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Performance analysis of finned tube and unbaffled shell-and-tube heat exchangers

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Abstract

This work considers an optimum design problem for the different constraints involved in the designing of a shell-and-tube heat exchanger consisting of longitudinally finned tubes. A Matlab simulation has been employed using the Kern's method of design of extended surface heat exchanger to determine the behavior on varying the values of the constraints and studying the overall behavior of the heat exchanger with their variation for both cases of triangular and square pitch arrangements, along with the values of pressure drop. It was found out that an optimum fin height existed for particular values of shell and tube diameters when the heat transfer rate was the maximum. Moreover it was found out that the optimum fin height increased linearly with the increase in tube outer diameter. Further studies were also performed with the variation of other important heat exchanger design features and their effects were studied on the behavior of overall performance of the shell-and-tube heat exchanger. The results were thereby summarized which would proclaim to the best performance of the heat exchanger and therefore capable of giving a good idea to the designer about the dimensional characteristics to be used for designing of a particular shell and tube heat exchanger.

Keywords: Fin height; Heat exchanger; Heat transfer rate; Longitudinal fins; Number of tube side passes; Pressure drop; Tube pitch layout

1. Introduction

Fins have long been recognized as effective means to augment heat transfer. The literature on this subject is sizeable. Shell and tube heat exchanger with its tube either finned or bare is extensively taught in the undergraduate level. Several text and reference books deal with the problems of longitudinal finned tube in a shell and tube heat exchanger [1–4]. It is well understood that with increase in fin height of a longitudinal fin, heat transfer area increases to increase the heat transfer and at the same time the driving force decreases to decrease the heat transfer. However, one important design aspect, which probably is not discussed, is presented here. For a particular shell diameter, capacity of tube numbers is decided depending on tube size and pitch arrangement. In case of finned tube, height of the fin also plays an important role. Therefore, with increase in fin height though surface area increases but number of tubes as well as efficiency of the fin decreases. So, there might be an optimum condition of tube number and fin height for a particular tube arrangement and a particular shell diameter for which the heat transfer rate is the maximum [5]. A Matlab coding has been designed to study the behavior of the overall performance of a heat exchanger on varying the important design features involved in it. The important constraints involved in the designing of a heat exchanger are studied here using the Matlab program. Several results and optimum conditions related to them are briefed out and tabulated in this literature to give a basic idea to the designer about the requirements and limitations to be included while designing a finned tube and unbaffled shell-and-tube heat exchanger.

In the present article, Kern's method of design [2] of extended surface heat exchanger is applied for a shell-and-tube heat exchanger problem. Optimum conditions of fin height and number of tubes in cases of triangular pitch and square pitch arrangements are found out along with the values of pressure drop. Other results concerning the various constraints of a heat exchanger like number of passes, tube outer diameter and tube pitch layout were also studied and compared in this literature.

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Nomenclature

a_s, a_t	fluid flow area	m^2
A_o, A_i	tube surface area	m^2
c_s, c_t	specific heat capacity	$J\,kg^{-1}\,K^{-1}$
d	thickness of each fin	m
d_e	equivalent diameter for heat transfer calculations	m
D, D_1	inner and outer diameter of tube	m
D_2	inner diameter of shell	m
D_b	tube bundle diameter	m
De_s	equivalent diameter for pressure drop calculations	m
f_s, f_t	friction factor of fluid	
h_f	heat transfer coefficient of fins	$W\,m^{-2}\,K^{-1}$
h_{fi}	heat transfer coefficient of outside tube surface and fins with respect to the inner tube surface	$W\,m^{-2}\,K^{-1}$
h_i	heat transfer coefficient of inside tube surface	$W\,m^{-2}\,K^{-1}$
H_f	height of each fin	m
G_s, G_t	mass velocity of fluid	$kg\,m^{-2}\,s^{-1}$
K_s, K_t	thermal conductivity	$W\,m^{-1}\,K^{-1}$
L	length of each tube	m
n	number of tube side passes	
N_f	number of fins per tube	
N_T	total number of tubes	
P_t	tube pitch	m
P_w	wetted perimeter	m
Pr_s, Pr_t	Prandtl number	
Q	overall heat transfer rate per unit LMTD	$W\,K^{-1}$
Re_s, Re_t	Reynolds number	
s_s, s_t	specific gravity of fluid	
U	overall heat transfer coefficient	$W\,m^{-2}\,K^{-1}$
w	tube side fluid's mass flow rate	$kg\,s^{-1}$
W	shell side fluid's mass flow rate	$kg\,s^{-1}$
ΔP_s, ΔP_t	pressure drop	Pa

