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Beneficial design of unbaffled shell-and-tube heat exchangers for attachment of longitudinal fins with trapezoidal profile

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ABSTRACT

A parametric variation followed with Kern's method of design of extended surface heat exchanger has been made for an unbaffled shell-and-tube heat exchanger problem. For this analysis, the rectangular and trapezoidal fin shapes longitudinally attached to the fin tubes are taken. In comparison with the attachment of trapezoidal fins, it is found that the heat transfer rate was lesser than the rectangular cross section by keeping a constant outer diameter of the shell along with all other constraints of a heat exchanger design, namely, number of passes, tube outer diameter, tube pitch layout, etc. But when the total volume of the fin over a tube was kept constraint, using trapezoidal fins the heat transfer rate is found to be increased and consequently the pressure drop decreases much more than in the case of fins with rectangular cross section. This optimization has shown beneficial results in all the cases of different constraints of heat exchanger design analysis.

1. Introduction

The shell and tube heat exchanger is suited for high pressure applications. This type of heat exchangers consists of a shell with a bundle of tubes inside it. One fluid runs through the tubes and another fluid flows over the tubes to transfer heat between the two fluids. The set of tubes is called a tube bundle, and may be composed of several types of tubes: plain, longitudinally finned, etc. From these categories, tubes with longitudinal fins have an ability to transfer more heat. Shell-and-tube heat exchangers are used in various applications for enhancing rate of heat transfer between two fluids.

The enhancement of heat transfer is critically important in industrial applications such as process cooling, refrigeration, chemical processing, air separation, etc. Fins or extended surfaces play an important role to augment the rate of heat transfer. In situations of combined conduction-convection effects, where the objective is to enhance the rate of heat transfer between a solid and an adjoining fluid, fins are commonly employed. The only way to increase the heat transfer rate in a heat exchanger with a given constant Log Mean Temperature Difference (LMTD) is made by increasing the surface area. Surface area can be increased by a number of ways, using finned surfaces being one of the oldest methods. In applications consisting of fluids (liquids, gases or halogen compounds), it can be mentioned that the heat transfer coefficient on the liquid side is much greater than that of the gas side. Fins are then used on the gas side so that heat transfer rate may be brought to same value on both sides of the boundary separating the two fluids and thus fins bring about equality in resistance to heat transfer. Broadly fins can be classified as those with constant cross section and another being those with varying cross section. It is well understood that as conductive heat transfer rate decreases along the length of the fin, taper fin is the better option for transferring heat effectively.

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Nomenclature

a_s	area of cross section in the shell side
a_t	area of cross section in the tube side
A_0	area defined in Eq. (14)
A_i	area expressed in Eq. (15)
c_s	specific heat capacity of fluid in the shell side
c_t	specific heat capacity of fluid in the tube side
d	thickness of rectangular fin
d_e	equivalent diameter for heat transfer calculations
D	inner diameter of tubes
D_1	outer diameter of tubes
D_2	inner diameter of shell
D_b	tube bundle diameter
De_s	equivalent diameter in the shell side
De_t	equivalent diameter in the tube side
f_s	friction factor in the shell side fluid flow
f_t	friction factor in the tube side fluid flow
h_f	heat transfer coefficient over the fin surface
h_{f_i}	heat transfer coefficient outside surface and fins with respect to the inner surface of tubes
h_i	heat transfer coefficient of inside tube surface
H_f	height of individual fin
G_s	mass velocity of fluid in the shell side
G_t	mass velocity of fluid in the tube side
$I_n(Z)$	modified bessel function of first kind order n and argument z
k_f	thermal conductivity of fin material
K	constant, see Table 1
$K_n(Z)$	modified bessel function of second kind order n and argument z
K_s	thermal conductivity of fluid in the shell side
K_t	thermal conductivity of fluid in the tube side
L	length of a tube
m	defined in Eq. (20)
M	constant used in Eq. (1)
n	number of tube side passes
N_f	number of fins per tube
N_T	total number of tubes
P	perimeter, see Eq. (16)
P_t	tube pitch
P_w	wetted perimeter
Pr_s	prandtl number based on shell side fluid
Pr_t	prandtl number based on tube side fluid
Q	heat transfer rate per unit lmtd
Re_s	reynolds number in shell side fluid flow
Re_t	reynolds number in tube side fluid flow
s_s	specific gravity in shell side fluid
s_t	specific gravity in tube side fluid
U	overall heat transfer coefficient
V	fin volume
w	tube side fluid mass flow rate
W	shell side fluid mass flow rate
y_b	semi-base thickness of trapezoidal fin
y_t	semi-tip thickness of trapezoidal fin
ΔP_s	pressure drop of fluid in the shell side
ΔP_t	pressure drop of fluid in the tube side

