance network shown in Fig. 1.

$$\frac{1}{US} = \frac{1}{\left(\eta_o hs\right)_c} + R_{f,c} + R_{hp} + R_{f,h} + \frac{1}{\left(\eta_o hs\right)_h} \tag{17}$$

$$\frac{1}{U_h S_h} = \frac{1}{\left(\eta_o hs\right)_h} + \frac{1}{2\pi k_w L_e}\ln\left(D_o/D_i\right) \tag{18}$$

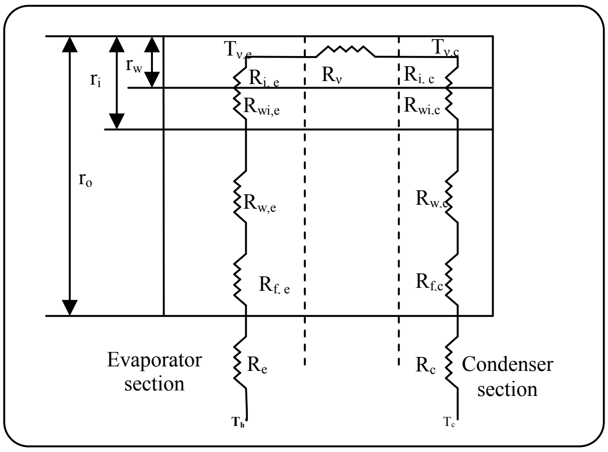

Fig. 1:Thermal resistance network of a "THPHE".

$$R_h = \frac{1}{(\eta_o hs)_h} \quad , \quad R_{w,h} = \frac{1}{2\pi k_w L_e} \ln(D_o/D_i) \tag{19}$$

In this research, it was assumed that fouling resistances due to corrosion or oxidation as well as resistances terms which occurred due to heat transfer through the liquid saturated wick are negligible.

Therefore, for the condenser section, we have:

$$\frac{1}{U_c S_c} = \frac{1}{(hs)_c} + \frac{1}{2\pi k_w L_c} \ln(D_o/D_i) \tag{20}$$

$$R_c = \frac{1}{(hs)_c} \quad , \quad R_{w,c} = \frac{1}{2\pi k_w L_c} \ln(D_o/D_i) \tag{21}$$

Pressure drop analysis

Heat pipe heat exchanger design is a complex problem which involves both quantitative calculations and qualitative judgments. Heat transfer between the high and low temperature fluids and the pressure drop of the fluids as they flow across the "THE" core are two major design criteria.

The amount of pressure drop is highly dependent on the geometry of the tubes and fins, mass flow rate across tube bundle, maximum velocity of flow through tube bundle and temperature of flow [18].

Kay's and London's correlation may be regarded as one of the best methods for calculating pressure drop across heat pipe heat exchanger with various tubes and fin geometries. The fluid flow configuration in the core of a heat pipe heat exchanger is normal to either a bare or a finned bank of tubes. The fractional pressure drop for flow normal to tube banks is given by:

$$\frac{\Delta P}{P_{in}} = \frac{G^2}{2P_{in}\rho_{in}}\left[\left(1+\left(\frac{A_c}{A_f}\right)^2\right)\left(\frac{\rho_{in}}{\rho_{out}}\right)+f\,\frac{A_o}{A_c}\frac{\rho_{in}}{\rho_m}\right] \tag{22}$$

Where $G = \frac{\dot{m}}{A_c}$

Friction factor correlation is empirically based on the Reynolds number and geometric parameters. For a bank of individually finned tubes with various geometries, it is obtained by:

$$f = \left\{0.44 + \frac{0.08(S_L/d)}{\left[\frac{(S_T-d)}{d^{0.43+1.13d/S_T}}\right]}\right\} Re_{max}^{-0.15} \tag{23}$$

For in-line configuration of a tube bank and

$$f = \left\{0.25 + \frac{0.118}{\left[\frac{(S_T-d)}{d}\right]^{1.08}}\right\} Re_{max}^{-0.16} \tag{24}$$

for triangular configuration.

