Understanding Finned Heat Exchangers

Fin geometry affects many aspects of boiler, evaporator and heater selection

Viswanathan Ganapathy
Boiler Consultant

FIGURE 1. Serrated (left) and solid (right) fins are used in boiler and heater applications

Solid and serrated fins (Figure 1) are widely used as heat transfer surfaces in boilers and heaters. The use of finned tubes makes heat exchanger equipment compact. The fluegas pressure drop is also decreased relative to a comparable plain-tube design, resulting in lower operating costs. Fin geometry, such as fin density (fins/in.), fin height and fin thickness, should be selected with care as it impacts the thermal design and performance of the exchanger. However, engineers often select fin geometry without understanding the implications on tubewall temperatures, heat flux, pressure drop or fin temperatures.

This article highlights the importance of selecting proper fin geometry for heat transfer equipment and the implications of poor fin selection. There are common misconceptions among engineers when evaluating fins regarding surface area. Many engineers assume that more fin surface area will always equate to a better design and that designs with smaller surface area will not be sufficient. Selecting a finned heat exchanger design based on surface area alone can lead to long-term problems including higher heat flux, increased tubewall temperatures and even tube failure. This article uses examples of boiler and steam superheater designs to illustrate the proper considerations when selecting fin geometry.

Why finned tubes?

Finned tubes are widely used in clean fluegas heat exchangers and heat recovery systems in petroleum refineries, chemical plants and power plants. In boilers or heaters, fin density ranges from 1 to 6 fins/in., height from 0.5 to 1 in. and thickness from 0.05 to 0.12 in. Fin or tube material can be made of carbon or alloy steel. Typical applications of finned tubes are in turbine exhaust heat-recovery steam generators (HRSG), incineration plant heat-recovery boilers, thermal fluid heaters or fired heaters that process natural gas or recover energy from clean fluegas.

Fin geometry should be carefully selected in fuel-fired applications, as the ash in fuel-oil can cause fouling or deposition of dust on finned tubes. Fins are generally avoided in solid fuel-fired applications, or tubes with very low fin density may be used based on experience. This article discusses heat recovery from clean fluegas applications where restrictions on fin geometry selection are minimal. An understanding of the thermal performance of finned tubes is helpful in optimizing the design and performance of the heat exchanger.

Users of finned heat exchangers experience many benefits. Using finned tubes results in a compact unit with low gas-side pressure drop. Fabricating smaller units results in lower labor costs. In gas turbine HRSGs, multiple pressure modules are used, such as superheaters, evaporators, economizers and condensate heaters. With finned-tube design (Figure 2, left), these modules are more compact, and can be easily assembled in a small space. With a traditional plain-tube design (Figure 2, right), the boilers or heat recovery equipment will be larger and may not fit into more confined spaces.

Example 1: Waste-heat boiler

Table 1 presents the performance of a waste-heat boiler with 150,000 lb/h of 1,000°F fluegas that must be cooled to 535°F in a waste-heat boiler generating saturated steam at 600 psig, with water entering near its saturation temperature. Table 1 shows six different exchanger design cases for this scenario. Some important observations can be made based on these results.

Fin geometry and the ratio of external surface area to tube internal area affects the overall heat-transfer coefficient. Figure 3 shows the behavior of gas-side heat transfer coefficients (h_g) at varying fin height (h) and fin density (n), as mass velocity (G) increases.

