Understanding Boiler Circulation

Proper arrangement of drum baffling, sizing and location of dowcomers and risers will ensure a good natural-circulation system

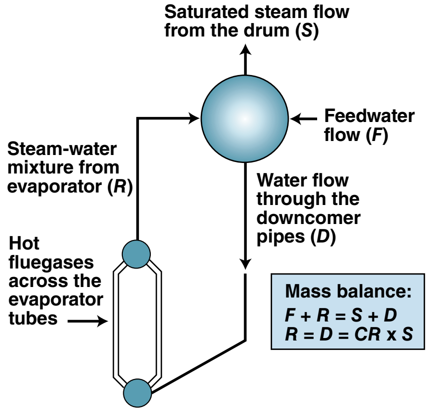

FIGURE 1. Boilers may operate using a variety of methods to circulate the steam-water mixture throughout the evaporator tubes. A typical natural-circulation system is shown here (CR = circulation ratio)

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Boilers generate steam using different methods to circulate the steam-water mixture through the evaporator tubes. These methods include natural circulation (Figures 1, 2a–2c), forced circulation (Figure 3a), and a once-through design. Figures 3b, 4 and 5 show examples of a waste-heat water tube and fire-tube boilers with external downcomers and risers. Most boiler systems have evaporator tubes heated by hot fluegases produced either from the combustion of fuels (such as oil, gas or solid fuels,) or by kilns, furnaces, gas turbines or catalytic crackers, or by other hot gas sources (such as waste-heat boilers). Boiling occurs in the evaporator tubes, and generates wet steam.

The density difference between the colder water in the downcomers and the hotter steam-water mixture in the evaporator tubes ensures the circulation of the steam-water mixture back into the drum. External downcomers, as shown in Figure 2c, are unheated, while in package boilers (Figure 2b), they are internally located and are slightly heated by the fluegases.

In the steam drum, the mixture is separated into saturated steam, which flows out of the drum, and remaining water, which mixes with the incoming feed water and flows through the downcomers to start the circulation process once again.

Circulation ratio

The ratio of the mixture that flows through the system and the amount of steam generated is called the circulation ratio (CR). CR is obtained by an iterative process, which is discussed in detail in Ref. 1;. It is a function of steam pressure and hydraulic resistance of downcomers, risers, evaporator tubes and thermal head.

Generally steam pressure and CR are inversely related, so the higher the steam pressure, the lower the CR, and vice versa. Another important parameter is the steam quality, x, exiting the evaporator, which is simply 1/CR. Thus, if CR = 8, then $x = 1/8 = 0.125$, meaning that 12.5% of steam is generated in the evaporator tubes while the rest of the mixture is water. CR may be in the range of 20 to 30 for low-pressure boilers (100 to 600 psig), and about 8 to 15 for higher-pressure units, depending on the design.

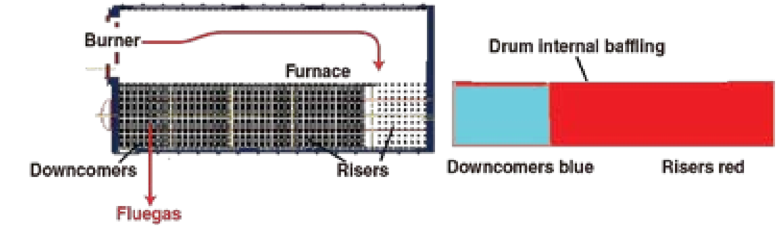
Note that CR usually represents an average circulation ratio. Keep in mind that CR varies with circuits depending on steam generation and any resistance offered by the system. In a boiler there can be several parallel circuits generating steam. For example, when hot fluegases flow across a bank of tubes, the first few rows will generate a large amount of steam due to the higher log mean temperature differential (LMTD) compared to the evaporator tubes at the cooler end. This will impact the CR.

Determining CR is only one part of the exercise. Once CR is estimated for a circuit, the engineer has to determine if the flow velocity inside the evaporator tubes will not cause separation of steam and water inside the tubes (particularly in horizontal evaporators) and must also determine if the heat flux in the circuit is low enough not to cause what is called departure from nucleate boiling (DNB) conditions.

