PERFORMANCE ANALYSIS OF A PLATE HEAT EXCHANGER AND ITS COMPARISON WITH DOUBLE PIPE EXCHANGER

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Abstract

The primary focus of this paper is to evaluate the performance of a Plate type heat exchanger. The evaluation is carried out for namely two heat transfer parameters, heat transfer coefficient and heat exchanger effectiveness for varying Reynolds number of hot and cold fluids. From the experimental values of inlet and outlet temperatures measured, the above two parameters were calculated.

The analytical approach for calculation is illustrated for one sample reading. The heat transfer coefficient and heat exchanger effectiveness was seen to increase with increasing Reynolds number of both the fluids. A PHE belongs to the family of compact plate heat exchangers. Therefore, the secondary objective was to compare the PHE with conventional heat exchanger i.e. Double pipe. The comparison was done on the basis of area required for a given temperature drop and heat transfer capacity. The numerical values of areas obtained confirmed that a PHE requires the least space for operation.

Keywords: *Plate heat exchanger, Reynolds number, heat transfer coefficient, effectiveness, compact*

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1. INTRODUCTION

Plate heat exchangers were introduced in the 1930s mainly for the food industries because of their ease of cleaning and their design reached maturity in the 1960s with the development of more effective plate geometries, assemblies, and improved gasket materials. The range of possible applications has widened considerably and, at present, under specific and appropriate conditions, overlaps and successfully competes in areas historically considered to be the domain of tubular heat exchangers. Therefore, they can be used as an alternative to tube-and-shell type heat exchangers for low-and medium-pressure liquid to-liquid heat transfer applications.

Fig -1: Disassembled view of PHE

To successfully incorporate a PHE, we need to have an understanding of the variation of a PHE's heat transfer characteristics with respect to changes in its input parameters. An attempt has been made in this paper to than end. Also, to demonstrate the space saving capability of a PHE, we determined the area required by a Plate heat exchanger, a double pipe heat exchanger for achieving the same temperature drop and heat transfer capacity.

2. EXPERIMENTATION

For experimentation, a working model of PHE with 9 plates was put on a test setup. The plates used were made of Stainless 3016L having a thermal conductivity (K) 16.2 W/mK. The plate dimensions are, 395 x 122 mm the channel diameter is 32 mm and the plate thickness is 0.5 mm.**The actual effective area of the PHE as specified by the manufacturer is 0.303751 m^2**. The chevron angle (ß) of the plates employed is 65º w.r.t the vertical. The gasket used is of Nitrile Butadiene with a temperature range of30 to 80 º C. The temperatures and flow rates were measured by alcohol thermometer and rotameter respectively. Water was used as both the cold and hot fluid and the source of hot fluid was an electric heater. The cold fluid was obtained from the city mains. CPVC pipes were used to convey the fluids having a maximum permissible temperature of 80º C.

The apparatus mentioned above was arranged in a way described below.

- The corrugated plates with gaskets were compressed between two end plates with the help of four compression bolts.
- The holes in the plates form cylindrical channels which are directly connected to flow lines.
- The hot and cold fluid flows through the channels formed due to corrugations on the plates.
- The plates are arranged such that the hot and cold fluid flow in alternate channels with the help of gaskets.
- The hot fluid from the heater flows via the C-PVC pipe into the upper right inlet and flows out of the lower right outlet.
- The cold fluid from the tap flows via the C-PVC pipe into the lower left inlet and flows out of the upper left outlet.
- The above arrangement allows the fluids to flow in counter flow in a single pass.
- A rotameter is used to measure the mass flow rates of the hot fluid while a beaker-stopwatch arrangement was used to measure the mass flow rate of the cold fluid.
- The temperature of the fluids was measured using 4 alcohol thermometers connected to the inlet and outlet ports.

The procedure of the experimentation for achieving the above was as follows

- Hot fluid flow rate was measured using a rotameter.
- Cold fluid flow rate was measured using a graduated beaker and a stop watch.
- The first set of observations were taken by keeping the flow rate of cold fluid constant at a certain value and the hot fluid flow rate was varied.
- For the second set of observations, the hot fluid flow rate was varied for a different value of cold fluid flow rate,
- The fluid flow rates were varied by using a ball valve.
- For each combination of hot and cold fluid flow rates, all the four temperatures, T_{hi}, T_{ho}, T_{ci}, T_{co} were noted down using the alcohol thermometers.
- Before taking the temperature readings, the apparatus was kept undisturbed for at least 10 minutes after setting the fluid flow rates for achieving steady state conditions.

3. OBSERVATIONS AND CALCULATIONS

As mentioned in the experimental procedure we recorded temperatures for different flow rates of the hot stream keeping the cold stream flow rate constant.