Greek symbols

μ_s, μ_t, μ_w	viscosity	Pa s
η_f	fin efficiency	

Subscripts

f	fin
i	inside of tube
o	outside of tube
s	shell side
t	tube side
w	wall

2. The mathematical program model of Kern's method and solution procedure: determination of tube bundle diameter and maximum number of tubes

A shell-and-tube heat exchanger with an internal shell diameter, D_2, consisting of finned tubes of outer diameter, D_1, inner diameter, D, length, L, with fins of height, H_f, and thickness, d, is considered here. Total number of fins per tube is N_f and total number of tubes is N_T. Tube bundle diameter is D_b:

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

The constants, K and M, are determined from Table 1 for different tube passes and tube pitch layouts for a tube pitch,

$$P_t = 1.25(D_1 + 2H_f) \quad [1] \tag{2}$$

Tube bundle diameter is first calculated by iterative process for bare tubes; henceforth maximum number of finned tubes, N_T, is calculated from the derived tube bundle diameter from Eq. (1).

3. Shell side calculations

The flow area, a_s, wetted perimeter, P_w, equivalent diameter, d_e, mass velocity, G_s, Reynold's number, Re_s and Prandtl number, Pr_s are calculated using Eqs. (3)–(8) as follows:

$$a_s = \left(\pi D_2^2/4\right) - N_T\left(\pi D_1^2/4 + N_f \times d \times H_f\right) \tag{3}$$

$$P_w = N_T(\pi D_1 - N_f \times d + 2N_f \times H_f) \tag{4}$$

$$d_e = 4a_s/P_w \tag{5}$$

$$G_s = W/a_s \tag{6}$$

$$Re_s = d_e \times G_s/\mu_s \tag{7}$$

$$Pr_s = c_s \times \mu_s/K_s \tag{8}$$

The heat transfer coefficient for the outside tube and fin surfaces can be calculated using Sieder–Tate correlation [4], Eqs. (9) and (10) as shown below:

$$h_f = 1.86(K_s/d_e) \times (Re_s \times Pr_s \times d_e/L)^{1/3} \tag{9}$$

for laminar flow;

Table 1
Values of constants, K and M [1]

	Triangular pitch					Square pitch				
No. of passes	1	2	4	6	8	1	2	4	6	8
K	0.319	0.249	0.175	0.0743	0.0365	0.215	0.156	0.158	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

$$h_f = 0.027(K_s/d_e) \times Re_s^{0.8} \times Pr_s^{1/3} \times (\mu_s/\mu_w)^{0.14} \quad (10)$$

for turbulent flow.

4. Tube side calculations

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 Eqs. (11)–(14). With the values of viscosity, μ_t, specific heat capacity, c_t and thermal conductivity, K_t, for tube side fluid and using the Sieder–Tate correlation, the heat transfer coefficient of inside tube surface, h_i, can be calculated.

$$a_t = (\pi N_T \times D^2)/4n \quad (11)$$

$$G_t = w/a_t \quad (12)$$

$$Re_t = DG_t/\mu_t \quad (13)$$

$$Pr_t = c_t \times \mu_t/K_t \quad (14)$$

5. Fin efficiency calculations

The process is assumed as a steady state one and there is a continuous flow of fluid in the axial direction (both in the shell and tube side). Therefore, for a particular value of radial location, the temperature for any location in the axial direction would be almost same. Further, the angular directional variation of temperature is also neglected. Thus, the problem is reduced to a one-dimensional heat conduction problem. Hence, the fin efficiency is represented as η_f and calculated using Eqs. (15) and (16).

