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An Experimental and Theoretical Investigation on Thermal Performance of a Gas-Liquid Thermosyphon Heat Pipe Heat Exchanger in a Semi-Industrial Plant

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Abstract
Waste heat recovery is very important, because it not only 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 used in waste heat recovery is heat pipe heat exchanger. An experimental and theoretical research is carried out to investigate heat performance of an air to water thermosyphon heat pipe heat exchanger according to ε-NTU method. The experiments were done according to the following procedure: cold water with 0.1kg/s flows through the condensation section and hot air in a closed cycle is blown into the evaporation section. A blower with varying frequency of current turns in the mass flow rate between 0.14-0.6 kg/s and a temperature range of 125-225℃. The results of the experiments show that as the ratio of C_h/C_c *rises, the rate of heat transfer goes up. The efficiency of the heat pipe heat exchanger remains constant as the temperature of the hot stream goes up, but the amount of heat transferred increases.*

Keywords: *Thermosyphon heat pipe heat exchanger, ε-NTU method, In-line configuration*

1- Introduction
Heat recovery, one method of energy conservation, can be successfully implemented when the investment cost of the 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^2k; and heat pipe thermal resistance

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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. Azad and Geoola [2] applied the ε-NTU model for gravity-assigned air-to-air heat pipe heat exchanger. They developed a new correlation for condensing water vapor on vertical carbon –steel and determined that the external thermal resistances in those cases limit the performance of heat pipe heat exchanger where the thermosyphon are operating below the sonic limit. Zhongliang Liu et al. [3, 4] have studied 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. In addition, they have studied the performance of the simultaneous charging/discharging operation modes of the heat pipe heat exchanger. The experimental results on the charging only mode and the discharging only mode of the system show that the new device performs the designed functions very well. It can both store and release the thermal energy efficiently, however, the device can be used as a conventional system in which the charging and discharging are operated independently. Also, the results for the simultaneous charging/discharging mode were unknown. Shah and Giovannelli [5] studied heat pipe heat exchanger performance from a comparative point of view. The performance of a single HPHE was modelled using both LMTD and the ε-NTU method. The thermal resistances were determined using many existing correlations and the results compared. In addition, 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 an evaporator, and condenser regions in a heat pipe heat exchanger were formulated by minimizing the total thermal resistance of the heat path. Their present results indicated that the optimal position of the partition plate of a heat pipe heat exchanger is the middle of HPHE. Nevertheless, it is not necessarily in the middle of it, because its position depends upon the relative magnitude of the flow rates of the hot and cold fluids. When the difference in flow rate of the two fluids is small, the optimal position is quite near the middle, but when the difference is large the position of the partition plate should be optimized. Wadowski and et al. [7] carried out an experimental study to investigate the performance of an air-to-air thermosyphon –based heat exchanger utilizing R-22 as the working fluid under different operating conditions. They have investigated thermal performance of HPHE, where the mass flow rate of hot air was limited to a range of 0.06