If DNB conditions are likely, overheating and failure of the tubes can occur. Variables such as tube size, tube orientation, steam pressure, mass flow through the tubes and quality of the steam affect DNB. Charts and correlations are available for estimating the critical heat flux that can cause DNB conditions.

Natural-circulation boilers, such as package D-type boilers that are fired with oil, gas or solid fuels (Figure 2a), are widely used in the chemical process industries, petroleum refineries and power plants. Their capacity ranges from 20,000 to 300,000 lb/h with steam pressures from 100 to 1,500 psig. These are typically shop-assembled boilers in which the downcomers are inside the boiler and are heated (Note: In this design, the downcomers and risers are all inside the boiler and heated by the fluegases. However, a portion of the heated tubes at the cooler end of the boiler are designated as downcomers and are baffled inside the drum. The tubes at the hotter end are the risers).

For larger capacities, an elevated drum modular unit, such as the one shown in Figure 2c, is often more suitable, due to shipping considerations. This unit has external downcomers and risers, and all the evaporator tubes in the boiler bank are risers, unlike the smaller D-type boilers, in which some of the evaporator tubes can act as downcomers. This discussion is limited to natural-circulation units only.



FIGURES 2a (left) AND 2b (right). The baffles inside the steam drum, and downcomer tubes inside a typical package D-type boiler, are shown here

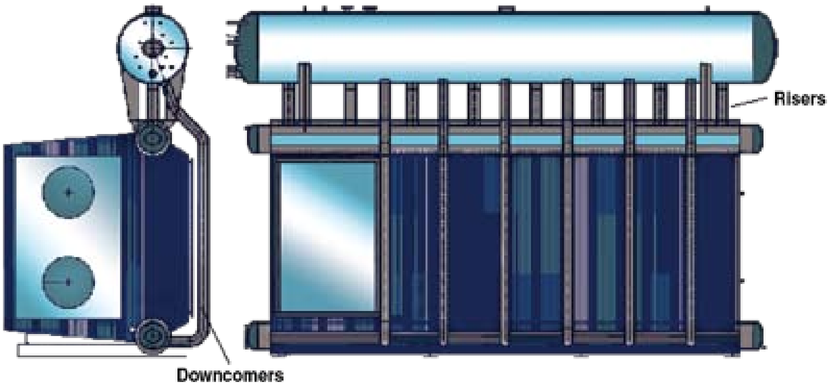


FIGURE 2c. In this D-type boiler with an external drum, downcomers and risers, feedwater from the economizer enters the steam drum and mixes with the water from the steam-water mixture from the riser tubes and flows through the eternal downcomers to the bottom of the evaporator tubes

Circulation calculations

For proper boiler operation, one has to ensure that there is adequate flow of the steam-water mixture inside the evaporator tubes to keep their tube-wall temperatures within metallurgical limits. If for some reason the flow is absent or inadequate or stagnation of flow has occurred, then the tubes can become overheated and fail. Users must carry out circulation calculations to ensure that there is proper circulation of the steam-water mixture through the evaporator. Ref. 1 describes the calculation procedure for determining CR in a natural-circulation boiler, and provides illustrative examples. Briefly, the basic steps are as follows:

Step 1. Thermal performance calculations must be carried out first, using the fuel analysis, excess air and the geometrical data of the furnace, superheater, evaporator and economizer tubes. The procedure for calculating thermal performance is also explained in Ref. 1. Typical tube-geometrty data for a D-type boiler and results for the thermal calculations can be found in Tables 1, 2 and 3, in the online version of this article (www.che.com).

Typical tube- geometry data for a D-type boiler is shown in Table 1, and results of the thermal calculations are shown in Table 2 (Note: All three tables for this article can be found in the online version of it, at www.che.com). Thermal calculations are carried out to obtain the gas-temperature distribution along the fluegas path and the energy transferred to each component in the system (such as the furnace, superheater, evaporator and economizer). The evaporator may be split into a few sections so that the energy transferred and steam generation in each section may be obtained. These data will be useful for carrying out the circulation-related calculations.

Step 2. Once the calculations in Step 1 are complete, a CR is assumed. The steam quality at the exit of the evaporator is then known, as well as the mass of steam-water mixture quantity flowing through the downcomer-evaporator-riser system, which is the product of CR and steam generated. If we assume a CR of 10 to start, and the steam generated in the boiler is 90,000 lb/h, then 900,000 lb/h of mixture flows through the downcomers, evaporators and risers, and the quality of steam at the exit of risers is 0.1.

