How Important is Surface Area.?

It's important, but it should not be the only criterion you use to size and specify boilers and safety valves. Consider the factors outlined here as well.

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Fire-tube and water-tube boilers are widely used in the chemical process industries (CPI), for example, to recover energy from flue gas streams and to generate steam in gas- or oil-fired packaged steam generators. One of the main criteria that engineers use to specify or evaluate boilers is surface area. Packaged firetube boilers, for instance, are often specified as requiring 5 ft= per boiler horsepower (one boiler horsepower is equivalent to 34,500 Btu/h of output).

However, surface area is a misleading variable because heat transfer depends on other factors as well, including gas velocity, the size of the tubes, the tube pitch and arrangement, the configuration of tube fins, fouling factors, and others. For the same duty or energy transferred, one can develop different designs with significant differences in surface areas, and the various designs can have widely different costs.

This article outlines how to size and specify boilers other than by simply stating surface area. In addition, it discusses the selection of safety valves, which is still done based on surface area, and describes a more practical approach.

FIRE-TUBE BOILERS

In fire-tube boilers (Figure 1), flue gas flows inside the tubes while the steam is generated outside the tubes. Depending on the cleanliness of the gas, tube sizes can vary from 1.5 to 3.5 in. O.D. If slagging is a concern, as in municipal solid waste incineration applications, the boiler should be of a multipass design, where the first pass is a pipe with a diameter ranging from 30 to 48 in. and subsequent passes consist of smaller diameter tubes. Packaged oil- or gas-fired boilers have a similar configuration. The specified gas velocities can vary, depending on the allowable gas pressure drop.

Both of these factors - tube size and gas velocity - influence the heat-transfer coefficients and, hence, the surface area.

Sizing procedure

The procedure for sizing a fire-tube boiler is as follows.

The required surface area, S, is calculated from:

$$S = Q/(UAT) \qquad (1a)$$

If U is based on the tube outer diameter, then the surface area is also based on the tube outer diameter; likewise, if U is based on the tube inner diameter, then the surface area should be based on the tube inner diameter. This can also be expressed as $U_o S_o = U_i S_i$, where $S_o = \pi d_o NL/12$ and $S_i = \pi d_i NL/12$. Thus, Eq. 1 a can be rewritten as either

$$S_i = Q/(U_i \Delta T) \qquad \text{(lb)}$$

or

$$S_o = Q/(U_o \Delta T) \qquad \text{(1c)}$$

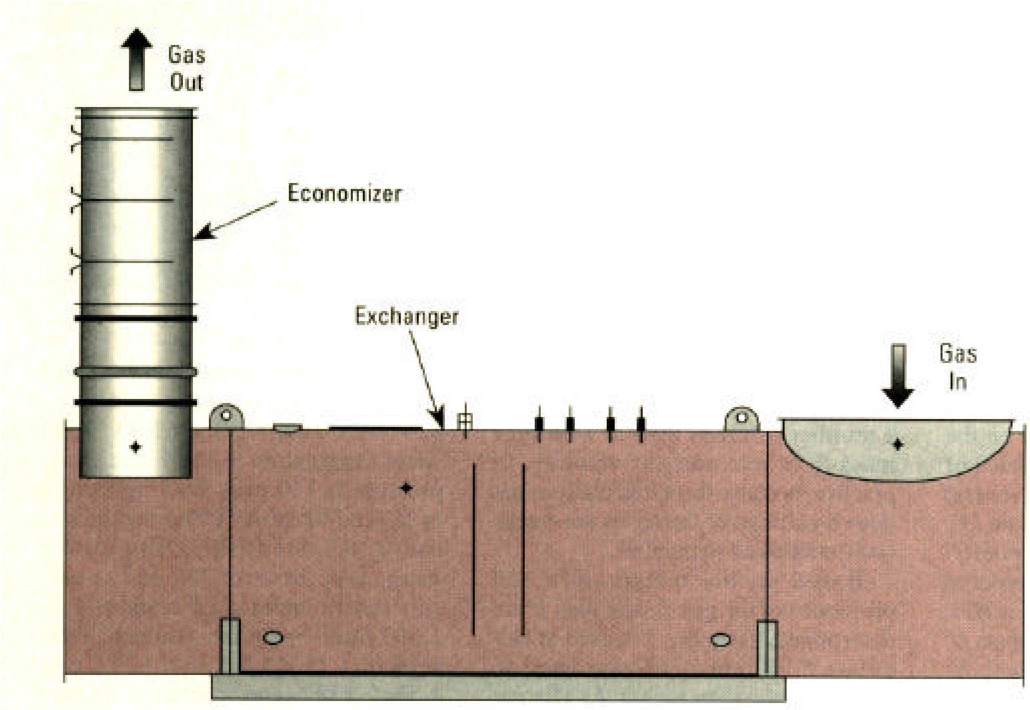
The energy transferred, Q, is:

