In a CFB, a primary solids collector is located immediately after the furnace and is designed to recycle all of the captured solids. As a result, sulfur capture and combustion efficiency are improved over bubbling beds. Some designs also include a secondary collector after the convection pass which further enhances sulfur capture and combustion efficiency by increasing the concentration of fine particles in the furnace. These improvements result from longer solids-gas contact times and higher specific surface of the fine particles in contact with the gas.

SO_2 reductions of 90% are typically achieved in a circulating bed with calcium to sulfur (Ca/S) mole ratios of 2 to 2.5, depending on the sulfur content of the fuel and the reactivity of the limestone. Slightly higher ratios are required in bubbling beds. The lower the sulfur concentration in the fuel the greater the calcium to sulfur mole ratio must be for a given SO_2 removal level.

For removal requirements greater than 90%, the amount of limestone needed increases rapidly and alternative SO_2 removal methods, such as conventional pulverized coal-fired boilers with scrubbers, may become the economic choice. (See Chapter 35.)

Nitrogen oxides

The nitrogen oxides that are present in the flue gas come from two sources, the oxidation of nitrogen compounds in the fuel and the reaction between nitrogen and oxygen in the combustion air. It is common to refer to the nitrogen oxides in the flue gas as NO_x, where the subscript x implies the presence of several different nitrogen/oxygen compounds. The NO_x formed by oxidizing fuel nitrogen compounds is referred to as *fuel* NO_x and the NO_x formed from nitrogen and oxygen in the combustion air is called *thermal* NO_x, because it is a product of a high temperature process above 2700F (1482C).

As a result of the low temperatures at which a fluidized bed operates, thermal NO_x makes only a minor contribution to overall emissions. A fluidized-bed boiler can also suppress the amount of fuel NO_x formed. This is accomplished by putting less than the theoretical amount of combustion air through the distributor plate and adding the remainder of the combustion air to the furnace above the dense bed. As a result, some of the fuel nitrogen compounds decompose into molecular nitrogen rather than forming NO_x. This process is referred to as *staged combustion* and is also used with other firing methods to reduce NO_x emissions.

Staged combustion is used for bubbling beds that do not contain in-bed surface and for circulating beds. However, it is not used for designs where cooling tube bundles are submerged in the bed. The reason is corrosion. By burning the fuel with less than theoretical air (substoichiometrically) the combustion gas contains many reduced chemical species that cause rapid metal loss from the furnace cooling tubes. When staged combustion is used, the furnace tube walls in the reducing zone are protected with a thin layer of refractory.

The combination of low temperatures and staged combustion permits fluidized-bed boilers to operate with significantly lower NO_x emissions. Typical values for NO_x emissions are within the 100 to 200 ppmdv (parts per million dry volume) range for a CFB boiler burning coal.

Carbon monoxide and hydrocarbons

When designing a boiler, it is necessary to maximize combustion efficiency by minimizing unburned carbon and the quantity of CO and hydrocarbons in the flue gas. This is done by choosing the proper number of fuel feed points, by proper design of the overfire air system, if used, and by providing sufficient furnace residence time for mixing and complete combustion (burnout).

Typical flue gas concentrations are less than 200 ppmdv for CO and 20 ppmdv for hydrocarbons in a CFB boiler burning coal.

Particulates

The ash contained in solid fuel is released during the combustion process. Some of this ash remains in the fluidized bed and is discharged by the bed material removal or *drain* system. This ash is normally larger than 140 mesh (105 microns) and is easy to handle and transport. The remaining ash leaves the boiler in the flue gas. This material is typically less than 325 mesh (44 microns) and requires a high efficiency collection device. Normally, a fabric filter is used with atmospheric pressure fluidized-bed boilers because it is less sensitive to the ash properties, such as size, concentration and resistivity, than electrostatic precipitators (ESP). (See Chapter 33 for a discussion of particulate collection.)

Fluidized-bed boiler furnace design

Numerous factors and parameters affect fluidized-bed furnace design. Some are specified by the boiler owner/operator. Many are selected by the designer and can only be derived from empirical data. Some are interrelated, making the design task more challenging.

The following list is typical of the type of information used when establishing the design of a fluidized-bed boiler.

Owner specified

1. unit capacity — steam flow requirements,
2. fuel — type, ash and moisture content, size, reactivity, chemical analyses, attrition characteristics, and fouling and sintering properties,
3. limestone — type, reactivity, size and attrition characteristics,
4. sulfur capture requirements,
5. NO_x emission limits, and
6. load turndown range.

Designer specified

1. type of fuel feed system — in-bed, under-bed or above-bed,
2. number and location of fuel feed points,
3. fuel combustion efficiency,
4. number and location of sorbent injection points,
5. primary/secondary air split when used and the location of the overfire air nozzles,
6. bed operating temperature,
7. bed operating velocity,
8. size of bed particles,
9. amount of solids that leaves through the bed drain and the amount that leaves with combustion gas to the final solids collector, and
10. amount, temperature and location of solids recycled to the furnace from particle separators and bed drain classifiers.

The following discusses some of the major areas involved with the functional design of fluidized-bed furnaces. The design of the convection pass surface is similar to a conventional boiler design process as discussed in Chapter 18.

Importance of bed material

Combustion and inventory requirements For a fluidized bed to operate properly it must be continuously supplied with a sufficient quantity of particles of the proper size distribution. If the particles are too coarse, the bed will defluidize and become fixed. If the particles are too fine, they will blow out of the furnace making it impossible to maintain an adequate bed inventory. There is a range of bed particle size which is needed to maintain a stable fluidized-bed process. The supply and retention of these particles must be controlled to provide the required inventory.

In the case of a bubbling bed, it is easy to visualize that a properly fluidized bed must be present to receive, suspend, mix and burn the fuel particles. If the bed density is too lean or if the bed is too shallow, an accumulation of fuel particles is possible in a localized area of the bed. This may lead to high temperature zones where the fuel and ash will turn into a sintered mass. At the other extreme, if too many large particles are fed into the bed it will defluidize with a similar result.

In circulating beds, even though the solids inventory is distributed over the furnace height, a dense bed is still required in the lower furnace to support and mix the fuel during combustion to avoid the problems just mentioned.

Normally, when coal is burned, most of the ash is liberated from the burning pieces as fine particles. These are so fine that they quickly blow out of the bed and are carried out of the furnace by the flue gas.

In bubbling beds, this material does not contribute to bed inventory. In circulating beds, this material is caught and returned to the furnace where it becomes part of the circulating mass. However, the fines do not make a significant contribution to the dense bed itself.

Because of the wide variation in fuel ash properties, the ash is not usually depended upon to form a stable bed, and a second inert material of the desired size distribution such as sand is added to the system. When SO_2 capture is required, limestone replaces the sand to achieve in-bed SO_2 capture.

Ideally, the size of the sorbent fed to the boiler would simply be that required to form a stable bed. However, in the process of heating, calcining and sulfating, the sorbent's size and physical properties change. The final size of the sorbent-derived bed material, after these chemical and physical changes, can not, in many cases, be reliably predicted. In addition, some limestones are softer than others and wear faster. For these reasons, limestone characteristics and some trial and error testing during initial operation are needed to establish the proper limestone feed size and makeup flow rate.