For individual circular finned tubes equation (22) can be written as:

$$\Delta P = 2nf'\frac{G^2}{\rho_{in}} + G^2\left(\frac{1}{\rho_{out}} + \frac{1}{\rho_{in}}\right) \tag{25}$$

Where, f′ is a modified friction factor per tube row. For flow normal to a circular finned bank of tubes the correlation of *Robinson* and *Briggs* [19] can be used to obtain f′:

$$f' = 9.465\, Re_d^{-0.316}\left(\frac{X_t}{d_o}\right)^{-0.937} \tag{26}$$

Here, the Reynolds number is based on the outside tube diameter.

EXPERIMENTAL SET UP AND PROCEDURE

In this research, we have designed and built a pilot plant for data acquisition purposes with following characteristics:

"THE" module is composed of 6 "rows", 15 "columns" of copper pipes with aluminum plate fins with dimensions of 130 cm (height), 47 cm (width), and 20 cm (depth) which have been filled with water of filling ratio of 30 %, 50 % and 70 %. The density and the thickness of fins are 300 fin/m and 0.4 mm, respectively. The configuration of tubes is in-line with 30mm pitch (Fig. 2, table. 1).

The test rig has two sections, top and bottom. The top section is condensation part of HPHE in which cold water is pumped into it with constant flow rate of 7 lit/min at about 17 °C; the bottom section is evaporation part of HPHE. The bottom duct is straight and forms a closed looped. A centrifugal blower and 90 electrical heaters were installed inside the duct to circulate hot air through the evaporator section. In the bottom duct mass flow rate varies by changing the input electrical frequency of the blower in the range of 20-70 HZ; resulting variation of the mass flow rate within the range of 0.15-0.55 kg/s. Thus the heat capacities ratio fluid streams (C_e/C_c) vary in the range of 0.34-1.27. The pressure drop between inlet and outlet of "THE" has measured by inclined manometer. The inlet hot air temperature has been controlled and to be kept constant at 100, 125, 150, 175, and 200 °C while the inlet heat into the evaporator section is in the range of 6-42 kw. The schematic of the test rig is illustrated in Figs. 3 and 4.

RESULTS AND DISCUSSION

Thermal performance and pressure drop across the tube bundle of a "THE" has been investigated, theoretically and experimentally.

Experimental results of thermal analysis

The effects of various parameters such as the heat capacity ratio of high- and low-temperature fluid streams "Ce/C_c", the inlet hot air temperature, and the mass flow rate or the inlet hot air velocity on thermal performance of a gas-liquid "THE" have been investigated, experimentally and theoretically. The following results have obtained.

Heat capacities ratio effect (C_e/C_c)

Fig. 5 shows the effectiveness and the rate of heat transfer vs. C_e/C_c for all of the inlet hot air temperatures. The heat capacity ratio of high- and low-temperature fluid streams affects on the effectiveness and the rate of heat transfer of "THE". When the heat capacity ratio of high- and low-temperature fluid streams is higher than unity the effectiveness increases due to the ability

Table 1: Specifications of thermosyphon heat pipe heat exchanger.

Aluminum plate Thickness 0.4 mm Density 300 fin/m	Fin
In-line $S_L = S_T = 30$ mm	configuration
$N_L = 6$, $N_T = 15$	Num. of heat pipe rows
$N_{Total} = 90$	Total num. of heat pipe
Copper-distillated water	Material & working fluid of HP

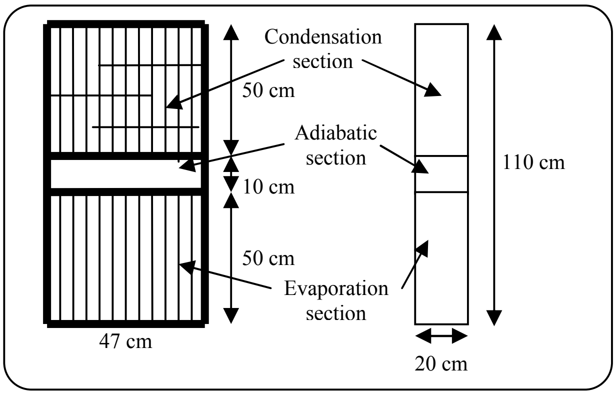

Fig. 2: Dimensions of the "THPHE".