$$Q = W_g C_p,(T_1-T_2)h_l = W_s H_s \qquad (2)$$

The term h, represents the heat loss factor and is equal to one minus the losses due to radiation and convection from the boiler surfaces. *A 2%* loss, or $h_l = 0.98$, is typical.

The log mean temperature difference, ΔT, is determined by:

$$\Delta T = (T_1-t_s)-(T_2-t_s)/\ln[(T_1-t_s)(T_2-t_s)] \qquad (3)$$



■ ***Figure 1. Fire-tube waste heat boiler.***

The overall heat-transfer coefficient, Uo, is given by:

$$\frac{1}{U_o} = \frac{d_o}{d_i h_i} + \frac{1}{h_o} + ff_i\left(\frac{d_o}{d_i}\right) + ff_o + \left(\frac{d_o}{24K}\right)\ln\left(\frac{d_o}{d_i}\right) \qquad (4)$$

The tube-side heat-transfer coefficient, hi, is the sum of the convective heat-transfer coefficient, h,., and the nonluminous heat-transfer coefficient, h_n. The value of h_n depends on the partial pressures of the tri-atomic gases in the flue gas (e.g., CO_2, H_2O) and is usually small - on the order of *5%* of h_c in fire-tube boilers. Thus, many designers are conservative and neglect h_n. (In water-tube boilers, however, h_n is very significant and cannot be neglected.) Further details on calculating h_n can be found in (1).

The value of h_i is obtained from the Dittus-Boelter equation:

$$Nu = 0.023Re^{0.8}\mathrm{Pr}^{0.4} \qquad (5)$$

where $Nu = h_c d_i/12k$, $Re = 15.2w/(\mu d_i)$, and $Pr = \mu Cp/k$. Substituting these expressions into Eq. 5 and simplifying yields:

$$h,. = 2.44w^{0.8}F/d_i^{1.8} \qquad (6)$$

where $F = (C_p/\mu)^{0.4}k^{0.6}$

The inside and outside fouling factors are denoted by ff_i and ff_o, respectively. For

NOMENCLATURE

- A_i, A_t, A_w = inside, total exposed (outside surface of the tube plus the total exposed fin surface), and average wall surface area, ft²/ft
- C_p = gas specific heat, Btu/lb
- d_i, d_o = tube inner and outer diameter, in.
- f = friction factor for pressure drop
- G = gas mass velocity, lb/ft²•h
- h_f = fin height, in.
- h_c, h_i, h_n, h_o = convective, inside, nonluminous, and outside heat-transfer coefficient, Btu/ft²•h•°F
- ΔH_s = enthalpy absorbed by steam, Btu/lb
- k, K = gas and metal thermal conductivity, Btu/ft•h•°F
- L, L_b, L_e = tube length, beam length, and effective length, ft
- n = fin density, fins/in.
- N = tube count
- N_d, N_w = number of rows of tubes deep, and wide
- Nu = Nusselt number
- ΔP_g = gas pressure drop, in. w.c.
- Pr = Prandtl number
- Q = heat-transfer duty, million Btu/h
- Re = Reynolds number
- S_i, S_o = inside, and outside tube surface area, ft²
- S_L, S_T = longitudinal and transverse pitch of tubes, in.
- T_1, T_2, t_s = inlet gas, exit gas, and steam saturation temperature, °F
- T_g, T_w = absolute temperature of the gas, and the tube wall, R
- ΔT = log mean temperature difference, °F
- U, U_i, U_o = overall heat-transfer coefficient, and overall heat-transfer coefficient inside and outside tubes, Btu/ft²•h•°F
- v = gas specific volume, ft³/lb
- V_g = gas velocity, ft/s
- w, W_g, W_s = gas flow per tube, total gas flow, and steam flow, lb/h

Greek letters

- ρ_g = gas density, lb/ft³
- μ = gas viscosity, lb/ft•h
- σ = Steffan-Boltzman constant = 0.174×10^{-8} Btu/h•ft²•°R⁴
- η = fin effectiveness, %

Table 1. Results of design calculations for fire-tube waste heat boiler.