Greek letters

η_f	fin efficiency
μ_s	viscosity of shell side fluid
μ_t	viscosity of tube side fluid

The description of shell-and-tube heat exchanger with its tube either finned or bare can be found elaborately in several text books [1–6]. The design of heat exchangers is a fairly complex thing to accomplish mainly owing to the fact that there are many qualitative decisions to be taken along with the quantitative aspects. The process and problem specification is one of the major steps in heat exchanger design. Information is needed on size, weight and other constraints like mass flow rates, inlet temperatures and pressures on both streams, maximum allowed pressure drops on both fluid sides, fluctuations in inlet temperature and pressure and also the environment parameters. Thermal and hydraulic design procedures, surface basic characteristics, surface geometrical properties and thermo physical properties are also taken into consideration. Mechanical design aspects and manufacturing considerations are also of importance here. Therefore, the common attempt is to create optimum design for maximum heat transfer rate with minimum space occupancy with given constraints.

During the heat transfer process in the shell and tube heat exchanger, a lot of constraints come into play in order to achieve maximum feasible rate of heat transfer for a given size of heat exchanger. It has been seen that the increase in fin height in a longitudinal fin, heat transfer area increases but at the same time, the driving force for the motion of the fluid increases. Both these two phenomena act simultaneously and counter to each other which may be a desirable condition. Hence there is an optimum dimension of the fin for a particular arrangement inside a heat exchanger with a number of variable constraints which gives best overall performance of the heat exchanger [4]. In addition, it has already been mentioned that taper fins are better with respect to heat transfer rate per unit fin volume. Due to variable cross section, the flow passage area for fluid flow increases and therefore pressure drop may be decreased in the shell side with adopting taper fins.

In the present study, design performance of longitudinal fins inside a shell and tube heat exchanger has been analyzed using Kern's method which may give easy and reasonably accurate measurement of heat transfer rate and pressure drop. It has been seen that with various constraints such as number of passes, tube outer diameter and tube pitch layout remaining constant, increase in fin height causes heat transfer rate of longitudinal fins with rectangular and trapezoidal profiles to decrease after attaining a maximum value. Upon further considerations with the fin shape, it can be mentioned that heat transfer rate per unit volume is the maximum if the profile of fins were changed to a parabolic one. Unfortunately parabolic

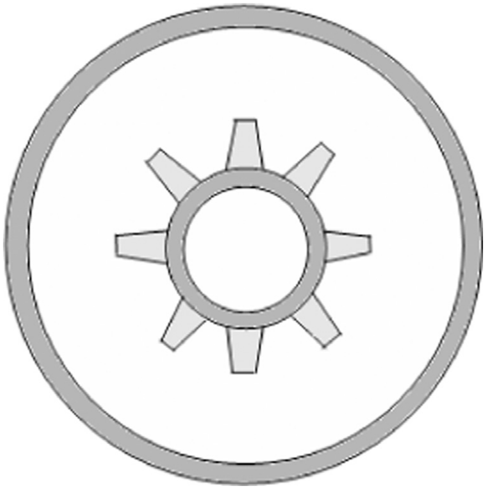

Fig. 1. Schematic diagram of a shell-and-tube heat exchanger with trapezoidal fins attached to a tube.

profile fins are difficult to manufacture. Alternatively, trapezoidal profile can be adopted in practical applications. In the present work, an extensive parametric study is performed in order to thermally characterize the two kinds of fins of an unbaffled shell-and-tube heat exchanger and to explore their superiority in conducting heat as compared to rectangular fins.