to 0.28kg/s and the temperature of hot air up to 70°C. Yang et al. [8] have built a thermosyphon heat pipe heat exchanger for recovery of the gas heat emitted from automobile exhausts, and investigated the thermal performance of "THE". Noie and Majidian [9] have built a "THE" for recovery of heat waste in hospitals and laboratories. The test rig used in their research consists of eight individual heat pipes and the following characteristics: three rows of heat pipe, methanol as a working fluid and heat pipes in a staggered equilateral triangle arrangement of 45 mm center. Since methanol is used as working fluid, the application of this heat pipe heat exchanger is limited to a range of 15-55°C. In addition, since the numbers of heat pipes used for manufacturing the exchanger are eight, the duty of heat transfer is limited to 150w, at least. In addition, Noie [10] has carried out an experimental study of the performance of an air-to-air thermosyphon-based heat exchanger utilizing water as working fluid to investigate its behavior under different operating conditions. Song Lin et al. [11] have presented a design method using CFD simulation of the dehumidification process with a heat pipe heat exchanger. Their studies illustrate that the CFD modeling is able to predict the thermal performance of the dehumidification solution with HPHE. The simulation results in this paper only reflect the overall performance of the system at different operating conditions. The details of the thermal characteristics and design parameters of the system cannot be published in this paper. The operating parameters used in this paper are: the inlet temperature of air 35-50°C, the inlet flow rate 4-8lit/s. The predicted results also indicate that the heat pipe unit can be further optimized to achieve similar or better heat transfer performance than the auxiliary condenser. However, the auxiliary condenser is necessary to maintain the temperature difference between the two ends of the heat pipe unit. In this research, we have investigated the effects of various parameters such as the heat capacity ratio of high- and low-temperature fluid streams "Ce/Cc", the inlet hot air temperature and the mass flow rate or the inlet hot air velocity on thermal performance of a gas-liquid "THE", experimentally and theoretically. In order to investigate the parameters affecting hydrodynamics and thermal performance of Thermosyphon heat pipe heat exchanger (THE), a semi industrial pilot was designed. The mass flow rate of hot air varies in the range of 0.15-0.55kg/s and the inlet hot air temperature controls at five quantities as 100,125,150,175,200°C.

2- Theory

The analysis of the heat transfer aspects of HPHE's based on the heat transfer rate equation obtained by an energy balance of the heat exchanger:

$$Q = U.S(T_h - T_c) \tag{1}$$

There are two main approaches used in the design of a HPHE:

1) The Log-mean temperature difference model (LMTD)
2) The effectiveness-number of transfer units model (ε-NTU)

2.1- ε-NTU method

The ε-NTU method based on the heat exchanger effectiveness, ε, which is defined as the ratio of the actual heat transfer in a heat exchanger to the heat transfer that would have occurred in a heat exchanger with an 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 [12]:

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

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

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

The ratio $\frac{U_t S_t}{C_{\min}}$ defines, as the number of transfer units (NTU),

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

$$C_{\min} = (\dot{m}C_p)_{\min} \tag{5}$$

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

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

Therefore, effectiveness can be calculated by the following correlations:

$$\varepsilon = \frac{T_{c,out}-T_{c,in}}{T_{h,in}-T_{c,in}}, if \quad C_e < C_c \tag{7}$$

$$\varepsilon = \frac{T_{h,in}-T_{h,out}}{T_{h,in}-T_{c,in}}, if \quad C_e > C_c \tag{8}$$

Due to phase change, the maximum heat capacity is several orders of magnitude larger than the minimum heat capacity. Therefore, the heat capacity ratio between minimum and maximum heat capacities are equal to zero $(\frac{C_{\min}}{C_{\max}} \approx 0)$ and the expressions for effectiveness are presented as follows:

$$\varepsilon = 1-\exp(-NTU) \tag{9}$$

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

$$\varepsilon_e = 1-\exp(-NTU_e) \tag{10}$$

$$\varepsilon_c = 1-\exp(-NTU_c) \tag{11}$$

Where, $NTU_e = \frac{U_e.S_e}{C_e}$ and $NTU_c = \frac{U_c.S_c}{C_c}$

These correlations are defined for a single row of pipes. The effectiveness of heat pipe heat exchanger with n rows of pipes is as follows:

$$\varepsilon_{e_n} = 1-(1-\varepsilon_{e_1})^n \tag{12}$$

$$\varepsilon_{c_n} = 1-(1-\varepsilon_{c_1})^n \tag{13}$$

The least overall effectiveness of heat pipe heat exchanger is obtained by the following correlations:

$$\varepsilon_o = \frac{1}{\frac{1}{\varepsilon_{c_n}} + \frac{C_c / C_e}{\varepsilon_{e_n}}} \qquad \text{If} \qquad C_e > C_c \tag{14}$$

$$\varepsilon_o = \frac{1}{\frac{1}{\varepsilon_{e_n}} + \frac{C_e / C_c}{\varepsilon_{c_n}}} \qquad \text{If} \qquad C_e < C_c \tag{15}$$