Size	1.75 × 1.521			2.0 × 1.77			2.5 × 2.238		
Velocity, ft/s	98	123	163	98	123	162	98	123	162
Number of tubes	1,000	800	600	750	600	450	470	375	280
Length, ft	15.75	16.75	18.0	18.75	20.0	21.5	24.75	26.0	28.5
S_i, ft²	6,269	5,333	4,298	6,513	5,558	4,480	6,812	5,710	4,673
U_i, Btu/ft²•h•°F	9.47	11.08	13.70	9.07	10.68	13.19	8.73	10.29	12.72
ΔP_i, in. w.c.	2.05	3.34	6.23	1.97	3.20	6.00	1.95	3.16	6.00

clean gases and boiler water, they can be assumed to be 0.001 ft²•h•°F/Btu. For gas streams that can cause fouling, ff can be much higher - on the order of 0.05 ft²•h•°F/Btu. Tables of fouling factors are available in several published sources, such as (2) and (3).

The boiling heat-transfer coefficient, *ho* is very high - on the order of 2,000 Btu/ft²h°F. Thus, even a 20% variation in its value will not impact U, because the tube-side coefficient, hi, which is typically on the order of 10-20 Btu/ft²h°F, governs U.

The last term in Eq. 4 is the resistance of the tube wall to heat transfer. The thermal conductivity of the tube material, K, is about 20-25 Btu/ft•h•°F for carbon steel, the typical material used for boilers.

To size the boiler, the mass flow per tube, ranging from 120 to 200 lb/h for a 2-in. tube, and the gas velocity, typically ranging ranging from 60 to 170 ft/s, are assumed and the tube count is calculated. The relationship between mass flow and velocity is:

$$V = 0.05wv/d_i^2 \qquad (7)$$

While it may seem easier to assume a number of tubes than to assume a mass flow rate and gas velocity, in practice, because these calculations are done by computer the terms are essentially conceived in parallel.

Based on the temperature and properties of the gas, h_c and then U are determined. Then Eq. 1 is used to calculate S, which is in turn used to determine the tube length, L. The gas pressure drop is then calculated based on geometry (1):

$$\Delta P_g = (93\times10^{-6}) f L_e v w^2/d_i^5 \qquad (8)$$

If the computed pressure drop is higher than that allowed by the specification, another mass flow rate per tube is assumed and the procedure is repeated.

Example 1

Consider a fire-tube waste heat boiler required to cool 100,000 Ib/h of flue gas from 1,300°F to 474°F. The gas is at atmospheric pressure and consists of (by volume) 12% CO2 12% H_2O, 70% N_2 and 6% O_2. Feed water temperature is 220°F and steam pressure is 150 psig.

Fouling factors of ff_i = 0.002 ft²h°F/Btu for the gas and ff_0 = 0.001 ft²h°F/Btu for the steam, heat losses of 2%, and an outside heat-transfer coefficient of h = 2,000 Btu/ft²hF are assumed. Tube sizes of 1.75 x 1.521, 2 x 1.773, and 2.5 x 2.238 (outer x inner diameter) will be considered. What are the effects on surface area requirements of tube size and gas velocity (which can range from 90 to 170 ft/s)?

For simplicity, most of the calculation details are omitted. The results of the calculations for the various tube sizes and velocities are summarized in Table 1.