The above discussion assumes a typical bituminous coal. In the case of low ash fuels like wood, sand is used for the bed. While sand properties are important, sand does not break down as rapidly as limestone, its makeup rates are lower and the size of the bed material is more predictable.

High ash waste coals require special consideration. Typically, these fuels contain large amounts of ash not tied to organic matter. This ash frequently consists of rock that has been generated by coal beneficiation and is called *tramp ash*. Because tramp ash does not break down into fine particles, it forms a large percentage of the bed material. For this reason, the size of the fuel feed must be chosen carefully so that the ash complements the bed material rather than causing fluidization problems. There are fluidized-bed boilers in operation where the size and consistency of the fuel ash will form a stable bed without the need for additional bed material makeup.

Bed drain classification systems are often used in circulating-bed units in addition to proper fuel and/or sorbent sizing to control the size and inventory of the bed material. These systems help to remove oversized noncirculating material while maintaining the required inventory of circulating particles.

Particle characterization and measurement In fluidized-bed boiler furnaces, particle motion is affected by the combined influence of gravitational and aerodynamic forces, as well as the impact with other particles and the walls of the boiler.

For an individual particle, the three most important characteristics are its size, density and shape. Particles of the same size but with different densities or with the same density but varying sizes do not act the same. The particle's shape, from spherical to flat, determines how it will react to the forces present in the furnace.

From an analytical standpoint, the ideal particle is a homogenous sphere and the ideal mixture consists of many homogenous spheres of equal diameter. In practice, mixtures that consist of a variety of particles with different sizes, densities and shapes are encountered.

Fluidized-bed heat transfer and pressure drop calculations assume that the mixtures of particles can be characterized by a mean particle diameter, a mean particle density and a mixture bulk density. The formulas for calculating mean diameters of mixtures assume spherical particles, and a sphericity correction term is applied to the calculated mean diameter to approximate the influence of the nonspherical nature of the actual material. Mixtures containing a significant quantity of particles that could be described as rods or flakes are difficult to characterize.

The equations that are used to calculate mean diameters are based on the weight fractions of incremental size groups. The first step of the determination is sieving a representative sample of the mixture through a stack of screens having a successively finer mesh and weighing the amount of solids retained on each screen. For calculation purposes, the average size of the particles is assumed to be the average of the mesh opening of the screen the particles were retained on and the next larger screen through which they all passed.

Two characteristic diameters are used in fluid-bed work, the weight mean diameter and the Sauter mean diameter. The Sauter mean diameter is also called the volume-surface mean diameter or the harmonic mean diameter. The particle size is typically given in microns. The weight mean diameter is calculated from the weight fraction of particles in each size cut as follows:

$$D_{WM} = D_1 X_1 + D_2 X_2 + \ldots + D_N X_N \qquad (1)$$

which can be rewritten as:

$$D_{WM} = \sum_{1}^{N} D_N X_N \quad (2)$$

where

D_{WM} = weight mean particle diameter, microns
D_1 to D_N = average diameter of first to last size cut, microns
X_1 to X_N = weight fraction of first to last size cut

The Sauter mean diameter for a mixture of particles is calculated from the ratio of the average volume to the average surface area. This diameter is used when predicting the hydrodynamic performance of particle mixtures. This is calculated from the weight fractions of particle mixtures using the following equation:

$$D_{VS} = \frac{X_1 + X_2 + \ldots + X_N}{\frac{X_1}{D_1} + \frac{X_2}{D_2} + \ldots + \frac{X_N}{D_N}} = \frac{1}{\sum_{1}^{N} \frac{X_N}{D_N}} \quad (3)$$

where

D_{VS} = Sauter mean particle diameter, microns

When all the particles have the same diameter, the weight mean and the Sauter mean diameters are equal. For mixtures with a fairly narrow size distribution of particles, the weight mean and Sauter mean diameters will be similar with the Sauter diameter being smaller. For particle mixtures with a wide range of diameters, the Sauter mean diameter will be considerably smaller than the weight mean diameter.

Pressure loss

Pressure loss in a fluidized-bed boiler furnace is of great importance because it defines the total amount of solids inventory in the furnace, which is a major variable that influences heat transfer. Because the concentration of solids and the pressure profile in a fluidized-bed furnace are closely related, the determination of the pressure loss is a primary task when establishing furnace performance.

Bubbling bed In the case of bubbling-bed boilers, pressure loss is of special interest only in the bed itself. For the remainder of the boiler, the pressure loss is calculated by conventional boiler furnace fluid flow equations. (See Chapter 21.) Fig. 5 shows the furnace density profile and identifies the different zones used in establishing pressure drop and heat transfer.

The following equation is used to calculate the dense bed pressure drop in a bubbling bed.

$$\Delta P = (C)(1-e)(\rho_s - \rho_g)(L) \quad (4)$$

where

ΔP = pressure loss
C = units conversion constant
e = bed void fraction
L = bed height
ρ_s = particle density
ρ_g = gas density at bed conditions

e is primarily a function of particle size, particle density, bed gas velocity and gas viscosity.

Various methods are used to predict bed voidage, including those proposed by Leva,[1] Babu, *et al.*,[2] and Staub and Canada.[3]

Circulating bed The density profile of a CFB boiler furnace is more complex than in a bubbling bed. It is normal practice to establish a dense bed, bubbling or turbulent, in the bottom of the furnace. This is achieved by staging the admission of air to the furnace and supplying between 50 and 70% of the total air flow to the distributor plate. This reduces the gas velocity in the primary zone below the overfire air ports and allows the maintenance of the dense bed with comparatively low solids recirculation rates. Injection of overfire air converts the solids-gas flow regime to a circulating bed above the overfire air ports. The upward flow of solids decreases with increased furnace height, resulting in a reduction of local furnace density. Fig. 7 shows the density profile and the location of the various density zones in the furnace.

Pressure loss in a CFB furnace conforms to the basic equation below:

$$\Delta P = (C)(\rho_b)(L) \quad (5)$$

where

ΔP = pressure loss
C = units conversion constant
ρ_b = average bulk density in the furnace or furnace section associated with L
L = height of the furnace or furnace part of interest

To use the equation above, a density profile as shown in Fig. 7 is developed. This curve is a function of many variables and has been derived from empirical data. The important variables are:

D_p = average particle size above the dense bed zone
D_{DB} = average particle size in the dense bed zone
V = nominal gas velocity
T = nominal furnace temperature
W_S = external solids flux, lb/h ft^2 (kg/s m^2)
ρ_s = particle density
ϕ = particle shape factor
D_e = furnace equivalent diameter

Because the mixture bulk density in a furnace varies significantly with height, the furnace is divided into a number of zones (see Fig. 7 for definitions): dense bed, disengaging, transition and freeboard, wherein an average density value is calculated for each based on experimental data. The pressure loss equation, above, is then applied to each zone and summed to get the total furnace pressure loss.

Heat transfer

In conventional furnaces, the combustion gas carries a portion of the fuel ash with it as it goes through and out of the furnace. In general, this fuel ash represents less than 10 lb (4.54 kg) of inert solids per 1000 lb (454 kg) of gas. Also, the heat transfer from the gas to the furnace enclosure walls is predominately by radiation.

