International Journal of Chemical Sciences and Applications
ISSN 0976-2590, Online ISSN 2278 – 6015
Vol 4, Issue 1, 2013, pp 77-83
http://www.bipublication.com

PERFORMANCE EVALUATION OF PLATE HEAT EXCHANGER IN LAMINAR AND TURBULENT FLOW CONDITIONS

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[Received-19/12/2012, Accepted-21/03/2013]

ABSTRACT:

Experimental data was generated by performing the experiment on a plate heat exchanger for water/water system, at different flow rate and different hot water inlet temperature, at steady state conditions. Overall heat transfer coefficient (U) is determined for the performance evaluation of plate heat exchanger. The thermal properties of water (thermal Conductivity, Specific heat and density) at different temperature were calculated from the linear correlation developed in Polymath 5. To determine the film heat transfer coefficient as a function of pressure drop we have developed the correlations from the graphs plotted h against $\Delta P^{0.35}$ for Turbulent flow and, h against $\Delta P^{0.2}$ for laminar flow. These correlations and graphs can be used to study the effect of pressure drop (ΔP) on film heat transfer coefficient (h). Graphs presented in this work were plotted using computer software polymath 5 and Microsoft excel. Correlations for the convective heat transfer coefficient as a function of Reynolds number (Re), and Prandtl number (Pr) is suggested.

Keywords:*Plate type heat exchanger,Performance evaluation of Plate heat exchanger, heat exchanger design*

[I] NTRODUCTION

In heat exchanger application the plate heat exchanger is the most widely used in chemical processing and food industries. The advantages of the plate heat exchanger over conventional heat exchanger are as superior thermal performance, availability of wide variety of corrosion resistance alloys, ease of maintenance, expendability and multiphase capability, compact design and high heat transfer area. These special advantages of plate heat exchanger over conventional heat exchanger make them perfect for the application of heat transfer in the chemical processes industries. Plate heat exchangers are described briefly in articles [1, 2, 3, 4, 5, 9 and 12]. Advantages of plate heat exchanger are mentioned in article [1 and 7].

The plate heat exchanger consist of a number of corrugated metal sheets **[Figure-1]**, it's also having the gaskets to separate the fluids. The hot fluid flows in one direction in alternating chamber while the cold fluid flow in the other alternating chamber in counter current manner. The gasket material selection depends upon the operating condition and type of fluids. The plate heat exchanger is basically a series of individual plates fixed between two heavy end cover, the entire assembly is tied by the tie bolt. The plate heat exchanger is most suitable for liquid-liquid heat transfer duty that requires uniform and rapid heating and cooling.

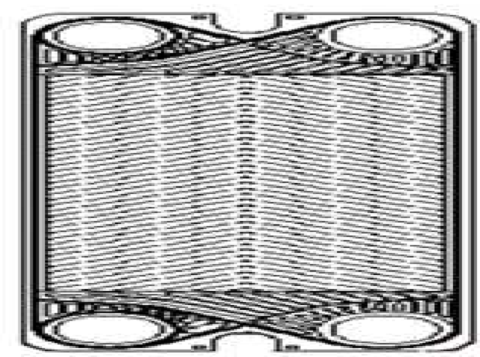
The aim of the present work is to determine the overall heat transfer coefficient for the performance evaluation of plate heat exchanger and suggest the correlation for the convective heat transfer [water-water duty] as a function of Reynolds number and pressure drop. Study on heat transfer and pressure drop in plate heat exchanger for water-water duty and Newtonian and Non-Newtonian fluid is presented in articles [1, 2, 3, and 6]

[II] MATERIALS AND METHODS

2.1. Experimental Procedure

The experiments were performed in a plate type heat exchanger having following geometrical configuration **[Table-1]**. Line diagram of experimental set up is shown in **[Figure-2]**.

Fig :1.



[Table-1].