For the same amount of' energy transferred, one can see significant variations in the surface area - by as much as 50%. As the gas velocity increases, U increases, which brings down the surface area,

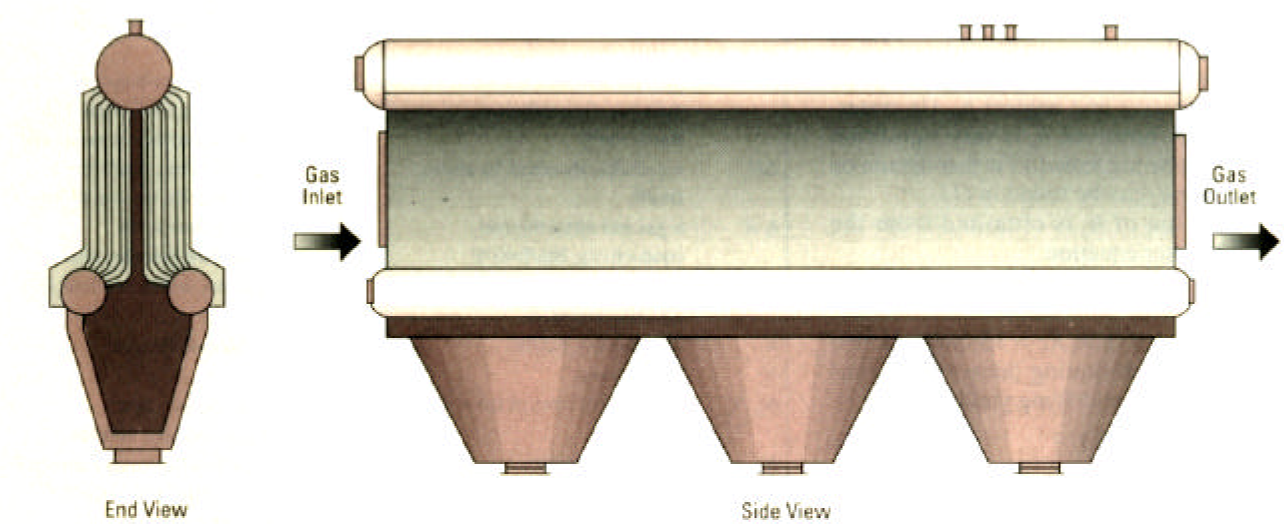


Figure 2. Water-tube waste heat boiler.

and the gas pressure drop increases. Also, as the tube size increases, U decreases for the same velocity. This, along with the fewer larger tubes, results in longer tube lengths. The main point to be noted is that for the same duty, the surface area can vary depending on the tube size and gas velocity.

These conclusions also apply to packaged fire-tube boilers firing oil or gas. A rule of thumb that, unfortunately, is still being used by specifying engineers is 5 ft^2 of surface per boiler horsepower. One can, by using a higher gas velocity or smaller tube size, develop a boiler design that will work fine with up to 10% to 20% less surface area. However, through lack of knowledge of heat-transfer design, several good designs are being overlooked by potential buyers, consultants, and end users.

Boiler cost generally increases with an increase in surface area. However, it does not rise proportionately because other items, such as boiler trim, controls, casing, insulation, and so on, account for a considerable part of the total cost and these may not increase proportionately. Labor costs are significant and may not be proportional to surface area. Each case, therefore, must be reviewed independently.

WATER-TUBE BOILERS

In water-tube boilers (Figure 2), if the gas stream is clean (such as with gas turbine exhaust gases), tubes with

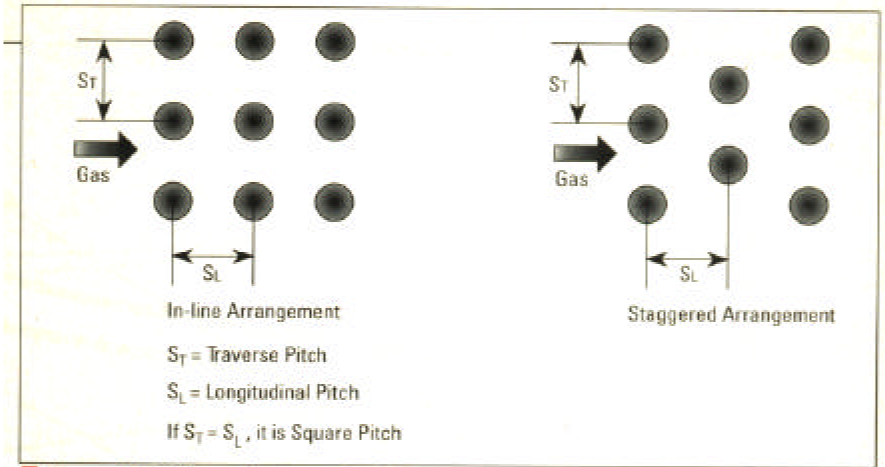


Figure 3. In-line and staggered tube arrangements.

extended surfaces, often called finned tubes, may be used. The use of finned tubes makes the design very compact. Other advantages include lower weight and lower gas pressure drop. If the gas stream is dirty, as in municipal solid waste incineration systems, only bare tubes should be used.