In a circulating fluidized-bed furnace, the amount of solids in the gas leaving the furnace may exceed 5000 lb (2268 kg) of solids per 1000 lb (454 kg) of gas. As a result of this high solids content, additional heat transfer mechanisms must be considered in the design. Heat transfer to the in-bed tubes of a bubbling bed and to the walls of a circulating bed includes solids and gas convection and solids and gas radiation in decreasing order of importance. In a conventional boiler furnace, gas radiation is the most important and solids convection is the least important to overall heat transfer.

The influence of the high solids concentration is significant. For equal temperatures, the heat transfer coefficients in a fluidized-bed boiler furnace are considerably higher than those in a conventional furnace. However, because the temperatures in the fluidized bed are between 1500 and 1600F (816 and 871C) the overall heat fluxes in the two systems are similar. Typical values for overall heat transfer coefficients in a fluidized-bed boiler furnace vary between 15 and 60 Btu/h ft^2 F (85 and 341 W/m^2 K).

Bubbling-bed heat transfer For heat transfer purposes, a bubbling-bed boiler is divided into three zones: bubbling bed or dense bed, disengaging, and upper furnace or freeboard. (See Fig. 5.)

Dense-bed heat transfer to tube banks The equation for the overall heat transfer coefficient for any tube is:

$$U_o = \frac{1}{\dfrac{1}{h_c + h_r} + R_m + R_{ft}} \quad (6)$$

where

U_o = overall heat transfer coefficient, Btu/h ft^2 F (W/m^2 K)
h_c = convection heat transfer coefficient for the tube bank, Btu/h ft^2 F (W/m^2 K)
h_r = radiation heat transfer coefficient for the tube bank and walls, Btu/h ft^2 F (W/m^2 K)
R_m = metal wall resistance, h ft^2 F/Btu (m^2 K/W)
R_{ft} = tube fluid film resistance, h ft^2 F/Btu (m^2 K/W)

The convection heat transfer coefficient h_c is given by Equation 7. Two single tube equations that are used are shown as follows. Equation 8 is a modified Vreedenberg form[4] and applies primarily to beds with particles less than 800 micron average. Equation 9, a Glicksman-Decker[5] type, applies well when the average particle size in the bed exceeds 800 microns.

$$h_c = (h_{st})(FAB) \quad (7)$$

$$h_{st} = 900\,(1-e)\left(\frac{k}{d_t}\right)\left[\left(\frac{G d_t \rho_s}{\rho_g \mu}\right)\left(\frac{\mu^2}{D_p^{\,3}\rho_s^{\,2} g}\right)\right]^{0.326} (\mathrm{Pr})^{0.3} \quad (8)$$

for $D_p < 800$ microns

$$h_{st} = \frac{k\,(1-e)}{D_p}\left[C_1 + (C_2)\left(\frac{3600\, D_p\, \rho_g C_p\, V}{k}\right)\right] \quad (9)$$

for $D_p > 800$ microns

where

h_{st} = convection heat transfer coefficient for a single tube, Btu/h ft^2 F
e = bed voidage, dimensionless
k = gas thermal conductivity, Btu/h ft F
d_t = tube outside diameter, ft
G = mass velocity or flux of the gas, lb/s ft^2
ρ_s = particle density, lb/ft^3
μ = gas viscosity, lb/ft s
ρ_g = gas density, lb/ft^3
D_p = average particle diameter, ft
g = acceleration constant, 32.2 ft/s^2
Pr = Prandtl number, dimensionless
C_1 = experimental constant, dimensionless
C_2 = experimental constant, dimensionless
C_p = gas specific heat, Btu/lb F
V = nominal bed gas velocity, ft/s

To convert the single tube heat transfer coefficients to those suitable for tube banks, the equations below are applied:

$$FAB = \left[1 - \left(\frac{D_o}{S_n}\right)\left(\frac{2D_o + S_p}{D_o + S_p}\right)\right]^{0.25} \quad (10)$$

where

FAB = bank arrangement factor (staggered arrangement only), dimensionless
D_o = tube outside diameter, in. (mm)
S_n = tube spacing normal to flow, in. (mm)
S_p = tube spacing parallel to flow, in. (mm)

Other variables are as defined previously. The equation for FAB is as derived by Gel'perin, *et al.*[6]

For the radiation heat transfer component, h_r, the following equation may be used:

$$h_r = (\sigma)(\epsilon)\left[(T_b)^4 - (T_w)^4\right]/(T_b - T_w) \quad (11)$$

where

ϵ = average overall emissivity, dimensionless
σ = 0.1713 $\times 10^{-8}$ Btu/h ft^2 R^4
T_b = absolute bed gas temperature, R
T_w = absolute wall temperature, R

The average overall emissivity in bubbling beds will be about 0.8 depending on wall emissivity and particle size. Typically, the overall heat transfer coefficient for an in-bed tube bundle is between 40 and 60 Btu/h ft^2 F (227 and 341 W/m^2 K).

Dense-bed heat transfer to wall Many equations for the vertical wall convection heat transfer coefficient have been proposed. An example is one proposed by Mickley,[7] which takes the following form:

$$h_{cw} = (C_3)\left[\frac{3600\,(\rho_s)(1-e)(\rho_g)(V)}{D_p^3}\right]^{0.263} \quad (12)$$

where C_3 is an experimental constant and the other variables are as previously defined. If the walls in the bubbling bed zone are coated with refractory, that resistance must be added to the other resistances in the equation for U_o. Refractory will also impact the calculation of h_r, but will not affect h_{cw} significantly.

Disengaging zone heat transfer to tube banks During periods of low bed level operation, the topmost tubes of the in-bed tube banks will be uncovered. While this portion of the bank is actually in the disengaging zone (called the splash zone in its lower portion), it is convenient to handle the heat transfer calculation as a special part of the bed. The solids content of the gas stream is much less than in the bed and decreases almost exponentially with height. As a result, the heat transfer rate to the exposed portion of the bank drops off rapidly. The heat transfer rates for this surface are based on experimental results.

As an example, Tang, *et al.*[8] developed the following empirical equation:

$$\frac{h}{h_m} = \exp\left(-\left(\frac{10+H_L}{25.8}\right)^{2.2}\right) \tag{13}$$

where

h = outside film heat transfer coefficient for an unsubmerged tube
h_m = outside film heat transfer coefficient for a fully submerged tube
H_L = unsubmerged tube height above the bed level, in.

Disengaging zone heat transfer to walls For the vertical walls in this zone, the convection rate conforms to an equation of the type shown below:

$$h_{cw} = C_4\,[1 - C_5(1-e)] + h_{cg} \tag{14}$$

where C_4 and C_5 are experimental constants, e is the dense bed voidage, and h_{cg} is given by the following equation:

$$h_{cg} = 0.023\frac{k}{D_e}\left(\frac{3600\,C_p\mu}{k}\right)^{0.3}\left(\frac{D_e G}{\mu\, e}\right)^{0.8} \tag{15}$$

where

k = gas thermal conductivity, Btu/h ft F
D_e = equivalent diameter of vessel, ft
C_p = gas specific heat, Btu/lb F
μ = gas viscosity, lb/ft s
G = mass velocity or flux of the gas, lb/s ft^2

Note that the voidage, e, is for the dense bed and not the disengaging zone.