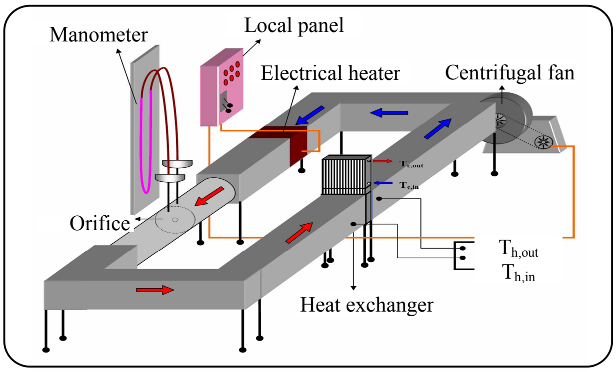

Fig. 3: 0verall schematic of pilot plant.

Fig. 4: Photo of pilot plant.

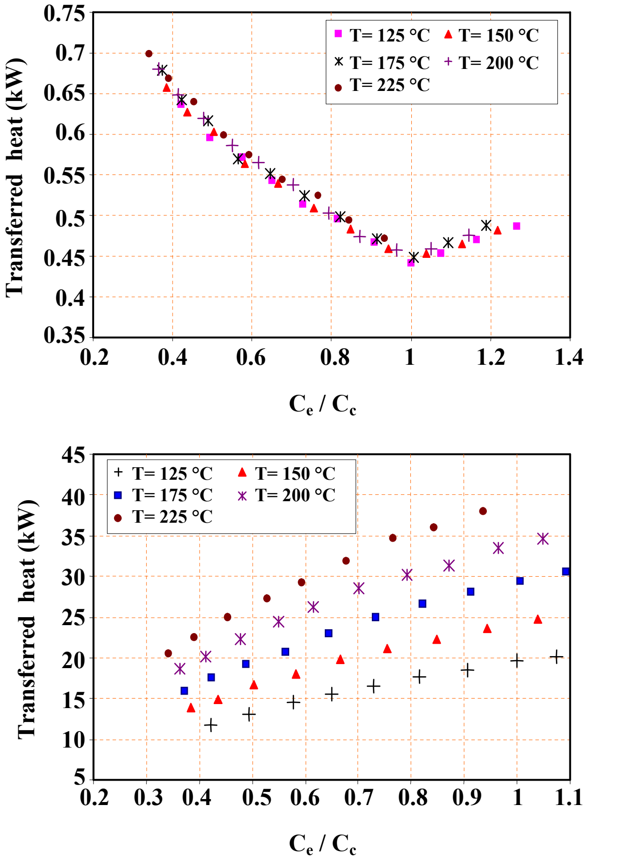

Fig. 5: The effectiveness and the rate of heat transfer of "THE" vs. the ratio of C_e/C_c (For the entire inlet hot air temperature).

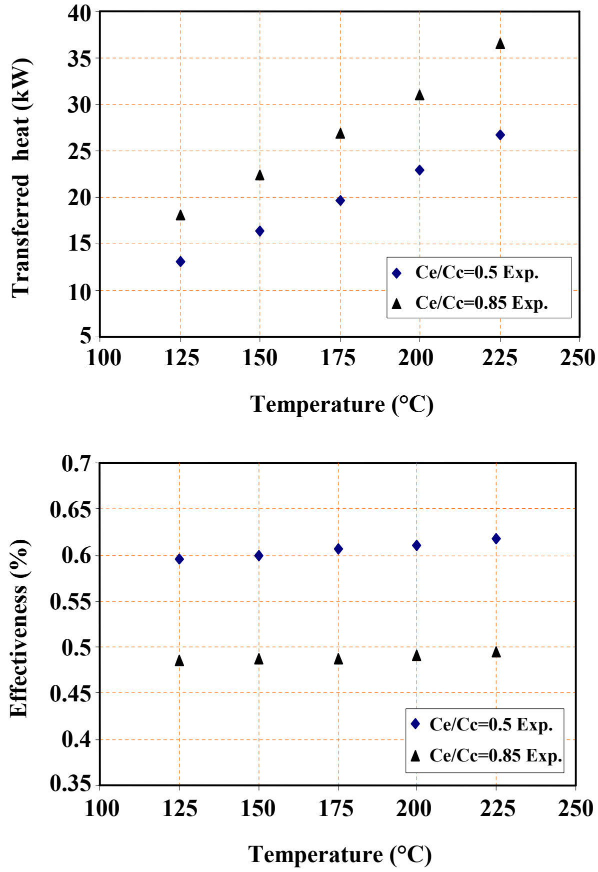

Fig. 6: The effectiveness and the rate of heat transfer vs. the inlet hot air temperature (For the two amount of C_e/C_c).

of the fluid streams to release and absorb more heat. At "C_e=C_c" the effectiveness is minimum, because of the releasing and the absorption heat is less. At "C_e<C_c", the effectiveness decreases by increasing the ratio of Ce/C_c, because of the sensible heat of high -temperature fluid stream is less than low -temperature fluid stream.