Step 3. An energy balance is then carried out at the drum, to estimate the enthalpy of the water entering the downcomer tubes. The density of the water is computed. Note that the density difference between the downcomer water and steam-water mixture in the riser is responsible for circulation.

In this example, the enthalpy of the steam-water mixture at the evaporator exit is: 0.1 × 1,203 + 0.9 × 474.8 = 547.6 Btu/lb, where 1,203 and 474.8 Btu/lb are the enthalpies of saturated steam (h_v) and water (h_{fw}), which are obtained from the steam tables. If the feedwater enters the drum at, say, 350°F, enthalpy = h_{fw} = 322.5 Btu/lb, then the heat balance in the drum is calculated as follows:

$$h_{fw} + h_e \times CR = h_v + CR \times h_d \qquad (1)$$

where:

h_{fw} = enthalpy of the feedwater, Btu/lb

h_e = the enthalpy of steam-water mixture leaving the evaporator, Btu/lb

h_d = the enthalpy of the steam-water mixture entering the downcomers, Btu/lb

h_v = enthalpy of saturated steam leaving the drum, Btu/lb

Note: In the above equation, the steam generation is taken as unity and thus will not affect the energy balance:

900,000 × 547.6 + 90,000 × 322.5 =
900,000 h_d + 90,000 × 1,203 =
h_d = 459.5 Btu/lb

With h_d = 459.5 Btu/lb, this corresponds to a water temperature of 476°F. This water will pick up additional energy in the heated downcomers. The downcomers are located in the cooler gas region of the evaporator section (where the energy pickup from the fluegas is not high). Thus, the downcomer water temperature is cooler than that of the mixture flowing in the riser and has a higher density, which forces the two-phase mixture through the evaporator tubes.

Nearly 95% of the energy from the fluegases is absorbed in the first 70–75% of the heating surface in the evaporator. As the last few rows do not absorb much energy, the water temperature in the downcomer tubes is cooler and the water density is higher than that of the hotter steam-water mixture in the riser tubes and this ensures the circulation process.

Step 4. Sizes, developed lengths and the number of bends of downcomer tubes and evaporator tubes are obtained from

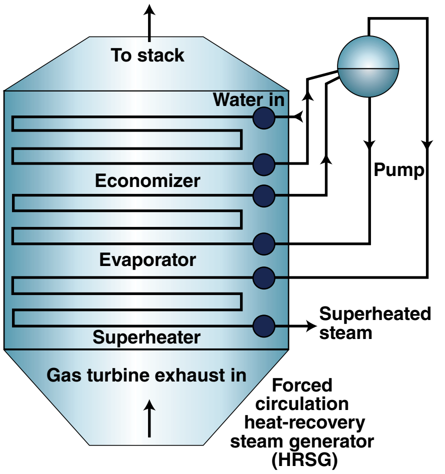


FIGURE 3a (above). **In this forced-circulation boiler (generally seen with horizontal evaporator tubes), a circulation pump is used to suck water from the drum and ensure its flow through the evaporator tubes and back into the drum. The pump capacity determines the CR, which can vary from 3 to 8, depending on the designer's choice**

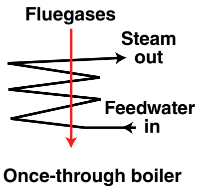


FIGURE 3b (right). **In this once-through boiler, there is no circulation system, so CR = 1. Water enters at one end and flows out as steam at the boiler exit. There are different considerations for designing this type of boiler, but that is beyond the scope of this article**

