Evaluate extended surface exchangers carefully

Finned tube exchangers are often purchased based on surface area only. Here's why this can be costly and how to evaluate extended surface exchangers correctly

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VARIOUS TYPES of fins, such as circumferential, rectangular, pegs and rods are used in heat transfer equipment. Fins can be used inside as well as on the outside of tubes. However, to illustrate the basic facts of heat transfer and how one should evaluate fins, these discussions will pertain to circumferential solid or serrated fins used widely in the energy equipment industry (Fig. 1).

Extended surfaces have the advantage of reducing the size and weight of heat transfer equipment. In addition they can result in lower gas pressure drop, thus reducing the operating costs. Appendix 1 compares the performance of an evaporator for a heat recovery steam generator (HRSG) using bare and finned tubes.

The important fact this article would like to bring out, however, is that extended surfaces should be evaluated and purchased based on performance and NOT based on surface area alone. Unfortunately, this is being done even today by well qualified engineers and purchasing managers.

Basic facts about extended surfaces. Extended surfaces are used in clean gas applications such as HRSGs for gas turbine plants, gaseous or liquid combustion, incineration systems, etc. A high fin density, say five or six fins/in. can be used in these cases. Cleaning becomes a concern if the gas stream contains particulates or is dirty. Extended surfaces have been used with flue gas streams containing particulates such as flue gases from combustion of coal. However, the fin configuration has to be carefully selected considering cleaning implications. A very low fin density, say two fins/ in., is recommended in such cases. Hence, fin configuration is decided by cleanliness considerations in several applications.

This article, however, will be devoted to clean gas situations where heat transfer alone is the basis for selecting the fin configuration and where one has a choice of using as much extended surface as desired.

Finned surfaces are attractive when the ratio between the heat transfer coefficients on the outside of the tubes to that inside is very small. In boiler evaporators or economizers for instance, the tube side heat transfer coefficient could be in the range of 1,500 to 3,000 Btu/ft^{2}h°F, while the gas side coefficient outside could be in the range of 10 to 20 Btu/ft^{2}h°F. A large fin density such as 5 or 6 fins/in is appropriate. As the ratio between the outside to inside heat transfer coefficients decreases, the effectiveness of using a large fin surface decreases. In air heaters or superheaters where the tube side coefficient is small (in the range of 20 to 250 Btu/Ft^{2}h°F) use of a large fin density such as 5 or 6 fins/in is not prudent. The combination of low tube side coefficient and a low gas side coefficient (by using a large fin density) is poor. It also will be shown later that the use of high density fins on surfaces with a low inside heat transfer coefficient will result in not only a low overall heat transfer coefficient, but also higher tube wall and fin tip temperatures, and higher gas pressure drop.

The important fact that engineers overlook while selecting extended surfaces is that the overall heat transfer coefficient will be lower as the fin density increases, even though the surface area will be several times greater. For heat transfer

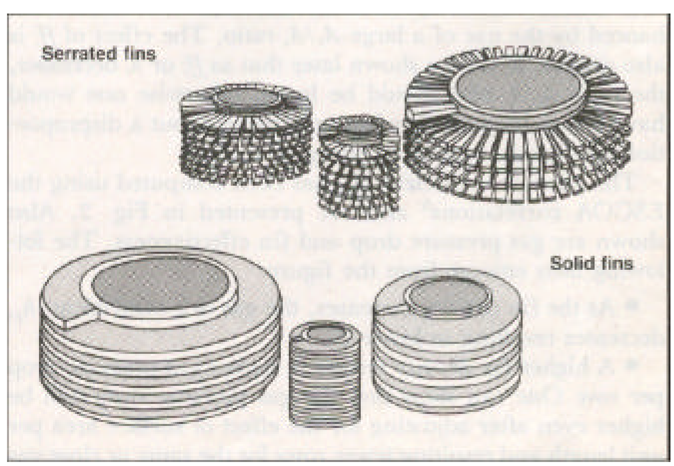

Fig. 1-Solid and serrated fins.

The next time you see a superheater or air heater design with five or six fins/in., please question the need for such a geometry. Ask for alternate designs with lower fin density. The results will be surprising.

purposes, one should be concerned with the product of *surface area and the overall heat transfer coefficient, not the surface area alone.* It will be shown that one can even transfer more energy with a lower surface area by proper use of fin configuration.