Material	Stainless steel
Plate no	8
Length 'L' (m)	0.425
Width 'w' (m)	0.125
Thickness 't' (m)	0.001
Distance between plates 'b' (m)	0.003
Thermal Conductivity 'k'	17

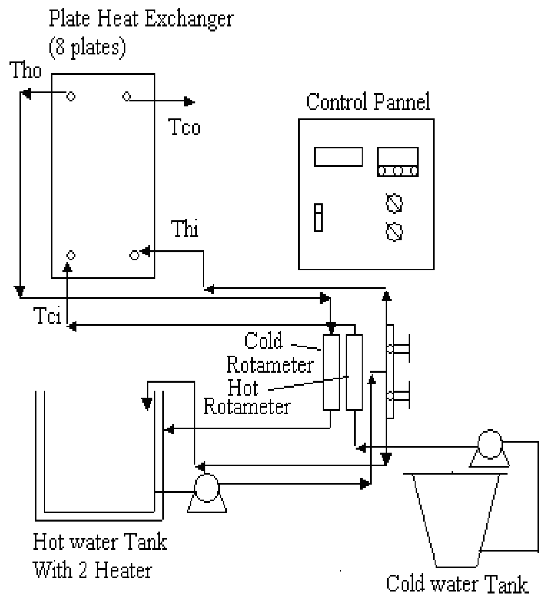
Table: 1. Geometrical characteristics of the Plate.

During experiment the temperature of inlet hot water was varied from 35 ^{0}C to 80 ^{0}C, by maintaining the temperature of cold water constant. The readings are noted at same flow-rate of hot and cold water.

Five probes of thermocouple connected to the multi-channel microprocessor device out of which two were connected to the hot water inlet and outlet connection, two were connected to the cold water inlet and outlet connection, and one was connected to a water bath. These temperatures were noted on display screen by rotating the multi-channel switch.

In this work the cold water and hot water were circulated through the heat exchanger plates at different temperatures and flow rates. The heat transfer takes place between hot and cold fluid. The Reynolds number varied from 178 to 1537. Cold water was pumped by a centrifugal pump. The flow rate was adjusted by means of control valve. Hot water from the hot water tank was pumped to the plate. The flow rates were determined by measuring exiting fluid in the 5 lit marked bucket. The flow measurement devices (rotameter) were also used to measure the flow rate

Fig: 2. Experimental Setup (line Diagram)



2.2. Heat Transfer Analysis

The determination of convective film heat transfer coefficient and overall heat transfer coefficient of water/water duty in a plate heat exchanger is based on the
following assumptions.

1. No internal heat generation.
2. No free convection.

The fluid properties are dependent on temperature. The properties of water are calculated by linear correlations using software polymath 5. As shown in **[Table-2].**

Viscosity of water is calculated by correlation:

μ=1 / (2.1482 (((T + 273)- 281.435) + ((8078.4+((T + 273) - 281.435)^2)$^{(1/2)}$)) – 120) [1]

In the present experimental work the plate heat exchanger was operated with a series arrangement and in counter-current flow manner. Therefore, the number of channel per pass is equal to unity

Mass flow rate per stream of fluid:

$$m = M / n \quad ; \quad n=1 \qquad [2]$$

Mass flow rate per unit area of fluid:

$$G = m / A_c \qquad [3]$$

The dimensionless Reynolds number and prandtl number are given by:

$$\mathrm{Re} = \frac{De.G}{\mu} \qquad [4]$$

$$\mathrm{Pr} = \left(\frac{Cp.\mu}{K} \right) \qquad [5]$$

[Table-2].

				Formula		
Specific Heat 'Cp' (J/kg.k)		$Cp = c_1 + (c_2.T) + (c_3.T^2) + (c_4.T^3) + (c_5.T^4) + (c_6.T^5)$				
Constant	C_1	C_2	C_3	C_4	C_5	C_6
	4220.6325	-3.989593	0.1199328	-0.0016129	0.00001056	-0.00000002504
Thermal Conductivity 'k' (w/m.K)		$k = c_1 + (c_2.T) + (c_3.T^2) + (c_4.T^3) + (c_5.T^4) + (c_6.T^5)$				
Constant	C_1	C_2	C_3	C_4	C_5	C_6
	0.5467126	0.0030777	-0.00002431	0.00000007425	0.0000000000113	-4.489E-13
Density 'ρ'(kg/m^3)		$\rho = c_1 + (c_2.T) + (c_3.T^2) + (c_4.T^3) + (c_5.T^4) + (c_6.T^5)$				
Constant	C_1	C_2	C_3	C_4	C_5	C_6
	999.85181	0.0613773	-0.0083206	0.00006448	-0.0000003977	0.000000001088

The film heat transfer coefficient of the two fluids can be estimated by using the following correlations.