2.2- Determination of the overall heat transfer coefficient

To determine the overall heat transfer coefficient, the heat transfer is modelled as a thermal resistance network shown in Fig.1.

$$\frac{1}{US} = \frac{1}{U_c S_c} = \frac{1}{U_h S_h} = \frac{1}{(\eta_o hs)_c} + R_{f,c} + R_{hp} + R_{f,h} + \frac{1}{(\eta_o hs)_h} \tag{16}$$

$$\frac{1}{U_h S_h} = [\frac{1}{(\eta_o hs)_h} + \frac{1}{2\pi k_w L_e} \ln(\frac{D_o}{D_i}) \tag{17}$$

$$R_h = \frac{1}{(\eta_o hs)_h} \quad , \quad R_{w,h} = \frac{1}{2\pi k_w L_e} \ln(\frac{D_o}{D_i}) \tag{18}$$

In this research, it is assumed that fouling resistances due to corrosion or oxidation are negligible and resistance terms that occurred due to heat transfer through the liquid saturated wick are negligible, too.

For the condenser section, we have:

$$\frac{1}{U_c S_c} = [\frac{1}{(hs)_c} + \frac{1}{2\pi k_w L_c} \ln(\frac{D_o}{D_i}) \tag{19}$$

$$R_c = \frac{1}{(hs)_c} \quad , \quad R_{w,c} = \frac{1}{2\pi k_w L_c} \ln(\frac{D_o}{D_i}) \tag{20}$$

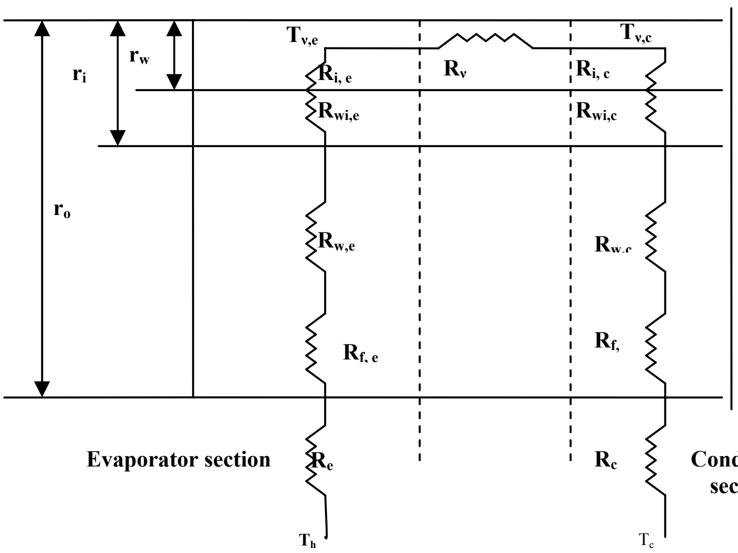

Figure 1. Thermal resistance network of a THPHE

3- Experimental set up and procedure

In order to investigate the parameters affecting the hydrodynamics and thermal performance of Thermosyphon heat pipe heat exchanger (THPHE), a semi industrial pilot based on Fig.2 is designed. This model consists of the following parts: Thermosyphon heat pipe heat exchanger, air channels, Rota meter, a centrifugal blower, electrical heaters, thermometers, electrical board, fan speed regulator, orifice and manometer. The test rig has two sections, top and bottom. The top section is the condensation part of the HPHE in which cooled water is drowned into it by a pump with a constant flow rate (7lit/min) at about 17°C. It then goes out from the top section after gaining the pipe's heat by passing above them. The water flow is regulated by Rota meter. The bottom section is the evaporation part of the HPHE. The bottom duct is straight and forms a closed loop. A centrifugal blower and 90 electrical heaters are installed in the duct to circulate hot air through the evaporator section. In the bottom duct, mass flow rate varies by changing the input frequency to the blower in the range of (20-70HZ), therefore the mass flow rate varies in the range of (0.15-0.55kg/s). The experimental data has been obtained by repeating the tests three times. Thermosyphon heat pipe heat exchanger (THPHE) module is composed of 6(rows)*15(columns) copper pipes with aluminum plate fins with dimensions of 130cm (height)*47cm (width)*20cm (depth) which have been filled with water at a filling ratio of 30%, 50% and 70%. The density and thickness of the fins are 300 fin/m and 0.4mm, respectively. The used exchanger has the following characteristics mentioned in Table 1 and Fig.3. The manufactured THPHE, which is fixed to the channel from the top and bottom, is placed inside the channel and is encompassed with cold water and hot air. Pressure drop between the inlet and outlet of the THPHE is measured by inclined manometer. The inlet hot air temperature is controlled at five quantities as 100,125,150,175,200°C.