$$\eta_f = \tanh(mH_f)/(mH_f) \quad (15)$$

where

$$m = (2h_f/K_f d)^{1/2} \quad (16)$$

6. Heat transfer calculations

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 as below using Eqs. (17), (21) and (22):

$$h_{fi} = (H_f \times P \times N_f \times \eta_f \times N_T + A_o)h_f/A_i \quad (17)$$

A_o and A_i are the outside bare tube surface area and inside surface area of tubes respectively, where P is the perimeter of a fin, as given by Eqs. (18)–(20).

$$A_o = (\pi D_1 - N_f d) \times N_T L \quad (18)$$

$$A_i = \pi D N_T L \quad (19)$$

$$P = 2(L + d) \quad (20)$$

The heat transfer coefficient for the inside tube surface can be calculated using Sieder–Tate correlation [4], Eqs. (21) and (22) as shown below:

$$h_i = 1.86(K_t/D) \times (Re_t \times Pr_t \times D/L)^{1/3} \quad (21)$$

for laminar flow;

$$h_i = 0.027(K_t/D) \times Re_t^{0.8} \times Pr_t^{1/3} \times (\mu_t/\mu_w)^{0.14} \quad (22)$$

for turbulent flow.

Thus the overall heat transfer coefficient, U, with respect to the inside tube surface is given by Eq. (23):

$$U = (h_{fi} \times h_i)/(h_{fi} + h_i) \quad (23)$$

Finally, the heat transfer rate with respect to the inside tube surface area, Q per degree LMTD is calculated using Eq. (24) as follows:

$$Q = U \times A_i \quad (24)$$

7. Pressure drop calculations

Equivalent diameter for pressure drop calculations in case of shell side fluid will be different from the diameter used for heat transfer calculations. This diameter is given by Eq. (25):

$$De_s = 4a_s/(P_w + \pi D_2) \quad (25)$$

The pressure drops for shell side and tube side fluid, ΔP_s and ΔP_t respectively are calculated using Eqs. (26)–(29) as follows:

$$\Delta P_s = (f_s \times G_s^2 \times L)/(5.22 \times 10^{10} \times De_s \times s_s) \quad (26)$$

$$f_s = 16/Re_s \quad (27)$$

for laminar flow and

$$f_s = 0.0035 + 0.24/Re_s^{0.42} \quad (28)$$

for turbulent flow.

Here, $Re_s = (De_s \times G_s)/\mu_s$

$$\Delta P_t = (f_t \times G_t^2 \times L \times n)/(5.22 \times 10^{10} \times D \times s_t) \quad (29)$$

where f_t is the tube side friction factor and can be calculated as shown above, Eqs. (27) and (28), using tube side Reynolds number, Re_t.

s_s and s_t are the specific gravities of shell side and tube side fluids respectively [2].

8. Solution basis

An exemplary problem discussed below is used to study the objectives as discussed. Hot fluid (3.8 $\mathrm{kg\,s^{-1}}$) in shell-side is to be cooled by a cold fluid (6.4 $\mathrm{kg\,s^{-1}}$) in tube side. Inner diameter of the shell and length of the shell are kept constant as 0.5 and 4.88 m respectively. Inner and outer diameters of the tube are varied. Number of fins with thickness 9×10^{-4} m per tube is 20 and is kept constant for all the calculations. Thermal conductivity of the fin material is 45 $\mathrm{W\,m^{-1}\,K^{-1}}$. Hot and cold fluids are oxygen gas and water respectively. The values for thermal conductivity, viscosity and heat capacity of oxygen gas and water are calculated at an average temperature of 353 and 305 K respectively.

