FULL ANALYSIS OF LOW FINNED TUBE HEAT EXCHANGERS

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Abstract In this paper, first the governing parameters characterizing low-finned tubes are reviewed. Second, the more important of the available performance correlations are compared with the available experimental data. The most reliable one can be employed to develop a pressure drop relationship, which has already been used in an algorithm for exchanger sizing. Also a means for the identification of advantages of low-finned tube heat exchangers over plain tube units has been developed. It has been recognized that for low-finned tube units there are some potential benefits to place certain liquids, particularly with high viscosities, in the shell side of heat exchangers rather than the tube side. These benefits can be obtained in both reduction of surface area and the number of shells required for a given duty. They result in heat exchangers, which are more compact and are also easier to construct. The performance evaluation of low-finned units, in terms of area benefits is not discussed in this paper. However, the results of this study will complete the author's investigation for low-finned tubes heat exchangers.

Key Words Heat Exchanger, Low-Finned, Performance Correlations, Heat Transfer, Shell and Tube, Heat Transfer Coefficient, Friction Factor

چکیده در این مقاله ابتدا پارامترهای حاکم و مربوط به مشخصه لوله های با پره های کوتاه مرور شده است. سپس مهمترین معادلات عملکردی تصحیح شده موجود با داده های آزمایشگاهی مقایسه شده اند. در ادامه مطمئن ترین معادلات به منظور توسعه روابط افت فشار در الگوریتم طراحی سریع مبدلهای حرارتی به کار گرفته شده اند. همچنین یک ابزار به منظور مقایسه میان طراحی مبدلهایی با لوله های پره دار کوتاه و مبدلهایی با لوله های صاف توسعه داده شده است. با استفاده از روابط توسعه داده شده همچنین تشخیص داده می‌شود که در مبدلهایی با لوله های پره دار کوتاه و در شرایطی که با سیالات با ویسکوزیته بالا کار می‌نمایند، مزایای بکارگیری سیال ویسکوز در طرف پوسته مبدل به جای لوله ها پتانسیل بالایی را از خود نشان می‌دهد. این مزایا صرفاً منجر به کاهش میزان سطح انتقال حرارت لازم (به مفهوم کوچکتر شدن مبدل) نمی‌باشد، بلکه به همراه آن کاهش تعداد پوسته ها و نهایتاً سادگی مبدل را در ساخت نیز در بر خواهد داشت. اگر چه ارزیابی بکار گیری مبدلهایی با پره های کوتاه بر سودمندی حاصل از کاهش سطح انتقال حرارت در این مقاله مورد بررسی قرار نگرفته است؛ ولی نتایج این مطالعه، بررسی های قبلی محققین این مقاله را در مورد مبدلهای حرارتی با پره های کوتاه تکمیل می‌نماید.

1. INTRODUCTION

The flow of viscous liquids through heat exchangers often results in very low heat transfer coefficients, particularly if these liquids are routed through the tube-side. As explained by Polley et al. [1] one way of enhancing heat transfer and consequently reducing exchanger size to pass the fluid through the tube side of exchangers fitted with appropriate type of tube inserts. An alternative means of reducing the size of a unit is the use of low–finned tubes to augment the shell side surface area. Low finned tubes might be significantly more expensive than the plain tube. Therefore, their use has to be justified by a significant exchanger size reduction and identification of other potential benefits in simplifying fabrication of a heat exchanger unit.

The main idea of passing viscous liquids to the shell side of a heat exchanger unit can be explained as follow. The heat transfer and pressure loss of a fluid flowing inside tubes can be estimated in a straightforward manner either by available nomographs or empirical correlations. However, the situation for the cross flow region on the shell-side is much more complex, especially if it is necessary to work with a viscous fluid (such as a lubricating oil for which the Reynolds number falls well below 2000). Fortunately, under cross flow conditions, the turbulent mixing induced by the irregular geometry is still substantial even at Reynolds numbers of 100 or less. Furthermore, the flow path length over the surface of a tube in cross flow is relatively small as compared to the effective hydraulic radius for the shell-side fluid. Therefore, as explained by investigators such as Fraas A.P. and Ozisik M.N. [2], comparing the heat transfer coefficients for shellside with those for flow through the inside of round tubes, it is apparent that it is advantageous to place the more viscous fluid, tending to give the lower Reynolds number, on the shell-side and the less viscous fluid, giving the higher Reynolds number, on the tube-side. In this way advantage can be taken of the higher heat transfer coefficients at lower Reynolds numbers given by cross flow conditions, and the two heat transfer coefficients can be made more nearly the same to give a well proportioned heat exchanger. Using low-finned tubes to enhance the heat transfer in cross-flow and to provide a greater surface area per unit volume than plain tubes further reduces the size of the unit.

Size reduction is not always the sole reason why the use of low–finned tubes results in more cost-effective heat exchanger designs. For instance, when highly viscous fluids such as lube oil or crude oil enters the shell of a heat exchanger with a plain tube bundle they need a high surface area due to the poor heat transfer coefficient in the laminar flow region. There is evidence that integral low-finned tubes foul at a lower rate than plain tube (Katz et al. [3], Mcluer H.K. and Knudsen J.G. [4]). They are also easier to clean for most types of fouling. For deposits which follow the contour of the fin, tube fin temperature differences cause expansions and contractions, which can loosen and in some cases shed these deposits.

Finned-tubes are now widely used in industrial shell and tube units used as boiler economizers, in water heaters and in air cooler heat exchangers. The application of finned tubes is not limited to single phase flow. They can also be used in condensation and boiling applications. In all application their use provides heat exchangers, which are more compact than a plain tube. The recent study of authors, Jafari Nasr M.R. and Polley G.T. [5], showed that mechanical constraints play a significant role in the design of shell and tube heat exchangers. For example, it is normal practice in exchanger design to restrict the length of the tubes to less than 6 meters. This restriction can lean to the use of multi-passing of tubes, which usually introduces the need to examine temperature cross considerations, a reduction in the effective mean temperature difference and therefore a need for increase in surface area. If the use of a single shell results in too low a value of F_t (temperature correction factor) the designer should move to a multiple shell in design.

The potential benefits of low finned tubes than plain tube units has already been investigated by Jafari Nasr M.R. and Polley M.R. [6]. The results clearly showed that using low finned tube units could provide a significant area reduction for a given duty. However, in this paper, the study is concerned with application of units of multiple shell arrangement with consideration of full use of allowable pressure drops, and the mechanical restriction for tube length.

In order to demonstrate the effect of low–finned tubes on reduction of number of shells required, for a given duty, a full analysis of interactions of tube side and fin side has to be considered.

The governing parameters required to execute the algorithm and also to characterize low-finned tubes are introduced. Then the more important of the available performance correlations are compared with the available experimental data. The most reliable one can be used to identify the other potential benefits of low finned tube units than the plain tube.

2. GOVERNING PARAMETERS

The basic dimensionless correlating parameters used in finned tube analysis are Reynolds number, Nusselt number (or Colburn j-factor) and the friction factor. The heat transfer and pressure drop are also affected by flow conditions, fin geometries, tube layout, and the number of tube rows. Since the above parameters can be found in literature just their results have been used in this analysis.

