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Heat Transfer Analysis of Gasketed Plate Heat Exchanger

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Abstract: Compact heat exchangers are most widely used for heat transfer applications in industries. Plate heat exchanger is one such compact heat exchanger, provides more area for heat transfer between two fluids in comparison with shell and tube heat exchanger. The present work deals with experimental heat transfer data performed on plate type heat exchanger which is used in hydraulic cooling system in an industry. The heat exchanger used for carrying out this work consists of thin metal welded plates of stainless steel with 0.5mm thickness; distance between two plates is 5mm, chevron angle 60° and counter flow arrangement. The total heat transfer area is 161.62 m^2. This consists of total 249 numbers of plates and it is designed to withstand with 65°C temperature with a flow rate of 64751 kg/h and cold fluid enters with a flow rate of 82366 kg/h at 35°C and leaves at 44.29°C, pressure drop is neglected. The inlet and outlet temperatures of cold and hot fluids are been observed and with that conditions performance evaluation is done. Based on the experimental data, a correlation will estimate for Nusselt number as a function of Reynolds number, Prandtl number and chevron angle and the outputs obtained are convective heat transfer coefficient, overall heat transfer coefficient, and exchanger effectiveness. From the obtained results, graphs are drawn to assess the performance of the Gasketed Plate heat exchanger.

Keywords: Plate heat exchanger, Convective heat transfer coefficient, Effectiveness, Overall heat transfer coefficient, Reynolds number.

1. Introduction

Plate Heat Exchangers (PHE) have been increasingly used in the past decades, not only in chemical and food processing industries for which it was originated in the 1930s, but also in wide range of industrial and energy application. This is due to their compactness, effectiveness in transferring heat, bio-fouling resistance and the ease of dismantling, cleaning, and also the ease to adapt to changes in thermal demand.

The plate heat exchanger is basically a series of individual plates pressed between two heavy end covers and compressed by tightening bolts. These plates are gasketed, welded or brazed together depending on the application of the heat exchanger. The basic geometry of plates used in plate heat exchanger is shown in Fig-1.Stainless steel is a commonly used metal for the plates because of its ability to withstand high temperatures, its strength, and its corrosion resistance. The sealing of the plates is achieved by gaskets fitted at their ends. The plates are fitted with a gasket which seals the interpolate channel and directs the fluids into alternate channels.

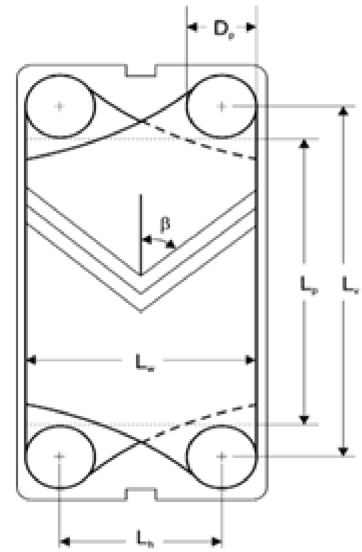

Fig-1: Corrugation features of the chevron type plate

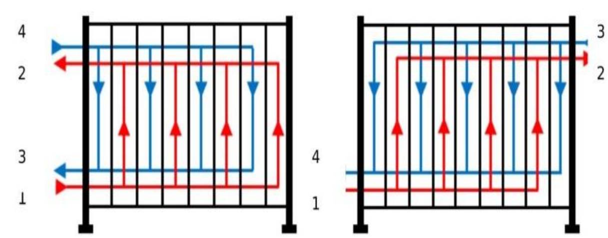

Fig-2: Fluid flow in plate heat exchanger 1. Hot oil inlet, 2. Hot oil outlet, 3. Cold water inlet, 4. Cold water outlet.

Fig-2, illustrates the nature of fluid flow through the plate heat exchanger. The primary and secondary fluids flow in opposite directions on either side of the plates. Multi-pass arrangements can be implemented, depending on the arrangement of the gaskets between the plates.