9. Results and discussions

The Kern's method of designing of shell and tube heat exchangers with extended surfaces was used for the designing of the heat exchanger concerned in this paper. The equations involved in this method are all simple and well established, and were incorporated in a Matlab program specially coded for the purpose of this paper. This program is simply a step-wise calculation and does not involve any iteration or any optimization technique that may lead to some numerical errors. However, the program was thoroughly checked and thereafter run to arrive at the reasonable conclusions as reported in the manuscript.

The results tabulated in Tables 2–5, and results shown graphically in Figs. 1 and 2 were found out for a tube outer diameter of 0.0254 m. Tables 2 and 3 present the maximum number of finned tubes of different fin heights for triangular pitch and square pitch arrangements respectively, which can be accommodated in the shell of inner diameter 0.5 m. They also reflect the obvious nature of variations of the shell-side and tube-side pressure drops with variation of fin height keeping one tube pass only. It is well understood that as the number of tubes decreases with the increase in fin height, the tube side fluid flow area is decreased thereby increasing the pressure drop. On the other hand, the shell side flow area increases leading to decrease in pressure drop, which is also shown through Figs. 1 and 2 for triangular pitch and square pitch arrangements respectively. The variations of the heat transfer rates for both the pitches with variations of fin height are reported in Tables 2 and 3 respectively. The nature of the variations is shown through Figs. 1 and 2 for triangular pitch and square pitch arrangements respectively.

It is observed from the figures that there exists an optimum fin height (0.4572×10^{-2} m for triangular pitch and 0.4826×10^{-2} m for square pitch arrangement), which gives the highest heat transfer rate. Corresponding to these optimum fin heights, optimum number of adjustable finned tubes is 78 and 60 respectively. Under these optimum conditions, heat transfer rates are 7798.4 and 5843.0 $\mathrm{W\,K^{-1}}$, tube side pressure drops are 0.2985 and 0.4723 kPa and shell side pressure drops are 1.3217 and 0.8343 kPa for triangular and square pitch arrangements respectively.

Tables 4 and 5 show the corresponding values of optimum fin height, total number of tubes, heat transfer rate and pressure drop for different values of tube side passes. We notice from these tables (Tables 4 and 5) that for a constant shell inner diameter, with increase in the number of tube-side pass the maximum heat transfer rate corresponding to the optimum value of fin height decreases. It is also noticed that as the total number of tubes decreases the tube side pressure drop values increases largely which is a major drawback from economic and

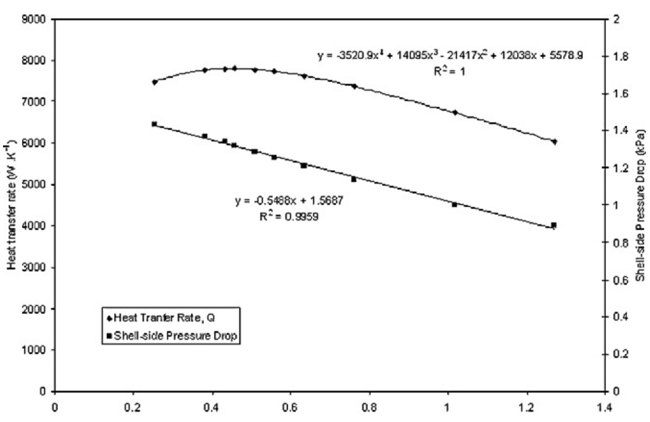

Fig. 1. Variation of heat transfer rate and shell-side pressure drop with increase in fin height for triangular pitch arrangement, one tube side pass and for tube outer diameter, 0.0254 m.