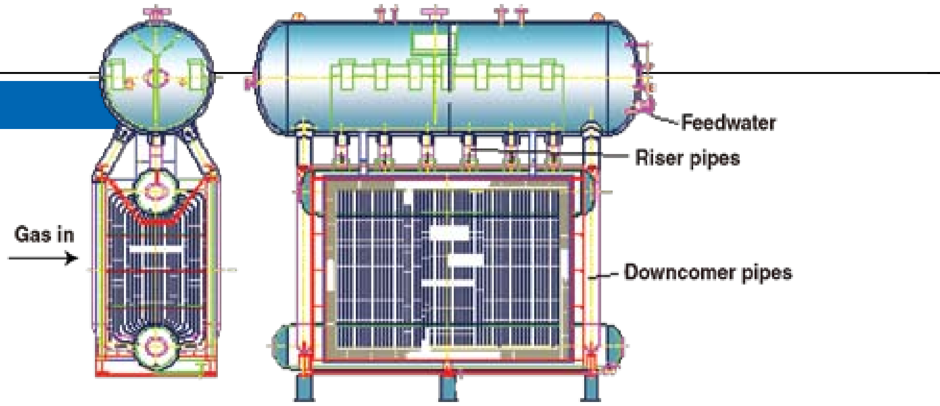


FIGURE 4. **In this waste-heat boiler with external downcomers and risers, feed water enters the drum from the boiler feed pump or from an economizer. The water mixes with the water from the evaporator tubes and the mixture flows down the downcomer pipes to the bottom of the evaporator tubes. The CR, established by an iterative process, can vary from 5 to 40, depending on a variety of variables**

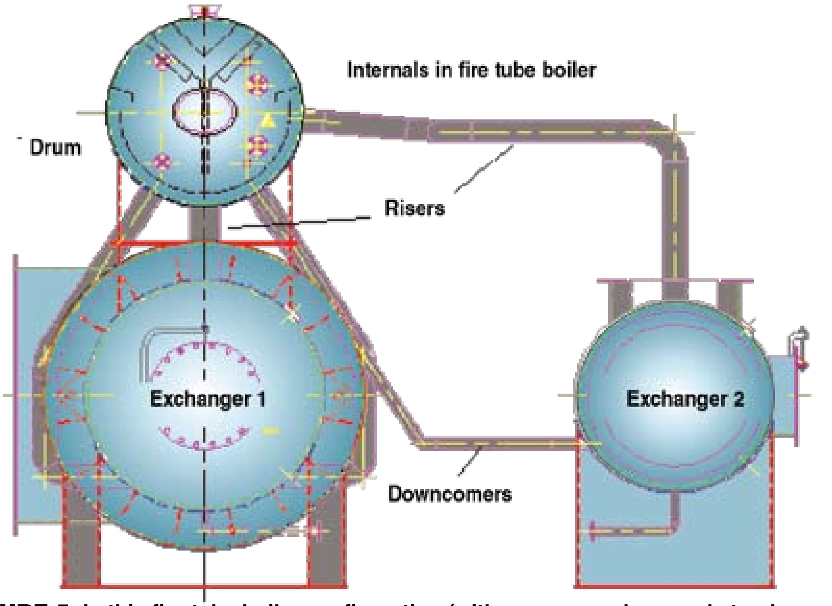


FIGURE 5. **In this fire-tube boiler configuration (with a common drum and standalone downcomers and risers), the feedwater is admitted into the drum, where it mixes with the hot water from the risers and flows through the external downcomer tubes**

the boiler drawings from which the various hydraulic losses in the circuits can be computed. These include:

- Thermal head available for circulation (this depends on the location of the drum)
- Friction loss in the downcomer tubes (this is due to the single-phase flow)
- Losses in evaporator tubes (this consists of friction losses, acceleration loss due to two-phase flow, and gravity loss due to varying quality along the evaporator height)
- Losses in the drum internals
- Losses in riser tubes in cases where external riser pipes are used

Charts for the calculation of various two-phase losses are available in Ref. 1. The total losses are matched with the available thermal head. If they match, then the CR assumed is correct. If they do not match, then another iteration is carried out to obtain the CR at which the available head matches the various losses. A computer program is usually used for these calculations.

If the CR is relatively low (say, in the single digits) and if the steam pressure is relatively high (say 1,500 psig or above), then DNB checks[1] may be carried out after obtaining the heat flux in each region. If the CR is low — for instance, in the single digits, such as 5–6, as in the case of high-pressure boilers operating above 1,500 psig — then efforts may be taken to revise the tube sizes of the risers and downcomers.

The actual heat flux in the evaporator tubes can be calculated from the thermal calculations results done earlier, using this relationship:

Actual heat flux = overall heat transfer coefficient × (the gas temperature – the saturated steam temperature)

One should ensure that the actual heat flux is far less than the allowable critical heat flux all along the evaporator tubes. Hundreds of correlations are available in the literature to determine the allowable critical heat flux.