For the first set of observations, the cold stream flow rate was kept at the following value.
$\dot{Q}_C$ = 2.3 LPM

Fig -2: PHE in working

Table -1: Observation table 1

T_{CI}	T_{CO}	T_{HI}	T_{HO}	$\dot{q}_C$	$\dot{q}_h$
°C	°C	°C	°C	LPM	LPM
29.7	38.8	53	41	2.3	2.3
29.7	38.5	49	39	2.3	3
29.7	38.2	45.6	38	2.3	3.4
29.7	37.8	43.5	37	2.3	4.4
29.7	37	42	36.5	2.3	5

For the second set of observations, the cold stream flow rate was kept at the following value.
$\dot{Q}_C$= 3.4 LPM

Table -2: Observation table 2

T_{CI}	T_{CO}	T_{HI}	T_{HO}	$\dot{q}_C$	$\dot{q}_h$
°C	°C	°C	°C	LPM	LPM
29	36	52.5	38.5	3.4	2.3
29	37.2	48.5	36	3.4	3
29	36.1	45.2	36.2	3.4	3.4
29	35.9	42.5	35.5	3.4	4.4
29	36.2	42.2	36	3.4	5

Heat transfer coefficient and effectiveness calculation-

Heat rejected by hot fluid,

$$Q_H = \dot{m}_h c_p (T_{HI} - T_{HO})$$

Heat accepted by cold fluid,

$$Q_C = \dot{m}_C c_p (T_{CO} - T_{CI})$$

The Q_H and Q_C ideally should be same but due to physical irreversibilties they are not equal. Therefore for calculation purposes, the averages of the two are taken.

$$Q_{avg} = \frac{Q_{C+}Q_H}{2}$$

The heat transfer is given by the following equation,

$$Q_{avg} = UA[LMTD]$$

Where **actual effective area of the plate as specified by the manufacturer (A) =0.303751 m^2** and [LMTD] is given by,

$$[LMTD] = \frac{\theta_i - \theta_e}{\ln\frac{\theta_i}{\theta_e}}$$

$$\theta_i = T_{HI} - T_{CO}$$

$$\theta_e = T_{HO} - T_{CI}$$

We get the heat transfer coefficient by the following equation,

$$\frac{1}{U} = \frac{1}{h_C} + \frac{1}{h_H} + \frac{t}{K}$$

Assuming$h_C = h_H$,

$$\frac{1}{U} = \frac{2}{h} + \frac{t}{K}$$

Now, effectiveness is given by,

$$\epsilon = \frac{Q_{avg}}{Q_{max}}$$

Where Q_{max} is given by,

$$Q_{max} = \left(\dot{m} \times C_p\right)_{min} (T_{HI} - T_{CI})$$

The above calculations were done for two readings which had the same flow rate for cold and hot fluid for the ease of calculations as their Reynolds number would be equal. The results are tabulated as follows-

$\dot{M}_h$	$\dot{M}_c$	q_a	U	h	ϵ
Kg/s	Kg/s	Kw	W/m^2.k	W/m^2.k	
0.038333	0.038333	1.693	439.12	890.31	0.4527
0.0567	0.0567	1.9104	775.225	1588.455	0.4967

Now, the area required by Double pipe HE and Shell & tube HE to achieve the same temperature drop as attained by PHE is found out. The formulas and relations to find out the area were referred from the book on Heat Exchangers by SadicKakac, HongtanLui and AnchasaPramuanjaroenkij.

Area requirement for Double pipe HE-

Dimensions of a commercially available double pipe heat exchanger have been found out. For that purpose, a set of dimensions for the diameters of inside and outside tubes are assumed from the TEMA standards. Also, the inside tube was assumed to have fins and the required fin dimensions were also taken from TEMA standards. They are as follows-

Shell ID	Tube OD	Tube wall	Tube ID	Fin height	Max. no. of fins
(mm)	(mm)	(mm)	(mm)	(mm)	
D_S	D_O		D_I	H_F	N_F
114	48.3	3.68	40.94	25.4	36

The material of the double pipes was taken to be the same as that of the PHE i.e. Stainless Steel 3016L having a thermal conductivity of 16.2 W/mk. The length was calculated for the temperatures and fluid flowrate values from the last reading of observation table 1.

The Hot fluid is assumed to flow from the inside tube and cold fluid through the annular space.

$$Q_{avg} = U_O A_O (LMTD)$$

Where outside pipe area $A_O = \pi D_O L$

$$1.5453 \times 10^3 = U_O \pi \times 48.3 \times 10^{-3} \times L \times 5.8539$$
$$U_O L = 1739.72$$

$$\frac{1}{U_O} = \frac{A_O}{A_I h_I} + \frac{1}{h_O} + A_O \frac{\ln\frac{D_O}{D_I}}{2\pi K L}$$

Where inside pipe area $A_I = \pi D_I L$,by substituting the values we get,

$$\frac{L}{1739.72} = 2.46 \times 10^{-4} + \frac{1.1797}{h_I} + \frac{1}{h_O}$$

Consider the above equation as (1)