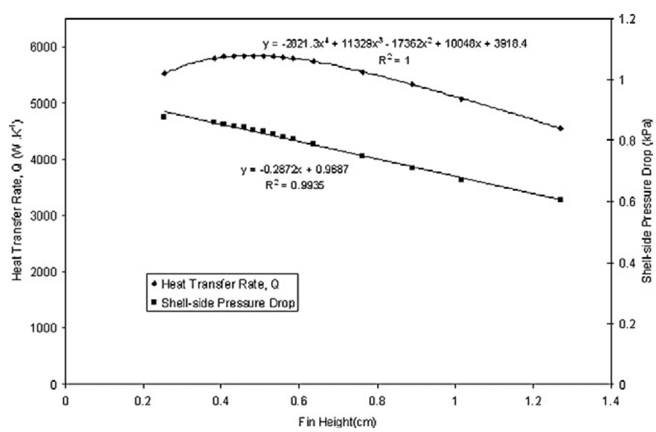

Fig. 2. Variation of heat transfer rate and shell-side pressure drop with increase in fin height for square pitch arrangement, one tube side pass and for tube outer diameter, 0.0254 m.

Table 2
Capacity of finned tubes of 0.0254 m outer diameter in the shell, pressure drops and heat transfer rate values for triangular pitch arrangement and for one tube side pass

Height of fin, $H_f \times 10^2$, m	0.254	0.381	0.4318	0.4572	0.508	0.5588	0.635	0.762
Total number of tubes, N_T	102	86	81	78	73	69	63	55
Shell side—pressure drop, ΔP_s, kPa	1.4341	1.3692	1.3383	1.3217	1.2893	1.2569	1.2093	1.1335
Tube side—pressure drop, ΔP_t, kPa	0.1882	0.253	0.2827	0.2985	0.333	0.3695	0.4295	0.546
Heat transfer rate per unit LMTD, Q, $\mathrm{W\,K^{-1}}$	7469.3	7767.1	7797.3	7798.4	7777.4	7730.5	7622.4	7371.2

Table 3
Capacity of finned tubes of 0.0254 m outer diameter in the shell, pressure drops and heat transfer rate values for square pitch arrangement and for one tube side pass

Height of fin, $H_f \times 10^2$, m	0.254	0.381	0.4064	0.4318	0.4572	0.4826	0.508	0.5334
Total number of tubes, N_T	82	68	66	64	62	60	58	50
Shell side—pressure drop, ΔP_s, kPa	0.8763	0.8605	0.8543	0.848	0.8411	0.8343	0.8274	0.8191
Tube side—pressure drop, ΔP_t, kPa	0.2765	0.3757	0.3985	0.422	0.4461	0.4723	0.4985	0.5268
Heat transfer rate per unit LMTD, Q, $\mathrm{W\,K^{-1}}$	5522.4	5797.5	5820.4	5835.1	5842.4	5843.0	5837.6	5826.8

optimization point of views. As expected, the shell side pressure drop decreases with decrease in tube number but the decrease is much less in comparison to the increase for the tube side pressure drop. So, in this case, the tube side pressure drop values bear more importance while selecting the number of passes. Hence, from the tabulated data obtained it can be said that one tube side pass is the best choice for the finest results of heat exchanger performance unless a constraint related to the number of tubes is faced when higher values of tube side pass could be considered. Moreover, it was also noticed that for a particular fin height, the total number of adjustable tubes varies for the pitch arrangements. As the number of tubes that could be adjusted in a square pitch arrangement were less in number than in triangular pitch arrangement so even the most optimum value of fin height in case of square pitch arrangement could not produce the same heat transfer rate as compared to the other. But the shell side pressure drop is higher in magnitude in triangular pitch than in square pitch arrangement, whereas the relation is just the opposite in case of tube side pressure drop values. So, in the absence of any pressure drop constraints, the triangular pitch arrangement with the optimum value of fin height will prove to be the best choice.

The other tables, i.e., Tables 6–9 give the values of different important parameters such as a_s, A_i, A_o, Re_s, Pr_s, h_f, η_f, h_{fi}, h_i and U used and determined during the calculations.