Figure 2. Photo of pilot plant

Table 1. Specifications of thermosyphon heat pipe heat exchanger

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

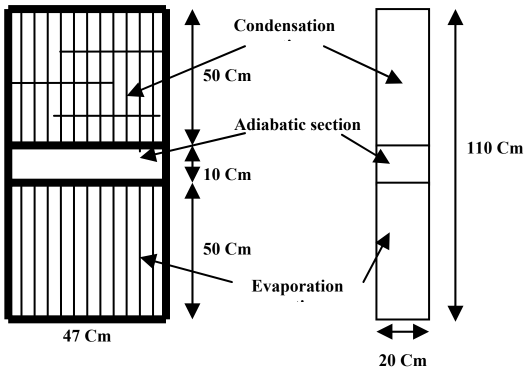

Figure 3. Dimensions of the "THPHE"

4- Results and discussion

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 been obtained.

4.1- Heat capacities ratio effect (C_e/C_c)

Heat capacities ratio is one of the most important factors influencing the effectiveness and the rate of heat transfer. The effectiveness and the rate of transferred heat vs. Ce/Cc is shown in Fig.4 for all inlet hot air temperature (Te,i). The heat capacity ratio of high-and low-temperature fluid streams affects 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 "Ce=Cc" the effectiveness is minimum because of the release, and the absorption heat is less. At "Ce<Cc", the effectiveness decreases by increasing the ratio of Ce/Cc, because the sensible heat of the high-temperature fluid stream is less than the low-temperature fluid stream.

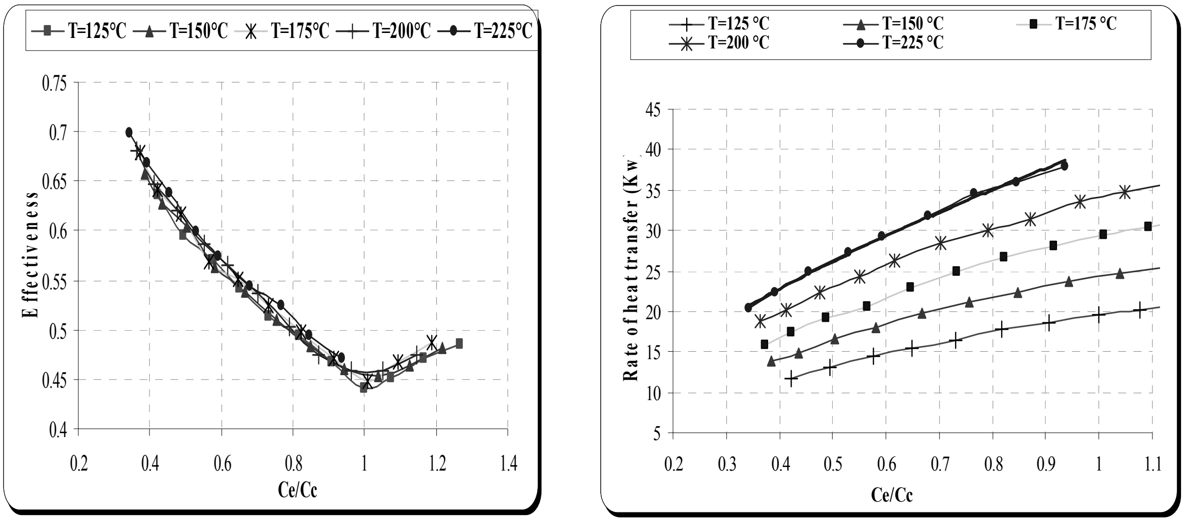

Figure 4. The effectiveness and the rate of heat transfer of "THE" vs. the ratio of Ce/Cc (For the entire inlet, hot air temperature)