The Macbeth correlation provided below in Equation (2) shows the relationship among the several variables and may give a high allowable flux. Using this correlation, two heat flux values are computed. One is the actual heat flux inside the tubes (determined from thermal calculations done by the boiler designer, based on the boiler fluegas velocity, heat transfer coefficient and tube geometry used). Then the allowable heat flux is estimated from charts or correlations available in the literature (based on the CR computed and the tube geometry, flow inside the evaporator tube and steam pressure.) One should ensure that the actual heat flux is far lower — for instance,

1. DNB checks ensure that the boiling process in the evaporator tubes is nucleate and not film boiling. Charts and correlations are available to do this evaluation, which is required if it is felt that the CR is low, as discussed in the text.

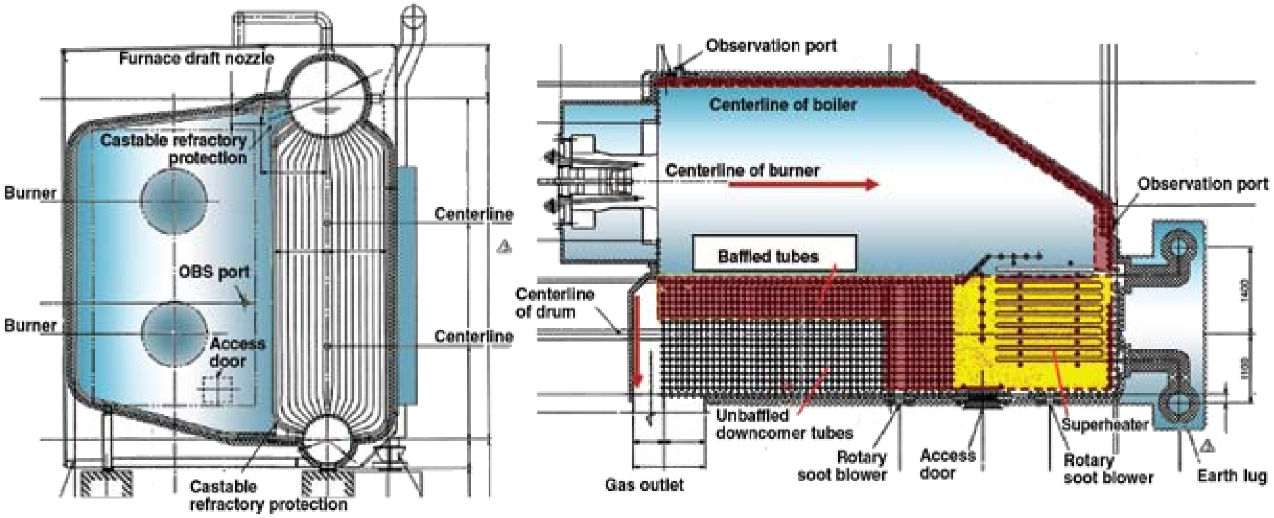


FIGURE 6a (top left) and b (top right). Shown is a typical arrangement of the boiler arrangement of a package D-type boiler with circulation problems. On the drum-baffling arrangment shown in Figure 6b, note the asymmetrical baffling with respect to gas-flow direction. This baffling system made it hard for many designated downcomer tubes to act freely as downcomers

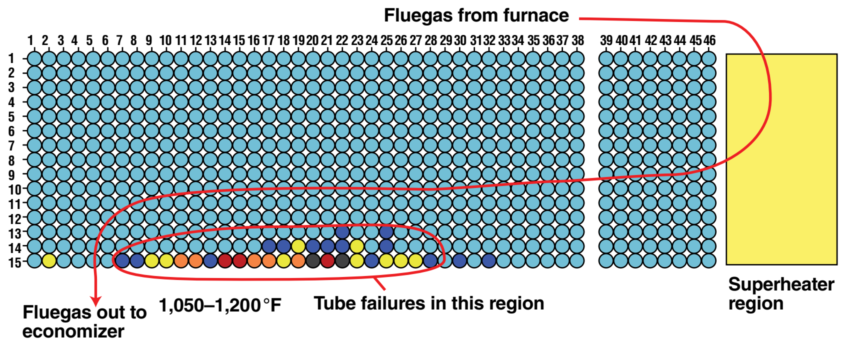


FIGURE 6c. This plan view shows the region where the tubes failed in the convection bank

at least 20–30% less than the allowable rate — to ensure that departure from nucleate boiler does not occur in the evaporator tubes. Boiler companies typically develop their own correlations based on their experience, the tube sizes and configuration used, and they routinely use safety margins when developing them.