Design procedure

As with fire-tube boilers, the heat-transfer duty, Q, is calculated by:

$$Q = W_g C_p (T_1 - T_2) h_l,$$
$$= W_s \Delta H_s. = U S \Delta T \qquad (9)$$

U refers to the overall heat-transfer coefficient, and it is usually based on the outside surface area of the tube. U_o is given by:

$$\frac{1}{U_o} = \frac{A_t}{A_i h_i} + \frac{1}{\eta h_o} + ff_i\left(\frac{A_t}{A_i}\right) + ff_o + \left(\frac{A_t}{A_w}\right)\left(\frac{d}{24K}\right)\ln\left(\frac{d_o}{d_i}\right) \qquad (10)$$

where A, and Ai, which refer to the total external and internal surface area per foot of tube, are used instead of do and di. (In the case of bare tubes, $A_t/A_i = d_o/d_i$). Fin effectiveness is represented by n, which equals 1 in the case of bare tubes. In water-tube boilers, h_o is the gas-side heat-transfer coefficient, which is the sum of h_c and h_n; h_i is the tube-side boiling heat transfer coefficient, which is in the range of 2,000-3,000 BtU/ft^2hF.

Bare tubes. The procedure for computing h_o for bare tubes is as follows. Grimson's correlation for convective heat transfer is used for tubes in either an in-line or staggered arrangement (depicted in Figure 3):

$$Nu = BRe^N \qquad (11)$$

where $Nu = h_c d_o/12k$ and $Re = Gd_o/12\mu$. Thus, Eq. 11 reduces to:

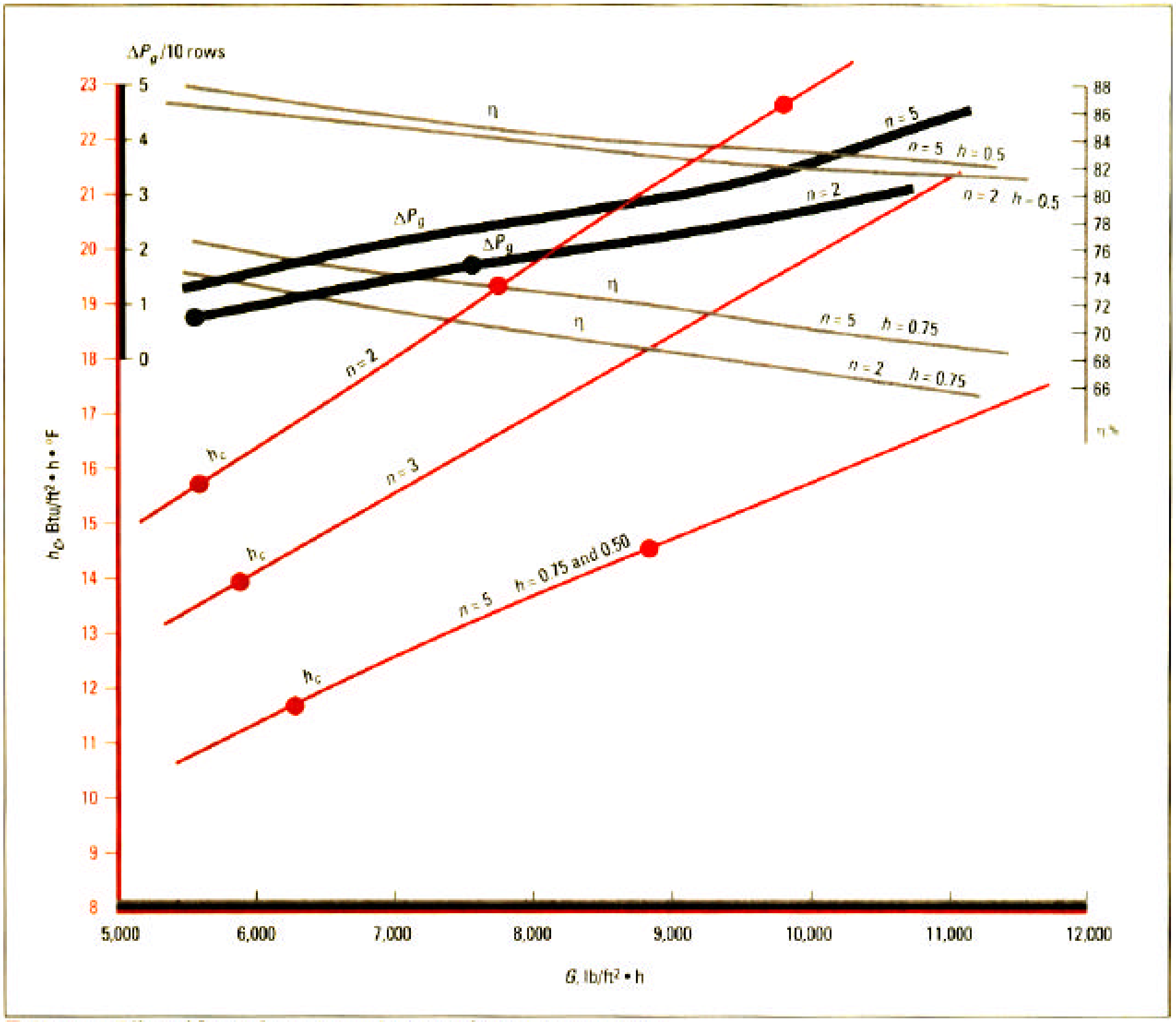
$$h_c = \left(\frac{12k}{d_o}\right)(B)\left(\frac{Gd_o}{12\mu}\right)^N \qquad (12)$$