3. THE OVERALL HEAT TRANSFER COEFFICIENT

The overall heat transfer coefficient "U" can be found once the fin side heat transfer coefficient has been obtained from the basic heat transfer equation;

$$Q = \alpha_{ft}.A_t(T_w - T_o) = \alpha_f(\varepsilon_f A_f + A_{unf}).(T_w - T_o) \tag{1}$$

Where Q is the total sensible heat transfer between finned tubes and shell-side fluid, A_t is the true heat transfer area, (ε_f A_f +A_{unf}) is total effective heat transfer area and is shell-side heat transfer coefficient. So, the relationship between heat transfer coefficients based on true heat transfer area and on effective area is (Saunders E.A.D. [7]);

$$\alpha_{ft} = (\frac{\varepsilon_f.A_f + A_{unf}}{A_t}).\alpha_f \tag{2}$$

It is also possible to define a heat transfer coefficient based on the inside surface of the tube A_i;

$$\alpha_{ft} = (\frac{\varepsilon_f.A_f + A_{unf}}{A_i}).\alpha_f \tag{3}$$

The inside tube heat transfer coefficient, the fouling resistances and the tube wall thermal resistance can be combined to yield an "opposing thermal resistances";

$$R_{OPR} = \frac{1}{\alpha_i} + R_D \tag{4}$$

Then, the overall heat transfer coefficient can be expressed (i.e. based on inside surface area);

$$\frac{1}{U_i} = R_{OPR} + (\frac{1}{\alpha_{fi}}) \tag{5}$$

Substituting of equation (3) in (5) gives;

$$\frac{1}{U_t} = R_{OPR} + \frac{A_t}{(\varepsilon_f.A_f + A_{unf})}.(\frac{1}{\alpha_f}) \tag{6}$$

4. COLBURN FACTOR (J) AND FRICTION FACTOR (F)

Because of the complexity of the flow structure, theoretical treatment of flow across a finned–tube bank is not easy. Therefore, many investigations have been carried out to determine the heat transfer coefficient and pressure drop. The most important works to date are Briggs D.E. and Young E.H. [8], Jameson S.L. [9], Schmidt Th.E. [10], Robinson K.K. and Briggs D.E. [11], Mirkovic Z. [12], Schack K. [13], the Engineering Service Data Unit (ESDU [14]), Rabas T.J. and Taborek J. [15]. A number of correlations for Nusselt number and friction factor (for both In-Line and staggered arrays) have been proposed (see Tables A1 and A2 in Appendix).

5. COMPARISON OF PREDICTIVE PERFORMANCE CORRELATIONS

As a matter of fact, it is not possible to directly compare the different correlations suggested for low-fin tubes. This was indicated by some investigators such as Webb R.L.[16]. They believe that it is difficult to recommend a single correlation. Such a recommendation would require that each correlation be compared using a wide range of standardized and acceptable data. Nevertheless, for example Webb R.L. recommended the heat transfer correlation of Briggs D.E. and Young E.H. [8] and the pressure drop correlation of Robinson K.K. and Briggs D.E. [11].

The accuracy of the Briggs and Young heat transfer correlation (standard deviation 5.1%) and the Robinson and Briggs pressure drop correlation (standard deviation 7.8%) is not confirmed by other experimental data. For example, the data of Rabas et al. [17] demonstrates significant deviation of their experiments from these correlations. These authors suggest that the Robinson and Briggs pressure drop correlation should not be used for low-finned tubes.

Based on his own experiments, Mirkovic Z. [12] derived correlations in which the tube pitch and the fin geometries were changed systematically. Zukauskas A. A. [18] also conducted experiments and established correlations based solely on his data.

Schmidt Th.E. [10] and Vampola I. [19] derived correlations using different sources of experimental data. However, Rabas et al. [17] claimed the Vampola pressure drop correlation should not be

TABLE 1. Comparison of Predictive Correlations for j-Factor (Equilateral Ideal Tube Banks).

n_f	Correlation	Re=1000	%Diff.	Re=8000	%Diff.	Re=25000	%Diff.	Re=200000	%Diff.
0.5	Briggs-Young	0.01763	+12	0.00882	+3.5	0.00604	+2	-	-
	Rabas et al.	0.01536		0.00852		0.0059		0.00285	
	ESDU	0.01802	+17	0.00966	+13	0.00686	+16	0.00368	+29
0.8	Briggs-Young	0.01535	+2.7	0.00788	+12	0.00525	+0.2	-	-
	Rabas et al.	0.01267	+5	0.00677	-4	0.00480	-8	0.00285	+3
	ESDU	0.01461	+20	0.00783	+11.2	0.00556	+6	0.00298	+8
	Exp.	0.01210		0.00704		0.00524		0.00276	
1.25	Briggs-Young	0.01334	-86	0.00659	+26	0.00451	+18	-	-
	Rabas et al.	0.09394		0.00525		0.00382		0.00285	
	ESDU	0.01164	-87	0.00624	+18	0.00443	+16	0.00237	-17

TABLE 2. Comparison of Predictive Correlations for f-Factor (Equilateral Ideal Tube Banks).

n_f	Correlation	Re=1000	%Diff.	Re=8000	%Diff.	Re=25000	%Diff.	Re=200000	%Diff.
0.5	Briggs-Young	-		-	-	-		-	-
	Rabas et al.	0.2615		0.1608		0.1233		0.07480	
	ESDU	0.2968	+13	0.1638	+2	0.1182	-4	0.0652	-14
0.8	Briggs-Young	-		-		-		-	
	Rabas et al.	0.3506	+12	0.2155	-9	0.1651	+7	0.1015	+19
	ESDU	0.3994	+27	0.2203	+11	0.1623	+6	0.0899	+5
	Exp.	0.3123		0.1976		0.1538		0.0854	
1.25	Briggs-Young	-		-		-		-	
	Rabas et al.	0.4881		0.3003		0.2301		0.1416	
	ESDU	0.5540	+13	0.3057	+2	0.2207	-0.2	0.1217	-14

used for low-finned tubes.

Webb R.L. [16] also states “all the correlations are empirical and attempt to define a power law dependence on five basic dimensionless groups involving the tube diameter, the fin parameters and the tube bundle geometry …”. Therefore, these correlations may be considered as interpolation formulas for the particular data bank rather than general correlations for a wide class of finned–tube geometries and flow conditions.

The above quotations clearly express that the evaluation of performance correlations for low-finned tubes is a controversial matter. However, the most reliable one can be determined relatively

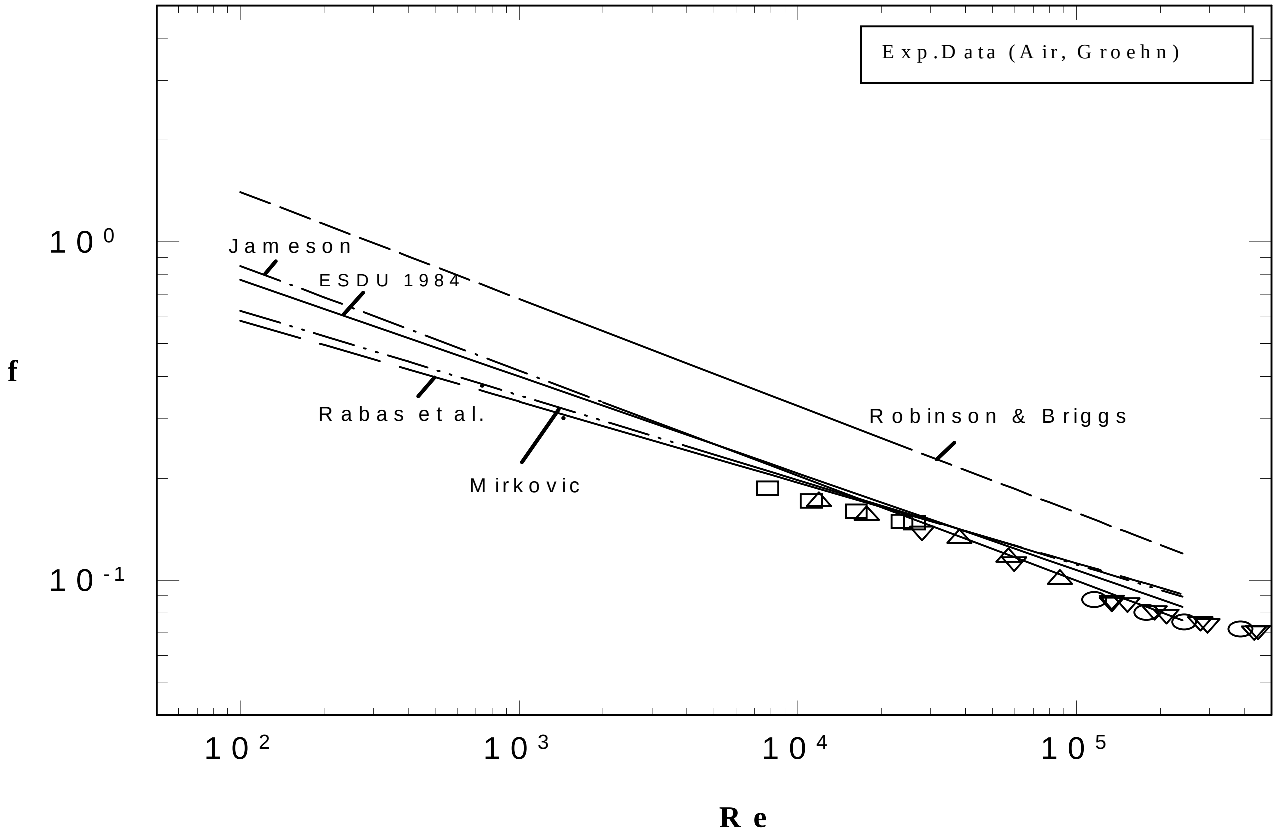

Figure 1. Comparison of friction factor correlations.

through regression analysis on available data such as evaluation of variances, measurement of standard deviations and so on.