For turbulent flow [6]:

$$Nu = (0.374)\,\mathrm{Re}^{0.668}\,\mathrm{Pr}^{0.333}(\mu / \mu_w)^{0.15} \qquad [6]$$

$$Nu = \frac{h.de}{k} \qquad [7]$$

de = 2.b

For laminar flow[12]:

$$j = \left(\frac{h}{Cp.G} \right) \mathrm{Pr}^{0.667} \left(\frac{\mu_w}{\mu} \right) = 0.742(\mathrm{Re}^{-0.62}) \qquad [8]$$

$$h = \frac{\left(0.742\,\mathrm{Re}^{-0.62}\,Cp.G\right)}{\mathrm{Pr}^{0.667}\left(\frac{\mu_w}{\mu} \right)^{0.15}} \qquad [9]$$

The Overall heat transfer coefficient *U* is calculated as:

$$\frac{1}{U} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{x_w}{k} \qquad [10]$$

written as:

$$\frac{1}{U} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{x_w}{k} + R_h + R_c \qquad [11]$$

Over all heat transfer coefficient can be calculated by the general design equation.

The amount of heat transferred *(Q)* depends on three factors: *U, A and LMTD*.

$$Q = U.A_t.F.LMTD \qquad [12]$$

$$U = \frac{Q}{(A_t.F.LMTD)} \qquad [13]$$

A_t is the total area needed for series flow exchanger.

$$A_t = A_p(Np-2) \qquad [14]$$

$$A_t = w.L.N \qquad [15]$$

F is the factor used to correct the mean logarithmic temperature difference. It was determined from the graph available in (4). It is a dimensionless number. It is assumed that, no significant heat exchange takes place between the equipment and the surrounding air.

For the true counter-current flow, the *LMTD* is determined by the formula;

$$LMTD = \left(\frac{(\Delta T_1 - \Delta T_2)}{\ln(\Delta T_1 / \Delta T_2)}\right) \qquad [16]$$

$$\Delta T_1 = T_{h_i} - T_{c_o}, \Delta T_2 = T_{ho} - T_{c_i} \qquad [17]$$

Q is the heat load calculated by using the energy balance:

$$Q = M.Cp.(T_i - T_o) \qquad [18]$$

Plate wall temperature

$$T_w = \frac{\left[T_h + \frac{h_c}{h_h}Tc\right]}{1 + \frac{h_c}{h_h}} \qquad [19]$$

The Pressure loss can be predicted from equations as follows [1,2 and 12]:

$$j_f = C.\mathrm{Re}^b \qquad [20]$$

$$\Delta P = 8.j_f.\left(\frac{G^2}{2.\rho}\right).\left(\frac{\mu}{\mu_w}\right)^{\gamma}.\frac{L}{de} \qquad [21]$$

C	*b*	*Γ*	*Re*
17.5	-0.8	0.25	≤ 200
1.26	-0.13	0.14	≥ 200

[III] RESULTS AND DISCUSSION

The Convective heat transfer coefficient of hot and cold water is a function of the Reynolds number. From the experimental data the graphs were plotted for Reynolds number *Re* against ($Nu/Pr^{0.333}$) as shown in **[Figure- 3and 4].**

Fig:3. Reynolds number Re against ($Nu_h/Pr_h^{0.333}$) For Laminar flow

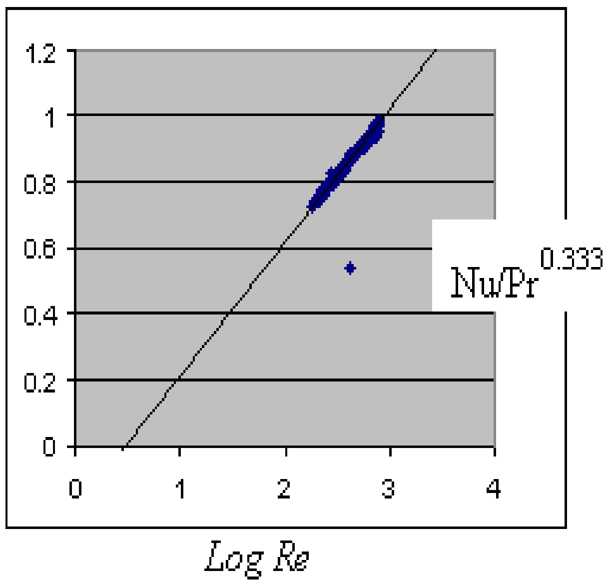
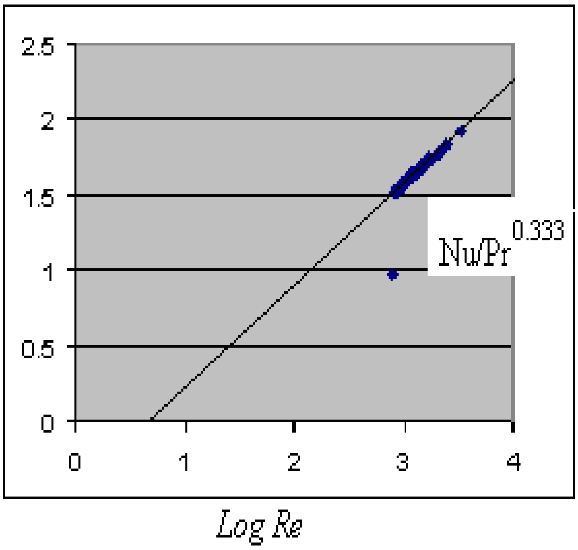