It is observed that the effectiveness decreases by increasing Ce/Cc, and by increasing the hot air mass flow rate (or Ce/Cc) the transferred heat increases. In fact, by increasing the hot air mass flow rate, the heat transfer coefficient increases and consequently the heat transfer rate increases.

4.2-Inlet hot air temperature effect (Te, i)

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

4.3- Constant heat capacities ratio or hot air mass flow

Now, the effectiveness and the rate of heat transfer is discussed at two constant heat capacities ratios, equal to 0.5 and 0.85, which are equal to hot air mass flows of 0.25 and 0.44 kg/s, respectively. Effectiveness vs. temperature is shown in Fig.5. It is found that by changing the inlet hot air temperature (Te, i), ε remains almost constant. It is clear, as the heat transfer coefficient varies by mass flow, at constant mass flow (or C_e/C_c) the heat transfer coefficient does not change. Therefore, the thermal resistance and the effectiveness remain constant.

4.4- Constant inlet hot air velocity

The effect of the hot air velocity at two constant values of one and 1.5 m/s on the effectiveness and the rate of heat transfer is taken into account and discussed. The results are shown in Fig.6. It is observed that, due to the decrement of the density of inlet hot air that results in less mass flow of hot air, the effectiveness and the rate of heat transfer increase.

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

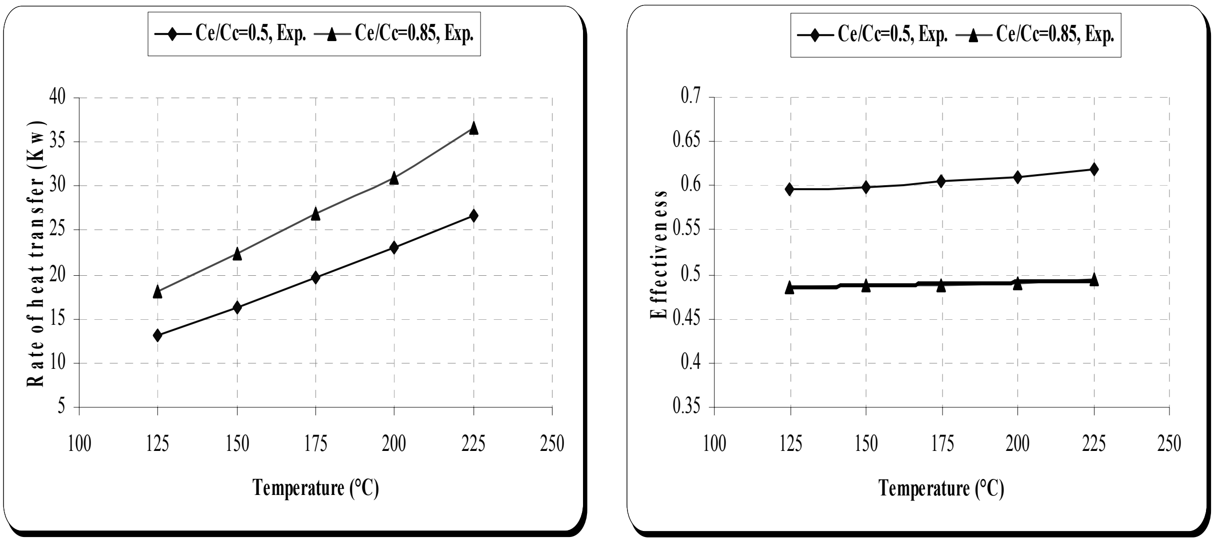

Figure 5. The effectiveness and the rate of heat transfer vs. the inlet hot air temperature (For the two amounts of Ce/Cc)