As a general rule, the height of this zone may be assumed to extend for a residence time of one second.

Upper furnace heat transfer (freeboard) This portion is handled similarly to conventional boilers. The main difference is the emissivity of the solids-gas mixture. The solids content of the gas will be quite high compared to conventional boilers that burn high ash coals and may alter the radiating properties.

Circulating-bed heat transfer Circulating fluidized-bed boilers do not incorporate tube bank surface and rely entirely on heat absorption of the containment walls and internal partitions, such as division walls and wingwalls. The heat transfer in CFB boiler furnaces is handled by breaking the furnace into two distinct regions. One comprises the dense bed and the other the remainder of the furnace.

Dense-bed heat transfer Dense bed heat transfer is similar to that described for the vertical walls of bubbling-bed boilers. There is some difference because the flow regime will usually be turbulent instead of bubbling.

Disengaging zone and upper furnace heat transfer The zone just above the dense bed, but below the point where secondary air is added to produce a circulating fluid bed, is defined as the disengaging zone. The upper furnace includes the transition zone and the freeboard as indicated in Fig. 7. Bed-to-wall heat transfer in these zones can be predicted by considering three parallel processes: particle convection, gas convection and radiation.

In particle convection, heat is removed from particles at the cooling wall surfaces by conduction. The energy loss is replenished by the material and energy exchange with the upwardly flowing central core of solids and combustion gases.

Gas convection is the dominant mode of heat transfer over the fractions of the surface not in contact with particles. This gas component is of minor consequence where the solids content is high. Even in the upper portions of a furnace, where solids content is relatively low, gas convection is usually small compared to the radiative component.

Radiant heat transfer occurs in a manner similar to that in conventional furnaces. Procedures are available to account for the combined emissivities and scattering effects of solids mixed with gases, as long as the void fraction is near or above 0.8. From a practical standpoint, this applies to the furnace above the dense bed.

The overall effective emissivity is a function of the pertinent radiative properties of the gases and solids found in a given mixture, as well as that of the heat absorbing surface. Typically, overall emissivities are about 0.5.

Various equations have been proposed to predict particle convection heat transfer coefficients. Some are complex and include many parameters. Two parameters have the most influence, particle size and mixture bulk density. Reliable convection equations have been developed considering only these two variables.

At full load conditions, where the solids circulation rate is high, a CFB furnace operates at approximately isothermal conditions from top to bottom. The overall heat transfer, in this case, is determined from the vertical bulk density distribution and a proper mean particle size. When the solids circulation rate is reduced, as occurs at low loads, the furnace becomes less isothermal, and more complex procedures are needed to calculate furnace heat absorption. One such procedure involves dividing the furnace into a larger number of vertically arranged zones. This permits the variation in temperature and bulk density to be properly handled in each zone wherein the variation is small.

The equations that follow are typical of those that predict furnace heat transfer:

$$U_o = \frac{1}{\dfrac{1}{h_c + h_r} + R_{ref} + R_m + R_{ft}} \tag{16}$$

where

$h_c = h_{cp} + h_{cg}$
$h_{cp} = C_6\,(\rho_b)^m / (D_p)^n$
$h_r = (\sigma)(\in)\,[(T_g)^4 - (T_w)^4] / (T_g - T_w)$

and where

T_g = absolute local bulk gas temperature, R
h_{cp} = particle heat transfer coefficient, Btu/h ft^2 F
C_6, m, n = experimentally derived constants

The value of h_{cg} is obtained with Equation 15 and all other variables are as previously defined.

Published literature provides a great deal of theoretical information, laboratory experimental data and test results from operating furnaces. Typical of the latter category is the paper of Kobro and Brereton.[9] Figs. 9 and 10 are from that paper. They show measured overall heat transfer rates for a range of bulk densities for two specific average particle sizes. Similar curves are made from commercial boiler field test data and are used to help make empirical corrections to the basic equations.

Heat and material balance

One of the first steps in designing a boiler is to calculate the overall heat and material balances. This establishes the flow quantities, compositions and temperatures of all streams entering and leaving the system. The boiler system is then subdivided and balances are calculated for the pieces. This process is repeated until all systems are defined.

The sensible heat of the solids, as a fraction of gas sensible heat, is quite large in fluidized-bed boilers. It represents the major difference between the heat and material balance for a fluidized-bed boiler and a conventional boiler. (See Chapter 9.) The information that follows describes the many flow streams that are considered when preparing the heat and material balance for a fluidized-bed boiler.

Material balance When limestone is used as a sorbent, the following chemical reactions which affect the solids balance occur in the furnace:

$$CaCO_3 \rightarrow CaO + CO_2$$
$$MgCO_3 \rightarrow MgO + CO_2$$
$$CaO + SO_2 + 1/2\ O_2 \rightarrow CaSO_4$$

There is a net solids weight loss during calcination as CO_2 in the carbonate is driven off endothermically and a solids weight gain as a result of the exothermic sulfation reaction. The Ca/S ratio selected, the amount of sulfur reacted and the ultimate analysis of the limestone will determine the net influence on boiler performance.

For the purpose of illustration, the material balance for an atmospheric pressure CFB boiler is shown in Fig. 11.

From an overall material balance standpoint, the solids that enter the furnace leave as relatively coarse material through the bed drain or as fine material collected

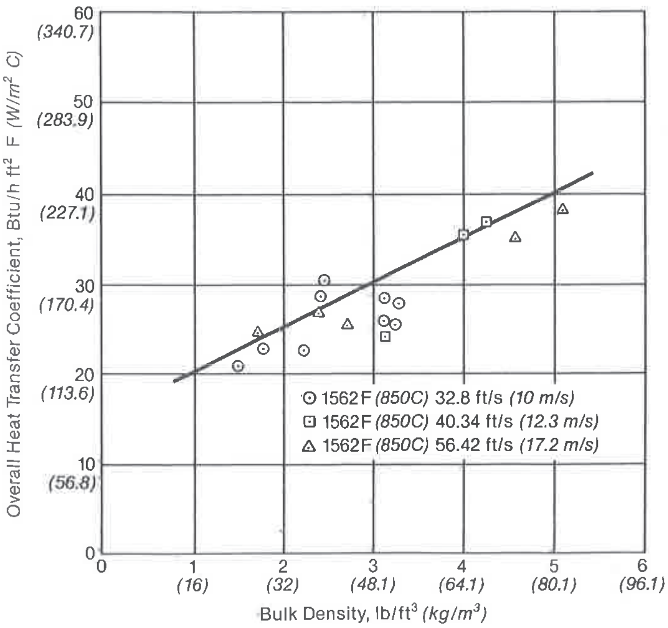

Fig. 9 Heat transfer coefficient versus density in a circulating fluidized bed, 250 micron mean diameter sand.