Tables 1 and 2 present typical comparisons of some experimental data reported by Rabas T.J. and Taborek J. [15] for " j " and " f " factors respectively. The comparisons include the results of correlations of Briggs D.E–Young E.H. [8], Rabas et al. [17], the Engineering Service Data Unit (ESDU [14]) and the experimental data of Groehn H.G. [20] obtained with an integral finned tube with 0.8 fins/mm and equilateral ideal tube banks. Two additional cases were also considered: having the same geometry except with lower and larger fin densities of 0.5 and 1.25 fin/mm.

In this comparison, the percentage difference (%Diff.) between the correlations and experimental data (given by Groehn H.G. [20] for a fin density of n_f = 0.8) and Rabas et al. [1981] has been calculated.

As Tables 1 and 2 show, the correlations of Rabas et al. [17] predict the measured data well. However, they recommend their correlations be used with an additional extrapolation procedure for Re > 25000. Unfortunately, as they themselves state, the method is not well suited for calculation by hand.

The j – factor correlation of ESDU [14] for n_f = 0.8, at Re = 1000 and 200,000, overpredicts the experimental data by 20% and 8%, respectively. Whilst the friction factors correlations are overestimates by 27% and 5%

Figures 1 and 2 compare the performance of the correlations provided by Rabas et al. [17], ESDU [14], Briggs D.E–Young E.H. [8], Jameson S.L. [9], Mirkovic Z. [12] , Schmidt Th.E. [10] and Robinson K.K.–Briggs D.E.[11] with the experimental data of Groehn H.G. [20] who studied the following fin geometry;

Layout: Staggered array

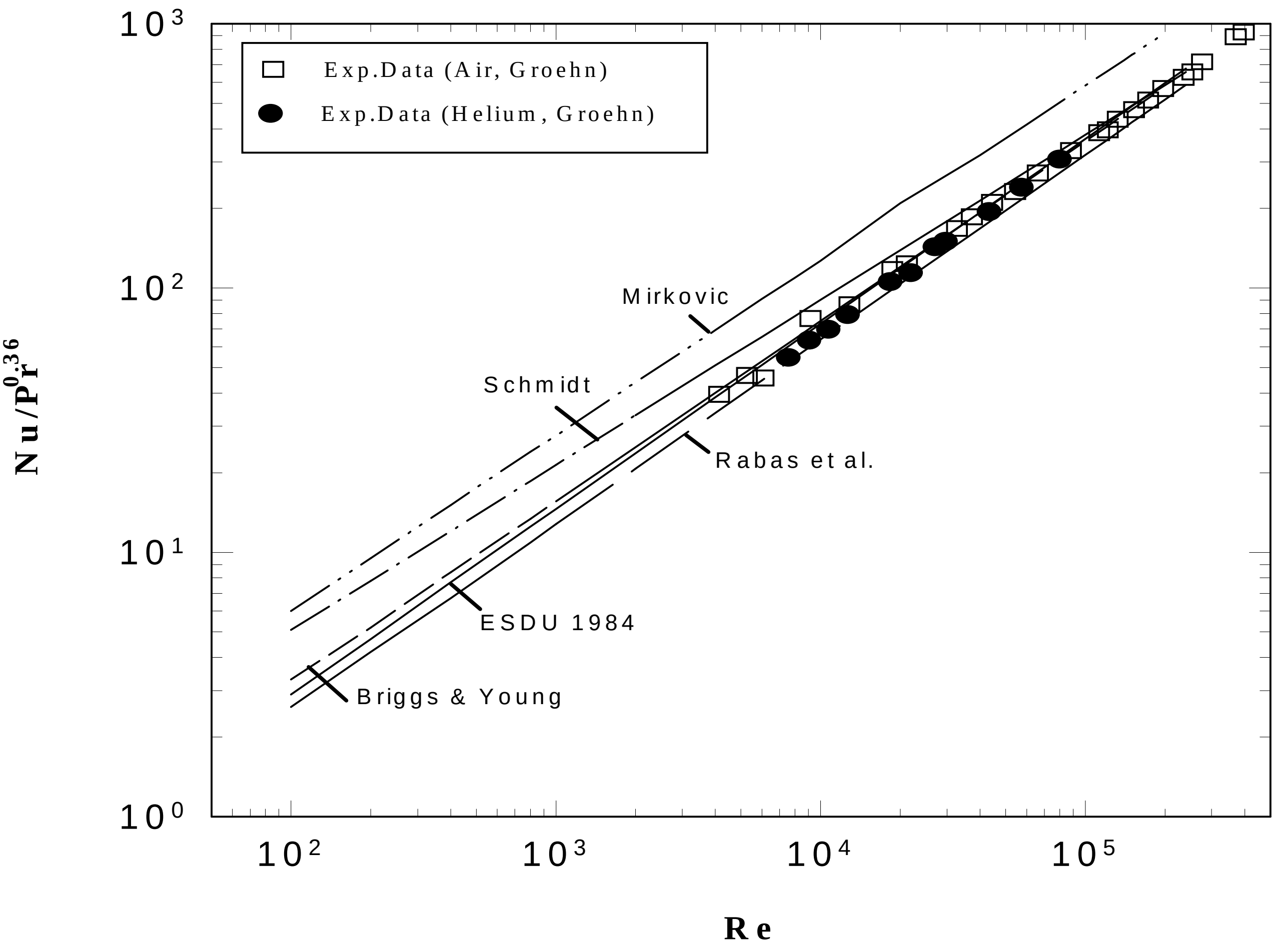

Figure 2. Comparison of heat transfer correlations.

d_r=0.02220 d_f = 0.02512
f_h = 0.00146 f_t = 0.0003
d_o=0.0254 P_f = 0.00125
P_x = 0.02685 P_Y = 0.0310

Figure 2 shows that, with the exception of the Robinson and Briggs's correlation, all of the friction factor correlations perform quit well. However, the correlation of ESDU [14] is the best.

Figure 3, indicates that the ESDU heat transfer correlation also performs well. Those of Briggs–Young and Rabas et al. [17] are also good. Those of Schmidt and Mirkovic perform poorly.

The analysis has been repeated for the above correlation and for other available experimental data given in literature. Finally, the correlations of ESDU [14] can be identified as reliable performance equations for low-finned tubes. Also, it is noteworthy that to identify the more reliable correlation a data analysis has been performed on all the correlations respect to available experimental data and for various fluids. The analysis involved the comparison of standard deviations, variances, and regression coefficients and also required statistical tests. Jafari Nasr, M.R. and Asadi, M. [21] has performed this analysis recently. Detailed results can be looked for in Reference 5.