Fig:4. Reynolds number Re against ($Nu_h/Pr_h^{0.333}$) For Turbulent flow



For the Reynolds number less than 800 the experimental data generates a straight line of slope approximately 0.333 which indicates the laminar flow conditions and the heat transfer correlation proposed as;

$$Nu = C.(\mathrm{Re})^{\alpha}.(\mathrm{Pr})^{0.33}$$

$$Nu = 0.977(\mathrm{Re}.\mathrm{Pr})^{0.33} \qquad [22]$$

For the Reynolds number greater than 800 the experimental data generates a straight line having a slope 0.68 and intercept equal to 0.338 .The correlation developed from these values is;

$$Nu = 0.338\,\mathrm{Re}^{0.68}\,\mathrm{Pr}^{0.33} \qquad [23]$$

From these equations the convective heat transfer coefficient (film heat transfer coefficient) can be calculated. In equation [22] and [23], no viscosity correction factors have been included because the temperature difference used during experiment was not much greater. So, the viscosity variation has been neglected

Fig:5. Film heat transfer coefficient against Pressure drop For Laminar flow "hot water side"

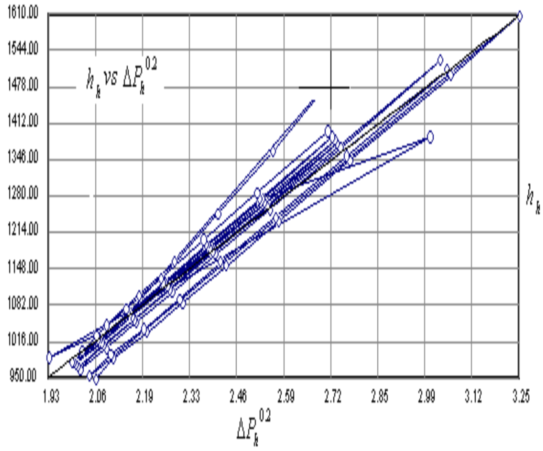


Fig:6. Film heat transfer coefficient against Pressure drop For turbulent flow "hot water side"

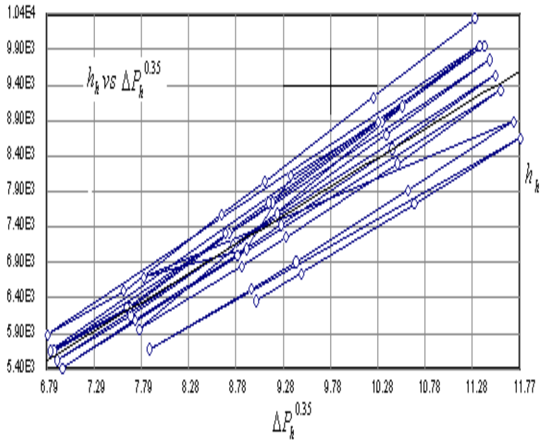


Fig:7. Film heat transfer coefficient against Pressure drop For Laminar flow "Cold water side"