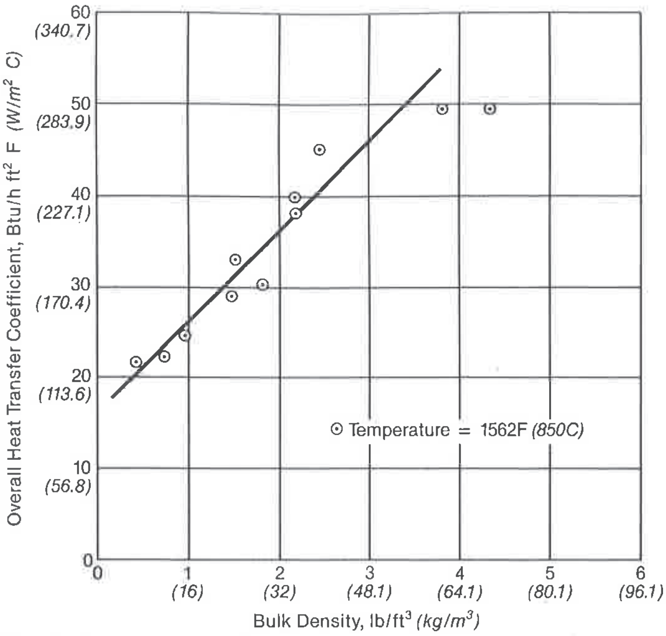

Fig. 10 Heat transfer coefficient versus density in a circulating fluidized bed, 170 micron mean diameter sand.

after the convection pass. There is a significant amount of internal solids recirculation that takes place within the system, and these streams and their effects must be considered. Also, as discussed in the heat transfer section, the distribution of the solids in the furnace influences overall furnace thermal performance.

The fine particles that leave the bed and circulate are called *elutriated solids*. These solids ultimately leave the system after the convection pass with the gas leaving the dust collector or as purged from the multi-clone, if required. Changes in this fraction only change the split between the quantity of solids that leave through the bed drain and the quantity that leave the back end. In the discussions that follow, the sorbent input is considered to have already undergone calcination and sulfation reactions, and the fuel has completely combusted, leaving only ash. These flows should not be confused with the nonreacted sorbent and fuel fed into the boiler.

Definitions of the terms shown in Fig. 11 and the equations which are used to define the mass balance of a B&W CFB boiler are listed in Table 1.

The elutriated fractions Ea, Es and Ei depend on the specific fuel ash, sorbent and inert material in the system. Numerical values are determined from laboratory testing or from operating boilers. N_{UB} and N_{MC} depend on the mechanical arrangement of the collectors plus the size distribution and physical properties of solids circulating (ash, sorbent and inert) in the system. The values are determined empirically. The values of N_{UB}, N_{MC} and E for each input component and for the total solids are interdependent as shown in the following material balance equation.

$$ESF(1-N_{UB})(1-N_{MC}) + MCP = E(ISF) \quad \textbf{(17)}$$

The external mass flux, W_S, is a variable controlled by the designer and is cited under the section on heat balance.

The outlines that follow describe the steps carried out when establishing the material balances for fluidized-bed boilers. For CFB boilers:

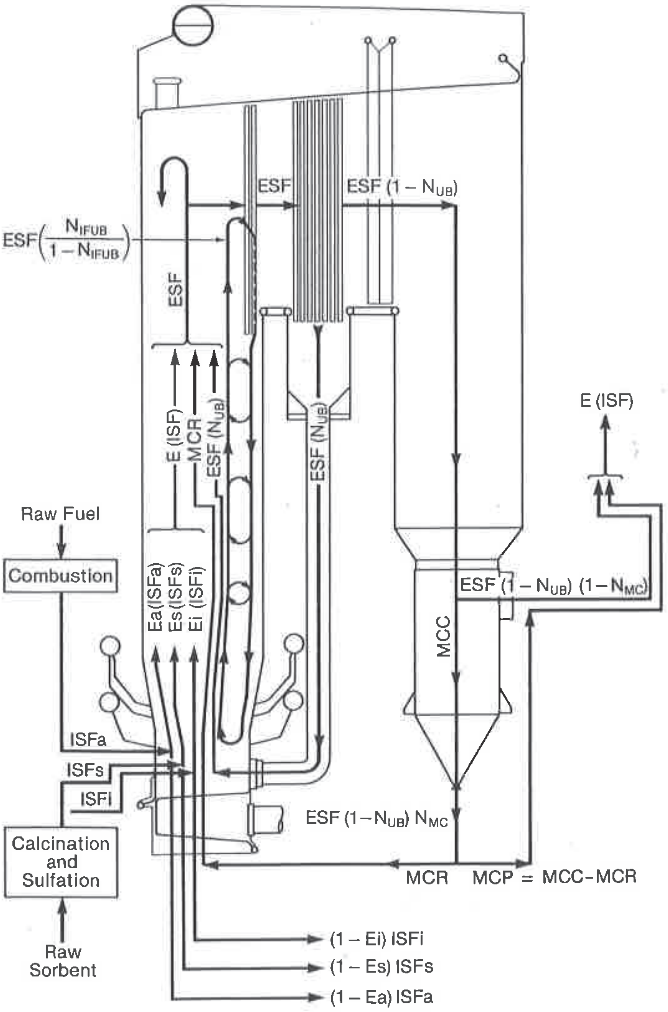

Fig. 11 Circulating fluidized-bed boiler material balance schematic.

1. ISF is determined from the required fuel and limestone inputs. Normally, the limestone input is proportional to the fuel input and is set to provide the required sulfur retention.
2. E is selected based on empirical data.
3. ESF is selected based on furnace design considerations.
4. N_{MC} and N_{UB} values are selected based on empirical data. It should be verified that the selected efficiency values are tied by Equation 17. All other solids flow, external and internal, can be determined from the information above.
5. BDF = (1-E) ISF
6. LVF = ESF (N_{UB})
7. CPSF = ESF (1-N_{UB})
8. BHC = ESF (1 - N_{UB}) (1 - N_{MC})
9. MCC = ESF (1-N_{UB}) N_{MC}
10. MCP = E(ISF) - BHC = MCC - MCR
11. MCR = ESF (1 - N_{UB}) - E(ISF) = CPSF - E(ISF)

Note that the baghouse loss is considered negligible. With properly selected E, N_{MC} and N_{UB} values, the CPSF will not exceed its maximum value corresponding to the allowable solids loading in the convection pass. In most cases, a circulating fluidized-bed boiler can be designed so the multi-clone purge (MCP) is zero. For bubbling-bed boilers:

1. ISF is as above
2. E is selected based on empirical data.
3. BDF = (1-E) ISF
4. Capacity of multi-clone recycle system (MCR_c) is selected based on empirical data in proportion to fuel input.
5. N_{MC} is selected based on empirical data and desired sorbent utilization and carbon conversion.
6. CPSF = ISF (E) + MCR_c
7. MCC = CPSF (N_{MC})
8. BHC = CPSF (1-N_{MC}) (assumes baghouse loss is negligible)
9. MCP = MCC - MCR_c
10. MCR = MCR_C
11. If MCP < 0, then set MCP = 0 and MCR = MCC.
12. BHC = E(ISF)
13. $MCR = \frac{E(ISF)\,N_{MC}}{1-N_{MC}}$
14. $CPSF = \frac{ISF\ (E)}{1-N_{MC}}$

Heat balance In furnaces fired by pulverized coal burners, Cyclones and stokers, little heat is removed from the combustion gases in the zone of maximum heat release. As a result, the flue gas reaches a high temperature before it is subsequently cooled by the furnace enclosure surface.

