A METHOD TO REDUCE THE COST OF PLATE HEAT EXCHANGER WITHOUT AFFECTING HEAT TRANSFER PERFORMANCE

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ABSTRACT
With the aim of improving heat exchanger compactness, this study investigates how the optimum configuration of plate heat exchanger can be carried out with reduction of cost of Plate heat exchanger in laminar flow. Square root of the flow depth reduction is possible proportionally to variables like plate thickness, the plate pitch, the fin thickness, and the fin pitch. The fin frequency also affects the heat transfer rate and pressure drop characteristic.

KEY WORDS: heat transfer, fin frequency, sensitivity analysis, plate heat exchanger cost.

INTRODUCTION
A plate heat exchanger is preferred in many applications because it offers a large heat transfer area and is highly effective with a comparatively small volume. Compact heat exchanger that is widely used in many industrial power generation plants, chemical, petrochemical, and petroleum industries due to their low weight, compactness and high effectiveness they weigh 95% less than comparable conventional shell and tube exchangers and provide 300-400 square feet of heat transfer per cubic foot of exchanger volume (1000-1500 m²/m³).

Although the construction cost of PFHEs due to the higher required details for manufacturing process is remarkably more than the conventional shell and tube heat exchangers, the rate of heat transfer which they can provide can reasonably justify using this type of heat exchangers.

Fins or extended surface elements are introduced to increase the heat transfer area the optimization of a plate heat exchanger occurs when the optimum dimensions of the fin pitch, the plate pitch, the fin thickness, and the plate thickness are found. This optimization has been performed based on the analysis of the effects of each design parameter. There are also other parameters that can be varied like cold stream flow length, no-flow length and hot stream flow length. Fins or extended surface elements are introduced to increase the heat transfer area some of commonly used fins are wavy, offset strip, louver, perforated, and triangle.

Gwi-Eun Song, Joohyun Lee, Dae-Young Lee found that after performing the experiment, if the geometric parameters, such as the plate thickness, the plate pitch, the fin thickness, and the fin pitch, are reduced proportionally to the square root of the flow depth reduction given that the flow remains laminar. They changed the size of plate heat exchanger without affecting the performance of PHE.

Gholap and Khan also studied air cooled heat exchangers by minimizing the energy consumption of fans and material cost as two objective functions. Some works were focused on the second law based modeling and optimization of heat exchangers. They derived the number of entropy generation units and exergy destruction or exergy efficiency as objective functions.

The temperature difference between the inlet and outlet fluids is considerably large as the result of the high compactness (usually 700-2500m2/m3) of PFHE. Some thermo physical properties of the fluids vary drastically with the changes.

REDUCTION IN HEAT TRANSFER WITH CONTINUOUS DIMENSIONS OF FIN AND PLATE
Including the transitional region between the developing and the fully developed regions, the Usagi–Churchill-type approximation is known to be adequately applicable:
\[
(Re)_{\text{trans}} = 3.2 \left( \frac{L}{D_e} \right)^{-0.57} (Re)_{\text{trans}}^{\text{app}} = \left[ (Re)_{\text{trans}}^{2} + (Re)_{\text{trans}}^{3} \right]^{1/2}
\]
\[
(Nu)_{\text{trans}} = 2.22 \frac{L}{D_e} (Re)^{-0.33}
\]
If the Reynolds number, which is defined, is larger than 2300, the flow is turbulent, and the following correlations are used:
For a turbulent, fully developed region
\[
\begin{align*}
(Re)_{\text{f,fd}} & = 0.079 Re^{0.25} \quad 10^{6} < Re < 4 \times 10^{6} \\
(Re)_{\text{f,fd}} & = 0.049 Re^{0.30} \quad 4 \times 10^{6} < Re < 6 \times 10^{6}
\end{align*}
\]
\[
(Nu)_{\text{f,fd}} = \frac{(Re)_{\text{f,fd}}}{1000 Pr \left( \frac{D_e}{L} \right)^{2} \left[ 1 + 12.7 \sqrt{Pr} \left( \frac{D_e}{L} \right)^{1/2} \right]}
\]
2300 < Re < 5 \times 10^{6}, \quad 0.5 < Pr < 2000.
For a turbulent, developing region
\[
\begin{align*}
(Re)_{\text{f,de}} & = 1 + \frac{1}{U} \left( \frac{L}{D_e} \right)^{2} \\
(Nu)_{\text{f,de}} & = 1 + \frac{1}{1.4} \left( \frac{L}{D_e} \right)^{2}
\end{align*}
\]
Because it is difficult to find accurate correlations for the friction factor in the transition region with a Reynolds number ranging from 2300 to 10,000, relations (17) are used with extrapolation as an approximation.