Fig. 3 shows the variation of optimum fin height with the change of tube outer diameters for a fixed number of tube side passes and for triangular pitch arrangement. The relation between them is found to be linear and can be expressed by Eq. (30):

$$H_f = 0.0852 \times D_1 + 0.0025 \tag{30}$$

Thus by using this equation, an approximate value of optimum fin height for the highest heat transfer rate can be pre-calculated for a tube of particular diameter. Fig. 4 shows a comparison of the performance of the heat exchanger for one and two tube side passes for triangular pitch arrangement. It is found out that the performance of the heat exchanger based on the heat transfer rate values for two-tube side passes could never meet up with the results for one tube side pass. Also after inspecting the pres-

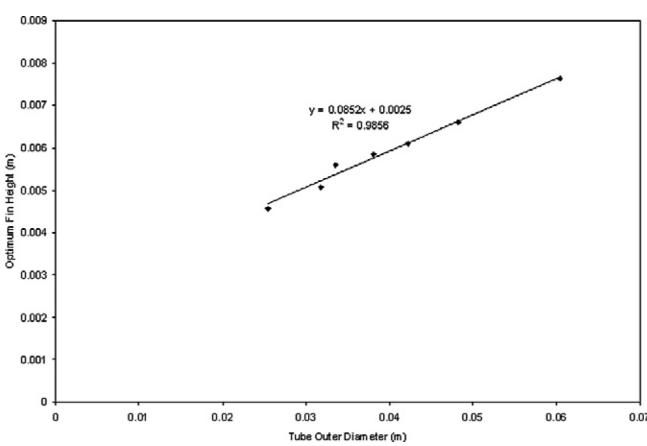

Fig. 3. Variation of optimum fin height with outer diameter of tubes.

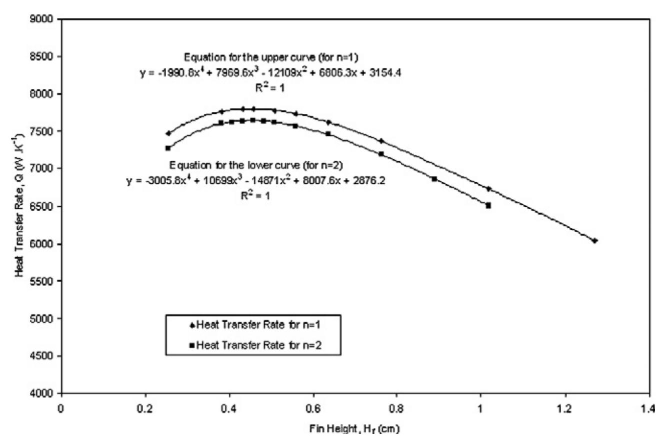

Fig. 4. Comparison of heat transfer rates with fin heights for one and two tubes side passes.

Table 4
Optimum fin height for maximum heat transfer rate, for different tube-side passes and corresponding values of total number of finned tubes and pressure drops for triangular pitch arrangement and for tube outer diameter as 0.0254 m

Number of tube side passes, n	1	2	4	6	8
Optimum fin height, $H_f \times 10^2$, m	0.4572	0.4572	0.4572	0.4318	0.38
Heat transfer rate per unit LMTD, Q, W K^{-1}	7798.4	7639.4	6523.8	4383.5	3166.4
Total number of finned tubes, N_T	78	72	62	47	40
Shell-side pressure drop, ΔP_s, kPa	1.3217	1.1204	0.8417	0.5083	0.3567
Tube-side pressure drop, ΔP_t, kPa	0.2985	2.291	20.0322	98.8108	297.322

Table 5
Optimum fin height for maximum heat transfer rate, for different tube-side passes and corresponding values of total number of finned tubes and pressure drops for square pitch arrangement and for tube outer diameter as 0.0254 m

Number of tube side passes, n	1	2	4	6	8
Optimum fin height, $H_f \times 10^2$, m	0.4826	0.4572	0.4826	0.4064	0.4064
Heat transfer rate per deg. LMTD, Q, W K^{-1}	5843	5476.7	5286.9	2911.7	2503.0
Total number of finned tubes, N_T	60	56	51	36	32
Shell-side pressure drop, ΔP_s, kPa	0.834	0.7031	0.6364	0.3252	0.2773
Tube-side pressure drop, ΔP_t, kPa	0.4723	3.5578	27.9617	160.236	441.324