2. Literature

Several researchers discussed performance analysis of Plate heat exchangers used for different purposes and developed correlations both numerically and experimentally.

Tambe Shahanwaj et al.[1] in their paper they mainly focused on thermal design of plate heat exchanger for one pass one

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1 December 2016

arrangement and water-water heat transfer and analyzed with regard to overall heat transfer coefficient, effectiveness.

Warnakulasuriya and Worek [2] investigated heat transfer and pressure drop of a viscous absorbent salt solution in a commercial plate heat exchanger. Based on the experimental data, correlations for Nusselt number and friction factor were proposed.

Vishal Naik et al. [3] carried out the experimental study of the effect of chevron angle with wide range of Reynolds Number on heat transfer characteristics of Gasketed oil-water PHE.

Jogi Nikhil et al. [4] reviewed the effect of plate geometry on heat transfer characteristics of corrugated PHE by conducting experiment for single phase flow (water-to-water) configurations and studied their effect. Based on that data, a simplified correlation estimated the Nusselt number as a function of Reynolds number, Prandtl number and chevron angle.

Akturk et al. [5] in their study experiments are performed with a commercial plate heat exchanger with 30° chevron angle. New Nusselt and friction factor coefficient correlations are found. The obtained correlations can be used between a Reynolds number range of 450 - 5250.

Dardour et al. [6] done a numerical analysis of the thermal performance of a plate type heat exchanger with parallel flow configuration.

Lin et al. [7] in their study they developed dimensionless correlations to characterize the heat transfer performance of the corrugated channel in a plate heat exchanger by using the Buckingham Pi theorem.

Gherasim Iulian et al. [8] presented an experimental investigation of the hydrodynamic and thermal fields in a two channel chevron-type plate heat exchanger for laminar and turbulent conditions.

Anil Kumar Khandale et al. [9] carried out performance evaluation of heat transfer enhancement by corrugated plate heat exchanger. They investigated the heat transfer characteristics and thermal performance of plate heat exchanger with a mixed plate configuration.

3. Analysis of Gasketed Plate Heat Exchanger

The performance of a plate heat exchanger can be analyzed by putting the necessary equations in order as below:

Heat duty is defined as the product of mass flow rate specific heat capacity and the temperature difference between inlet and outlet fluid temperatures.

$$Q = mC_p\Delta T \qquad (1)$$

Heat rejected by hot fluid

$$Q=m_h C_{ph}\Delta T_h \qquad (2)$$

Heat absorbed by cold fluid

$$Q=m_c C_{pc}\Delta T_c \qquad (3)$$

The logarithmic mean temperature difference is

$$\Delta T_{lm}=\frac{\Delta T_1-\Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}} \qquad (4)$$

$$\Delta T_1=T_{h1}-T_{c2} \qquad \Delta T_2=T_{h2}-T_{c1}$$

3.1 Governing parameters and calculation procedure

Total heat transfer area of a plate (effective heat transfer area) is given by

$$A_1=\phi\times A_{1p} \qquad (5)$$

The enlargement factor is given by equation .It generally lies between $1.15<\phi<1.25$. A_{1p} is approximated from fig-2 as:

$$A_{1p}=L_p\cdot L_w \qquad (6)$$

$L_p \& L_w$ can be estimated from the port distance $L_v \& L_h$ and port diameter D_p are given as below

$$L_p=L_v-D_p \qquad L_w=L_h+D_p$$

As specified by fig-2 the mean channel spacing (b) can be taken as the difference between plate pitch (p) and plate thickness (t) thus

$$b=p-t \qquad p=\frac{L_c}{N_t} \qquad (7)$$

The equivalent diameter of the channel, D_e, is defined as:

$$D_e=\frac{4\times b\times L_w}{2(b+L_w\phi)}\approx\frac{2b}{\phi} \qquad (8)$$

With the approximation that $b \ll L_w$

Single-phase heat transfer

The heat transfer correlation is in the form

$$Nu=C_1(Re)^m(Pr)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.17} \qquad (9)$$