The average heat transfer coefficient based on root fin heat transfer area of the finned tube, is therefore given by;

$$Nu = \frac{\alpha_f d_r}{k_s} = 0.183.Re^{0.7}.(\frac{f_s}{f_h})^{0.36}.(\frac{P_Y}{d_f})^{0.06}.(\frac{f_h}{d_f})^{0.11}.Pr^{0.36}.F \quad (7)$$

With the correlation covering a Reynolds number

TABLE 3. Problem Specification for Water in Shellside.

Data	Units	Shellside	Tube side
Flow rate	Kg /s	88.6	77.3
Thermal Conductivity	W/m .C	0.61	0.696
Density	Kg/m^3	995	888
Heat Capacity	J/ kg .C	4180	4312
Viscosity	Ns/m^2	0.86×10^3	0.75×10^3
Range of Allowable ΔP	KPa	10-200	10-200
Heat load	KW	6665	
Thermal Resistances R_D	$m^2.C/w$	0.0006	
Log-Mean Temperature Difference ΔT_{LM}	^oC	14.6	
Tube Geometry	Based on standard values given in ref.[16], code : 195083		

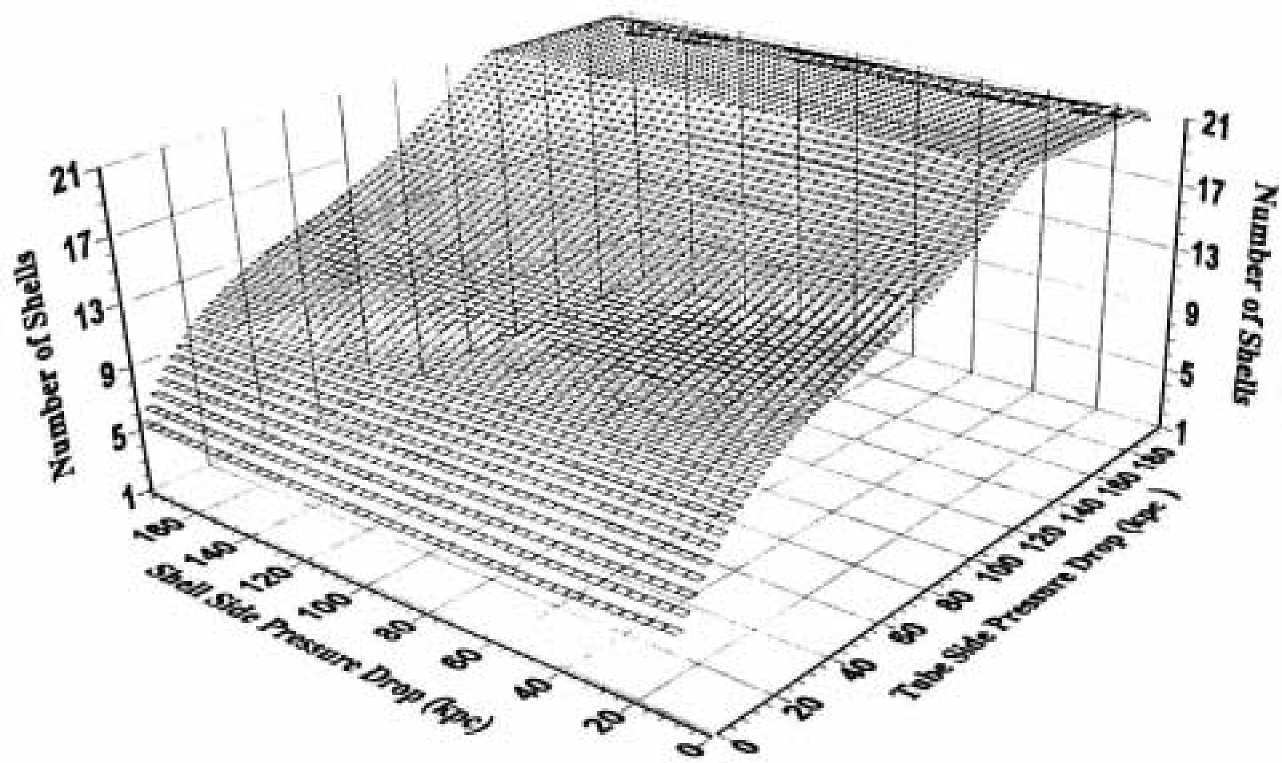

Figure 3. Number of shell for a plain tube unit with single tube pass (water).

range: $10^3<Re<8\times10^5$.

The ESDU friction factor correlation is;

$$f = 4.71Re^{-0.286}.(\frac{f_h}{f_s})^{0.51}.(\frac{P_Y - d_r}{P_X - d_r})^{0.536}.(\frac{d_r}{P_Y - d_r})^{0.36} \quad (8)$$

As mentioned these correlations were used by M.R. Jafari Nasr and G.T. Polley [5] to derive a pressure drop relationship for heat exchangers with low-finned tube bundles. Such relationships can be seen in Equations 9 and 10 for shellside and tubeside. The result of that investigation showed

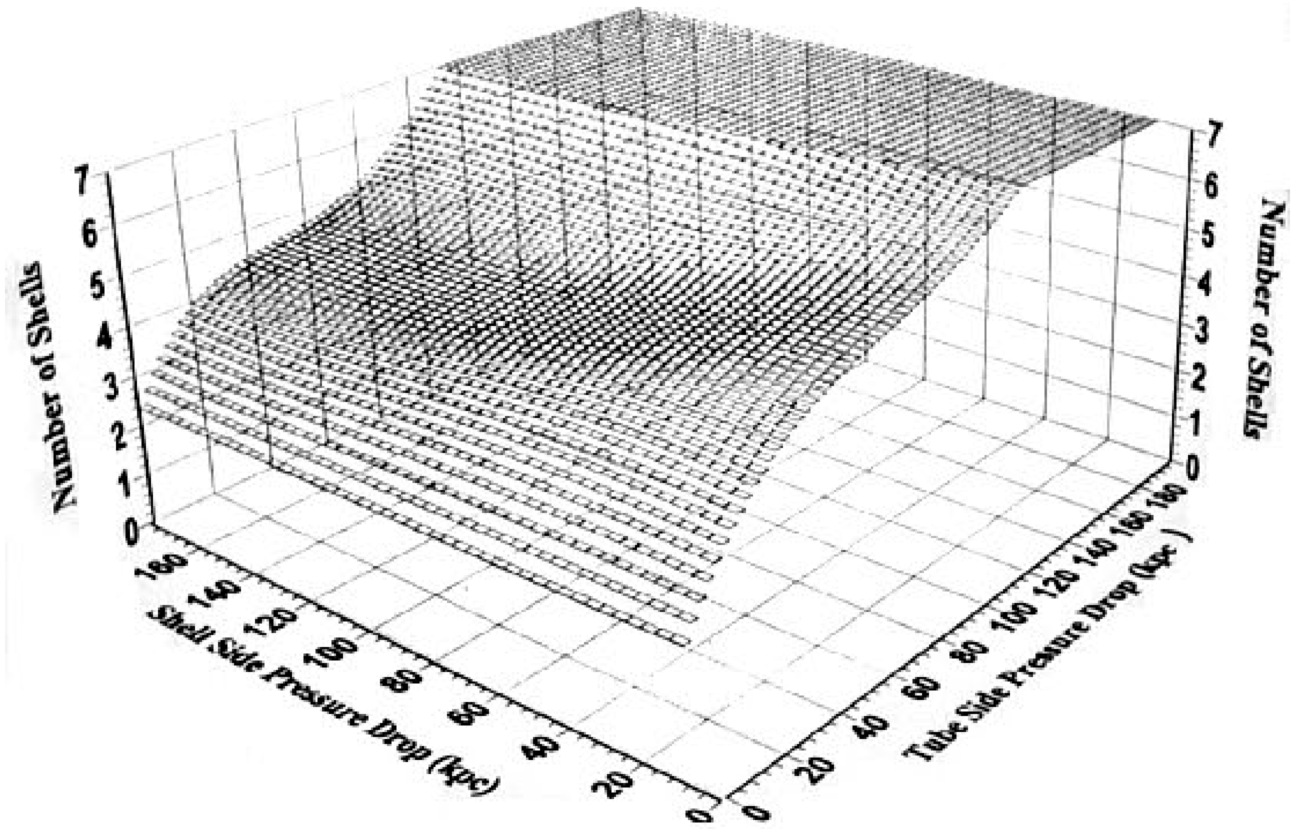

Figure 4. Number of shell for a finned tube unit with single tube pass (water).

that using low finned tubes significantly reduce the size of heat exchanger compare with plain tube units. However, here, it is shown that the potential benefit of low-finned tubes cannot only be restricted to a size reduction.