Table 6
Values of various parameters involved in determining the important variables of Table 2

Height of fin, $H_f \times 10^2$, m	0.254	0.381	0.4318	0.4572	0.508	0.5588	0.635	0.762
Shell side flow area, a_s, m^2	1.41	1.485	1.511	1.523	1.546	1.567	1.596	1.637
Inside tube surface area, A_i, m^2	31.3	26.37	24.71	23.94	22.5	21.18	19.41	16.9
Outside tube surface area, A_o, m^2	31.101	26.21	24.56	23.793	22.36	21.05	19.28	16.8
Shell side Reynolds number, $Re_s \times 10^{-4}$	3.988	3.612	3.521	3.483	3.423	3.377	3.33	3.294
Shell side Prandtl number, Pr_s	0.701	0.701	0.701	0.700	0.701	0.701	0.701	0.701
Fin heat transfer coefficient, h_f, W m^{-2} K^{-1}	111.8	108.26	106.96	106.34	105.14	104.01	102.42	100.05
Fin efficiency, η_f	0.9882	0.9746	0.968	0.9645	0.9571	0.9492	0.9364	0.9133
Heat transfer coefficient of fins and outside tube surface with respect to inside tube surface, h_{fi}, W m^{-2} K^{-1}	291.03	365.342	392.96	406.34	432.25	457.06	492.27	545.82
Inside tube surface heat transfer coefficient, $h_i \times 10^{-3}$, W m^{-2} K^{-1}	1.325	1.52	1.601	1.642	1.726	1.811	1.943	2.17
Overall heat transfer coefficient with respect to inside tube surface, U, W m^{-2} K^{-1}	238.63	294.55	315.52	325.75	345.68	364.97	392.74	436.11

Table 7
Values of various parameters involved in determining the important variables of Table 3

Height of fin, $H_f \times 10^2$, m	0.254	0.381	0.4064	0.4318	0.4572	0.4826	0.508	0.5334
Shell side flow area, a_s, m^2	1.532	1.595	1.606	1.6162	1.626	1.636	1.645	1.654
Inside tube surface area, A_i, m^2	25.03	20.98	20.283	19.62	18.98	18.38	17.81	17.26
Outside tube surface area, A_o, m^2	24.88	20.85	20.157	19.5	18.87	18.27	17.7	17.16
Shell side Reynolds number, $Re_s \times 10^{-4}$	4.987	4.54	4.484	4.434	4.392	4.355	4.323	4.296
Shell side Prandtl number, Pr_s	0.701	0.701	0.701	0.701	0.701	0.701	0.701	0.701
Fin heat transfer coefficient, h_f, W m^{-2} K^{-1}	98.37	96.32	95.91	95.5	95.09	94.7	94.3	93.92
Fin efficiency, η_f	0.9896	0.9773	0.9744	0.9713	0.9681	0.965	0.9613	0.9577
Heat transfer coefficient of fins and outside tube surface with respect to inside tube surface, h_{fi}, W m^{-2} K^{-1}	256.3	325.67	338.81	351.72	364.38	376.81	389.0	400.96
Inside tube surface heat transfer coefficient, $h_i \times 10^{-3}$, W m^{-2} K^{-1}	1.585	1.825	1.875	1.926	1.977	2.028	2.081	2.133
Overall heat transfer coefficient with respect to inside tube surface, U, W m^{-2} K^{-1}	220.62	276.36	286.96	297.4	307.67	317.78	327.73	337.52

sure drop values (Table 4), it can be well concluded that the best option would be to select a heat exchanger with one tube side pass if there is no tube number constraint involved. Hence it can be well summarized by mentioning that a combination of triangular pitch arrangement, one tube side pass and a value of fin height calculated from Eq. (30), when incorporated in the designing of a shell-and-tube heat exchanger with no baffles would certainly proclaim to give the best performance until and unless some restriction is being levied on in terms of pressure drop or number of tubes.