C_1 and m values are obtained from literature. The Reynolds number, Re, based on channel mass velocity and the equivalent diameter, D_e, of the channel is defined as:

$$Re=\frac{G_c D_e}{\mu} \qquad (10)$$

Where, G_c = channel mass velocity which is given by

$$G_c=\frac{\dot{m}}{N_{cp} b l_w} \qquad (11)$$

Where N_{cp} is the number of channel per pass and obtained by following equation

$$N_{cp} = \frac{N_t - 1}{2N_p} \quad (12)$$

Pressure drop

The total pressure drop is composed of the frictional channel pressure drop, Δp_c and the port pressure drop Δp_p.

The channel pressure drop is given by the equation

$$\Delta p_c = \frac{4fL_{eff} N_p}{D_e} \frac{G_c^2}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17} \quad (13)$$

The friction factor (f) in equation 15 is given by:

$$f = \frac{C_2}{\mathrm{Re}^p} \quad (14)$$

C_2 and p values are taken from literature.

The pressure drop in the port ducts, Δp_p, can be roughly estimated as 1.4 velocity head

$$\Delta p_p = 1.4 N_p \frac{G_p^2}{2\rho} \qquad G_p = \frac{\dot{m}}{\frac{\pi D_p^2}{4}} \quad (15)$$

The total pressure drop is then given by

$$\Delta p_t = \Delta p_c + \Delta p_p \quad (16)$$

The overall heat transfer coefficient for a clean surface:

$$\frac{1}{U_c} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k_w} \quad (17)$$

And under fouling conditions (fouled or service overall heat transfer coefficient):

$$\frac{1}{U_f} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k_w} + R_{fh} + R_{fc} \quad (18)$$

The relationship between U_c, fouled U_f, and the cleanliness factor, CF, can be written as:

$$U_f = U_c(CF) = \frac{1}{\frac{1}{U_c} + R_{fh} + R_{fc}} \quad (19)$$

The heat balance relations in PHE are the same as for tubular heat exchangers. The required heat duty, Q_r, for cold and hot streams is obtained from equations 2 and 3.

On the other hand, the actually obtained heat duty, for fouled conditions is defined as

$$Q_f = U_f A_e F \Delta T_{lm} \quad (20)$$

A comparison between Q_r and Q_f defines the safety factor, C_s of the design:

$$C_s = \frac{Q_f}{Q_r} \quad (21)$$

3.2 Thermal performance

Heat exchanger has some heat transfer equations, resulting in the following dimensionless groups.

Heat capacity ratio:

$$C^* = \frac{C_{min}}{C_{max}} \quad (22)$$

Where C_{min} and C_{max} are the smaller and larger of the two magnitudes of C_h and C_c, respectively.

Heat transfer effectiveness:

It is defined as ratio of actual heat transfer rate to the thermodynamically limited maximum possible heat transfer rate in a counter flow heat exchanger.

$$\varepsilon = \frac{Q}{Q_{max}} \quad (23)$$

The actual heat transfer rate is obtained by either the energy rejected by the hot fluid or energy absorbed by the cold fluid, from equations (2) or (3).

The fluid that might undergo the maximum temperature difference is the fluid with the minimum heat capacity rate $C_{min.}$ Therefore, the maximum possible heat transfer is expressed as

$$Q_{max} = (\dot{m} c_p)_c (T_{h1} - T_{c1}) \text{ if } C_c < C_h$$

$$Q_{max} = (\dot{m} c_p)_h (T_{h1} - T_{c1}) \text{ if } C_h < C_c \quad (24)$$

This can be obtained with a counter flow heat exchanger if an infinite heat transfer area were available. Heat exchanger effectiveness, ε, is therefore written as

$$\varepsilon = \frac{C_h (T_{h1} - T_{h2})}{C_{min} (T_{h1} - T_{c1})} = \frac{C_c (T_{c2} - T_{c1})}{C_{min} (T_{h1} - T_{c1})} \quad (25)$$

The first definition is for $C_h = C_{min}$, and the second is for $C_c = C_{min}$

4. Results and Discussion

The values got from the above are shown plotted as graphs depicting their behavior as mentioned below.