6. CALCULATION OF NUMBER OF SHELLS

There are two approaches to have heat exchangers with the higher compactness. One is utilization of techniques to enhance heat transfer coefficient and the other by increasing the amount of area per unit volume. The use of low-finned tubes is an effective means of achieving the latter. There is, therefore, a need for sizing procedures that avoid the recourse to the full detailed design. The key to the procedure is a relationship between the shellside pressure drop, shellside heat transfer coefficient and overall exchanger surface area.

Following the work of Polley G.T. and et al. [22] and the recent work of Jafari M.R. and Polley G.T. [5], there is a Possibility to develop such a relationship for finned tube units. In the cases in which the tube side heat transfer coefficient cannot be assumed and is a function of pressure drop the following three equations must be solved simultaneously;

$$\Delta P_s = kp_s.A.\alpha_s^m \qquad (9)$$

$$\Delta P_t = kp_t.A.\alpha_t^n \qquad (10)$$

$$A = \frac{Q}{F_t.\Delta T_{LM}}.(\frac{1}{U}) \qquad (11)$$

Where $U = f(\alpha_t,\alpha_s,R_D)$ and kp_t , kp_s are functions of tube geometry and physical properties of hot and cold streams and were given in reference [6]. The values of m and n are equal to 3.877 and 3.5,

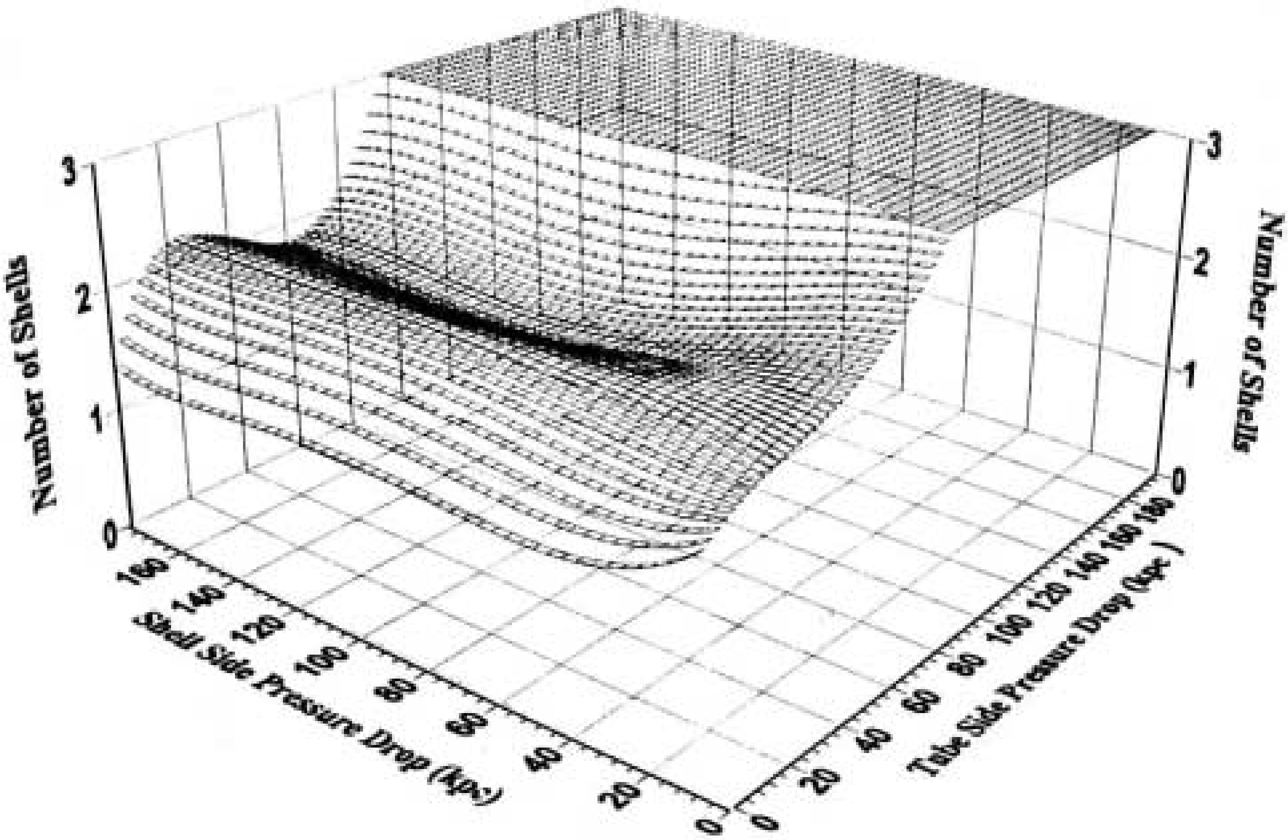

Figure 5. Number of shells for a plain tube unit with eight tubes passes (water).

respectively.

Having the values of heat transfer coefficients it becomes possible to calculate the tubeside and shellside velocities. Consequently, the number of tubes, tube length, shell diameter and the bundle geometries, such as baffle spacing and number of baffles, can be calculated. If the tube length exceeds the maximum allowable value, as mentioned earlier, a need for an additional shell in series is indicated.

Since the above comparisons are normalized base on definitions of 'j' and 'f' factors, the execution of algorithm can be extended to the other fluids. The procedure has also been examined for a number of fluids and units equipped with plain and low-finned tubes.

7. EXAMPLE APPLICATIONS

a-Water The first example examined is the processing of water on the shellside. Table 3 shows a problem specification for this example. The results are presented in Figures 3 and 4 for both plain tube and finned–tube units with a single tube pass.

The comparison clearly reveals that by using a finned–tube bundle the number of shells required for the given duty can be significantly reduced. Also, it can be found that the numbers of shells required for a given duty cannot be isolated from tube side pressure drop. Returning to the plain tube unit, a designer constrained by a need for multiple shells would generally look at the use of multiple tubes passes. Therefore, the analysis is also repeated for both units of plain and low-finned tubes with two, four and eight tube passes. The results for eight tubes passes are shown in Figures 5 and 6. The results indicate that in such a situation units with finned–tube bundles need just a single shell. At the same duty the use of a plain tube bundle requires more than one shell in series in order