2. Analysis

Fig. 1 shows a schematic diagram of a shell and tube heat exchanger [3] consisting of a tube. Longitudinal trapezoidal fins are attached to the outer surface of a tube. An internal shell diameter D_2 having fined tubes of outer diameter D_1, inner diameter D, and length L with fins height H_f, is considered in the analysis. The bundle sheet diameter D_b is first calculated for bare tubes which can be accommodated within a given shell diameter by iterative process and then the maximum number of finned tubes can be calculated from the following relationship:

$$D_b = \left(D_1 + 2H_f\right) \times (N_T/K)^{1/M} \tag{1}$$

The constants, K and M are given in Table 1 for different tube passes and tube pitch layouts

$$P_t = 5\left(D_1 + 2H_f\right)/4 \tag{2}$$

Here it may be mentioned that two different fins shape are considered where for rectangular fin the fin thickness is d and for trapezoidal fins the base thickness is $2y_b$ and the tip thickness is $2y_t$. The tube side flow area a_t, mass velocity G_t, Reynolds number Re_t, and Prandtl number Pr_t, for tube side fluid are calculated from the following equations:

$$a_t = \left(\pi N_T \times D^2\right)/4n \tag{3}$$

$$G_t = w/a_t \tag{4}$$

$$\mathrm{Re}_t = D\left(G_t/\mu_t\right) \tag{5}$$

and

$$\mathrm{Pr}_t = c_t(\mu_t/K_t) \tag{6}$$

The shell side flow areas a_s, wetted perimeter P_w, equivalent diameter d_e, mass velocity G_s, Reynolds number Re_s, and

Table 1
Values of constants, K and M.

No. of passes	Triangular pitch					Square pitch				
	1	2	4	6	8	1	2	4	6	8
K	0.319	0.249	0.175	0.0743	0.0365	0.215	0.158	0.156	0.0402	0.0331
M	2.142	2.207	2.285	2.499	2.675	2.207	2.291	2.263	2.617	2.643

Prandtl number Pr_s are calculated from the following expressions:

$$a_s = \begin{cases} \pi\, D_2^2/4 - N_T\left(\pi\, D_1^2/4 + N_f \times d \times H_f\right) & \text{rectangular fin} \\ \pi\, D_2^2/4 - N_T\left(\pi\, D_1^2/4 + N_f \times (y_b - y_t) \times H_f\right) & \text{trapezoidal fin} \end{cases} \tag{7}$$

$$P_w = \begin{cases} N_T\left(\pi\, D_1 - N_f \times d + 2N_f \times H_f\right) & \text{rectangular fin} \\ N_T\left(\pi D_1 - N_f \times 2yb + 2N_f \times H_f \times \sqrt{1 + (y_b - y_t)^2/H_f^2}\right) & \text{trapezoidal fin} \end{cases} \tag{8}$$

$$d_e = 4a_s/P_w \tag{9}$$

$$G_s = W/a_s \tag{10}$$

$$\mathrm{Re}_s = d_e\left(G_s/\mu_s\right) \tag{11}$$

and

$$\mathrm{Pr}_s = c_s\left(\mu_s/K_s\right) \tag{12}$$

The heat transfer coefficient for the outside tube and fin surfaces can be calculated from the Sieder–Tate Correlation [6]

$$h_f = \begin{cases} 1.86(K_s/d_e) \times (\mathrm{Re}_s \times \mathrm{Pr}_s \times d_e/L)^{1/3} & \text{laminar flow} \\ 0.027(K_s/d_e) \times \mathrm{Re}_s^{0.8} \times \mathrm{Pr}_s^{1/3} \times \left(\mu_s/\mu_w\right)^{0.14} & \text{turbulent flow} \end{cases} \tag{13}$$