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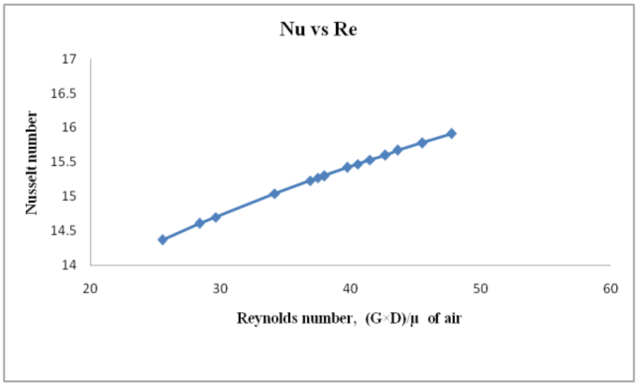

Fig-3: Effect of Variation of Reynolds Number to Nusselt Number for 60° chevron plate

The above fig-3 indicates that, as Reynolds number increases from 20 to 60, the Nusselt number also being increases from 14 to 17. Similarly as hot oil inlet temperature increases the Nusselt number goes on increasing. Hence Nusselt number is directly proportional to Reynolds number and inlet temperature. By correlations from literature, it is clear that as Reynolds number and inlet temperature of oil increases, the Nusselt number goes on increasing automatically.

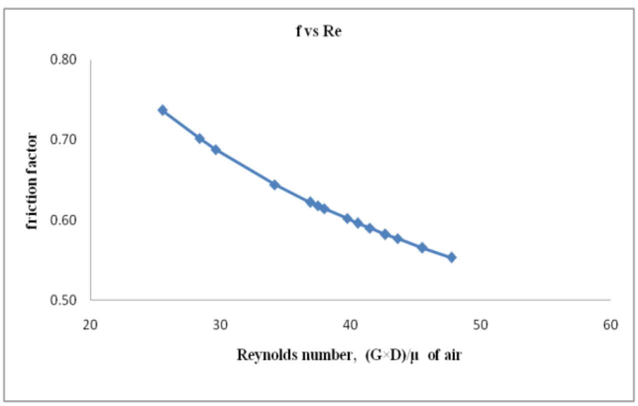

Fig-4: Effect of variation of friction factor with Reynolds number of oil

The fig-4 indicates the graph between friction factor and Reynolds number. It is concluded from the graph that the friction factor f, decreased with increase in Reynolds number. However, increasing Reynolds number can be achieved with high flow rates which will increase pressure drop dominantly more than f factor because of the component G_c^2 in equation.

The below graph shows the variation of effectiveness to Reynolds Number and NTU. It is clearly that as the increase of mass flow rate, Reynolds number increases and thus the effectiveness increases. The maximum effectiveness observed in this performance evaluation is 0.949. The effectiveness increases with increase of heat transfer rate.