$$q = 6{,}330\, h_{fg}\, d^{-0.1} (G/10^6)^{0.51} (1-x) \quad \textbf{(2)}$$

where:

q = critical heat flux, Btu/ft^{2}h
d = tube inner dia., in.
G = mass velocity of steam water mixture through tubes, lb/ft^{2}h
x = steam quality, fraction

Circulation issue, D-type boiler

Figures 6a and 6b show a boiler, fired with oil and refinery gas, generating 130,000 lb/h of superheated steam at 630 psig, and 750°F, which had an interesting problem. Tubes were thinning and failing at a location in the boiler bank shown in Figure 6b. There were two identical units in this plant and both were having this problem. Engineers were wondering why this region alone was facing this problem — not the hotter zone ahead of these tubes, or the tubes at the other end of the same cross-section (that is, in the same plane perpendicular to the gas flow direction), where one would expect the same gas temperature. As a consultant, I was asked to evaluate the design and suggest suitable solutions.

Analysis. The first step was to simulate the boiler performance using the tube geometry and furnace dimensions provided, to see if the exit gas temperature, superheated steam temperature, and water temperature leaving the economizer were all close to the measured field data. This step was designed to confirm that the model used to carry out the calculations of gas temperatures, overall heat transfer coefficients and actual heat flux were reasonable, and that the boiler was properly sized, as that may provide some indication of other issues such as fouling. The simulation gave the exit gas temperature from the evaporator and economizer as 798°F and 388°F, respectively, which matched the field data. Hence the simulation program results were used to study the circulation issue.

The evaporator section was broken up into four sections and the gas temperature, duty and steam generation in each section was calculated. It was found that the gas temperatures in the region where tubes were failing were ranging from 1,050°F to 1,200°F, as seen in Table 1b (found in the online version of this article, at www.che.com). Circulation calculations were carried out using these data. Typical results are shown in Table 2 (online). The heat flux q in the hottest gas zone of 1,300°F, given by Equation (3):

$$U(T_g - T_s)A_o/A_i = 9.5 \times (1{,}300-498) \times 2.5/2.24 = 8{,}503 \text{ Btu/ft}^2\text{h} \quad \textbf{(3)}$$

where:

q = heat flux, Btu/ft^{2}h
U = overall heat transfer coefficient, Btu/ft^{2}hF
T_g/T_s = gas and steam temperatures, °F
A_o, A_i = the tube outer and inner surface area, ft^2

This value of U is not high enough to cause DNB in normal boiling situations. Heat fluxes have to be in the range of 150 to 200,000 Btu/ft^{2}h before we can attribute DNB to the tube-failure problems in natural-circulation boilers with vertical tubes at this pressure.

Next, the drum internal arrangement was reviewed (Figure 6a). This diagram provided a clue to the problems that the plant was experiencing. In a typical D-type boiler with fluegases flowing parallel to the drum over the boiler bank tubes, the drum baffling is designed as shown in Figure 2b. This will ensure that all the risers are in the hot gas section and downcomers are in the cooler gas section. Also, this type of baffling ensures that the tubes in any cross-section are acting either as risers or downcomers. In some large boilers, it may be necessary to have the drum at a separate location and feed the boiler evaporator tubes using an external downcomer system and collect the steam from the evaporator using an external riser system as shown in Figure 2c.

Figure 4 shows a waste-heat boiler with external downcomers and risers. The layout can vary depending on the boiler design adopted.

In a typical boiler, the tubes perpendicular to gas flow direction are symmetrically baffled. However in this boiler, by using this non-symmetrical baffling system, a portion of the evaporator tubes in the hot gas section in the same cross-section are forced to be downcomers, while some tubes in the same cross-section are under the baffle acting as risers. This was determined to be the source of the problem.