Theory. To understand the heat transfer implications of using a low or high finned surface, one has to understand the basic equation for U, the overall heat transfer coefficient. The equation below gives U based on outside surface area. (To convert this U to a tube side area basis, simply multiply by the ratio of external to internal surface areas.)

$1/U = 1/h_i(A_t/A_i) + ff_i A_t/A_i + ff_o + (A_t\, d/A_w\, 24\, K_m)\ln(d/d_i) + 1/\eta h_g$

(where: A_t, A_i, A_w = Total outside surface, tube inside surface, average tube surface, ft²/ft

d, d_i = Tube O.D., I.D., in.

h_g, h_i = Gas side heat transfer coefficient (outside) and inside heat transfer coefficient, Btu/ft²h°F

ff_o, ff_i = Fouling factors, outside and inside, ft²h°F/Btu

K_m = Metal thermal conductivity, Btu/ft h °F η = Fin effectiveness

The importance of fin efficiency and effectiveness and their method of computation are explained in the literature cited [1, 2, 3, 4]. Several variables are involved in the computation of U, such as the fin configuration (which affects h_g), the ratio of outside to inside surface area, A_t/A_i and the tube side coefficient, h_i. The tube side coefficient effect is enhanced by the use of a large A_t/A_i ratio. The effect of ff is also similar. It will be shown later that as ff_i or h_i decreases, the ratio A_t/A_i also should be lower, otherwise one would have a very large surface but only bring about a disproportionately small increase in energy transfer.

The gas side coefficient, h_g, has been computed using the ESCOA correlations[5] and are presented in Fig. 2. Also shown are gas pressure drop and fin effectiveness. The following facts emerge from the figure:

As the fin density increases, the gas side coefficient, h_g, decreases resulting in lower U.

A higher fin density results in higher gas pressure drop per row. One can show that the gas pressure drop will be higher even after adjusting for the effect of surface area per unit length and resulting fewer rows for the same or close gas mass velocities.

Fin effectiveness decreases with an increase in gas velocity. It increases with a decrease in fin height. Since the product of fin effectiveness and h_g enters into the calculation for U, by choosing proper fin height and density more energy can be transferred with a lower surface area.

Effect of tube side heat transfer coefficient. A simple calculation may be done to show the effect of tube side coefficient and fin density on U.

Rewriting equation 1 based on tube side area and neglecting other resistances, we have the following expression for U_i, the tube side overall heat transfer coefficient:

$1/U = 1/h_i + (A_i/A_t)/\eta h_g \quad (2)$

True comparison of surface area requirements can be made if U is computed based on a tube side area basis. Note that U will be higher with higher fin density while U, based on external surface area, will be lower with higher fin density.

Using the data from Fig. 2, U values have been computed for different fin densities and for different h_i values. These are shown in Table Ia. Also shown are the ratio of U values between the five and two fins/in. tubes as well as the ratio of their surface areas. The following conclusions can be drawn:

As the tube side coefficient decreases, the ratio of U values (between five and two fins/in.) decreases. With h_i = 20, U ratio is only 1.11. With an h_i = 2,000, the U_i ratio is 1.74. What this means is that as the tube side coefficient decreases, the benefit of adding more external surface becomes less attractive. We have 2.325 times the surface area but only 1.11 times the increase in U. However, with a higher h_i of 2,000, the ratio is better; 1.74. The same trend can be shown to be true at different gas velocities.

A simple estimation of tube wall temperature can tell us that the higher the fin density, the higher the wall temperature.

For example, with n = 2, U_i = 39.28, gas temperature = 900°F and fluid temperature = 600°F, the heat flux q = (900 - 600)39.28 = 11,784 Btu/ft²h. The temperature drop across the tube side film (h_i = 100) = 11,784/ 100 = 118°F. The wall temperature is then 600 + 118 = 718°E

With n = 5, U = 53.55, q = (900 - 600)53.55 = 16,065. The tube wall temperature = 600 + 16,065/100 = 761°F.