**Similarity in optimum geometries**
The non-dimensionalized air velocity is defined as follows:
\[
U = \frac{u}{\sqrt{\Delta p / \rho}}
\]
Using the above dimensionless velocity, the friction factor and the NTU can be rewritten as
\[
\begin{align*}
(Re)_{\text{f,de}} & = \frac{1}{2} U \\
(Nu)_{\text{f,de}} & = \frac{4 \rho \nu}{\Delta p U}
\end{align*}
\]
In these equations, C is a geometry-dependent, dimensionless parameter defined as
\[
\Gamma = \frac{\sqrt{\mu \Delta p}}{\mu} \left( \frac{D_e}{L} \right)
\]
A similar derivation and conclusion can be obtained for developing flow regions.
\[
\begin{align*}
(Re)_{\text{f,de}} & = C_1 \left( \frac{L}{D_e} \right)^{m} \\
(Nu)_{\text{f,de}} & = C_2 \left( \frac{L}{D_e} \right)^{m}
\end{align*}
\]
If reductions of geometric parameters are carried out then the heat exchanger depth can be decreased maintaining heat transfer rate constant related to the channel cross section on the square root of the depth ratio. With some practical limitations to the thickness of material, thickness of fin and plate cannot be selected according to specific requirement. To realize the effect of an un-scaled thickness, the case of a decreased heat exchanger depth without a reduced material thickness was analyzed, and the results are plotted in Fig. 2. In this unscaled case, the depth was 18 mm as in the scaled case while the fin thickness was 0.15 mm and the plate thickness was 0.8 mm as in the baseline. The heat transfer rate shown in Fig. 2 is found to decrease. As a result, both the optimum plate pitch and fin pitch with maximum heat transfer become slightly larger. Detailed examination shows that the optimum channel configuration changes insignificantly in terms of the internal width and height of the channel. The above mentioned findings are closely related with the fact that the heat and fluid flow in the finned channels are determined by the internal dimensions of the channel, assuming that the flow is laminar and the fin effectiveness is sufficiently large. Consequently, the heat and fluid flow in channels are expected to be approximately equal regardless of the material thickness internal width and heights of the channels are equal. The heat transfer performance can remain unchanged if the geometric parameters, such as the plate thickness, the plate pitch, the fin thickness, and the fin pitch, are reduced proportionally to the square root of the flow depth reduction given that the flow remains laminar. Because the heat transfer features in the channels are similar if the internal dimensions are equal, the reduction in the total heat transfer rate in Fig. 2 is presumed to be related to the decrease in the volume fraction of the channels, i.e., the porosity of the heat exchanger. As shown in Fig. 3, the porosity in this...
un-scaled case certainly decreases compared with the scaled case due to the relatively large thickness of the plate and the fin materials.

For a more quantitative analysis, the heat transfer reduction is expressed as follows from Equation.

$$\frac{Q}{Q_0} = 1 - \exp(-\text{NTU})$$

The subscript 0 refers to the baseline case. The heat transfer ratio is determined from the ratios of the porosity, the velocity, and the NTU. The effects of the relevant ratios on the heat transfer rate ratio are examined when the heat exchanger depth is reduced from the baseline case, with a constant fin thickness of 0.15 mm and a plate thickness of 0.8 mm, while adjusting the internal width and height of the channel as decided.