The fluegas temperature in the first row of downcomers is close to 1,400°F and the intense bubbling inside these tubes will force these tubes to act as risers, even though they are baffled as downcomers, due to the high heat transfer rate and the lower density of the steam-water mixture inside the tubes. Hence, these tubes — though not baffled — will still function as risers. Flow will continue through these tubes and circulation will be good.

In essence, these tubes have not failed, even though they are in the hot gas region. However, as the gas cools in the next nine rows, the gas temperature drops to about 1,200°F. This is the region where tubes at the far end of the furnace have failed.

These failed tubes have difficulty acting as downcomers due to the high gas temperatures; vapor formation at the inlet to the tube will still be intense and thus will prevent the flow of water in the downcomers. Also, the head available in the colder end of the boiler is not high enough to force the steam-water mixture through the tubes, since they have longer lengths and hence greater resistance to flow. The thermal head at the cool downcomer end is nearly the same as in these hotter tubes, and hence there is difficulty in forcing the steam-water mixture through these tubes.

Also, if you look at the typical gas-velocity profile in any cross section, it will be as shown in Figure 7, with the tubes in the middle receiving higher energy transfer compared to the end tubes. Thus, the end tubes that have failed do not generate sufficient steam-water mixture for the cold downcomer tubes at the rear to force the circulation of the steam-water mixture through these tubes. Simply put — these failed tubes are neither acting as good risers nor as good downcomers, and thus a stagnant vapor mixture may have formed in the tubes.

The vapor heat-transfer coefficient is very small compared to the two-phase boiling coefficient and hence the tube wall temperature will reach fluegas temperature, as there is not much of cooling inside these tubes due to the stagnant column of vapor in this region. Measurement of tube-wall temperatures near the failed region also confirmed the high temperatures.

The tubes within the same cross-section in the riser section (under the baffles and closer to the furnace) face no problem, as the head available for circulation is higher in baffled regions. The normal water level in the drum is typically at the drum center line and is at a higher elevation than the baffled section, so this ensures circulation. Hence, these tubes near the furnace act as risers. The mid-section is likely to have a higher steam formation than the ends due to the velocity profile and higher heat flux and hence have lower mixture density resulting in some circulation through these tubes.

Solution. The failed tubes have difficulty acting either as risers or downcomers due to the skewed baffling inside the drum. This is mainly due to the fact that many so-called downcomers are located in a very hot section. Stagnation of the flow is suspected in the failed tubes. If baffling inside the drum is done as shown in Figure 2a, the evaporator tubes under the baffles in the hot gas zone will be forced to act as risers and the cooler unbaffled section where the gas temperature drops to less than 900°F will act as downcomers, which will ensure good circulation of the steam-water mixture through the evaporator tubes. This solution is currently being implemented at the facility.

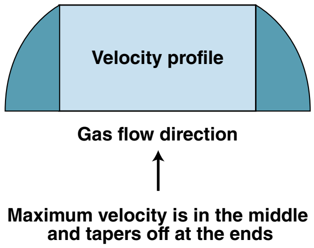


FIGURE 7. Shown here is a typical gas-velocity profile in a boiler bank cross-section. Tubes in the mid-section usually operate at higher heat flux compared to end section

Final thoughts

Before buying a boiler plant, engineers should review the boiler thermal performance, including circulation issues. Many plant engineers review information such as construction details, painting, duct thickness, structural integrity, code documents and so on, but few review the boiler's thermal performance calculations, drum-baffling system details or thermal performance and circulation issues. However, engineers should review these things before buying a new boiler plant, using either in-house expertise or third-party consultants. Getting a second opinion on thermal, process and performance issues can help the facility to purchase a better boiler and thus avoid unnecessary plant shutdowns and costly modifications later — an important consideration, given the fact that a boiler has a life of over thirty years. ■