It is observed that by increasing the hot air mass flow rate (or C_e/C_c); the rate of heat transfer also increases. In fact, by increasing the hot air mass flow rate, heat transfer coefficient increases and consequently heat transfer rate increases.

Inlet hot air temperature effect ($T_{e,i}$)

In this section, the effects of the inlet hot air temperature on the effectiveness and the rate of rate of heat transfer have been investigated for two constant hot air mass flows (or C_e/C_c) and velocities of hot air stream.

Constant heat capacities ratio or hot air mass flow

Now, the effectiveness and the rate of heat transfer are discussed at two constant heat capacity ratios of 0.4 and 0.71 and hot air mass flows of 0.25 and 0.44 kg/s, respectively. The effectiveness and the rate of heat transfer versus the temperature of inlet hot air are shown in Fig. 6. It is found that by changing the inlet hot air temperature, the effectiveness remains almost constant. Clearly, because of the heat transfer coefficient varies by mass flow rate. Thus the heat transfer coefficient does not change at constant mass flow or constant "C_e/C_c", and the thermal resistance and also the effectiveness remain constant.

Constant inlet hot air velocity

The effect of the hot air velocity at two constant values of 1 and 1.5 m/s on the effectiveness and the rate of heat transfer are taken into account and discussed. The results are shown in Fig. 7.

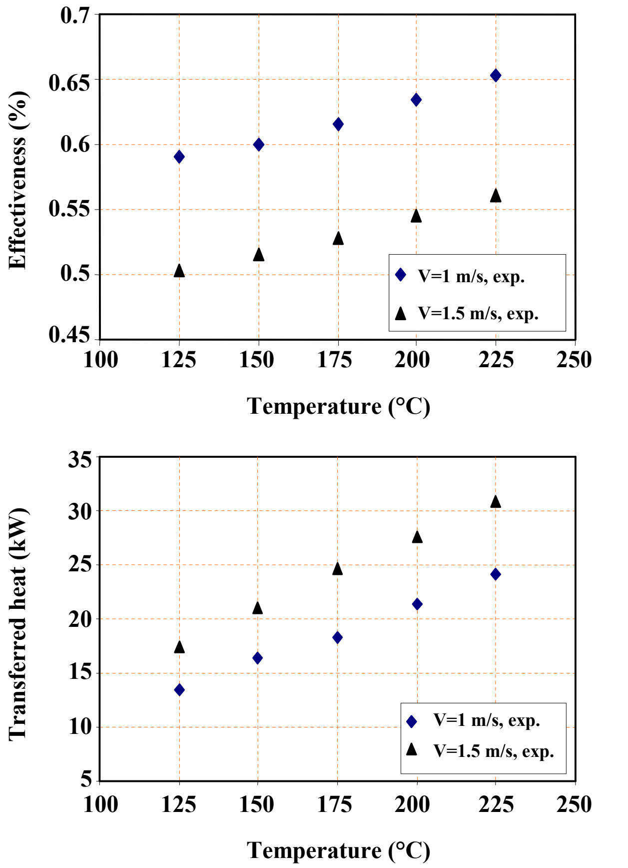

Fig. 7: The effectiveness and the rate of heat transfer vs. the inlet hot air temperature (For the two amount of fluid velocity).