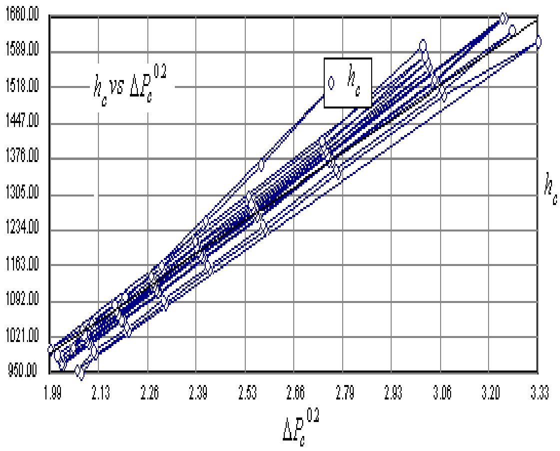
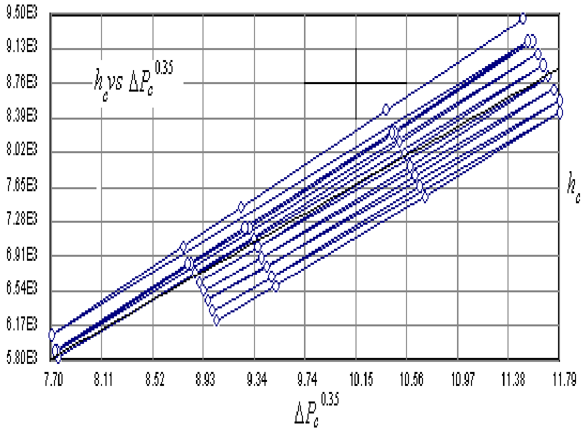


Fig:8. Film heat transfer coefficient against Pressure drop For Turbulent flow "Cold water side"



From the above **[Figure 5, 6, 7 and 8]** it is clear that, higher the pressure drop, greater the film heat transfer coefficient. The graphs of film heat transfer coefficient against Pressure drop were plotted using software polymath 5.The nature of line is straight and passes through the origin and slope is obtained as;

For laminar flow cold water side slope= *497.49*

For turbulent flow cold water side slope= 757.85

For laminar flow hot water side slope= *495.17*

For turbulent flow hot water side slope= 817.11

Therefore the film heat transfer values can be obtained by correlation as:

$$h_c = 497.49\Delta P_c^{0.2} \qquad [24]$$

$$h_c = 757.85\Delta P_c^{0.35} \qquad [25]$$

$$h_h = 495.17\Delta P_h^{0.2} \qquad [26]$$

$$h_h = 817.11\Delta P_h^{0.35} \qquad [27]$$

The pressure drop exponent 0.35 for turbulent flow is available in [11]. Pressure drop exponent 0.2 for laminar flow is adjusted in software polymath 5 for the experimental data.

[IV] CONCLUSION

Simple correlation for film coefficient of heat transfer is developed in terms of pressure drop the values of overall heat transfer coefficient for laminar flow and turbulent flow are shown in **[Table-3].** The values are in agreement with the values reported in literature [11].

Table-3.

S.No	***Mass flow rate***	***h_h***	***h_c***	***U***
Laminar flow				
1	0.0083	1059.73	1064.19	514.89
2.	0.0093	1103.59	1108.47	535.58
3.	0.0104	1153.36	1158.53	558.97
4.	0.0119	1211.94	1217.35	586.37
5.	0.0139	1283.88	1289.67	619.92
6.	0.0167	1373.74	1379.49	661.52
7.	0.0208	1492.69	1498.96	717.39
8.	0.0278	1664.03	1671.39	794.86
9.	0.0333	1780.85	1788.69	847.87
Turbulent flow				
10.	0.0417	8872.98	8223.36	3411.45
11.	0.0455	9392.21	8704.33	3569.13
12.	0.0556	10710.29	9925.29	3953.44
13	0.0658	11964.51	11087.37	4299.29

[V] NOMENCLATURE

A_c Flow area [m^2]

A_t Total area [m^2]

Ap Projected area [m^2]

Cp Specific heat capacity [J/kg°C]

de Hydraulic diameter [m]

F Log mean temperature difference correction factor

G Mass flow rate per unit area [kg/m^2.s]

h Film heat transfer coefficient[W/m^2 ^{0}C]

Jf Friction factor

k Thermal conductivity [W/m°C]

L Length (flow length) [m]

LMTD Log mean temperature difference

M Mass flow rate [kg/s]

m Mass flow per stream [kg/s]

N Number of Plates

Q Heat load [W]

T Temperature [°C]

t Plate thickness [m]

U	Overall heat transfer coefficient. [W/m^2 ^{0}C]
V	Velocity [m/s]
w	Flow width [m]
ρ	Density [kg/m3]
μ	Viscosity [N.s/m^2]
ΔP	Pressure Drop [N/m^2]

Subscripts and indices

c	Cod
h	Hot
i	Inlet
o	Outlet
w	Wall

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