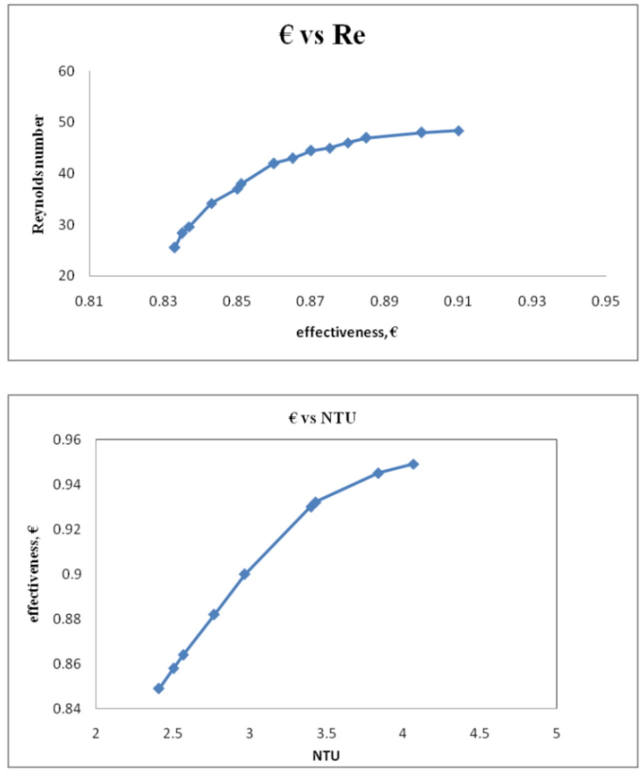

Fig-5: Effect of variation of effectiveness with Reynolds number of oil and number of transfer units

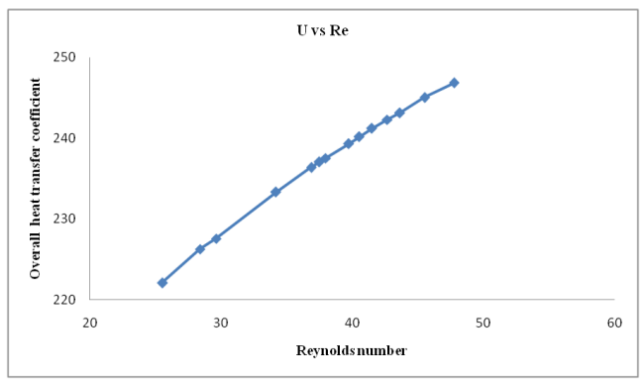

Fig-6: Effect of variation of Reynolds number to Overall heat transfer coefficient

The fig-6 shows the variation of overall heat transfer coefficient with Reynolds number. From the graph, it is seen that, overall heat transfer coefficient gets increased from 220 to 250 W/m^2-K with increase of Reynolds number. Overall heat transfer coefficient is also dependant on convective heat transfer coefficient so increase in Reynolds number results into higher heat transfer rates.

The fig-7 shows variation of convective heat transfer coefficient with respect to mass flow rate. Increase in mass flow rate results into increase in flow velocity of fluid, so Reynolds number increases which ultimately increases heat transfer rate. The mass flow rate of oil increase from the heat transfer co-efficient of hot

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side also increase from 240 to 280 W/m^2-K by keeping the mass flow rate of water constant.

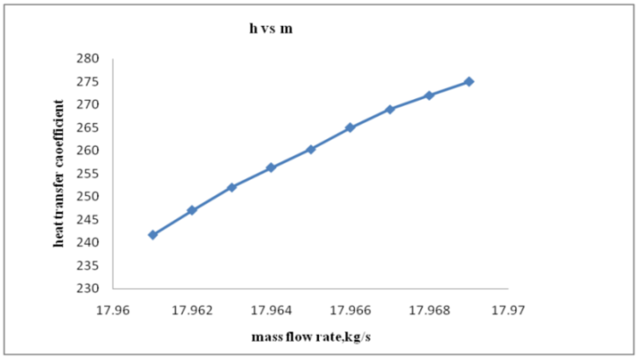

Fig-7: Effect of variation of heat transfer coefficient to mass flow rate

5. Conclusions

- A detailed procedure was given for estimating the effectiveness of Plate heat exchanger. Maximum effectiveness achieved with counter flow arrangement is 0.949.
- Varying hot fluid (oil) mass flow rate from 17.96to 17.98 kg/s and cold fluid (water) rate is almost constant at 22.8 kg/s, with inlet temperatures taken on hot side and cold side, the thermal performance of the exchanger was analyzed and results obtained are with increase in Reynolds number from (20 to 60) on hot side, Nusselt number increased by 10.01%, friction factor on air side was decreased by 25.7%, overall heat transfer coefficient increases by 10.44% and effectiveness of the exchanger was increased by 12.53%.With increase in mass flow rate of air from 17.96 to 17.98 kg/s heat transfer coefficient is increased by 17.85%.

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