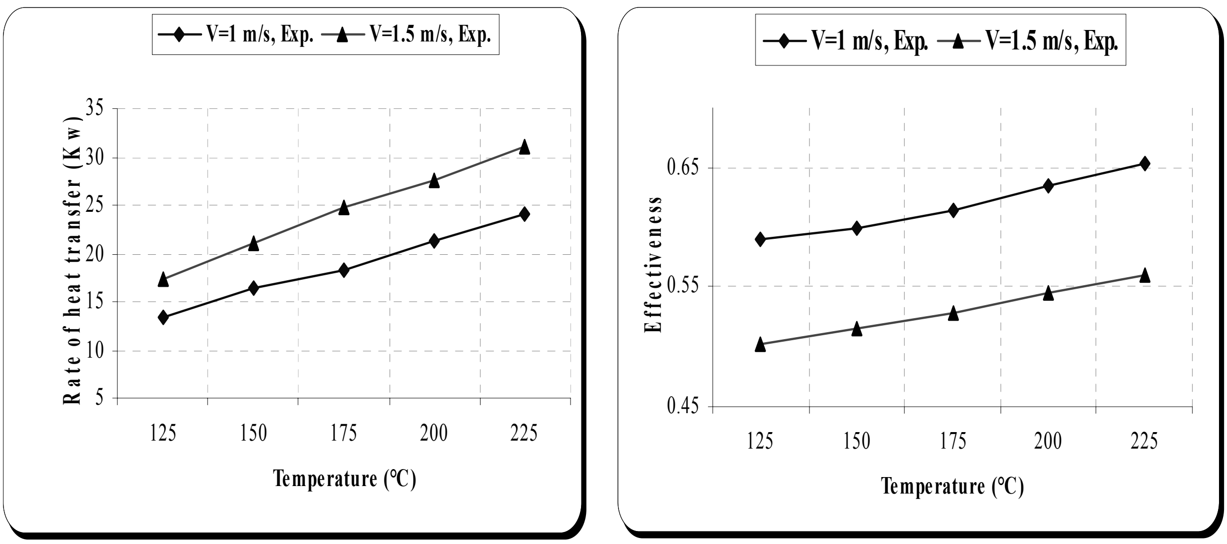

Figure 6. The effectiveness and the rate of heat transfer of a "THE" vs. the temperature of inlet hot air

4.5- Comparison between experimental results and theoretical model

In this section, comparison between experimental and theoretical results of effectiveness and transferred heat of a gas to liquid THPHE are carried out. This has been done for inlet hot air in 125-225°C range, Because of the similarity of results only the comparison of inlet hot air at 125°C is presented here (Tables 2, 3, 4 and Figs. 7, 8). A good agreement between the experimental results of the effectiveness and the rate of heat transfer with the theoretical model has been achieved.

Table 2. Experimental and theoretical results of inlet hot air for Thi=125°C

$\dot{m}_h$ (kg/s)	Vh (m/s)	Ce/Cc	Tho(theo.) (°C)	Th,o(exp.) (°C)	exp.ε	Qexp.(kw)	εtheo.	Qtheo..(kw)
0.175	0.800	0.421	63.5	58	0.636	11.795	0.663	12.292
0.239	1.096	0.577	72.25	65	0.571	14.495	0.588	14.913
0.302	1.386	0.73	75.25	71	0.514	16.477	0.53	16.984
0.413	1.898	1.00	77.75	78	0.441	19.644	0.45	20.039
0.481	2.209	1.165	79.5	82	0.47	20.929	0.48	21.361
0.523	2.400	1.266	82.5	84	0.486	21.682	0.496	22.144

Table 3. %AAD of experimental and theoretical results for outlet air temperatures (Thi=125°C)

