An experimental & 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 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 and theoretical research has been 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 to the evaporation section. A blower with varying frequency of current turns in the mass flow rate between 0.14-0.6 kg/s and temperature range of 125-225˚C.the results of experiments show that as the ratio of $\frac{C_h}{C_C}$ rises the amount of heat being transferred goes up. The efficiency of heat pipe heat exchanger remains constant as the temperature of hot stream goes up but the amount of heat transferred increases.

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

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. They may be used to transfer heat between two fluid phases. Features include no cross-contamination between streams, no moving parts, compactness and no need for any external power supply. Since the mid-1970's, research dealing with heat pipe heat exchanger has steadily increased. 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 HPHE's was applied by Azad and Geoola. They developed a new correlation for condensing water vapor on vertical carbon –steel and determined that the performance of HPHE's is limited by the

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external thermal resistances in those cases where the thermosyphon are operating below the sonic limit **[1]**. Shah and Giovannelli 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 final results compared **[2]**. Noie and Majidian built a THPHE for recovery the waste heat in hospital and laboratories **[3]**. 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 and Shokri **[4]**.

Theory

The analysis of the heat transfer aspects of HPHE's is 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)

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 in a heat exchanger to the heat transfer that would 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 **[5]**:

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

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

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

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

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

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

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

The C_e, C_c are the heat capacity of fluid in evaporator and condenser sections of a heat pipe heat exchanger, respectively. Therefore, effectiveness can be obtained by following correlations:

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

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

Due to phase change, The maximum heat capacity is several orders of magnitude larger than the minimum heat capacity, actually $\frac{C_{\min}}{C_{\max}} \approx 0$. The expressions for effectiveness will be presented as:

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

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

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

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

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

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

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

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

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

$$e_o = \frac{1}{\frac{1}{e_{c_n}} + \frac{C_c/C_e}{e_{e_n}}} \quad \text{If} \quad C_e > C_c \tag{14}$$

$$e_o = \frac{1}{\frac{1}{e_{e_n}} + \frac{C_e/C_c}{e_{c_n}}} \quad \text{If} \quad C_e < C_c \tag{15}$$

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 **Fig1**.

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

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

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

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

For the condenser section, we have:

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

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

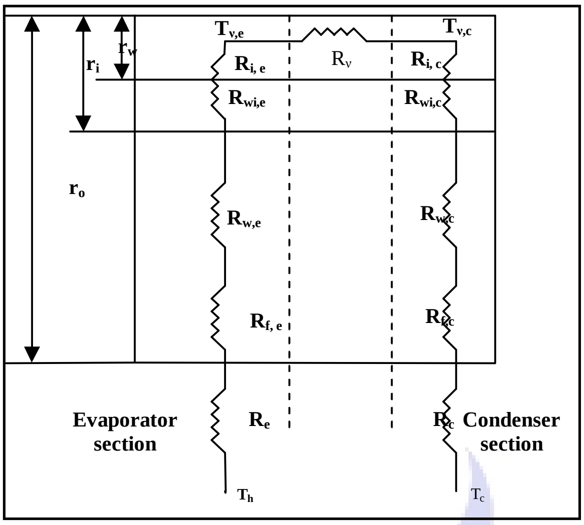

Fig1. Thermal resistance network of a THPHE

Experimental set up and procedure

In this research we have manufactured a pilot plant for the data acquisition purposes with below conditions:
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 by water with filling ratio of 30%, 50% and 70% .the density and thickness of fins are 300 fin/m and 0.4mm, respectively. The test rig has two sections, top and bottom sections. The top section is condensation part of HPHE in which cooled water is drowned into it by a pump with constant flow rate (7lit/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 in the duct to circular hot air through the evaporator section. In the bottom duct mass flow rate varies by changing the input frequency to blower in the range of (20-70HZ), therefore the mass flow rate varies in the range of (0.15-0.55kg/s). Pressure drop between inlet and outlet of THPHE was measured by inclined manometer. The inlet hot air temperature was controlled at five quantities as 100,125,150,175,200°c.The schematic of test rig is illustrated in **Fig.2**

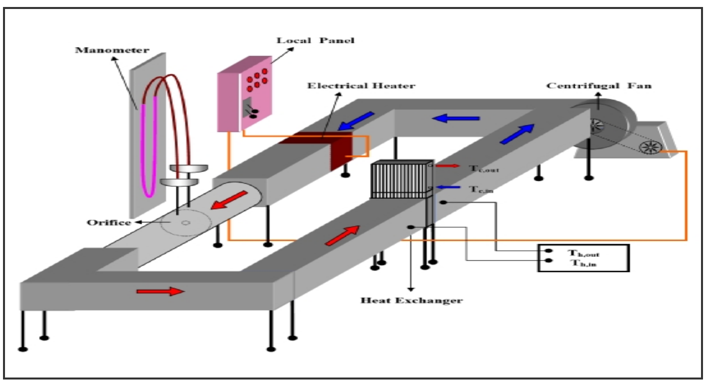

Figure2. 0verall schematic of pilot plant

Results and discussion

Heat capacities ratio effect (C_e/C_c)

Heat capacities ratio is one of the most important factors which have influence on effectiveness and the rate of heat transferred. Effectiveness and transferred heat vs. C_e/C_c is shown in Fig.3 for all inlet hot air temperature ($T_{e,i}$). It is observed that the effectiveness decreases by increasing C_e/C_c and by increasing the hot air mass flow rate (or C_e/C_c) transferred heat 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, effects of inlet hot air temperature ($T_{e,\,i}$) on two parameters, consist of effectiveness and transferred heat 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 transferred heat rate are discussed at two constant heat capacities ratios, equal to 0.4 and 0.71 which are equal to hot air mass flows of 0.25 and 0.44 kg/s, respectively. Effectiveness vs. temperature is shown in **Fig.4**. It found that by changing the inlet hot air temperature ($T_{e,\,i}$), ε remains almost constant. It's clear, because heat transfer coefficient varies by mass flow, thus at constant mass flow (or C_e/C_c) heat transfer coefficient doesn't change. Therefore, 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.5**.