The outside bare tube surface area, inside surface area and the perimeter of a fin are calculated as follows:

$$A_o = \begin{cases} \left(\pi\, D_1 - N_f d\right) \times N_T L & \text{rectangular fin} \\ \left[\pi\, D_1 - N_f(y_b + y_t)\right] \times N_T L & \text{trapezoidal fin} \end{cases} \tag{14}$$

$$A_i = \pi D L N_T \tag{15}$$

and

$$P = \begin{cases} 2(L + d) & \text{rectangular fin} \\ 2\left(L + y_b + y_t\right) & \text{trapezoidal fin} \end{cases} \tag{16}$$

The fin efficiency is calculated with the assumptions that the process is in steady state and one-dimensional and there is a continuous flow of fluid in axial direction. Hence the temperature distribution in the radial direction is taken to be same. Thus it is assumed the problem to be one dimensional in character and fin efficiency can be determined from the followings [7–12]:

$$\eta_f = \begin{cases} \dfrac{\tanh\left[\left(2h_f/K_f d\right)^{1/2} H_f\right]}{\left(2h_f/K_f d\right)^{1/2} H_f} & \text{rectangular} \\ \dfrac{K L y_b (y_b - y_t)\ (m \times y_b)^{-1/2}\left[C_1 I_1\left(2m\sqrt{y_b}\right) - C_2 K_1\left(2m\sqrt{y_b}\right)\right]}{H_f L\sqrt{H_f^2 + (y_b - y_t)^2} \times h_f} & \text{trapezoidal} \end{cases} \tag{17}$$

where

$$C_1 = \frac{K_1\left(2m\sqrt{y_t}\right)}{I_0\left(2m\sqrt{y_b}\right)K_1\left(2m\sqrt{y_t}\right) + I_1\left(2m\sqrt{y_t}\right)K_0\left(2m\sqrt{y_b}\right)} \tag{18}$$

$$C_2 = \frac{I_1\left(2m\sqrt{y_t}\right)}{I_0\left(2m\sqrt{y_b}\right)K_1\left(2m\sqrt{y_t}\right) + I_1\left(2m\sqrt{y_t}\right)K_0\left(2m\sqrt{y_b}\right)} \tag{19}$$

and

$$m = \frac{h_f H_f \sqrt{H_f^2 + (y_b - y_t)^2}}{k_f (y_b - y_t)^2} \tag{20}$$

The heat transfer coefficient of outside surface and fins with respect to the inner surface of tubes h_{fi} and heat transfer coefficient of inside surface h_i are given below

$$hf_i = \begin{cases} \left(H_f \times P \times N_f \times \eta_f \times N_T + A_o\right) h_f / A_i & \text{rectangular} \\ \left(2(L + yb + yt) \times H_f \times N_f \times \eta_f \times N_T + A_o\right) h_f / A_i & \text{trapezoidal} \end{cases} \tag{21}$$

The heat transfer coefficient for the inside tube surface can be calculated from the Sieder–Tate Correlation [6]

$$h_i = \begin{cases} 1.86(K_t/D) \times (\mathrm{Re}_t \times \mathrm{Pr}_t \times D/L)^{1/3} & \text{laminar low} \\ 0.027(K_t/D) \times \mathrm{Re}_t^{0..8} \times \mathrm{Pr}_t^{1/3} \times \left(\mu_t/\mu_w\right)^{0.14} & \text{turbulent flow} \end{cases} \tag{22}$$

The overall heat transfer coefficient U with respect to the inside tube surface is given by

$$U = \left(h_{fi} \times h_i\right) / \left(h_{fi} + h_i\right) \tag{23}$$

The heat transfer rate with respect to the inside tube surface area Q per degree LMTD is calculated using the relation given below

$$Q = U \times A_i \tag{24}$$

The pressure drop for shell side and tube side fluid ΔP_s and ΔP_t respectively, are calculated using the relationship in the followings:

$$\begin{bmatrix} \Delta P_s \\ \Delta P_t \end{bmatrix} = \begin{bmatrix} \left(f_s \times G_s^2 \times L\right) / \left(5.22 \times 10^{10} \times De_s \times s_t\right) \\ \left(f_t \times G_t^2 \times L \times n\right) / \left(5.22 \times 10^{10} \times D \times s_t\right) \end{bmatrix} \tag{25}$$