$\dot{m}_h$ (kg/s)	Tho(theo.) (°C)	Th,o(exp.) (°C)	%AAD of Th,o
0.175	63.5	58	8.66
0.245	72.25	65	10
0.319	75.25	71	5.64
0.395	77.75	78	0.3
0.473	79.5	82	3.1
0.543	82.5	84	3.03

Table 4. %AAD of experimental and theoretical results for the rate of heat transfer and the effectiveness of THPHE (Thi=125°C)

exp.ε	Qexp.(kw)	εtheo.	Qtheo..(kw)	%AAD of Q	%AAD of ε
0.636	11.795	0.663	12.292	4.04	4.24
0.571	14.495	0.588	14.913	2.8	2.9
0.514	16.477	0.53	16.984	3	3
0.441	19.644	0.45	20.039	2	2
0.47	20.929	0.48	21.361	2	2.1
0.486	21.682	0.496	22.144	2.1	2

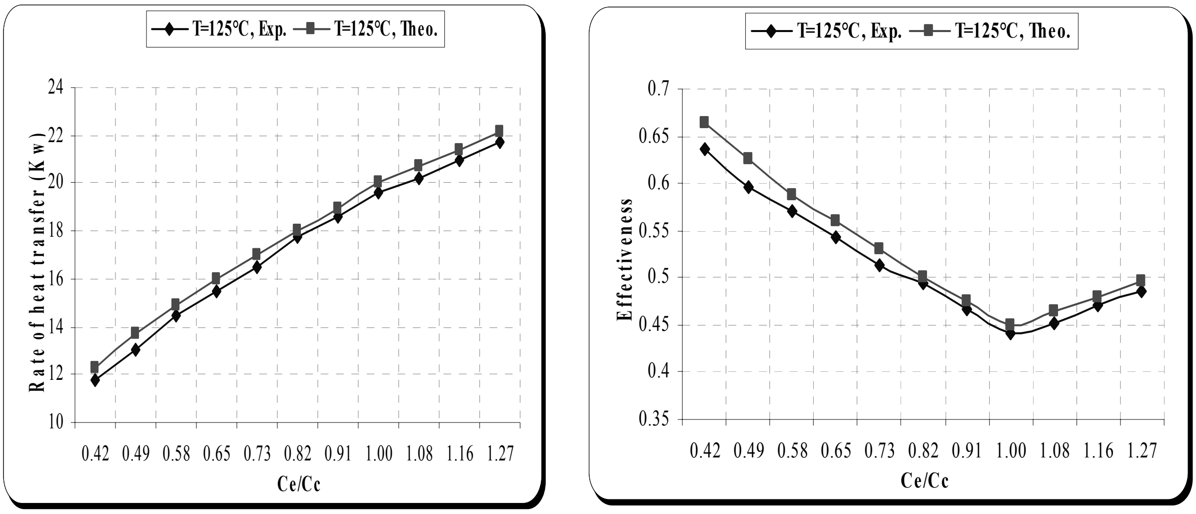

Figure 7. The effectiveness and the rate of heat transfer of a "THE" vs. the temperature of inlet hot air (Comparison between experimental and theoretical results)