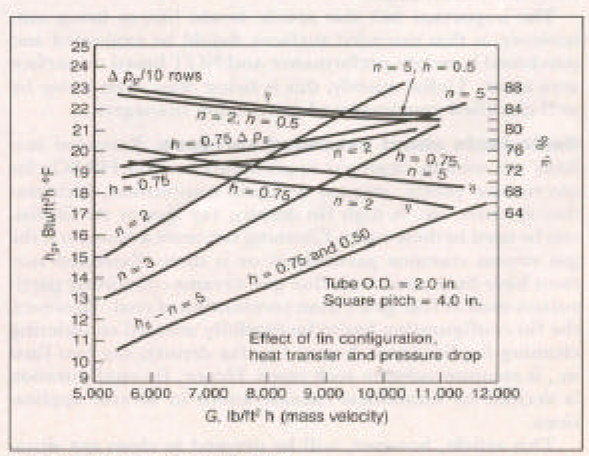

Fig. 2—Heat transfer coefficients.

Note that we are comparing for the same fin height.

The ratio of the gas pressure drop (after adjusting for the effect of U, values and consequential different surface area for the same energy transfer) shows that as the tube side coefficient decreases, the gas pressure drop ratio between the five and two fins/in. increases. It is *1.3* for $h_i = 100$, *1.02* for $h_i = 2,000$, and *1.6* for $h_i = 20$.

Effect of fouling factors ff, and ff.. The effect of fouling factors, ffi and ff_o are shown in Tables 1b and 1c for two and five fins/in. The following observations can be made:

With a smaller fin density the effect of ff is smaller. With 0.01 fouling and two fins/in., *Uo = 6.89* compared to *10.54* with 0.001 fouling. The ratio is *0.65*. With five fins/in., the corresponding Uo values are *4.01* and *7.46*. The ratio *is 0.53*.

The effect of ff_o is less significant as it is not enhanced by the ratio of surface areas. The U_o values for fouling factors of 0.01 and 0.001 are about *90* to *93%* of each other for two and five fins/in. cases.

In summary, a high fin density or a large ratio of outside to inside areas not only results in lower thermal performance, but also in higher gas pressure drop and tube wall temperature and hence, fin tip temperatures. Also, it is less prudent to use a high fin density when the tube side coefficient is small, as in superheaters or air heaters. While with boiler evaporators or economizers, which have a tube side coefficient of *1,500* to *3,000* Btu/ft^{2}h°F, a high fin density may be justified, but this is not the case with superheaters or air heaters, which have tube side coefficients in the range of *20* to *250* Btu/ft^{2}h°F. The same comment is valid if the tube side fouling factor is high.

Examples of superheater performance. Let us study the performance of a boiler superheater designed for the following conditions with different fin configurations:

Gas flow=200,000 lb/h at 1200 F
(% volume flue gas:CO2=7,H2O=12,N2=75,O2=6)

Steam flow, pph = 100,000

Steam temp, °F = *491* (sat.)

Steam press, psig = *600* (exit)

Tube size = *2 x 0.120;* tubes/row = *22;* length = 10 ft; tubes arranged in square pitch = *4.0* in.; streams = *22*, counterflow.

Fouling factors (in/out) = 0.001 (ft^{2}h°F/Btu)

Vary fin density from two to five fins/in., height from *0.5* to *0.75* in. with fin thickness = *0.075* in. and number of rows deep to obtain a duty of *14* to *18* MMBtu/h.

Results and discussions. The performance of the superheater with six rows deep is shown in Fig. *3* for various fin configurations. This study shows the effect of increasing the fin density on a given bundle configuration. The following can be noted:

1. As the fin density increases, the surface area increases more steeply than the duty. That is, the rate of change of duty with respect to surface area decreases.