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TABLE 1. GEOMETRIC DATA USED FOR THE BOILER							
Geometry	Suph	Evap	Evap	Evap	Evap	Evap	Econ
Tube O.D, in.	2.000	2.50	2.50	2.50	2.50	2.50	1.50
Tube I.D, in.	1.550	2.240	2.240	2.240	2.240	2.240	1.240
Tubes/row	31	16.00	16.00	16.00	16.00	16.00	34.00
Number of rows deep	18	9	9	9	9	10	60.00
Length, ft	6.000	15.00	15.00	15.00	15.00	15.00	10.00
Transverse spacing, in.	4.250	4.33	4.33	4.33	4.33	4.33	2.36
Longitudinal spacing, in.	4.000	4.50	4.50	4.50	4.50	4.50	2.30
Streams	31	0.00	0.00	0.00	0.00	0.00	34
C=counterflow P= parallel flow	C	P	P	P	P	P	C

Furnace dimensions: Length = 25 ft, height = 14 ft and width = 13 ft.
O.D. = outer diameter, I.D. = inner diameter, Suph = Superheater,
Evap = Evaporator, Econ = Economizer, Number of rows deep = Number of tubes along gas-flow direction, Traverse spacing = Distance between tube centers perpendicular to gas-flow direction, Longitudinal spacing= Distance between tube centers along gas-flow direction

TABLE 2. SUMMARY OF THERMAL PERFORMANCE							
Gas flow, lb/h	156,109	Combustion temperature, °F	3,439	Gas mol. wt.	29.16	Gas pressure, psia	14.50
Drum pressure, psig	655	Saturation temp., °F	498	Blow down-%	3.0	Furnace duty, million Btu/h	66.63
Design pressure, psig	775	Excess air,%	15	Feed-water temp., °F	298	Steam-lb/h	133,000
Fuel	Oil/gas						
Component	Suph	Evap	Evap	Evap	Evap	Evap	Econ
Gas temperature, inlet, ±10°F	2,162	1,698	1,398	1,181	1,020	899	798
Gas temperature, outlet,-±10°F	1,698	1,398	1,181	1,020	899	798	388
Gas specific heat, Btu/lb°F	0.3131	0.3028	0.2949	0.2884	0.2834	0.2793	0.2698
U,Btu/ft^{2}h°F	9.91	9.59	8.96	8.52	8.2	7.96	10.69
Surface area, ft^2	1752	1414	1414	1414	1414	1571	8011
Duty, million Btu/h	22.70	14.15	9.97	7.21	5.32	4.35	17.25
Steam generation, lb/h	133,000	19,500	13,700	10,000	7,400	Down-comer	Feed-water
Steam temp., inlet, °F	498	418	418	418	418	418	298
Steam temp., outlet, °F	742	498	498	498	498	498	418

(Furnace generates about 82,400 lb/h of steam)
Suph = Superheater, Evap = Evaporator, Econ = Economizer,
U = The overall heat transfer coefficient required for circulation of heat flux

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TABLE 3. CIRCULATION CALCULATIONS					
Steam pressure, psia	665	Water specific volume, ft^3/lb	0.02	Riser friction losses, psi	0.18
Saturated steam temperature, °F	497	Average CR for the entire system	21.5	Riser gravity, psi	0.377
Feedwater temperature	418	Total static head available, ft	18	Drum internals loss, psi	1
Total steam flow, lb/h	133,100	Total static head available, psia	6.204	Heated tubes loss, psi	4.333
Steam specific volume, ft^3/lb	0.692	Downcomer losses, psi	0.314	Mixture temperature from drum, °F	493

Downcomer flows, lb/h

Level	Stream	Inner dia., in.	Length, ft	Bends	Flow, lb/h	Velocity, ft/s	Path	
1	160	2.24	17	2	2,854,995	3.6	1	

Heated tubes/riser sections

Riser number	Steam flow, lb/h	Quality, %	Acceleration	Gravity	Friction	Mixture flow, lb/h	Number of tubes	Tube inner dia., in.	Boiling height, ft.	Developed length, ft	CR	Swaged dia., in.
			← Losses, psi →									
1	82,500	0.049	0.028	4.124	0.181	1,690,574	140	2.7	16	40	20.5	2.7
2	19,500	0.04	0.004	4.318	0.02	482,673	144	2.24	16	16	24.8	2.24
3	13,700	0.039	0.002	4.348	0.01	349,490	144	2.24	16	16	25.5	2.24
4	10,000	0.039	0.001	4.348	0.005	255,102	144	2.24	16	16	25.5	2.24
5	7,400	0.039	0.001	4.348	0.003	188,776	144	2.24	16	16	25.5	2.24