FIGURE 2. A cross-flow boiler with finned tubes (left) is very compact, while a longitudinal gas boiler with plain tubes (right) has a large footprint

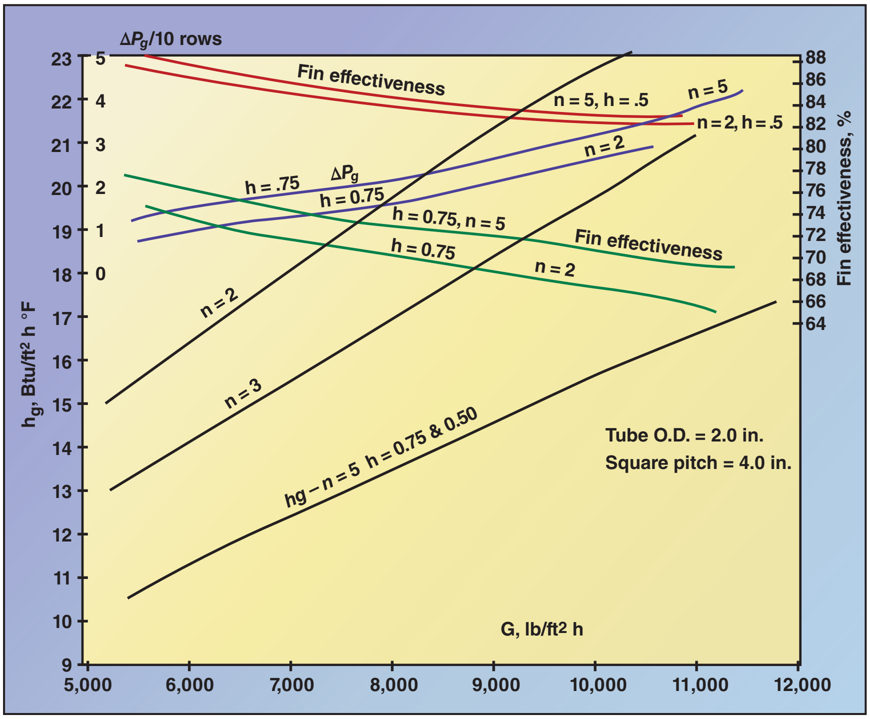

FIGURE 3. Geometry affects gas-side heat-transfer coefficients; as fin density (*n*) increases, the gas-side heat transfer coefficient (h_g) will decrease. In this figure, *h* is fin height in inches

Most notably, as fin density increases, the gas-side heat transfer coefficient will decrease. Also, as the ratio of external to internal area increases, the heat-transfer coefficient decreases. This further demonstrates that more surface area does not necessarily translate into a better design. The results in Table 1 show that the surface area of the designs in Case 2 (2 fins/in.) and Case 4 (6 fins/in.) varies by about 50%, and yet the exchanger performance is similar. The surface area of the plain-tube boiler for the same duty (Case 1) is much lower due to the higher overall heat-transfer coefficient compared to the finned-tube boiler. The lower surface area exhibited in Case 1 when compared to finned-tube designs also means that more tubes are required to achieve the same heat transfer. Table 1 shows that the plain-tube design will need 72 tubes, while the finned-tube designs only require 14 to 24 tubes. Thus, plain-tube designs have a much larger footprint than finned designs. This illustrates how crucial it is that purchasing managers and engineers are aware of the effect that fin geometry has on surface area and exchanger performance.

As seen in Table 1, finned tubes have a higher tubewall temperature compared to plain-tube designs. Tubewall temperature also increases as fin density increases. In this example, tubewall temperatures reached nearly 650°F, so it is important to choose an appropriate material of construction. Carbon-steel construction is assumed for this example. Higher heat flux causes the increased tubewall temperatures in finned tubes, since heat flux is a function of surface area.

$$q_i = U_o \frac{A_t}{A_i}\left(T_g - T_i\right) \quad (1)$$

where:

q_i = heat flux, $\frac{\text{Btu}}{\text{ft}^2\text{h}}$

U_o = overall heat transfer coefficient, $\frac{\text{Btu}}{\text{ft}^2\text{h°F}}$

A_t = surface area of finned tube, ft^2

A_i = surface area inside tube diameter, ft^2

T_g = gas temperature, °F

T_i = fluid temperature, °F

It is also apparent in this example that the pressure drop changes with fin geometry. In comparing Case 1's plain-tube design with Case 4 (6 fins/in.), the pressure drop is much lower for the finned design. This is because the finned exchangers require fewer tubes, even though the resistance of the finned tube to gas flow is much higher on a per-tube basis.