The gas properties are evaluated at

Table 2. Grimson's coefficients for calculating hc for bare-tube water-tube boilers.

S_T/d_o	1.25		1.5		2		3	
S_L/d_o	B	N	B	N	B	N	B	N
Staggered								
1.25	0.518	0.556	0.505	0.554	0.519	0.556	0.522	0.562
1.50	0.451	0.568	0.460	0.562	0.452	0.568	0.488	0.568
2.0	0.404	0.572	0.416	0.568	0.482	0.556	0.449	0.570
3.0	0.310	0.592	0.356	0.580	0.440	0.562	0.421	0.574
In-line								
1.25	0.348	0.592	0.275	0.608	0.100	0.704	0.0633	0.752
1.50	0.367	0.586	0.250	0.620	0.101	0.702	0.0678	0.744
2.0	0.418	0.570	0.299	0.602	0.229	0.632	0.198	0.648
3.0	0.290	0.601	0.357	0.584	0.374	0.581	0.286	0.608

Source: (1)



■ *Figure 4. Effect of fin configuration on heat transfer and pressure drop.*

the gas film temperature, and the coefficients *B* and *N* are obtained from Table 2. (The ratio of transverse pitch to outside diameter *(S_t/d_o)* and of longitudinal pitch to outside diameter (S_l/d_o) are computed. *(S_t/d_o)* is read across the top of the table, and S_l/d_o down the side, under "Staggered" or "In-line," as appropriate. The values of *B* and *N* are then read from the chart.)

Gas mass velocity, G, is calculated by:

$$G = \frac{12W_g}{N_w L(S_T - d_o)} \qquad (13)$$

The nonluminous heat-transfer coefficient, h,,, could be significant depending on the tube pitch, the partial pressures of water vapor and other triatomic gases present, and the beam length, L_b. *b* is given by:

$$\begin{aligned} L_b &= 1.08(S_T S_L - d_o^2)/d_o \\ &= 3.4(\text{volume/surface}) \qquad (14) \end{aligned}$$

where "surface" refers to the total external surface area touched by the gas. Hottels' charts (2) are used to determine the gas emissivity, s, from the above data. One can then calculate *h* from:

$$h_n = \frac{\varepsilon\sigma\left(T_g^4 - T_w^4\right)}{T_g - T_w} \qquad (15)$$

Once *ho* is computed (here, too, *ho = hc + h_n)*, Eq. 10 can be used to calculate U_a. S is then obtained from:

$$S = A_t N_w N_d L \qquad (16)$$

where $A_t = \pi d_o/12$ for bare tubes.