In conclusion, for an un-scaled case without reduced material thicknesses, the heat transfer rate degrades as the heat exchanger decreases proportionally to the decrease in the porosity.

$$\frac{Q_{\text{rel}}}{Q_{\text{rel},0}} \approx \frac{\epsilon_{\text{eff}}}{\epsilon_{\text{eff},0}}$$

2. Sensitivity analysis of Plate heat exchanger

To have a good insight into the study, a sensitivity analysis should be performed. This analysis, which is carried out based on the change in a related parameter helps us to predict the results while some modifications are necessary in modeling and optimization. Therefore, the sensitivity analysis of the optimum design variables related to the heat exchanger geometries are presented.

The variation of optimum value of effectiveness with the total pressure drop for various values of design parameters for design point C (final optimum point) are shown in Fig.

It is observed for design point C, the increases of total heat transfer surface area due to variation of fin cores (fin pitch and fin height) causes conflicting results between objective functions. But increase of total heat transfer surface area due to variation of heat exchanger configurations (no-flow length, hot stream flow length and cold stream flow length) have no conflicting behavior and improve both pressure drop and effectiveness simultaneously.

Thermal modeling

NTU method was applied here for predicting the heat exchanger performance. The effectiveness of cross-flow heat exchanger with both fluids unmixed is proposed as

$$\epsilon = 1 - \exp\left[-\left(1 + C^*\right)\text{NTU}\right] \times \left[1 - \frac{2\text{NTU}}{C^*} \left(1 + C^*\right) \text{erfc}\left(\sqrt{\text{NTU}/C^*}\right)\right]$$

where I is the modified Bessel function. Number of transfer units (NTU) and heat capacity ratio (C*) are defined as follows

$$\text{NTU}_{\text{max}} = \frac{U_{\text{gen}}}{C_{\text{min}}}$$

$$C^* = \frac{C_{\text{gen}}}{C_{\text{max}}}$$

U is the overall heat transfer coefficient and Atom is total heat transfer surface area computed from

$$U A_{\text{tot}} = \left(\frac{1}{A_{\text{hot},c} \eta_c} + \frac{1}{A_{\text{hot},h} \eta_h} \right)$$

$$A_{\text{hot}} = (\beta V_p)_{c} + (\beta V_p)_{h}$$

In the above equations, h is the heat transfer coefficient and b is the heat transfer surface area per unit volume defined as follow for triangle fin geometry

$$\beta = \frac{2a + 4h}{a(b + b_w)}$$

where h is fin length: $\delta_w$

$$l_f = \sqrt{b^2 + \frac{a}{2}^2}$$

where b, $\delta_f$ and a are height, thickness and pitch of the fin, respectively. $V_p$ is the volume between plates for hot and cold stream sides of the heat exchanger:

$$V_{p,c} = L_c L_b b_c (N_p + 1)$$

$$V_{p,h} = L_c L_b b_h N_p$$

Where,

Figure 5 and 6. Total pressure drop vs Effectiveness [3]

Figure 7 and 8. Total pressure drop vs Effectiveness. [3]

3. Effect of fin frequency and fin layers.

Modeling of the system

The e-NTU method is utilized in order to determine the value of the total rate of heat transfer for a cross
flow PFHE. It should be noted that in this work, the heat exchanger is working under steady state condition and both fluids are assumed to be air in the ideal gas.

\[ A_h = \frac{\Delta h - T_h}{\gamma_h} (1 + n_h H_h - t_h) \]

\[ A_c = \frac{\Delta c - T_c}{\gamma_c} (1 + n_c H_c - t_c) \]

And the flow free areas can be calculated as:

\[ A_{eff,h} = (\Delta h - T_h)(1 - \gamma_h) L_c N_c \]

\[ A_{eff,c} = (\Delta c - T_c)(1 - \gamma_c) L_h N_h \]

The total heat transfer area is the sum of the heat transfer areas at both sides:

\[ A_T = A_h + A_c \]

The rate of the heat transfer can be determined as:

\[ Q = \frac{\Delta h - T_h}{\gamma_h} L_c N_c \]

Here effectiveness, total heat transfer and fins characteristics are considered. After increasing the length of fin and reduction of heat transfer area what type of effect occurs are considered and the different equations and graphics are plotted as per the suitability of fins characteristics.