Finally, the effects of fouling are also examined. Cases 5 and 6 show the effect of tubeside fouling on the tubewall temperature and boiler duty. With plain tubes, the effect of increasing the tubeside fouling proved insignificant. Small changes are seen in exit gas temperature, duty and tubewall temperature. However, compared to the finned-tube design, the effect is minimal. With the finned-tube boiler, the duty decreases significantly with tubeside fouling; duty increases by 5.5% with fins, compared to only 1% with plain tubes. The tubewall temperature in the plain-tube design increases by only 24°F while in the finned bundle it

TABLE 1. BOILER DESIGNS WITH VARYING FIN GEOMETRY							
		Design with plain and finned tubes					
Case		1	2	3	4	5	6
Description	Units	Plain tubes	2 fins/in.	4 fins/in.	6 fins/in.	6 fins/in. with fouling	Plain tubes with fouling
Gas flowrate	lb/h	150,000	150,000	150,000	150,000	150,000	150,000
Inlet gas temperature	°F	1,000	1,000	1,000	1,000	1,000	1,000
Outlet gas temperature	°F	537	536	535	536	564	542
Specific heat	Btu/lb°F	0.2791	0.2791	0.2791	0.2791	0.2796	0.2792
Heat loss (assumed)	%	1	1	1	1	1	1
Heat duty	MM Btu/h	19.21	19.25	19.26	19.21	18.12	18.99
Gas pressure drop	inch wc	3.46	2.22	2.20	2.47	2.52	3.47
Steam flowrate	lb/h	26,250	26,307	26,307	26,331	24,769	25,959
Steam pressure	psia	600	600	600	600	600	600
Heat transfer coefficient (overall)	Btu/ft^{2}h°F	15.95	9.90	7.82	6.51	5.31	15.27
Tube outer diameter	in.	2	2	2	2	2	2
Tube inner diameter	in.	1.773	1.773	1.773	1.773	1.773	1.773
Fin density	fins/in.	n/a	2	4	6	6	n/a
Fin height	in.	n/a	0.75	0.75	0.75	0.75	n/a
Fin thickness	in.	n/a	0.05	0.05	0.05	0.05	n/a
Fin serration	in.	n/a	0.172	0.172	0.172	0.172	n/a
Fin conductivity	Btu/fth°F	n/a	25	25	25	25	n/a
Pitch	in.	4	4	4	4	4	4
Tubes per row		20	20	20	20	20	20
Number of tubes		72	24	17	14	14	72
Effective length	ft	8	8	8	8	8	8
Tubewall temperature	°F	500	526	543	554	647	524
Fin tip temperature	°F	n/a	726	709	696	760	n/a
Surface area	ft^2	6,032	9,796	12,453	14,797	14,797	6,032
Heat flux inside tubes	Btu/ft^{2}h	16,391	49,608	70,306	84,432	68,909	15,694
Fouling factor	ft^{2}h°F/Btu	0.0005	0.0005	0.0005	0.0005	0.003	0.003

increases by 93°F. These results demonstrate that finned-tube designs are much more susceptible to fouling and that care must be taken to ensure that tubeside fluids are clean and devoid of deposits. Otherwise, high tubewall temperatures — and even failures — can occur. This is especially important with superheater design, as tubeside heat-transfer coefficients are smaller, and a large fin density will contribute to higher heat flux and possible failure. Heat flux does not create as many issues for steam-generating evaporators since the tube-side boiling coefficient is very high. This further proves that careful examination must take place when selecting finned exchangers — one cannot judge on surface area alone. Heat flux and fouling considerations must also be taken into account when evaluating fin geometry.

The next example presents the importance of heat flux and fin geometry in an HRSG superheater.