where

$$f_s = \begin{cases} 16/\mathrm{Re}_s & \text{laminar flow} \\ 0.0035 + 0.24/\mathrm{Re}_s^{0.42} & \text{turbulent flow} \end{cases} \tag{26}$$

$$f_t = \begin{cases} 16/\mathrm{Re}_t & \text{laminar flow} \\ 0.0035 + 0.24/\mathrm{Re}_t^{0.42} & \text{turbulent flow} \end{cases} \tag{27}$$

Here the equivalent diameter for pressure drop calculations for both shell and tube side fluid is given by

$$\begin{bmatrix} De_s \\ De_t \end{bmatrix} = \begin{bmatrix} 4a_s/(P_w + \pi D_2) \\ 4a_t/(P_w + \pi D_2) \end{bmatrix} \tag{28}$$

and the Reynolds number is modified as

$$\begin{bmatrix} \mathrm{Re}_s \\ \mathrm{Re}_t \end{bmatrix} = \begin{bmatrix} (De_s \times G_s)/\mu_s \\ (D \times G_t)/\mu_t \end{bmatrix} \tag{29}$$

3. Results and discussion

In this paper the comparison between the heat transfer rate for two different shaped finned tubes in a shell and tube heat exchanger is made based on the above analysis. Hot oxygen gas at an average temperature of 80 °C with a flow rate of 3.8 kg/s flowing in shell side is to be cooled by the cold water at 32 °C flowing in the tube side with a flow rate of 6.4 kg/s. The number of fins per tube is kept to be 20 for the part where the volume of the fin varies with the height and it varies when the total volume of the fin over a tube is kept as a constant. The outer diameter of the tube is 0.0254 m. Thermal conductivity of the fin material is 45 $\mathrm{Wm^{-1}\,K^{-1}}$and the inner diameter and length of the shell are taken as 0.5 m and 4.88 m, respectively. The base thickness of the fin is 9×10^{-4} m.

The result is obtained followed with the Kern's method of designing of shell and tube heat exchanger. For the heat

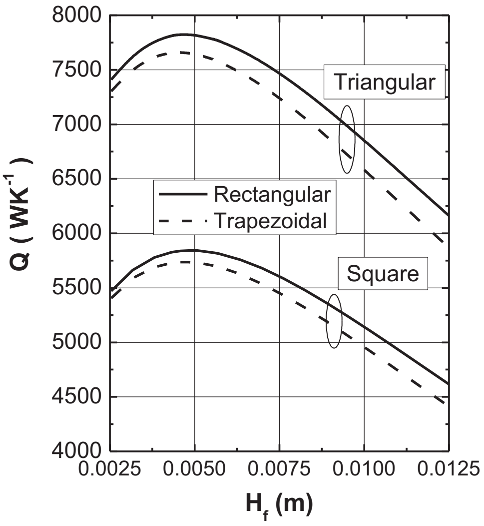

Fig. 2. Comparison of the heat transfer rate for the rectangular and the trapezoidal fin $(y_b/y_t = 5)$ for triangular pitch and square pitch arrangement and a tube side pass for a design condition.

exchanger design, heat transfer and pressure drop play an important role and thus the present investigation concentrates on the effect of the said variables. The variation of heat transfer rate and the pressure drop in the shell and tube side are compared for rectangular and trapezoidal fins for both the triangular and square pitch arrangements. The exact variation of heat transfer rate for trapezoidal fin shape of both triangular and square pitch arrangements with the fin height is depicted in Fig. 2. From this figure, it can be demonstrated that heat transfer rate increases initially with the increase in fin height, reaches to a maximum value at a particular fin height and then decreases with the further increase in fin height. Therefore, there is an optimum value of fin height for which heat transfer rate becomes a maximum. In comparison, the above variation has been studied for the rectangular fin attachment with tubes which is plotted in the same figure. For every pitch arrangement, heat transfer rate through shell-and-tube heat exchanger with consideration of rectangular fins is higher than that with trapezoidal fin. This phenomenon comes due to constraint of inner shell diameter taken and hence the volume of rectangular fin associated with the analysis differs than the trapezoidal shape. From the figure, it can also be mentioned that the heat transfer rate for triangular pitch arrangement is always higher in comparison to that of square pitch arrangement.