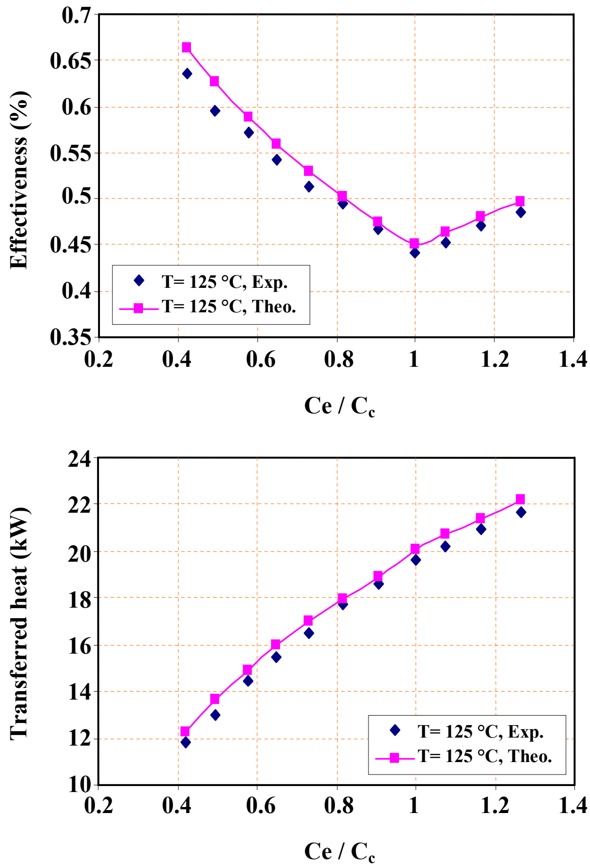

Fig. 8: The effectiveness and the rate of heat transfer of a "THE" vs. the heat capacity ratio of fluid streams (Comparison between experimental and theoretical results).

It is observed that due to decrement of the density of inlet hot air which results in less hot air mass flow rate, the effectiveness and the rate of heat transfer increase.

Comparison between experimental results and theoretical model

The comparison between experimental and theoretical results of the effectiveness and the rate of heat transfer of a gas to liquid "THE" has been studied for inlet hot air in the range of 125-225 °C. Because of the similarity of results only the comparison of inlet hot air at 125 °C is presented here (Figs. 8, 9, 10). A good agreement between experimental results of the effectiveness and the rate of heat transfer with theoretical model has been achieved.

EXPERIMENTAL RESULTS OF PRESSURE DROP ANALYSIS

In this paper, the pressure drop across a "THE" has been investigated experimentally and theoretically. The experiments were done in the range of 125-200 °C. Since the measured pressure drop of hot air has been alike in various temperatures, we have used experimental data for inlet hot air at 125, and 200 °C (table 2).

Geometrical dimensions of the heat pipe heat exchanger have used in this experiment are:

$$A_C = (W - N_b d)(L_e - t_f n_f L_e) \tag{27}$$

$$A_f = 2N(Wb - \frac{\pi}{4} d_o^2) n_f L_e \tag{28}$$

$$A_o = 2N(Wb - \frac{\pi}{4} d_o^2) n_f L_e + (L_e - n_f t_f) \pi d_o N$$

$$A_c = 0.099 m^2, A_f = 22.278 m^2, A_o = 23.84 m^2 \tag{29}$$

Since calculation of the pressure drop across "THE" needs data on other physical properties of the inlet hot air and some parameters, experimental data and their

Table 2: Experimental data for pressure drop.

V(HZ)	$T_{h,out}$ (°C)	$\Delta P_{exp.}$ (mmH_2O)
20	58	6
30	68	14
40	74	23
50	78	34
60	81	48
70	84	60

a) Inlet hot air at 125 °C.

V(HZ)	$T_{h,out}$ (°C)	$\Delta P_{exp.}$ (mmH_2O)
20	74	6
30	90	14
40	100	24
50	107	34
60	114	46
70	120	62

b) Inlet hot air at 200 °C.