The gas pressure drop, ΔP_g, is then obtained from:

$$\Delta P_g = \frac{\left(9.3\times10^{-10}\right) f G^2 N_d}{\rho_g} \qquad (17)$$

where *f is* the friction factor. For an inline arrangement, *f is:*

$$f = Re^{-0.15}\left[0.044 + \frac{0.08(S_T/d_o)}{(S_T/d_o - 1)^{(0.43+1.13d_o/S_L)}}\right] \qquad (18a)$$

and for a staggered arrangement, *f is:*

$$f = Re^{-0.16}\left[0.25 + \frac{0.1175}{(S_T/d_o - 1)^{1.08}}\right] \qquad (18b)$$

Finned tubes. The correlations for heat transfer in finned tubes are more complex, so only the chart technique for computing hc will be discussed here, since the objective of this article is only to show the effect of a few variables on surface area and not the complete design procedure for finned tubes. [The interested reader can find further details in (1).[Figure 4 can be used to estimate h,., the convective heat-transfer coefficient, for in-line arranged tubes having a 2-in. O.D. and a 4-in. square pitch.

The heat-transfer coefficient, fin effectiveness (11), and gas pressure drop are shown for 10 tube rows based on a gas turbine exhaust at an average temperature of 600°F.

The real value of the chart, however, is not in estimating hc. Rather, it illustrates the effect of fin configuration (that is, fin density, n, and fin height, *h)* on h_c. From this figure, one can see that:

• as fin density increases, *h* decreases;

• the higher the fin density, the higher the gas pressure drop will be, even after adjusting for the effect of the different number of rows required; and

• fin effectiveness decreases with fin height.

Hence, the simple conclusion that can be drawn from Figure 4 is that a higher fin density (or surface area per unit length) results in a lower *hc* and a lower Uo, which in turn means that more surface area is required.

Let us now look at two specific examples to see how different fin configurations, and how the difference between bare and finned tubes, can significantly affect surface area.

This example also illustrates the advantages of using extended surfaces, particularly in clean gas applications.

A boiler evaporator needs to be designed for a gas turbine exhaust. Gas data: Flow = 150,000 lb/h. Iinlet gas temperature = 1,000°F. Exit gas temperature = 382°F. Feed water temperature = 240°F. Inside and outside fouling factors = 0.001 ft²h°F/Btu. *ho* = 2,000 Btu/ft²•h•°F. The geometry is as follows: 18 tubes/row, tube O.D. = 2 in., I.D. = 1.77 in., length = 10 ft, in-line arrangement with transverse and longitudinal (square) pitch = 4 in., and material of construction is carbon steel. How do designs using bare tubes and serrated finned tubes (fin density = 5 fins/in., height = 0.75 in., thickness = 0.05 in.) compare?

Again, the calculation details are omitted. [The complete procedure can be found in (1).] Here we will discuss the most

Table 3. Comparison of bare and finned-tube boilers (Example 2).

	Bare	Finned
Gas flow, lb/h	150,000	150,000
Gas inlet temperature, °F	1,000	1,000
Gas exit temperature, °F	382	382
Duty, million Btu/h	24.4	24.4
U_o, Btu/ft²•h•°F	12.64	7.08
Surface area, ft²	11,500	20,524
Number of rows deep	122	20
Gas ΔP_g, in. w.c.	4.5	3.2
Total tube weight, lb	48,000	26,000

The convective and nonluminous heat-transfer coefficients, hc and hn are computed for bare tubes using the procedure described above, and Uo is evaluated from Eq. 10. The outside tube area, So is then computed and is used, along with the assumed tube length (L) and number of tubes wide (Nw), to determine N_d, the required number of rows of tubes deep. Then Eq. 17 is used to calculate ΔP_g.

For finned tubes, the gas mass velocity is calculated and is used to obtain h . from Figure 4. The fin effectiveness, η and the gas pressure drop for 10 rows are also read from the chart. A_t and A_w can be obtained from standard reference charts or can be calculated based on the fin geometry (1). Beam length is calculated and used to determine h„. Then Eq. 10 is employed to compute U(), which is used to determine N_d and ΔP_g.

Table 3 summarizes the results and compares the bare-tube and finned tube cases. The advantages of using finned tubes are clear: The finned tube boiler is more compact (it has only 20 rows deep, vs. 122 rows of bare tubes), has a lower gas pressure drop (3.2 vs. 4.5 in the bare tube boiler), and weighs less (26,000 lb vs. 48,000 lb). On the other hand, the sur

Fire tube boiler.