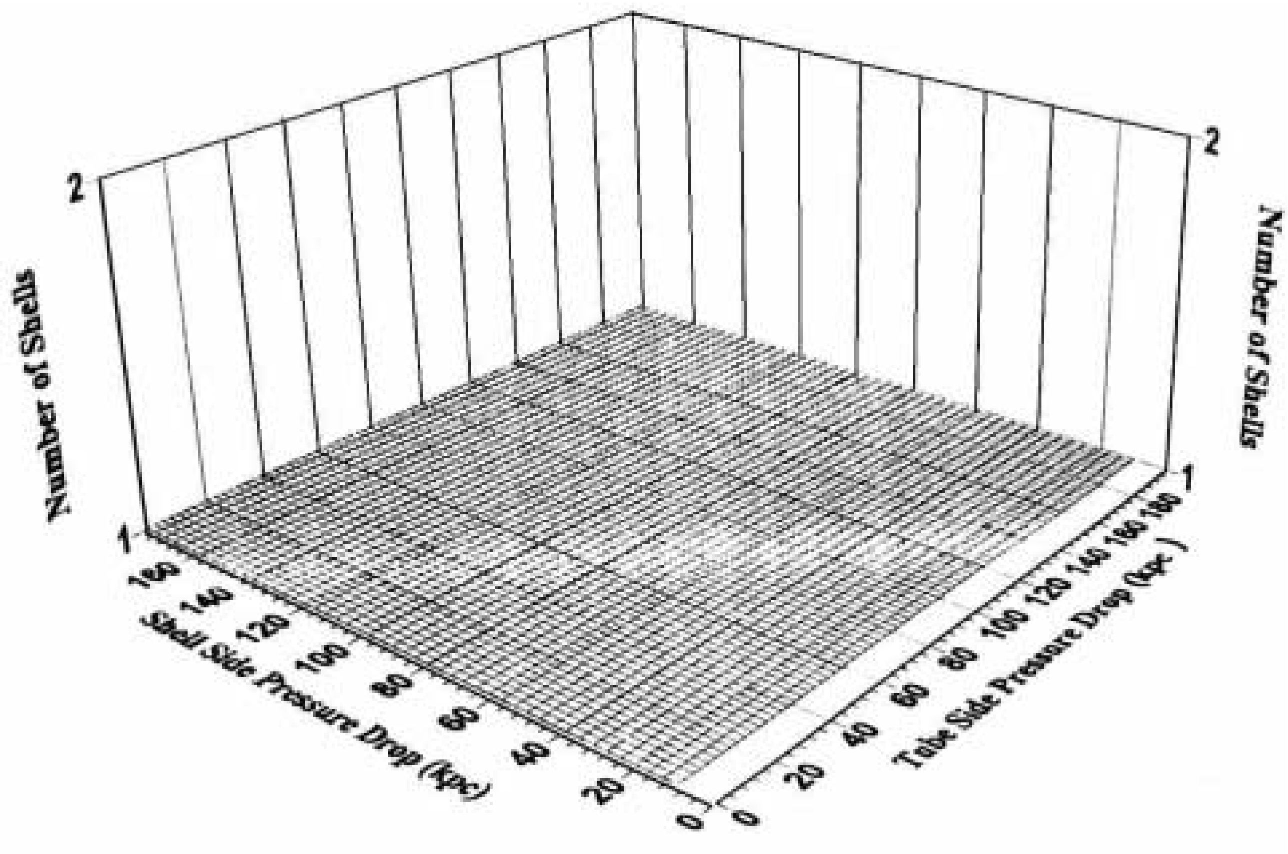

Figure 6. Number of shells for a finned tube unit with eight tubes passes (water).

TABLE 4. Problem Specification for Crude Oil in Shell Side.

Data	Units	Shell side	Tube side
Flow rate	Kg/s	192	135
Thermal Conductivity	W/m.C	0.122	0.59
Density	Kg /m^3	786	995
Heat Capacity	J/ kg .C	2177	4187
Viscosity	Ns/m^2	5×10^3	0.72×10^3
Range of Allowable ΔP	KPa	10-300	10-200
Heat load	kW	5295	
Thermal Resistances R_D	m^2.C/w	0.00036	
Log-Mean Temperature Difference ΔT_{LM}	^{o}C	43.8	
Tube Geometry	Based on standard values given in Reference 16, code: 197083		

to absorb the same pressure drops. This also clearly demonstrates the simpler arrangement of a unit with a finned tube bundle in comparison with the plain tube.

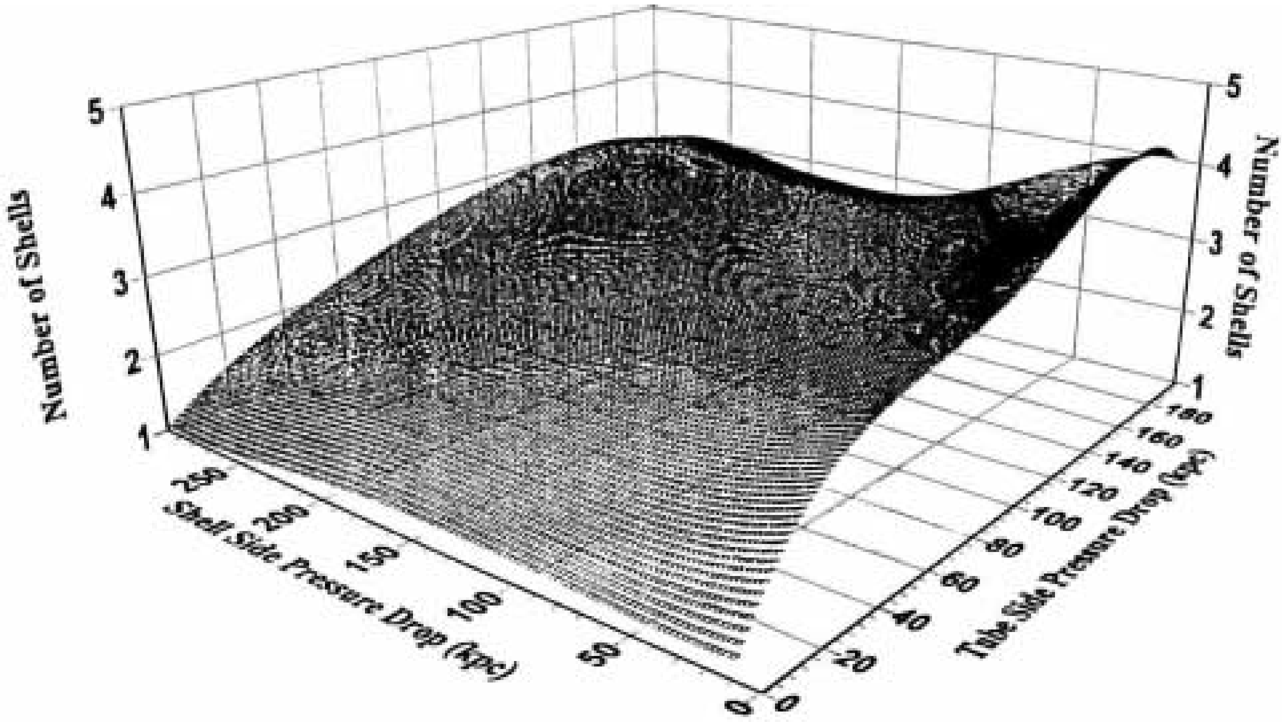

Figure 7. Number of shells for a plain tube unit with single tube pass (Crude oil).