In fluidized-bed combustion, heat is removed from the zone of maximum heat release at a much higher rate. As a result, the maximum temperature of the flue gas is limited to a predetermined level.

Table 1
Mass Balance Terms

Input solids flow (ISF), lb/h (kg/s)

ISF	=	ISFa + ISFs + ISFi
ISFa	=	Fuel ash and unburned carbon
ISFs	=	Postcalcination and postsulfation sorbent
ISFi	=	Inert bed material (SiO_2 and H_2O)

Elutriated fractions leaving the boiler, dimensionless

Ea	=	Ash
Es	=	Sorbent
Ei	=	Inert material
E	=	[Ea(ISFa) + Es(ISFs) + Ei(ISFi)]/ISF
N_{IFUB}	=	In-furnace U-beam efficiency

Additional terms

BDF	=	Bed drain flow, lb/h (kg/s)
BHC	=	Baghouse catch flow, lb/h (kg/s)
CPSF	=	Convection pass solids flow, lb/h (kg/s)
ESF	=	External solids flow, lb/h (kg/s)
LVF	=	L-valve flow, lb/h (kg/s)
MCC	=	Multi-clone catch flow, lb/h (kg/s)
MCP	=	Multi-clone recycle purge flow, lb/h (kg/s)
MCR	=	Multi-clone recycle flow, lb/h (kg/s)
MCR_c	=	Multi-clone recycle system capacity, lb/h (kg/s)
N_{IFUB}	=	In-furnace U-beam efficiency
N_{MC}	=	Multi-clone dust collector efficiency
N_{UB}	=	Hot particle collector efficiency
W_S	=	External mass flux based on plan area of the upper furnace shaft (ESF/A), lb/h ft^2 ($kg/m^2 s$)

In bubbling-bed systems, the heat removal process is accomplished by having the cooling surface immersed in the active bed of hot solids and burning fuel.

In circulating beds, the large quantity of solids circulating in the system removes heat from the active combustion zone and transfers it to the heating surface throughout the furnace.

The ability to change the amount of heat removed from the combustion process to achieve the desired bed temperature provides flexibility when designing a fluidized-bed boiler. In a bubbling bed, the heat balance around the bed itself determines the bed temperature. The excess heat, which is the difference between the quantity of heat entering the bed with the fuel plus air and the amount of heat leaving the bed in the combustion products and through radiation to the rest of the furnace, must be absorbed by the in-bed tubes and the enclosure surrounding the bed. If the heating value of the fuel decreases, less in-bed surface is required to achieve the same bed temperature. If the heating value of the fuel increases, more in-bed surface is required. At the lower extreme of fuel heating value, no in-bed surface is used and the enclosure is often covered with refractory to minimize heat loss. This latter example is typical when high moisture sludges are burned.

In the CFB designs, all furnace heat transfer surfaces must be considered when making a heat balance. The amount of material circulating from the furnace to the primary collector and back to the furnace determines the inventory or average density of solids in the furnace. The heat transfer coefficient in the furnace is proportional to bulk density. Therefore, in the case of CFB boilers, the furnace heat absorption depends on the total furnace surface and the external recycle rate (which is a direct function of W_S, the external mass flux).

The furnace heat balance is also influenced by the way a particular fuel burns. When fuel is introduced into a fluidized bed, the majority of it burns in the dense bed. However, some of the fuel burns above the dense bed and the heat release pattern of the above-bed burning must be considered when calculating the furnace exit gas temperature. The actual split between in-bed and above-bed burning depends on fuel properties such as type, volatility, size and feed system. Because of the large amount of recirculated solids, the heat release pattern in a CFB furnace does not have a strong influence on the temperature distribution. For this reason, CFB furnaces are more tolerant of fuel changes than bubbling beds.

The heat balance around the convection surface is calculated similarly to conventional boilers. However, it is necessary to include the effects of the solids in the gas because they can have a substantial influence on the convection pass heat balance.

Fluidized-bed boiler arrangements

Boiler subsystems

The unique features discussed include the distributor plate, the bubble caps and the overfire air system. Also discussed are some of the features that are common to all water-cooled boilers designed by B&W.

Distributor plate and bubble caps The distributor plate is located at the bottom of the furnace and separates the windbox from the furnace. The distributor plate is fitted with bubble caps to provide a uniform distribution of combustion air to the entire furnace cross-section over the boiler load range. (See Fig. 12.) For this reason, distributor plates designed by B&W have a pressure drop of about 16 in. (406 mm) of water column across the bubble caps at full load and a minimum drop of 4 in. (102 mm) of water column at minimum load. Typically, 50 to 70% of the combustion air flows through the distributor plate of a CFB boiler at full load. A bubbling fluidized-bed boiler distributor plate is designed for 85 to 100% of the combustion air when staged combustion is not used.

The distributor plate must form an air-tight seal, other than the bubble caps between the furnace and the windbox and must support the weight of a slumped bed and resist the uplift generated from the air pressure drop across it during operation. In general, most distributor plates used in boilers are constructed from water-cooled membraned tubes.

In addition to the items discussed above, the bubble caps are designed to split the air into small streams for good flow distribution, avoid the formation of large bubbles in the bed, minimize erosion and prevent back sifting of bed solids into the windbox.

Overfire air system The overfire, or secondary, air system is part of the larger combustion and emission control system. As previously described under *Emissions*, all CFB boilers and some bubbling fluidized-bed boilers, when no tube bundle is present, use a staged combustion process to control the emission of NO_x. Overfire air is sometimes used in bubbling-bed boilers to improve free-

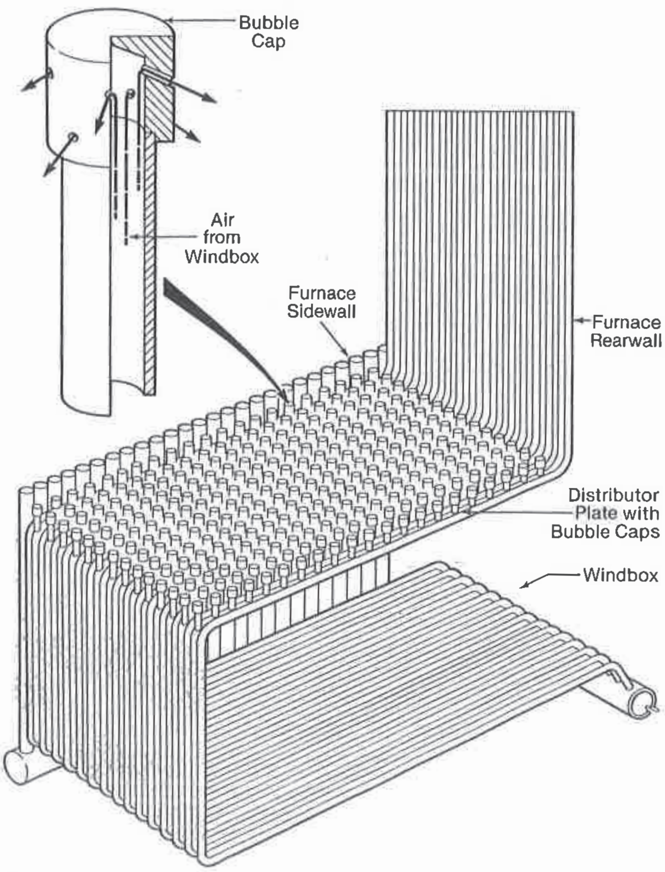

Fig. 12 Furnace distributor plate and bubble caps.

board mixing and burning. This technique is used when fuel is fed over-bed or when there are excessive fuel fines.