Fig. 3 indicates the variation of shell side pressure drop for both the triangular and square pitch arrangement of tubes. The pressure drop decreases monotonically as the height of the fin is increased. It can be explained in this way that as the fin height increases the number of tubes that can be accommodated in the fixed area of shell by keeping constant inner diameter decreases resulting in the increase in the shell side flow area leading to decrease in the pressure drop. In additional observation, pressure drop in the shell side with trapezoidal fin is less in comparison to the rectangular fin due to increase in

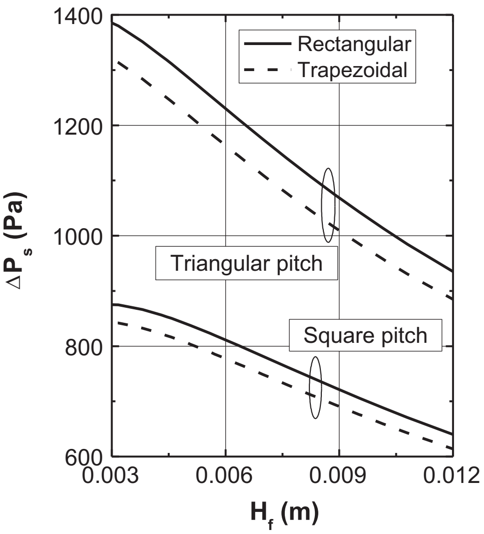

Fig. 3. Comparison of the shell side pressure drop for the rectangular and the trapezoidal fin $(y_b/y_t = 5)$ for rectangular pitch and a tube side pass.

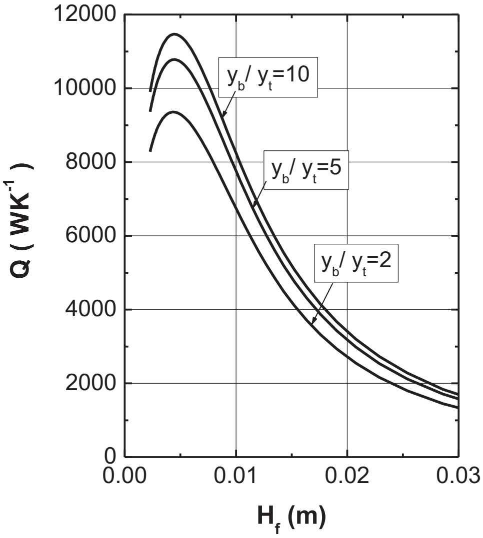

Fig. 4. Comparison of heat transfer rate for different y_b/y_t ratios of trapezoidal fin of a constant fin volume for triangular pitch and one tube side pass.

flow area. Nevertheless, the pressure drop in the shell side for square pitch of tube is less in comparison to the triangular pitch. On the contrary the tube side, it can be highlighted that an increase in fin height decreases the number of tubes and therefore the flow area in the tube side decreases which affects an increasing pressure drop.

The effect of geometric ratio $\left(y_b/y_t\right)$ of a trapezoidal fin on the heat transfer of shell-and-tube heat exchanger is exhibited in Fig. 4 with the variation of fin height. For each geometric ratio $\left(y_b/y_t\right)$ there is an optimum fin height clearly shown in this figure. With the increase in this geometric ratio, the heat transfer for an optimum condition increases more in comparison to other design conditions. Therefore, it can be indicated that the higher heat transfer can be achieved for triangular geometric fins. However from manufacturing point of view, zero tip thickness can not be made. In addition, with the small thickness near the tip, fin becomes fragile. To avoid these circumstances, trapezoidal shape is chosen in most of the applications. The trapezoidal fins have a particular ratio between the base thickness and the tip thickness selected according to the manufacturing point of view. Thus, it can be concluded that more the geometric ratio more the heat transfer rate but as the ratio increases the fin shape tends to become triangular and manufacturing of those fins are very difficult so the ratio is to be within limits from the practical point of view.