2. The overall heat transfer coefficient decreases with an increase in fin density.

TABLE 1a-Effect of h_i on UI

[Calculations based on:2.0 x 0.105 tubes, 29 tubes/row, 6 ft long, 0.05 in. thick serrated fins; tubes on 4.0 in. square pitch; fin height = in.; gas flow = 150,000 pph; gas inlet temp =

1. h_i	- 20 -		- 100 -		- 2,000	
2. *n*, fins/in.	2	5	2	5	2	5
3. G, lb/ft^{2}h	5,591	6,366	5591	6,366	5,591	6,366
4. A_t/A_i/eta h_9	0.01546	0.00867	0.01546	0.00867	0.01546	0.00867
5. U_o	2.73	1.31	7.03	4.12	11.21	8.38
6. U;	15.28	17.00	39.28	53.55	62.66	109
7. Ratio S	- 2.325					
8. Ratio U;	- 1.11 -		- 1.363 -		- 1.74	
9. Ratio 0 P_9	- 1.6 -		- 1.3 -		- 1.02	

(Surface area of 2 fins/in. tube =2.59 ft^2/ft and for 5 **fins/in.=6.02)**

TABLE 1 b-Effect of ff,, tube side fouling factor

	(Tube side coefficient 2,000)			
1. Fins/in., *n*	2	2	5	5
2. U_o clean	11.21	11.21	8.38	8.38
3. *ffi*	*0.001*	*0.01*	*0.001*	*0.01*
4. U_o dirty	10.54	6.89	7.56	4.01
5. U_o as %	100	65	100	53

TABLE 1 c-Effect of ff_o, outside fouling factor

	(Tube side coefficient 2,600)			
1. Fins/in.	2	2	5	5
2. U_o clean	11.21	11.21	8.38	8.38
3. ff_o	*0.001*	*0.01*	*0.001*	*0.01*
4. U_o dirty	11.08	10.08	8.31	7.73
5. U_o as %	100	91	100	93

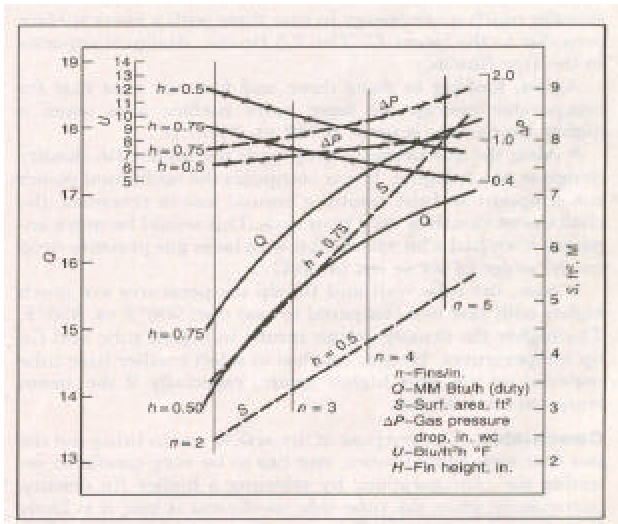

Fig. 3—Superheater performance (for six rows).

3. Gas pressure drop increases rapidly with fin density.

4. The lower fin height results in higher U.

5. It is also obvious from the chart that one can transfer the same energy with a lower surface area.

Q = 16.5 with *h = 0.75*, *n = 3* and S = *5,100* ft2, while *Q = 16.35* with h = *0.5*, *n = 4* and S = *4,250* ft^2. This shows that by proper selection of fin configuration, one can transfer comparable energy with a lower surface area.

TABLE 2-Effect of fin geometry on superheater performance

	1	2	3	4
1 Case no.	1	2	3	4
2. Duty, MMBtu/h	14.14	14.1	17.43	17.39
3. Steam exit temp, °F	689	691	747	747
4. Gas a P. in. WC	0.65	1.20	1.15	1.37
5. Gas exit temp, °F	951	950	893	893
6. Fins/in.	2.0	5.0	2.5	4.0
7. Fin height, in.	0.50	0.75	0.75	0.75
8. Fin thickness, in.	0.075	0.07	0.075	0.075
9. Surf. area, ft^2	2,471	5,34	5,077	6,549
10 Max. tube wall temp, °F	836	908	905	931
11 Fin tip temp, °F	949	1,03	1,064	1,057
12 U_o, Btu/ft^2	11.79	5.50	8.04	6.23
13 Tube side AP, psi	9.0	6.5	11.0	9.0
14 No. of rows deep	6	4	7	6
15 Fin effectiveness, %	84	72.3	67.7	70.4

Table 2 shows that study performed by varying the number of rows for the same problem. This was done to obtain several alternate designs. More revealing are the following facts:

0 The energy transferred is the same for both the two and five fins/in. designs in cases 1 and 2. However, due to a large difference in U values, - 11.79 vs. 5.5, the surface area is nearly 2.16 times; 5,342 vs. 2,472 ft^2!