With overfire air, the combustion air that does not flow through the bubble caps is injected into the combustion gases shortly after initial coal ignition to complete combustion. The overfire air system must provide adequate penetration and thorough mixing of the overfire air with the combustion gases to achieve complete fuel burnout and minimize the amount of CO discharged to the atmosphere.

Overfire air penetration depends on the size of the overfire nozzles and the air and gas velocities and densities. In circulating beds, the solids density at the point of injection must also be considered.

Boiler enclosure The furnace and convection pass enclosures are conventional. (See Chapter 22.) They are constructed of water-cooled membraned tubes welded together to form a gas-tight enclosure. This enclosure is also used to support the fluidized bed, windbox, superheater and other components because the boilers are usually top supported.

Auxiliary equipment

Fuel feed systems The fuel feed system has had a greater impact on the evolution of fluidized-bed boilers than any other auxiliary or support system. As a result, three classifications of fuel feed systems have been developed and are in use today. These are the under-bed, over-bed and in-bed systems.

The under-bed feed system is usually applied to bubbling-bed boilers burning bituminous coal. An under-bed feed system is essentially a pneumatic transport system that moves the coal from a storage silo to the bed.

When burning bituminous coal and less reactive fuels, it is necessary to spread the coal as evenly as possible throughout the bed for good fuel-air mixing. To do this, fuel feed pipes are spaced on about 4 ft (1.2 m) centers throughout the bed.

When compared with the over-bed feed system described below, a well designed under-bed feed system can provide between 2 and 4% better combustion efficiency with bituminous coals.

On the negative side, an under-bed feed system is complicated. The fuel, typically, must be crushed to less than 0.25 in. (6.4 mm); the fuel must be dried to less than about 6% moisture or the transport pipes will plug; transport piping erosion can be a problem; the equipment needed to pressurize the coal for transport is a high maintenance item; and costs are generally higher than for an over-bed feed system.

Over-bed feed systems are also used on bubbling beds. They are used for reactive fuels and bituminous coals where the simplicity of the system can justify any reduction in carbon conversion.

The feeders are located above the bed where the furnace gas pressure is slightly below atmospheric. This location simplifies feeding since the coal stream does not have to be pressurized. An over-bed system uses the same fuel feed equipment as a spreader stoker. (See Chapter 15.)

While over-bed feed systems are simpler than under-bed systems, they still have requirements and limitations. The coal for an over-bed feed system is crushed to a top size of 1.25 in. (31.8 mm), and the amount of fine coal (less than 30 mesh) is limited to prevent excessive burning in the freeboard zone immediately above the bed. Also, the amount of fines in the fuel must be relatively consistent.

Fig. 13 compares the combustion efficiency for under-bed and over-bed coal feed systems for two coals, Sarpy Creek and Kentucky No. 9. Sarpy Creek is a subbituminous coal and Kentucky No. 9 is a bituminous coal. It can be seen from Fig. 13 that the Sarpy Creek coal is much more reactive than the Kentucky coal. Also, under-bed feed results in higher combustion efficiency than over-bed. Recycling solids collected in a multi-clone to the furnace also increases combustion efficiency.

In-bed feed systems are used for CFB boilers and some sludge-burning bubbling beds. Typically, some form of a feed screw or air assisted chute is used. The fuel is injected through an enclosure wall just above the distributor plate. Because the furnace pressure at this point can be as high as 50 in. (1270 mm) of water gauge, the feed system must seal against this pressure to allow the fuel to be introduced to the furnace. The pressure seal is accomplished by the use of a rotary seal or a head of coal between the coal silo and a belt feeder.

A CFB with its higher fluidizing velocity is a better fuel mixing process than a bubbling bed with in-bed tubes. As a result, fewer feed points are required.

Sorbent feed systems As previously discussed under *Emissions*, limestone or dolomite is added to a fluidized-bed boiler to capture SO_2. In B&W designs, the sorbent is usually fed into the dense bed area of the furnace. While good limestone distribution is desired to reduce sorbent requirements, it is not nearly as critical as distribution in the case of under-bed coal feed.

The design of the sorbent feed system must consider where the material will be injected into the furnace, the furnace gas pressure at that point and the method of addition. In the case of under-bed and in-bed feed, the sorbent can be mixed and injected with the fuel. Sorbent can not be fed properly with a spreader due to its fineness. Sorbent has been blown into the furnace through separate pneumatic feed points and by gravity from a storage silo.

Bed ash removal systems When an ash bearing fuel is burned in a fluidized-bed boiler, the ash is liberated from the coal in the boiler furnace. Also, sorbent or inerts are fed to the boiler. Therefore, means must be provided to remove the solids from the system to prevent accumula-

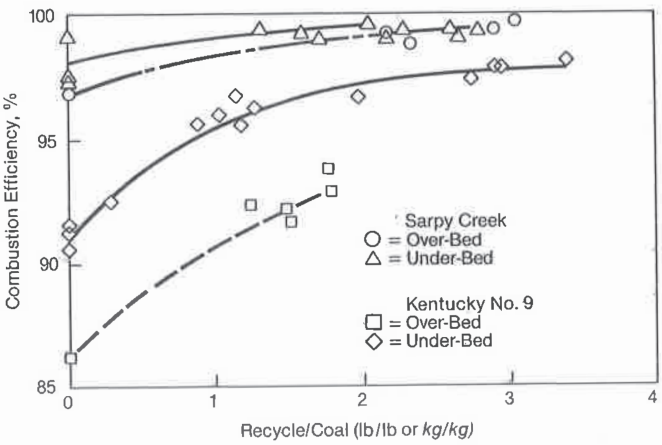

Fig. 13 Comparison of under-bed and over-bed coal feed.

tion. There are two major places in a fluidized-bed boiler system where solids are removed, the bed drain and the baghouse or electrostatic precipitator.

Normally, the design of the bed ash removal system includes a large safety margin because of the uncertainty as to where the solids will exit the system and to handle changes in fuels. Operating results have shown that the bed drain flow can range from 0 to more than 50% of the total solids output.

In the case of fuels with a high alkali content in the ash, it may be necessary to drain the bed material from the boiler at a rate higher than the solids buildup rate to prevent a concentration buildup of the alkali in the bed. Under these conditions, it is also necessary to make up any excessive bed material lost from the system. Typically, when the concentration of alkali exceeds 5 to 6% of the bed weight, the probability of forming agglomerates increases significantly.

A second case that requires greater than normal bed drain flow rates is created when the fuel contains a high percentage of large rocks and ash, greater than 0.5 in. (12.7 mm). There is a natural tendency for these large pieces of noncombustible material to accumulate in the bed and cause defluidization. To avoid this, it is necessary to move the oversize material to the bed drain for removal. This is done by maintaining the bed drain flow rate above a minimum experimentally-determined value. It is necessary to make up any excess bed material lost in the rock removal process.