An optimization scheme has been performed with a constant fin height rather than the fin volume. The optimum heat transfer rate with a constraint fin height is depicted in Fig. 5. At a particular fin height, the optimum heat transfer rate is a maximum. Therefore, the imposition of constraint fin height does not always make guaranty to obtain the maximum heat transfer rate. Therefore, caution has to be taken for the selection of a constraint fin height along with a design condition.

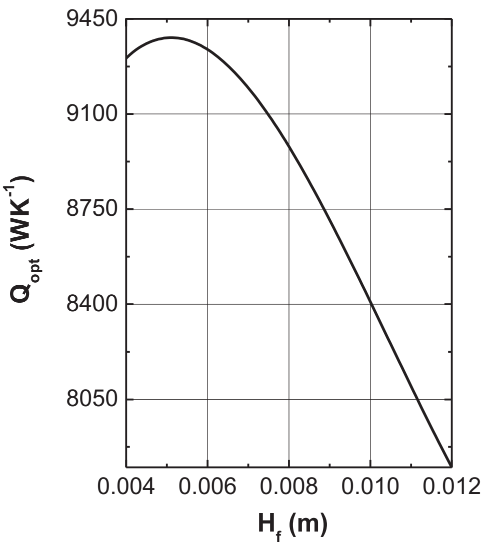

Fig. 5. The optimum heat transfer rate as a function of fin height for trapezoidal fin ($y_b/y_t=5$).

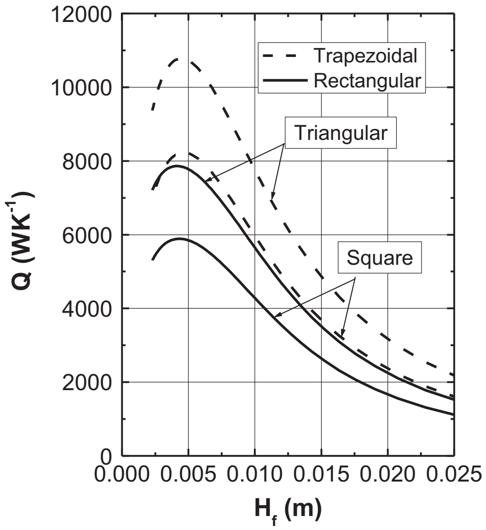

Fig. 6. Comparison of the heat transfer rate of the rectangular and the trapezoidal fin $\left(y_b/y_t=5\right)$ for the triangular pitch and square pitch.

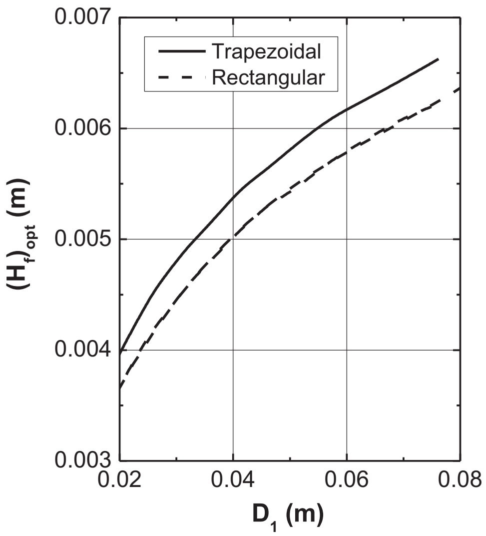

Fig. 7. Comparison of the optimum fin heights of the rectangular and the trapezoidal fin for the triangular pitch with outer diameter of tubes $\left(H_f=0.00254m;\ \ y_b/y_t=5\right)$.