Where the effectiveness (\( \epsilon \)) can be found as

\[ \epsilon = 1 - \exp \left( \frac{1}{Cr} \right) N_{Tu}^{0.23 - \exp \left( - Cr \right) NTU^{0.78}} - 1 \]

Where \( Cr = \frac{C_{min}}{C_{max}} \) and the value of NTU calculated as follows:

\[ \frac{1}{NTU} = \frac{C_{min}}{UA} - \frac{1}{(h/a)_h} + \frac{1}{(h/a)_c} \]

By substituting \( h \), the heat transfer coefficient, the above equation can be expressed as:

\[ \frac{1}{NTU} = C_{min} \left[ \frac{1}{jC_{Pr}^{1/3} R_h^{1/3} \frac{Ag h}{A_h}} + \frac{1}{jC_{Pr}^{1/3} R_t^{1/3} \frac{Ag c}{A_c}} \right] \]

Where \( j \) is the thermal performance of the surface of the PFHE and the empirical correlation for this term can be given as

\[ j = 0.53(Re)^{-0.5} \left( 1/D_h \right)^{0.15} \left( s/H + t \right)^{-0.14} \text{ (for } Re \leq 1500) \]

\[ j = 0.21(Re)^{-0.4} \left( 1/D_h \right)^{0.24} \left( s/H + t \right)^{0.02} \text{ (for } Re > 1500) \]

Where \( D_h \), the hydraulic diameter and the Reynolds number can be calculated as follow:

\[ D_h = \frac{2(s - t)(H - t)}{(s + (H - t))(H - t)} \]

\[ Re = \frac{G D_h}{\mu} = \frac{m H_h}{\Delta f h \mu} \]

\[ s = (1/n - t) \]

CONCLUSION

Sensitivity analysis shows the increases of heat transfer surface area necessarily do not increases the pressure drop because it is case sensitive. It represents that whenever manufacturer increases heat exchanger area for increasing heat transfer rate, the parameter of pressure drop also be considered to not allow it to exceed up to its predefined limit.

Plate heat exchanger optimized considering two objective functions including the total rate of heat transfer and the total annual cost of the system. Several geometric variables including the total length of the hot and cold side of the heat exchanger, fin height, fin frequency, lance length of the fin, fin thickness and the number of fin layers are considered as optimization parameters. Optimization of all parameters is possible if they are reduced proportionally to the square root of the flow depth reduction given that the flow remains laminar. This analysis is utilized for optimization of the system and

Figure 9. Effect of fin frequency on total rate of heat transfer

Figure 10. Effect of fin frequency on the total annual cost

The variation of the total rate of heat transfer versus the fin frequency. The increment in the fin frequency causes a reduction in the free flow area and raise in the total heat transfer area which in turn increases the value of NTU. As it shows in Fig.10

Increasing the number of fins per meter results in increasing the pressure drop and consequently the operational cost. By raising the total heat transfer area, the initial cost is also increased. These effects resulted in increment in the total annual cost, as it shows in Fig. 10.

It should be noted that the relation between the number of fin layers for the cold side and the hot side is \( N_c = Nh + 1 \). From the given equations can be found out, any increment in the number of fin layers leads to an increase in both the free flow area and the total heat transfer area. Where they have a conflicting effect on the value of NTU and consequently Q. As it is shown in Fig. 11, increasing the number of fin layers resulted in raising the total rate of heat transfer.

Figure 11. Effect of the number of fin layers on the total rate of heat transfer.
achieving set of optimal solutions each of which is a trade-off between the highest total of heat transfer and the least total annual cost. The principal advantage of this work is providing a wide range of optimal solutions which allows the user to choose the best design parameters regarding the application and the total annual cost of the system. This analysis is useful to obtain the configuration of a more compact heat exchanger from the existing configuration.

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