Many purchasing managers presented with both designs will fall into the trap of thinking that the design with the larger surface area is superior. What a trap!

Also, by looking at cases two and three we see that we transfer much more energy in case three with a lower surface area due to the larger U. The 2.5 fins/in. design is superior to the five fins/in.

Again, looking at cases three and four we note that for comparable energy, we have more surface area when a higher fin density is used; 6,549 vs. 5,077 ft^2.

Also, the gas pressure drop with the higher fin density design is much higher. If one computes the additional power consumption and the resulting annual loss in revenues, the choice is obviously a very poor one. This would be more apparent if we had a bundle design with large gas pressure drop on the order of six to ten in. WC.

Also, the tube wall and fin tip temperatures are much higher with case two compared to case one; 908°F vs. 836°F. The higher fin density design results in higher tube and fin tip temperatures. Hence, one has to select costlier base tube material or the next higher grade, especially if the steam temperature is higher.

Conclusion. The purpose of the article was to bring out the fact that with finned tubes, one has to be very careful in selecting the configuration. By selecting a higher fin density, particularly when the tube side coefficient is low, it is likely that the same energy will be transferred with more surface area, at a higher gas pressure drop and with higher tube wall and fin tip temperatures. Purchasing managers should never fall into the trap of selecting a design simply because it has more surface area. On the contrary, they should ask for alternate designs with better fin configurations. It is possible to have a design which transfers comparable energy with a lower surface area, at a lower gas pressure drop and with lower metal temperature. The next time you see a superheater or air heater design with five or six fins/in., please question the need for such a geometry. Ask for alternate designs with lower fin density. The results will be surprising.

APPENDIX 1-Why finned tubes?

This example will illustrate the advantage of using extended surfaces.

A boiler evaporator for a gas turbine heat recovery system has to be designed for the following conditions:

1. Gas flow = 150,000 pph
2. Gas inlet temp. = 1,000 °F
3. Steam pressure = 150 psig
4. Feed water temp. = 240°F
5. Fouling factors (in/out) = 0.001 $ft^2 h$ °F/Btu
6. Width = 6 ft; tube length = 10 ft; 18 tubes/row, square pitch = 4.0 in.; 2 x 0.105 in. tubes

Compare the designs for bare and finned tubes using 5 fins/in. x 0.75 in. high x 0.05 in. thick serrated fins.

Table 3 shows the performance for bare and finned tubes. See references 1 and 5 for calculation procedure.

TABLE 3-Comparison of bare and finned tube performance

1. Case	bare	finned
2. Duty, MMBtu/h	24.4	24.4
3. Gas exit temp, °F	382	382
4. U_o, Btu/ft^1h°F	12.64	7.08
5. Surface area, ft^2		
6. No. of rows deep	122	20
7. Gas pr. drop, in. WC	4.50	3.2
8. Tube weight, lb	48,000	26,000

The finned bundle is more compact, requires less floor space, has lower gas pressure drop, weighs less and also has lower cost. Some variations in performance could result if the bundle cross section was different. The trend, however, would remain unchanged. The basic fact remains that the design with bare tubes would be uneconomical for this appliction in terms of cost; both operating and installed.

Please do not evaluate bids using a spreadsheet which shows only the surface area of alternate designs. You can have designs with even 100 to 150% difference in surface area but transfer the same energy. With finned tubes there are several variables and purchasing a design simply because it has more surface area may be detrimental to the performance in the long run.

LITERATURE CITED

1. V. Ganapathy, "Applied Heat Transfer," Pennwell Books, Tulsa, 1982, pp. 494519.
2. V. Ganapathy "HRSG features and applications," Heating Piping, Air-Conditioning, Jan. 1989.
3. V. Ganapathy, "Charts help evaluate finned tube alternatives," Oil and Gas Journal, Dec. 3, 1979.
4. V. Ganapathy, "Charts simplify spiral finned tube calculations," Chemical Engineering, April 25, 1977.
5. Escoa fin tube manual, Escoa Corp, Tulsa.

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