It is observed that due to decrement of the density of inlet hot air which results in less mass flow of hot air and Ce/C_c the $T_{e,i}$ increase, effectiveness and transferred heat increase.

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 has been carried out. This has been done for inlet hot air in 125-225°c range, but it has been compared with inlet hot air only in 125°c, because the results were alike **(Fig. 6).**

Conclusion

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

1) A good agreement between experimental results and theoretical has been achieved.
2) At constant C_e/C_e, by increasing the inlet hot air temperature, the effectiveness and transferred heat increase.
3) At constant inlet hot air velocity, by increasing the inlet hot air temperature, the effectiveness and transferred heat increase.
4) As a result, the number of experimental tests that needed to be carried out on a large scale plant is quite limited.

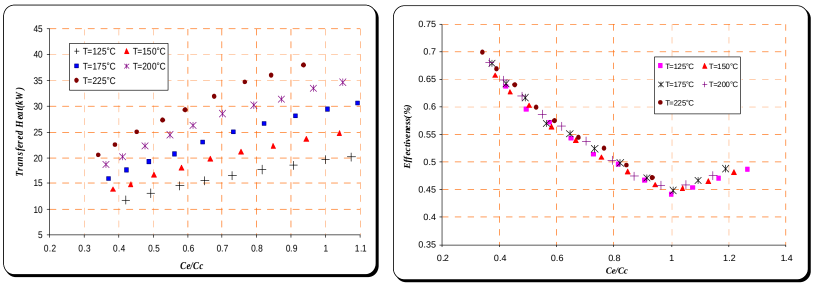

Fig3.effectiveness and transferred heat vs. inlet hot air temperature For two measure of C_e/C_c

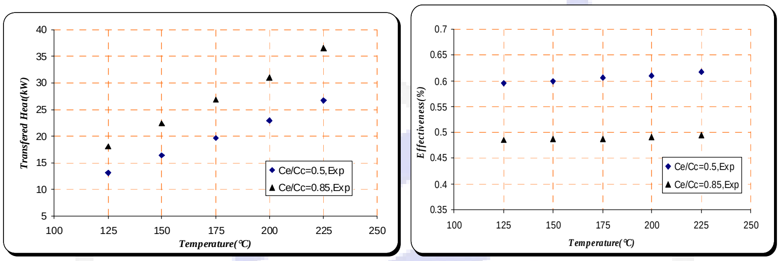

Fig4.effectiveness and transferred heat vs. inlet hot air temperature for two measure of C_e/C_c

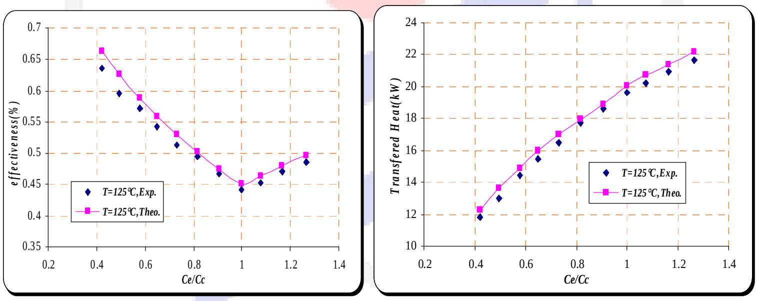

Fig5.effectiveness and transferred heat of gas- liquid THPHE vs. heat capacity ratio

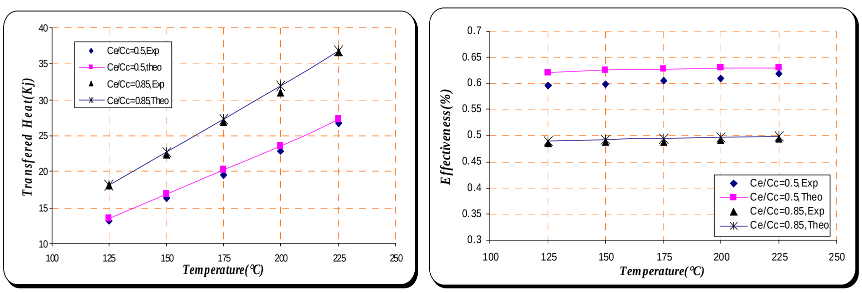

Fig6. Effectiveness and transferred heat of gas- liquid THPHE vs. temperature

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/m2)
R Thermal resistance, $(m^2{}^{\circ}C/W)$
S_c Minimum free-flow area in the core, (m2)
S_f Surface area of fins, (m2)
S_o Total frontal area of HPHE, (m2)
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

Re_{max} , $\mathrm{Re}_{max} = \frac{r_m U_{max} d_o}{m}$ 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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