For inner pipe-
Reynolds number is given by,

$$Re = \frac{\rho V_m D_I}{\mu}$$

Where velocity is given by,

$$V_m = \frac{\dot{m}_c}{\frac{\pi}{4} {D_I}^2 \rho}$$

Taking properties of water at an average temperature of hot inlet and outlet T_H=39.5 ºC

Density $(\rho) = 992.5$ Kg/m^3
Viscosity $(\mu) = 0.000663$
Prandtl Number (Pr) $= 4.39$
Thermal conductivity $(K) = 0.631$ W/mK

We get,

$$V_m = 0.06375\, m/s$$

And

$$Re = 3907.01$$

Hence the flow is turbulent and the heat transfer will be governed by the empirical relation given byPetukhov-Kirillov which gives the value of Nusselt Number as,

$$\mathrm{Nu} = \frac{\frac{f}{2} \times Re \times Pr}{1 + 8.7(\frac{f}{2})^{0.5}(Pr - 1)}$$

Where$f = (1.58 \ln Re - 3.28)^{-2}$

$$Re = 3907.01$$

$$Pr = 4.39$$

We get,

$$f = 0.01044$$

And Nusselt Number as

$$Nu = 28.595$$

Now Nusselt Number is also given by,

$$Nu = \frac{hD_e}{K}$$

Where,
Equivalent diameter $D_e = \frac{4A_C}{P} = D_I = 0.04094\, m$
Thermal conductivity $(K) = 0.631$ W/mK

Hence substituting we get, heat transfer coefficient at inlet as,

$$h_I = 440.64\, \mathrm{W}/m^2.\mathrm{k}$$

For outer pipe-
Reynolds number is given by,

$$Re = \frac{\rho V_m D_h}{\mu}$$

Now the hydraulic diameter for annular space is given by,

$$D_h = \frac{4A_C}{P_w}$$

Where,

$$A_C = \frac{\pi}{4}\left[\left(D_S{}^2 - D_O{}^2 N_t\right) - \left(\delta H_f N_f N_t\right)\right]$$

$$A_C = 7.551 \times 10^{-3} m^2$$

And

$$P_W = \pi(D_I + D_O N_t) + \left(2H_f N_f N_t\right)$$

$$P_W = 2.338\, m$$

Substituting, we get

$$D_h = 0.0129\, m$$

And also

$$V_m = \dot{m}_h / \rho A_C$$

$$V_m = 0.005103 \frac{m}{s}$$

Hence we get Reynolds number as,

$$Re = 87.88$$

Hence the flow is laminar and the heat transfer will be governed by the empirical relation given by Sieder and Tate.

$$Nu = 1.86(Re \times Pr \times \frac{D_h}{L})^{1/3}$$

Here the Length (L) is unknown. Substituting other parameters by taking properties of water at an average temperature of cold inlet and outlet T_c=33.35 ºC

Density $(\rho) = 994.59$ Kg/m^3
Viscosity $(\mu) = 0.000745$
Prandtl Number (Pr) $= 4.7246$
Thermal conductivity $(K) = 0.6239$ W/mK

We get,

$$Nu = \frac{125.212}{L^{\frac{1}{3}}}$$

Now,

$$Nu = \frac{hD_e}{K}$$

Where

$$D_e = \frac{4A_C}{P_h}$$

$$P_h = \pi D_O N_t + 2N_f H_f N_t = 1.84\ m$$

Hence $D_e = 0.164\ m$

$$\frac{0.0164 \times h_c}{0.631} = \frac{3.254}{L^{1/3}}$$

$h_O = \frac{125.212}{L^{1/3}}$

Substituting the values of heat transfer coefficient of both inlet and outlet in equation 1, we get the value of Length (L) as

$$\frac{L}{1739.72} = 2.46 \times 10^{-4} + \frac{1.1797}{440.64} + \frac{1}{125.212}$$

Solving for unknown L, we get

$$L = 59.2416\ m$$

The area of heat transfer for the standard diametrical dimensions and the above calculated length is,

$$A = N_t L(\pi D_O + 2N_f H_f)$$

$$A = 1 \times 59.2416 \times ((\pi \times 0.0483) + (2 \times 36 \times 0.0254))$$
$$A = 117.33\ m^2$$

This is the area required by a standard Double pipe heat exchanger to attain the same temperature drop and heat transfer capacity. The area required by the Plate type heat exchanger is 0.303751 m^2.

4. CONCLUSION

From this experimentation, the heat transfer coefficient and effectiveness were calculated. From the values it can be seen that the heat transfer coefficient increases as both the hot and cold fluid Reynolds number increases. Also, from the calculation of area required for double pipe heat exchanger, it can be seen that it requires far more area than Plate type heat exchanger to achieve same temperature drop and heat transfer capacity.

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