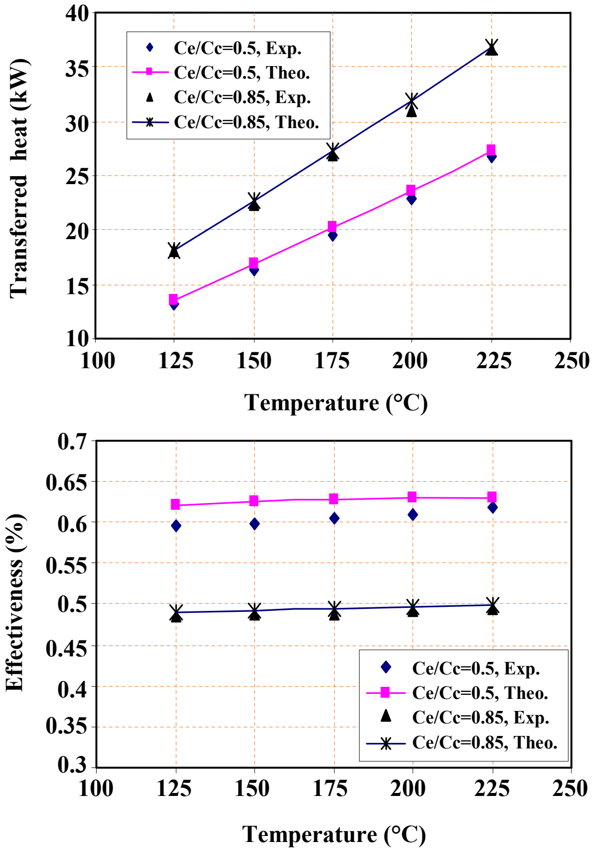

Fig. 9: The effectiveness and the rate of heat transfer of a "THE" vs. the temperature of inlet hot air (Comparison between experimental and theoretical results).

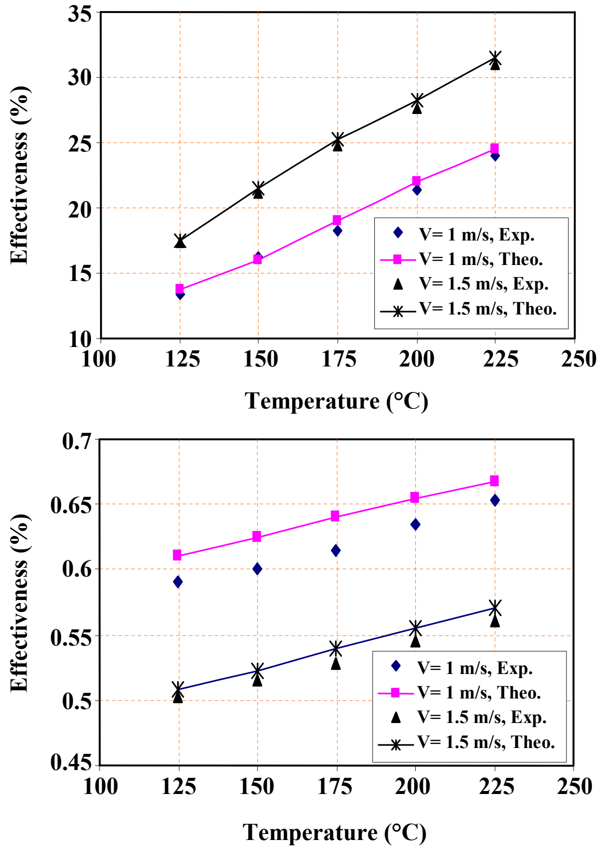

Fig. 10: The effectiveness and the rate of heat transfer of a "THE" vs. the temperature of inlet hot air (At constant fluid velocity) (Comparison between experimental and theoretical results).

estimation have been used to determine the hot air physical properties.

We have obtained the parameters of fluid flow in the bottom duct of the test rig as follows:

$$\dot{m} = C\sqrt{\rho_m \Delta P} \tag{30}$$

Where C is the calibration factor of orifice meter in the pilot plant which is equal to 0.067 and ΔP is the pressure drop of the fluid flow through the orifice meter.

$$U_\infty = \frac{\dot{m}}{\rho_m A} \tag{31}$$

Table 3: Pressure drop obtained from various Correlations.

$\dot{m}_h(\frac{kg}{s})$	$\Delta P_{cal.}$ (mmH_2O) New Correlation	$\Delta P_{cal.}$ (mmH_2O) Kays & London	$\Delta P_{cal.}$ (mmH_2O) Shah & Giovanelli
0.175	8	8	6
0.245	14	15	11
0.319	24	25	17
0.396	36	37	25
0.473	50	52	34
0.543	64	68	43

a) Inlet hot air at 125 °C.

$\dot{m}_h(\frac{kg}{s})$	$\Delta P_{cal.}$ (mmH_2O) new correlation	$\Delta P_{cal.}$ (mmH_2O) Kays & London	$\Delta P_{cal.}$ (mmH_2O) Shah & Giovanelli
0.165	8	7	7
0.231	15	14	12
0.294	24	25	18
0.366	35	37	27
0.43	48	51	35
0.493	62	66	44

b) Inlet hot air at 200 °C.