Table 2
Heat transfer rate per unit LMTD for square and triangular pitch arrangement of tubes for rectangular and trapezoidal fin shapes and one tube side pass (average gas temperature=90 °C, gas flow rate=4.0 kg/s, cold water temperature=30 °C, water flow rate= 6.4 kg/s, outer diameter of the tube=0.025 m, fin material thermal conductivity =50 Wm^{-1} K^{-1}, shell inner diameter=0.5 m, shell length= 5.0 m, and fin base thickness=1×10^{-3}m).

Fin volume $V\times10^5$ m^3	0.254	0.381	0.4445	0.4826	0.5588	0.635	0.762	0.889
Q, WK^{-1} (Triangular pitch and trapezoidal fin)	7409.6	7765.6	7818.8	7822.1	7778.3	7682.4	7443.6	7142.7
Q, WK^{-1} (Triangular pitch and rectangular fin)	7303.5	7623.7	7660.1	7653.6	7591.1	7477.8	7212.6	6889.2
Q, WK^{-1} (Square pitch and trapezoidal fin)	5469.7	5778.1	5833.1	5843.0	5821.8	5757.9	5585.9	5361.8
Q, WK^{-1} (Square pitch and rectangular fin)	5406.0	5690.8	5734.0	5737.0	5702.1	5625.1	5432.7	5190.7
Total no. of tubes, N_T (Triangular pitch)	102	86	79	75	69	63	55	48
Total no. of tubes, N_T (Square pitch)	82	68	63	60	54	46	40	35

Next, an exercise has been devoted to compare heat transfer rate for two pitches of arrangement of tubes with the separate attachment of rectangular and trapezoidal fin shape on the outer surface of the tubes. For this observation, fin volume is taken as a constant. Fig. 6 is plotted for the aforementioned comparison. Form this figure, it can be undoubtedly demonstrated that the heat transfer rate with the attachment of trapezoidal fin is always higher than that of rectangular fin. This figure also shows that the heat transfer rate for triangular pitch is more compared to that for square pitch. Again for a particular fin height, it transfers maximum heat for a constant fin volume. From the design point of view, this height may be recommended to construct fin shape easily.

Fig. 7 depicts the variation of optimum fin height with the tube outer diameter D_1 for both rectangular and trapezoidal fins of constant fin volume. The optimum fin height is calculated from the principle of maximization of heat transfer rate. The optimum fin height is incremented with the increase in tube outer diameter. In other words, for a high value of tube outer diameter, fin height is required more to transfer maximum amount of rate of heat. In comparison to the rectangular fin, the optimum fin height for trapezoidal fin is shown a larger because of a constant fin volume employed.

For knowing the design parameters determined in the present study, Table 2 has been illustrated for a design condition and it shows heat transfer rate and number of tubes as a design parameter of a shell and tube heat exchanger for square and triangular pitches under a constant fin volume. From this table, it can be seen that the triangular pitch with trapezoidal fin of shell and tube heat exchangers transfers maximum heat rate compared to the square pitch with rectangular fins. It is also highlighted from the table that the trapezoidal fin has more effectiveness in heat transfer under a design constant.

4. Conclusions

In the present paper, a parametric variation followed with Kern's method of design of extended surface heat exchanger has been made for an unbaffled shell-and-tube heat exchanger. The rectangular and trapezoidal fin shapes longitudinally attached to the fin tubes, separately, are considered during this analysis. When the result has been compared with the attachment of trapezoidal fins, it has been found that the heat transfer rate was lesser than the rectangular cross section keeping the outer shell diameter is a constant along with all other constraints of a heat exchanger, namely, tube outer diameter, number of passes, tube pitch layout, etc. But when the total volume of the fin over a tube was kept constraint only, using trapezoidal fins the heat transfer rate is found to be increased in comparison to the rectangular and the pressure drop decreases much more than that in the case of fins with rectangular cross section. From the result, it can be highlighted that the heat transfer rate and pressure drop are not only dependent upon the arrangement of tube but also depends upon the geometric of the fin as well as the constraint associated with the design process. This optimization result may be helpful to a designer for selecting fin geometry in heat exchangers on the basis of constraints involved in the design process.

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