Example 2: HRSG superheater

Steam superheaters (Figure 4) present a unique situation when determining fin geometry, since they exhibit much lower tubeside heat-transfer coefficients than evaporators and economizers, whose heat transfer coefficients can be up to ten times higher than those seen in a superheater. Consider two designs for a superheater that is to heat 100,000 lb/h of saturated steam at 600 psia to 840°F, with 150,000 lb/h clean fluegas at 1,300°F available from an incinerator. Case 1 features a design with a fin density of 5 fins/in., while Case 2's design has 2 fins/in. Table 2 presents all of the data required to evaluate this scenario.

As expected, the most significant difference between Cases 1 and 2 is surface area. The surface area required in Case 1 is 68% higher than that in Case 2, due to the higher fin density. Once again, the design should not be judged on surface area alone. Upon further examination of the results in Table 2, it is seen that Case 1 exhibits a much larger heat flux inside the tubes, resulting in a higher tubewall temperature of 1,007°F, compared to only 952°F for Case 2. Higher tubewall temperature is detrimental to the superheater's predicted operating life.

Figure 5 shows the typical Larson-Miller parameter (LMP) chart for estimating the life of superheater tubes based on stress. LMP is defined as follows:

$$\frac{(T+460)(20+\log t)}{1{,}000} \qquad (2)$$

where:

T = tubewall temperature, °F

t = predicted tubelife, h

LMP is useful for estimates of tube life or for studying the effect of tube temperature on equipment lifetime. First, the stress must be calculated. As stress is a function of operating

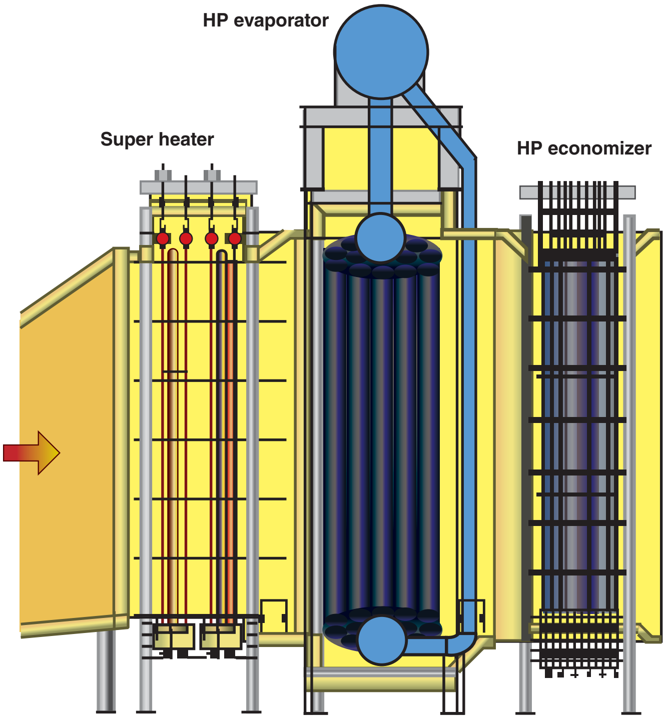

FIGURE 4. A typical HRSG consists of several elements, including a superheater

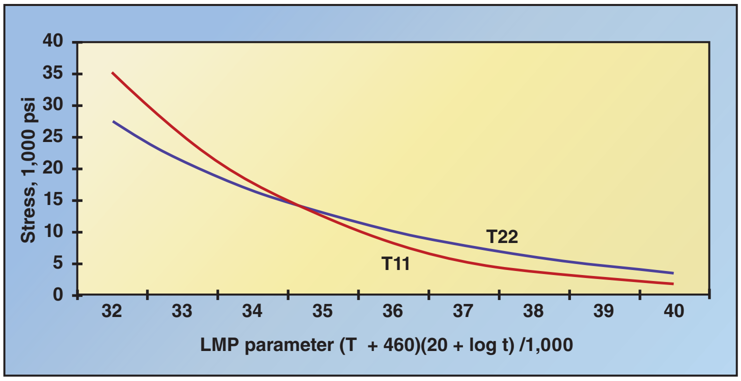

FIGURE 5. The Larson-Miller parameter allows for calculation of superheater tube life