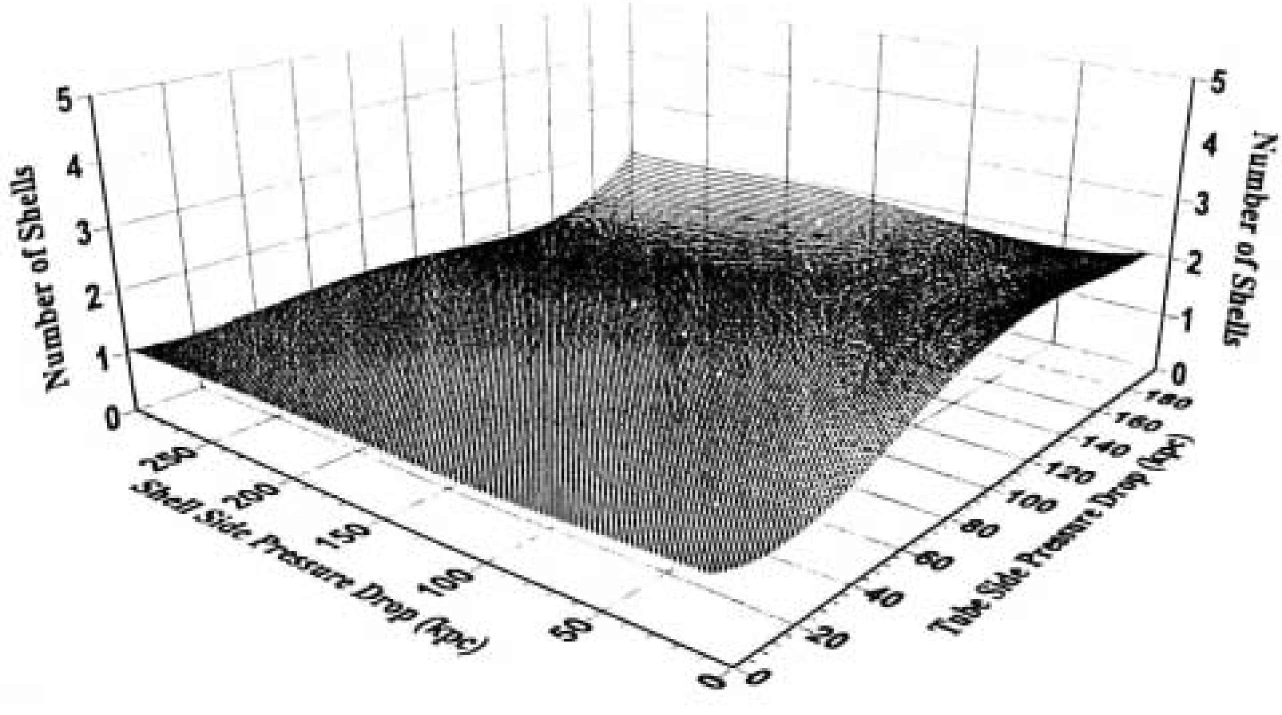

Figure 8. Number of shells for a finned tube unit with single tube pass (Crude oil).

b-Crude Oil: The procedure has been applied for crude oil on the shellside. The problem specification is presented in Table 4.

Figures 7 and 8 show the number of shells required

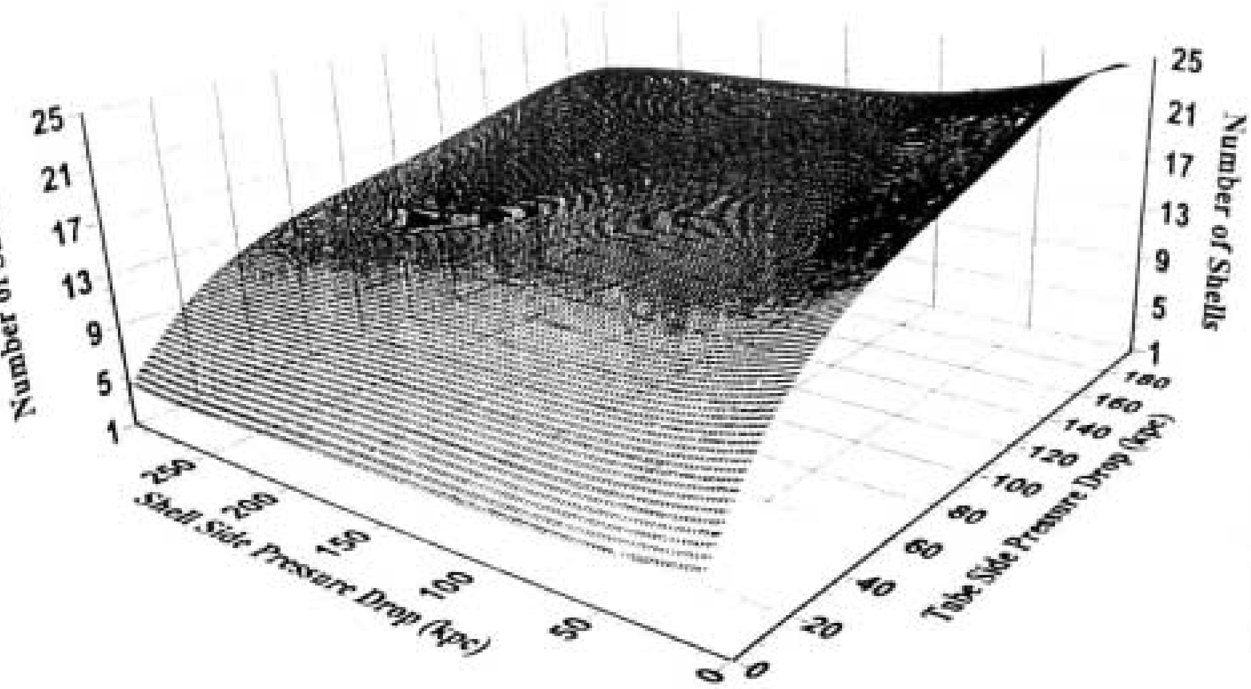

Figure 9. Number of shells for plain tube unit with eight tube passes (Crude oil).

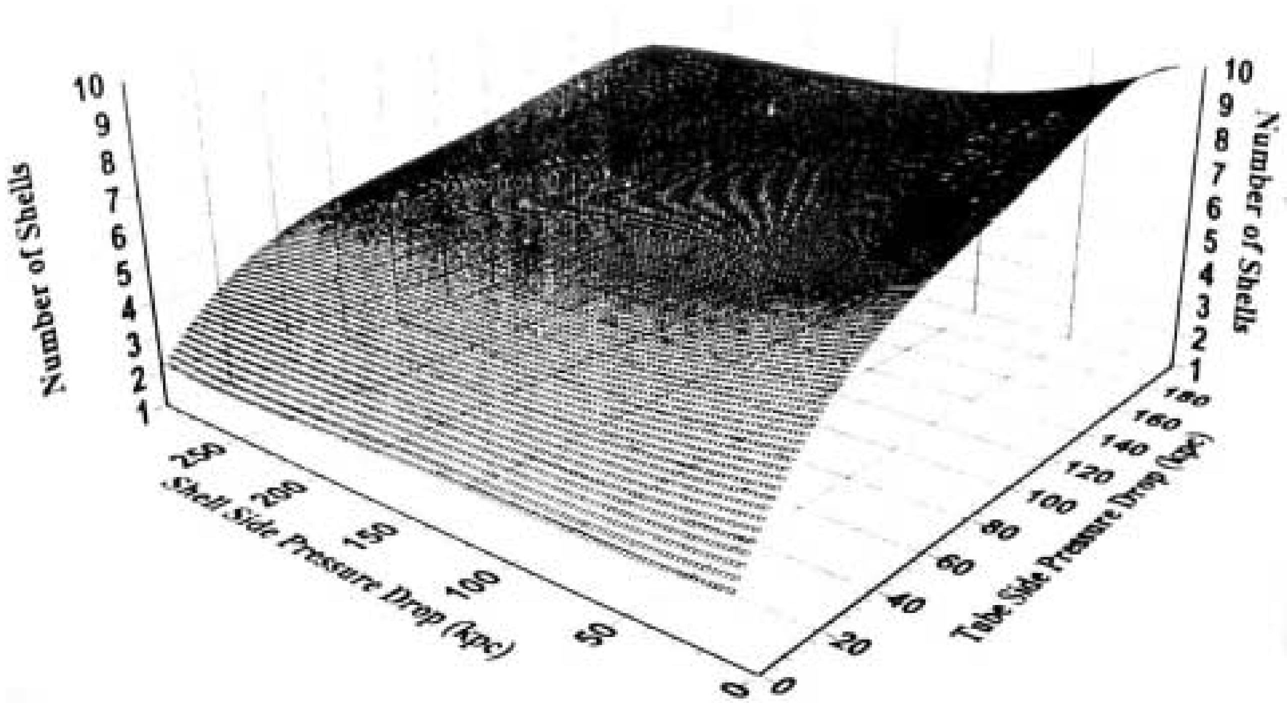

Figure 10. Number of shells for a finned tube unit with eight-tube pass (Crude oil).

for plain tube and finned–tube units with a single tube pass. Again, the reduction in the number of shells for finned tube units is considerable.

Also, in cases with multiple tube passes the potential benefit of heat transfer augmentation on the shell side using low-finned tubes would be significant. Figures 9 and 10 compare the results of such an analysis for units of plain and low-finned tubes. These units are considered with eight tube passes. As seen, the low-fin bundle needs about half the number of shells used in the plain tube bundle.

Since distribution of such liquids into the shellside of a unit with plain tubes will usually lead to laminar flow even under high-pressure drops, therefore, substitution of tubes with low-finned tubes may not only improve the performance of the heat exchanger but also reduce the number of shells in series. This means that the benefits of using low-finned tubes can go beyond just size reduction.