Where A is the cross sectional area of the duct which is equal to 0.3 m^2 for test rig.

$$U_{max} = U_{\infty}\frac{S_L}{S_L - d} \tag{32}$$

The configuration of the tubes is in-line with 30 mm pitch. The pressure drop across "THE" has been obtained by using experimental data and equations 22 to 32 and Fig. 11. The results of various correlations are shown in table 3.

NEW CORRELATION

The authors present a new correlation to predict the pressure drop in a thermosyphon heat exchanger with continuous fins and in-line configuration based on the experimental data obtained in the pilot plant.

$$\Delta P = 0.345 f \frac{A_f}{A_c}\frac{G^2}{\rho_m} \tag{33}$$

We can calculate the fanning friction factor for various configurations of tube banks by using the following correlations [20]:

- In-line configuration of tube bank

$$f = [0.176 + \frac{0.32b}{(a-1)^{0.43+1.13/b}}]Re_{max}^{-0.15} \tag{34}$$

-rectangular configuration of tube bank

$$f = [1 + \frac{0.47}{(a-1)^{1.08}}]Re_{max}^{-0.16} \tag{35}$$

where; $a = \frac{S_L}{d}$, $b = \frac{S_T}{d}$

The results show that the new correlation has a high degree of accuracy to predict the pressure drop in the heat pipe heat exchanger with continuous fins and in-line configuration of tubes (Figs. 12, 13).

CONCLUSIONS

The effects of various parameters such as the heat capacity ratio of high- and low-temperature fluid streams "C_e/C_c", the inlet hot air temperature, and the mass flow rate or the inlet hot air velocity on thermal performance of a gas-liquid "THE" and the pressure drop across tube bundle of it have been investigated. The following conclusions have obtained:

1- A good agreement between experimental results of the effectiveness and the rate of heat transfer with theoretical model has been achieved.

2- At constant C_e/C_c, by increasing the inlet hot air temperature the effectiveness almost remains constant but the rate of heat transfer increases.

3- At constant the inlet hot air temperature, by increasing the inlet hot air velocity, the effectiveness and the rate of heat transfer increase. Due to the heat transfer coefficient of high- temperature fluid stream increment.

4- The heat capacity ratio of high- and low-temperature fluid streams affects on the effectiveness and the rate of heat transfer of "THE". When the heat capacity ratio of high- and low-temperature fluid streams is higher than unity the effectiveness increases due to the ability of the fluid streams to release and absorb more heat. At C_e=C_c the effectiveness is minimum, because of the releasing and the absorption heat is less.

5- The average variation between experimental and theoretical results obtained by *Kay's* and *London's*

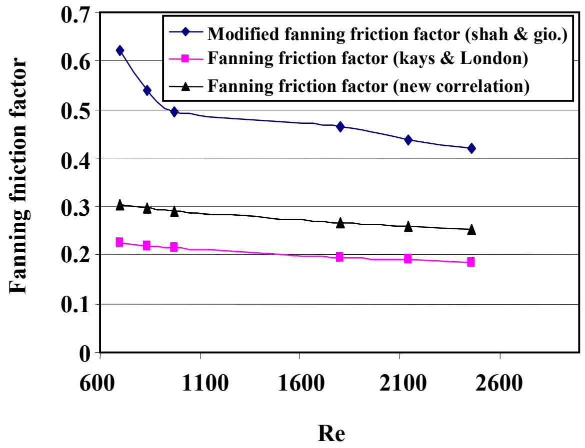

Fig. 11: Fanning friction factor versus the Maximum Reynolds number of flow across "THE".

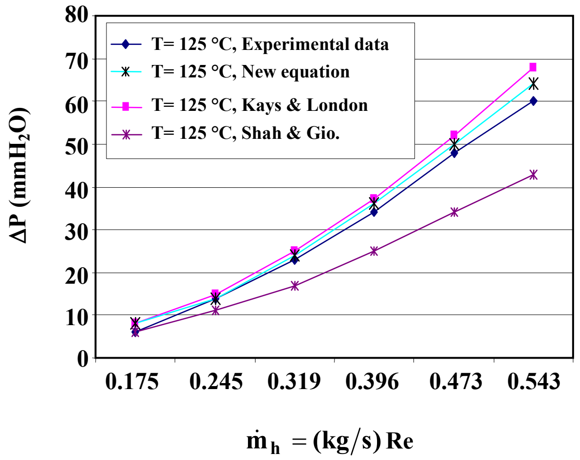

Fig. 12: Pressure drop vs. mass flow rate of inlet hot air to "THE", ($T_{h, in}$=125 °C).