TABLE 2. EFFECT OF FIN GEOMETRY ON SUPERHEATERS

		Case 1	Case 2
Gas flowrate	lb/h	150,000	150,000
Inlet gas temp.	°F	1,300	1,300
Outlet gas temp.	°F	773	773
Specific heat	Btu/lb°F	0.2884	0.2885
Heat duty	MM Btu/h	22.59	22.58
Heat transfer coefficient (overall)	Btu/ft^{2}h°F	5.74	9.66
Tubeside coefficient	Btu/ft^{2}h°F	254	254
Gas pressure drop	inch wc	2.23	1.76
Tubeside conditions			
Tubeside flowrate	lb/h	100,000	100,000
Inlet temperature	°F	486	486
Outlet temperature	°F	840	839
Pressure drop	psia	16	23
Operating pressure	psia	600	600
Inlet wall temp.	°F	590	556
Outlet wall temp.	°F	1,007	952
Inlet fin temp.	°F	655	631
Outlet fin temp.	°F	1,111	1,073
Tubeside velocity	ft/s	80.17	80.16
Surface area	ft^2	10,737	6,374
Tube outer dia.	in.	2	2
Tube inner dia.	in.	1.738	1.738
Fin density	fins/in.	5	2
Fin height	in.	0.75	0.625
Fin thickness	in.	0.05	0.05
Fin serration	in.	0.172	0.172
Fin conductivity	Btu/fth°F	25	25
Pitch	in.	4	4
Tubes per row		20	20
Number of tubes		12	18
Effective length	ft	8	8
Number of streams		20	20
Arrangement		inline	inline
Configuration		counter-flow	counter-flow
Heat Flux	Btu/ft^{2}h	32,467	21,625

temperature and pressure, this value will be the same for Cases 1 and 2. For this example, T11 is the selected material of construction. Assuming a stress of approximately 5,000 psi (based on operating pressure of 600 psig), the resulting LMP for T11 material is 37,500. Using the tubewall temperatures shown for the two cases in Table 2 and applying a 25°F margin, the predicted life can be calculated. For Case 1, the predicted lifetime is approximately 135,000 h, while for Case 2, the predicted lifetime is 1.2 million h. The LMP's logarithmic scale means that even incremental changes to tubewall temperature can greatly affect tube lifetime. This example further illustrates the dramatic difference fin geometry can have on heater operation and lifetime. Obviously, despite its smaller surface area, Case 2's fin geometry presents a much more economical option.

Final remarks

Finned tubes are an excellent option to achieve efficient heat transfer in evaporators, boilers and superheaters, but a clear understanding of finned exchanger design is important. Fin geometry affects the surface area of heat-transfer equipment significantly. The heat flux inside the tubes, the tubewall temperature and the equipment lifetime are also impacted. Hence a better understanding of the thermal performance aspects of finned tubes will help plant engineers to select a better HRSG or boiler and also to ask proper questions of vendors. ■

Edited by Mary Page Bailey

References

1. Ganapathy, V., "Industrial Boilers and Heat Recovery Steam Generators: Design, Applications and Calculations", CRC Press, FI, 2003

Author

Viswanathan Ganapathy is a boiler consultant from Chennai, India (v_ganapathy@yahoo.com), who specializes in thermal design and performance aspects. He has over 40 years of experience in the thermal design and performance aspects of steam generators and waste-heat boilers. He has authored over 250 articles on boiler-related subjects that have been published in a number of U.S., Indian and U.K. magazines. He has also authored several books and conducts courses on boilers. He graduated from I.I.T Madras with a degree in mechanical engineering.