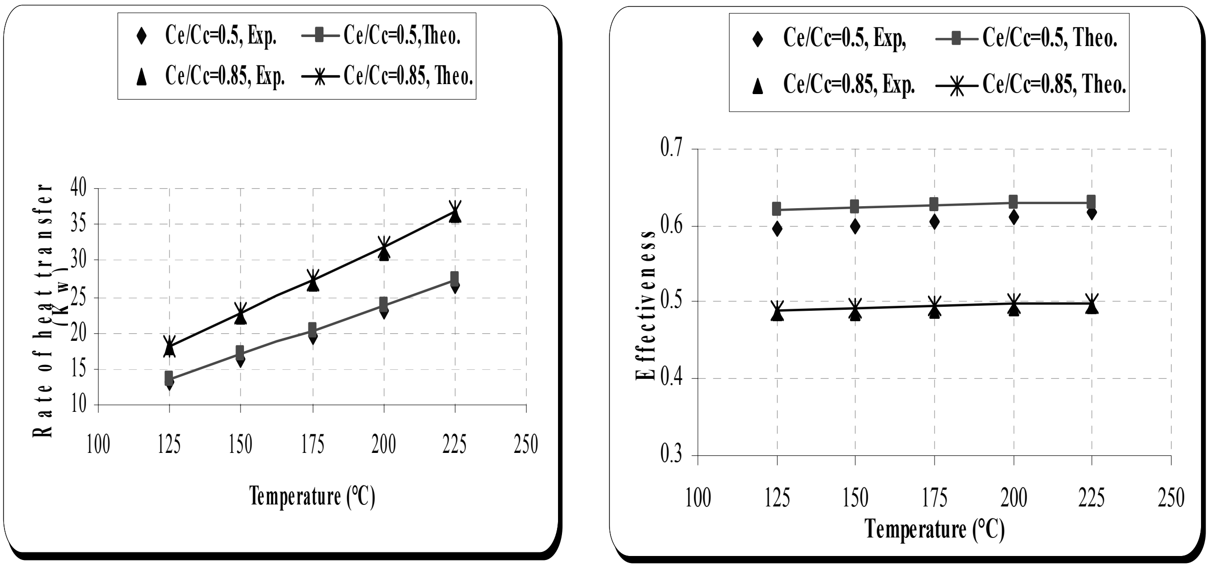

Figure 8. The effectiveness and the rate of heat transfer of a "THE" vs. the temperature of inlet hot air (Comparison between experimental and theoretical results)

5- Conclusion

The effect of various parameters on the thermal performance of a gas-liquid THPHE was investigated. The following conclusions were obtained from the present study:

At constant Ce/Cc, by increasing the inlet hot air temperature the effectiveness remains almost constant, but the rate of heat transfer increases. Clearly, this is because the heat transfer coefficient varies by mass flow rate. Thus, the heat transfer coefficient does not change at constant mass flow or constant

"Ce/Cc", and the thermal resistance and the effectiveness remain constant.

1) The heat capacity ratio of high- and low-temperature fluid streams affects 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 Ce=Cc the effectiveness is minimum because of releasing, and the absorption heat is less.
2) At constant the inlet hot air temperature, by increasing the inlet hot air velocity, increases effectiveness and the rate of heat transfer, due to the heat transfer coefficient of high-temperature fluid stream increment.
3) As a result, the number of experimental tests that needs to be carried out on a large-scale plant is quite limited.

Nomenclature

C Heat capacity of fluid $(W/^{\circ}C)$

C_h Heat capacity of hot fluid $(W/^{\circ}C)$

C_c Heat capacity of cold fluid $(W/^{\circ}C)$

D_i Inside diameter of heat pipe (m)

D_i Outside diameter of heat pipe, (m)

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)

R Thermal resistance, $(m^2 {^{\circ}C}/W)$

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_T Transverse tube pitch, (mm)

$T_{h,in}$ Temperature of flow, inlet of HPHE, (°C)

$T_{h,out}$ Temperature of flow, outlet of HPHE, (°C)

$U_{\max}$ Maximum flow velocity in tube bank, (m/s)

U Heat transfer coefficient, $(W/m^2 {^{\circ}C})$

$U_{\max}$ Maximum flow velocity in tube bank, (m/s)

U Heat transfer coefficient, $(W/m^2 {^{\circ}C})$

U_t Total Heat transfer coefficient, $(W/m^2 {^{\circ}C})$

Subscribes

c Condenser

e Evaporator

f Fin, Fouling

i Inside

o Outside, Overall

p Pipe

w Wick

Dimensionless groups

$\mathrm{Re}_{\max}$, $\mathrm{Re}_{\max} = \frac{\rho_m U_{\max} d_o}{\mu}$ Reynolds number

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

Greek letters

μ Dynamic viscosity of fluid, N.s/m2

ν Frequency of current, HZ

ρ Density of fluid, kg/m3

ε Effectiveness

η Fin effectiveness

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