10. Conclusions

In this work the variation of heat transfer rate with fin height for a finned tube shell-and-tube heat exchanger was studied for two different pitch arrangements. It was found out that for par-

Table 8
Values of various parameters involved in determining the optimum fin height and other important variables of Table 4

Number of tube side passes, n	1	2	4	6	8
Optimum fin height, $H_f \times 10^2$, m	0.4572	0.4572	0.4572	0.4318	0.38
Shell side flow area, a_s, m^2	1.523	1.5619	1.6261	1.722	1.7725
Inside tube surface area, A_i, m^2	23.94	22.08	18.99	14.5	12.238
Outside tube surface area, A_o, m^2	23.793	21.94	18.875	14.41	12.163
Shell side Reynolds number, $Re_s \times 10^{-4}$	3.483	3.778	4.391	6.001	7.798
Fin heat transfer coefficient, h_f, $W\,m^{-2}\,K^{-1}$	106.34	102.04	95.1	84.37	77.781
Fin efficiency, η_f	0.9645	0.9659	0.9681	0.9746	0.9817
Heat transfer coefficient of fins and outside tube surface with respect to inside tube surface, h_{fi}, $W\,m^{-2}\,K^{-1}$	406.34	390.3	364.4	311.5	263.33
Inside tube surface heat transfer coefficient, $h_i \times 10^{-3}$, $W\,m^{-2}\,K^{-1}$	1.642	3.051	5.992	1.0285	1.4827
Overall heat transfer coefficient with respect to inside tube surface, U, $W\,m^{-2}\,K^{-1}$	325.75	346.03	343.51	302.34	258.74

Table 9
Values of various parameters involved in determining the optimum fin height and other important variables of Table 5

Number of tube side passes, n	1	2	4	6	8
Optimum fin height, $H_f \times 10^2$, m	0.4826	0.4572	0.4826	0.4064	0.4064
Shell side flow area, a_s, m^2	1.636	1.6644	1.692	1.795	1.8206
Inside tube surface area, A_i, m^2	18.38	17.15	15.71	11.037	9.788
Outside tube surface area, A_o, m^2	18.27	17.05	15.613	10.97	9.728
Shell side Reynolds number, $Re_s \times 10^{-4}$	4.355	4.826	5.097	8.24	9.291
Fin heat transfer coefficient, h_f, $W\,m^{-2}\,K^{-1}$	94.7	91.04	88.72	75.96	73.118
Fin efficiency, η_f	0.965	0.9694	0.967	0.9796	0.9803
Heat transfer coefficient of fins and outside tube surface with respect to inside tube surface, h_{fi}, $W\,m^{-2}\,K^{-1}$	376.81	349.19	353.61	269.38	259.44
Inside tube surface heat transfer coefficient, $h_i \times 10^{-3}$, $W\,m^{-2}\,K^{-1}$	2.028	3.734	6.974	1.28	1.773
Overall heat transfer coefficient with respect to inside tube surface, U, $W\,m^{-2}\,K^{-1}$	317.78	319.3	336.54	263.82	255.69

ticular shell and tube diameters an optimum value of fin height exists, which gives the highest heat transfer rate. Moreover it was also found out that on increasing the number of tube side passes while keeping the shell diameter constant, though the number of tubes could be decreased but the performance on the basis of heat transfer rate kept on decreasing and tube side pressure drop values increased substantially. The optimum fin height also increased linearly with the increase of tube outer diameter.

It is worth mentioning here that the Matlab coding designed for this problem and the results obtained on using it, might prove quite beneficial in choosing the most appropriate fin height, total number of tubes, tube dimensions, arrangements, number of tube side passes and fin dimensions for a known value of shell diameter as well as keeping the pressure drops in check. In this problem the physical properties of the fluids were assumed constant, tube and fin dimensions were assumed uniform, throughout the entire system.

It can be further stated that no experimental verification could be possible due to lack of such experimental data. However, it would be highly appreciated to carry experimental work in this regard.

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