It should be noted that the non-integer values for the number of shells in the figures results from the use of the Stanford Graphic package and is due to interpolation and fitting a surface to integer raw-data within the software.

8. CONCLUSION

The more reliable correlations for analysis of thermal and hydraulic performance of low finned tubes are identified. It is shown that correlations of ESDU [14] still more reliable to use. It is demonstrated that by using low-finned tubes rather than plain tubes, not only can the performance of the heat exchanger be improved but also the number of shells required for a given duty can be reduced. This study can be considered as a supplementary work of the recent investigation of author.

9. NOMENCLATURE

A	Heat transfer surface area, m2
A_f	Fin surface area of tube, m2
Ai	Inside Surface Area, m2
At	True Heat Transfer Area, m2
A_{unf}	Outside surface area of excluding fins, m2/m
deH	Equivalent (hydrodynamic) diameter, m
d_f	Fin tip diameter, m
d_i	Tube I.D., m
d_o	Tube O.D., m
d_r	Fin root diameter, m
d_s	Shell diameter, m
f	Fanning's friction factor
F	Correction factor for Nusselt number in Equation 7
f_h	Fin high
f_s	Spacing between fins, m
f_t	Fin thickness, m
j	Colburn j-factor
k_f	Fin thermal conductivity , W/m.K
k_s	Shellside fluid thermal conductivity , W/m.K
Nu	Nusselt number
P_x	Longitudinal tube pitch, m
P_Y	Transverse tube pitch, m
P_t	Tube pitch, m
P_f	Fin pitch, m
Pr	Prandtl number
Q	Heat load, W
R_D	Fouling resistance, m^2.K/W
R_{opp}	Opposing resistance, m^2.K/W
Re	Reynolds number
T	Temperature, ^oC or K
U	Overall heat transfer coefficient, W/m^2K

Greek Symbols

α_f	Mean heat transfer coefficient over the finned-tube bank,W/m^2.K
α_{ft}	Effective mean heat transfer coefficient over the finned-tube bank,W/m^2.K
α_s	Shellside heat transfer coefficient, W/m^2k
ΔP	Pressure drop, N/m^2
ΔT	Temperature changes of fluid, ^oC
ΔT_m	Corrected log mean temperature difference, ^oC
ΔT_{LM}	Log mean temperature difference, ^oC
ρ	Density, kg/m^3
μ	Viscosity, kg/m.s
ε_f	Fin efficiency

TABLE A1. j-Factor Correlations for Low-Finned Tube Banks.

j-factor	Re	Ref.
1- $j = 0.292\, Re_r^{-n} . \Phi_{j1}$ $\Phi_{j1} = (\frac{f_s}{d_f})^{1.115} . (\frac{f_s}{f_h})^{0.257} . (\frac{f_t}{f_s})^{0.666} . (\frac{d_f}{d_r})^{0.473} . (\frac{d_f}{f_t})^{0.772}$ $n = 0.415 - 0.0346\, \mathrm{Ln}(\frac{d_f}{f_s})$	1000-25000 In line/staggered Array	[15]
2- $j = 0.183\, Re_r^{-0.3} . Pr^{0.027} . \Phi_{j2}$ $\Phi_{j1} = (\frac{f_s}{f_h})^{0.36} . (\frac{f_h}{d_f})^{0.11} . (\frac{p_y}{d_f})^{0.06} . F$	10^3-80×10^4 In line/staggered Array	[14]
3- $j = 0.3\, Re_r^{-0.375} . \Phi_{j3}$ $\Phi_{j3} = (1 + 2(\frac{f_h}{P_f}) + 2(\frac{f_h}{d_r}))^{-0.375}$	200-500×10^3 In line array	[10]
4- $j = 0.134\, Re_r^{-0.319} . \Phi_{j4}$ $\Phi_{j4} = (\frac{f_s}{f_h})^{0.2} . (\frac{f_s}{f_t})^{0.1134}$	1100-18000 Staggered array	[11]
5- $j = 0.45\, Re_r^{-0.375} . \Phi_{j5}$ $\Phi_{j5} = 1 + 2(\frac{f_h}{p_f}) + 2(\frac{f_h}{d_r})$	2000-500×10^3 $5 < \Phi j_5 < 12$ Staggered array	[12]
6- $j = 0.224\, Re_r^{-0.338} . \Phi_{j6}$ $\Phi_{j6} = (\frac{1 - f_t / p_f}{f_h / p_f})^{-0.25} . (\frac{p_t}{d_r} - 1)^{0.1} . (\frac{p_y}{d_r} - 1)^{-0.15} . (\frac{d_{eH}}{dr})^{-0.338}$ $d_{eH} = \frac{\left[\frac{(d_f^2 - d_r^2)}{2P_f} + \frac{d_f\, f_t}{P_f} + d_r(1 - \frac{f_t}{P_f})\right]}{\left(\frac{(d_f - d_r)}{P_f} + 1\right)}$	3000-56000 Staggered array	[10]

TABLE A2. f-Factor Correlations for Low-Finned Tube Banks.

f-factor	Re	Ref.
1- $f = 3.805\, Re_r^{-0.234}.\Phi_{f1}$ $\Phi_{f1} = (\frac{f_s}{d_f})^{0.251}.(\frac{f_h}{f_s})^{0.759}.(\frac{p_x}{p_y})^{0.379}.(\frac{d_r}{d_f})^{0.729}.(\frac{d_r}{p_x})^{0.709}$	1000-25000 In line/staggered Array	[15]
2– $f = 4.71 Re_r^{-0.286}.\Phi_{f2}$ $\Phi_{f2} = (\frac{f_h}{f_s})^{0.51}.(\frac{p_y - d_r}{p_x - d_r})^{0.536}.(\frac{d_r}{p_y - d_r})^{0.36}$	$1000\text{-}10^5$ In line/staggered Array	[14]
3- $f = 0.565\, Re_r^{-0.16}.\Phi_{f3}$ $f = 0.09.\Phi_{f3}$ $\Phi_{f3} = (\frac{d_f}{d_r} - 0.8)^{-0.32}.(\frac{p_1}{d_r})^{0.59}.d_{eH}^{-0.67}$	$1000\text{-}10^5$ Re> 10^5 In line array	[10]
4- $f = 9.465\, Re_r^{-0.316}.\Phi_{f4}$ $\Phi_{f4} = (\frac{p_t}{d_r})^{-0.927}.(\frac{p_t}{q_1})^{0.515}$ $q_1 = (p_t^2 + p_1^2)^{1/2}$	Re=200-5×10^5 Staggered array	[11]
5- $f = 1.98\, Re_r^{-0.31}.\Phi_{f5}$ $\Phi_{f5} = (\frac{1 - f_t / p_f}{f_h / p_f})^{-0.20}.(\frac{p_t}{d_r} - 1)^{0.14}.(\frac{p_y}{d_r} - 1)^{-0.18}.(\frac{d_{eV}}{d_r})^{-0.31}$ $d_{eV} = (4\, p_t\, p_y/\pi\, d_r - d_r)/\Phi_{j3}^{0.375}$	1600-31000 Staggered array	[12]
6- $f = 1.532\, Re_r^{-0.25}.\Phi_{f6}$ $\Phi_{f6} = (d_{ej}/d_r)^{-0.25}$ $d_{ej} = d_{eH}(f_h/2 f_s)^{-0.4}.\left\{\frac{1}{2\sqrt{\frac{p_t}{d_{eH}} - 1}} + \frac{1}{2\sqrt{\frac{p_3}{d_{eH}} - 1}}\right\}^{-4}$	Re> 5×10^3 Staggered array	[10]

Subscripts

i	Refer to inside
o	Refer to outside
r	Based on fin root diameter
t	Refer to tubeside
s	Refer to shellside
w	Refer to tube wall

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11. APPENDIX A

Some of the more important thermal and hydraulic correlations characterizing performance of low-finned tubes are tabulated in Tables A1 and A2 as follow.

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