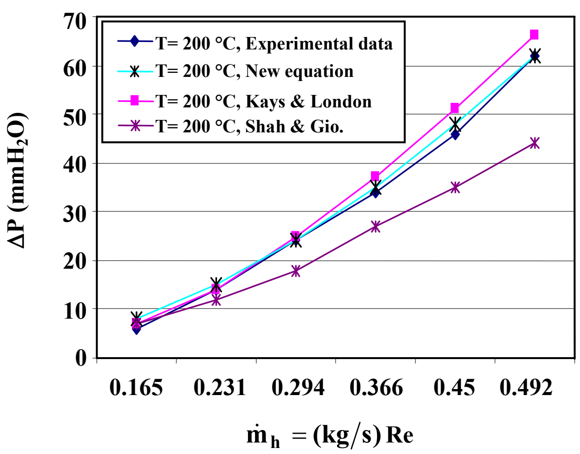

Fig. 13: Pressure drop vs. mass flow rate of inlet hot air to "THE", ($T_{h, in}$=200 °C).

correlation, *Shah* and *Giovanelli's* correlation and the new correlation are 13 %, 22 %, and 10 %, respectively.

6- *Shah* and *Giovanelli's* correlation has low accuracy for a "THE" with continuous fins when the Reynolds number is higher than 2000, however it is proposed for the thermosyphon heat exchanger with individual circular fins.

7- *Kay's* and *London's* correlation is one of the best methods for calculating the pressure drop in a heat pipe heat exchanger with various tube and fin geometries, though this correlation is very complicated and bears numerous variables. More than eight parameters are needed to predict the pressure drop. The new correlation has a high degree of accuracy to predict the pressure drop. This correlation is very easy and there are five parameters on it, at least.

8- The accuracy of the new correlation should be studied for a "THE" with various fins when the Reynolds number of the flow is higher than 5000.

Nomenclatures

A_c	Minimum free-flow area in the core
A_f	Surface area of fins
A_o	Total frontal area of heat pipe heat exchanger
C	Heat capacity of fluid (w/°C)
C_h	Heat capacity of hot fluid (w/°C)
C_c	Heat capacity of cold fluid (w/°C)
D_i	Inside Diameter of heat pipe (m)
D_i	Outside Diameter of heat pipe, (m)
F	Fanning friction factor
f'	Modified fanning friction factor
G	Maximum mass velocity in the core
L_c	Length of condensing section, (m)
L_e	Length of evaporator section (m)
$\dot{m}$	Mass flow rate of fluid in duct, (kg/s)
N	Number of rows of tubes
Q	Heat transfer flux, (w/m^2)
$\Delta P_{exp.}$	Experimental pressure drop
$\Delta P_{cal.}$	Theoretical pressure drop
R	Thermal resistance, (m^2 C/W)
S_c	Minimum free-flow area in the core, (m^2)
S_f	Surface area of fins, (m^2)
S_o	Total frontal area of HPHE, (m^2)
S_L	Longitudinal tube pitch, (mm)
S_o	Total frontal area of HPHE, (m^2)
S_T	Transverse tube pitch, (mm)

$T_{h,in}$ Temperature of flow, inlet of THE, (°C)
$T_{h,out}$ Temperature of flow, outlet of THE, (°C)
U_{max} Maximum flow velocity in tube bank, (m/s)
U Heat transfer coefficient, (W/m^2 °C)
U_t Total Heat transfer coefficient, (W/m^2 °C)

Subscribes

c Condenser
e Evaporator
f Fin, Fouling
i Inside
o Outside, Overall
p Pipe
w Wick

Dimensionless groups

Re_{max}, $Re_{max} = \frac{\rho_m U_{max} d_o}{\mu}$ Reynolds number

NTU, $NTU = \frac{u_t s_t}{c_{min}}$ Number of transfer unit

Greek letters

ν Frequency of current, HZ
ρ Density of fluid, kg/m^3
ε Effectiveness
η Fin effectiveness
μ Dynamic viscosity of fluid, $N.s/m^2$

Received : 15th July 2007 ; Accepted : 27th January 2008

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