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face area is much higher - nearly twice the area of the bare-tube boiler - because of the lower overall heattransfer coefficient.

Example 3

The choice between bare and finned tubes is not the only factor that affects surface area. One can also see significant variations in surface area for the same duty with finned tubes of different configurations.

Consider a finned-tube superheater being designed for the following conditions: Gas flow = 200,000 lb/h. Gas inlet temperature = 1,200°F. Gas analysis (vol. %) = 7% CO_2,12% H_2O, 75% N_2 and 6% O_2. Steam flow= 100,000 Ib/h. Steam inlet temperature = 491°F (saturated) at 600 psig. Fouling factors for the gas and steam= 0.001 ft^2-•h•°F/Btu. The tube configuration is: tube O.D. = 2 in., I.D. = 1.738, 22 tubes/row, length = 10 ft, inline arrangement with square pitch = 4 in., countercurrent flow, and 22 streams (100,000 lb/h of steam flows through 22 tubes). The duty is between 14 and 18 million Btu/h. What happens when the number of rows deep is varied, the fin density is varied from 2 to 5 fins/ in., and the fin height is varied from 0.5 to 0.75 in. (fin thickness is constant at 0.075 in.)?

The procedure is very similar to that used in Example 2. In this case, though, the tube-side heat-transfer coefficient, h_i must be computed using Eq. 6. The results are presented in Table 4.

Though cases l and 2 transfer the same energy, the surface areas are significantly different, varying by nearly 100%. The reason is that the high fin density coupled with a smaller *hi* (on the order of 200-250 Btu/ft²h°F compared with 2,000 BtU/ft²h°F for boiling water) results in a much lower U . Hence, more surface area is required for the same duty. Similar results are obtained for cases 3 and 4, which have the same duty.

Comparing cases 2 and 3, we see that case 3 transfers more energy with less surface area. This is due to a better fin configuration [as explained fully in (l)].

Thus, the bottom line is that simply relying on surface area for specifying a boiler is simplistic and can lead to wrong decisions by eliminating designs that can transfer the same duty but with a lower surface area.

Table 4. Effect of fin geometry on superheater performance (Example 3).

Case number	1	2	3	4
Duty, million Btu/h	14.14	14.18	17.43	17.39
Steam exit temperature, °F	689	691	747	747
Gas ΔP_g, in. w.c.	0.65	1.20	1.15	1.37
Gas exit temperature, °F	951	950	893	893
Fins/in.	2	5	2.5	4
Fin height, in.	0.5	0.75	0.75	0.75
Fin thickness, in.	0.075	0.075	0.075	0.075
Surface area, ft²	2,471	5,342	5,077	6,549
Max. tube wall temperature, °F	836	908	905	931
Fin tip temperature, °F	949	1,033	1,064	1,057
U_o, Btu/ft²•h•°F	11.79	5.50	8.04	6.23
Tube side ΔP, psi	9.0	6.5	11.0	9.0
Number of rows deep	6	4	7	6
Fin effectiveness, %	84	72.3	67.7	70.4

SAFETY VALVE SIZING AND *ASME* CODE

Section 1 of the American Society of Mechanical Engineers' Code (4) on boilers recommends that safety valves should be sized based on the surface area used in the boiler. This is not a prudent way of sizing safety relief valves. As we have just seen, one can transfer the same duty with variations in surface area of 50 to 100%.

It is more practical to use the steaming capacity as the criterion for relief valve sizing. For example, in gas turbine exhaust applications, one can generate anywhere from 20% to 100% of the steam simply by varying the fuel input to the burner. To illustrate, a heat recovery boiler for a typical 3-5 MW gas turbine can generate about 20,000 lb/h of steam in the unfired mode but can be designed to generate up to 100,000 Ib/h of steam with the same surface area simply by firing auxiliary fuel. That is, the same boiler with the same surface area can generate a maximum of only 30,000 lb/h at one site because the demand is only 30,000 Ib/h or 90,000 Ib/h at another location because that site needs 90,000 lb/h of steam. Now, according to ASME code, the safety valve sizing remains unaffected, as the surface area is the same!

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