Sootblowers Chapter 23 discusses the application of sootblowers to various types of firing systems. Because the fuel in a fluidized-bed boiler is burned at temperatures below the ash softening point, the flyash never reaches the plastic state. As a result, it forms a dry powder that is easy to remove from the heating surfaces.

There are fluidized-bed boilers in operation that have no sootblowers. As long as the heat transfer surface is properly spaced and the dusting that does occur is accounted for in the design of the unit, the boiler will perform satisfactorily, achieving the desired values of superheat temperature and boiler efficiency. However, when fuels with high sodium and/or potassium ash contents (low ash fusion temperatures) are fired, especially with an over-bed feed system, there is a higher likelihood for convection pass fouling and, therefore, sootblowers may be required.

Typical fluidized-bed boiler designs

At the time of this writing, the majority of applications for bubbling fluidized beds had been directed at installations of less than 200,000 lb/h (25.2 kg/s) of steam. Most of the new, large capacity, coal-fired fluidized-bed boilers have been of the circulating type. However, several large boilers have been retrofitted with bubbling beds because the bubbling bed was physically more compatible with the changes, as compared with circulating beds. In some of these cases, bubbling beds were used to reduce SO_2 emissions. In other cases, fuel-related operating problems have provided the incentive for change.

Another area where bubbling beds, both new and retrofitted, are being used is for burning sewage sludge and pulp mill and recycled de-inking paper sludge.

The design of a boiler and its auxiliaries is a complex task. Because other chapters describe the conventional aspects of boiler design, the following descriptions are limited to unique fluidized-bed boiler features.

Bubbling-bed retrofit During the winter of 1986, the moving grates were removed from Unit No. 2 at Montana-Dakota Utilities' R. M. Heskett station and replaced with a bubbling fluidized bed. (See Fig. 14.) This boiler was originally designed to generate 650,000 lb/h (81.9 kg/s) steam at 1300 psig (89.6 bar gauge) and 950F (510C) superheat burning Beulah lignite. However, high sodium content in the fuel ash caused severe furnace slagging and superheater fouling. Before the bubbling-bed modification, long term output was limited to about 50 MW of the rated capacity of 72 MW. The fluidized bed was installed to reduce furnace operating temperatures, to avoid slagging and fouling problems, and to achieve full power operation.

The new fluidized bed measured 40 × 26 ft (12.2 × 7.9 m) and was fitted under the boiler with only minor pressure part changes to the furnace division walls. The distributor plate and enclosure walls are water-cooled. In this unit, both superheat and boiling surface are placed in the bed to meet steam generation and turbine superheat temperature requirements and to limit the bed operating temperature to 1500F (816C). The superficial bed velocity is 12 ft/s (3.7 m/s) and the bed is 54 in. (1372 mm) deep in the expanded condition. Eight windbox compartments are used for unit turndown and startup. Because Beulah lignite is a highly reactive fuel, no flyash recycle was installed. Additionally, because of the low sulfur and high alkali content of this fuel, sand is used as the bed material.

The boiler was restarted in May 1987. This unit is now producing 80 MW and avoiding fuel related slagging and fouling problems. However, operators must avoid high bed and furnace temperatures because of the fuel's fouling tendencies.

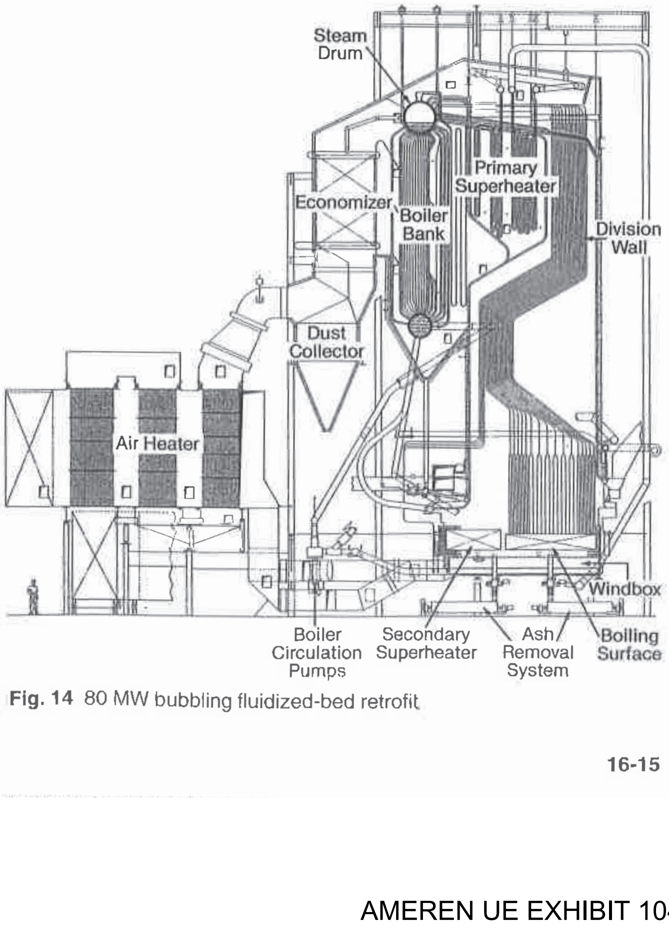

Fig. 14 80 MW bubbling fluidized-bed retrofit

Circulating fluidized-bed boiler The B&W circulating fluidized-bed boiler design is based on a completely water-cooled setting. This feature provides a modern gas-tight enclosure suitable for operating with a positive pressure in the furnace. It has no high temperature refractory lined flues in the vicinity of the primary particle collector and therefore requires minimal building space and reduces furnace refractory maintenance. This construction is possible due to the use of an impingement-type primary solids separator (U-beams) which can easily be integrated into the boiler enclosure.

Fig. 15 shows the side view of a 55 MW CFB boiler designed to burn low volatile bituminous coal. The unit produces 465,000 lb/h (58.6 kg/s) steam at 1550 psig (107 bar gauge) and 955F (513C). The furnace is 30 ft (9.1 m) wide, 15 ft (4.6 m) deep, 85 ft (25.9 m) high and contains full height, water-cooled division walls and steam-cooled wingwalls in the upper furnace.

Fuel and sorbent are fed to the bed through the lower furnace frontwall. The ash and spent sorbent are removed through drain pipes in the floor. The solids collected by the U-beams and multi-clone are returned to the lower furnace through the rearwall.

The primary air enters the furnace through the distributor plate and secondary air is injected at elevations approximately 6 and 12 ft (1.8 and 3.7 m) above the distributor plate.

The entire lower furnace, up to 22 ft (6.7 m) above the distributor plate, is covered by a thin layer of highly conductive refractory held to the water tubes by pin studs. Refractory is used in the lower furnace to protect the tubes from corrosion and erosion. The remaining portion of the furnace enclosure consists of bare tubes.

The B&W CFB boiler design uses an impact separator to collect and recycle solids to the furnace. The primary solids separation system consists of staggered rows of U-shaped channel members, or U-beams, suspended from the boiler roof. (See insert, Fig. 15.) Material strik-

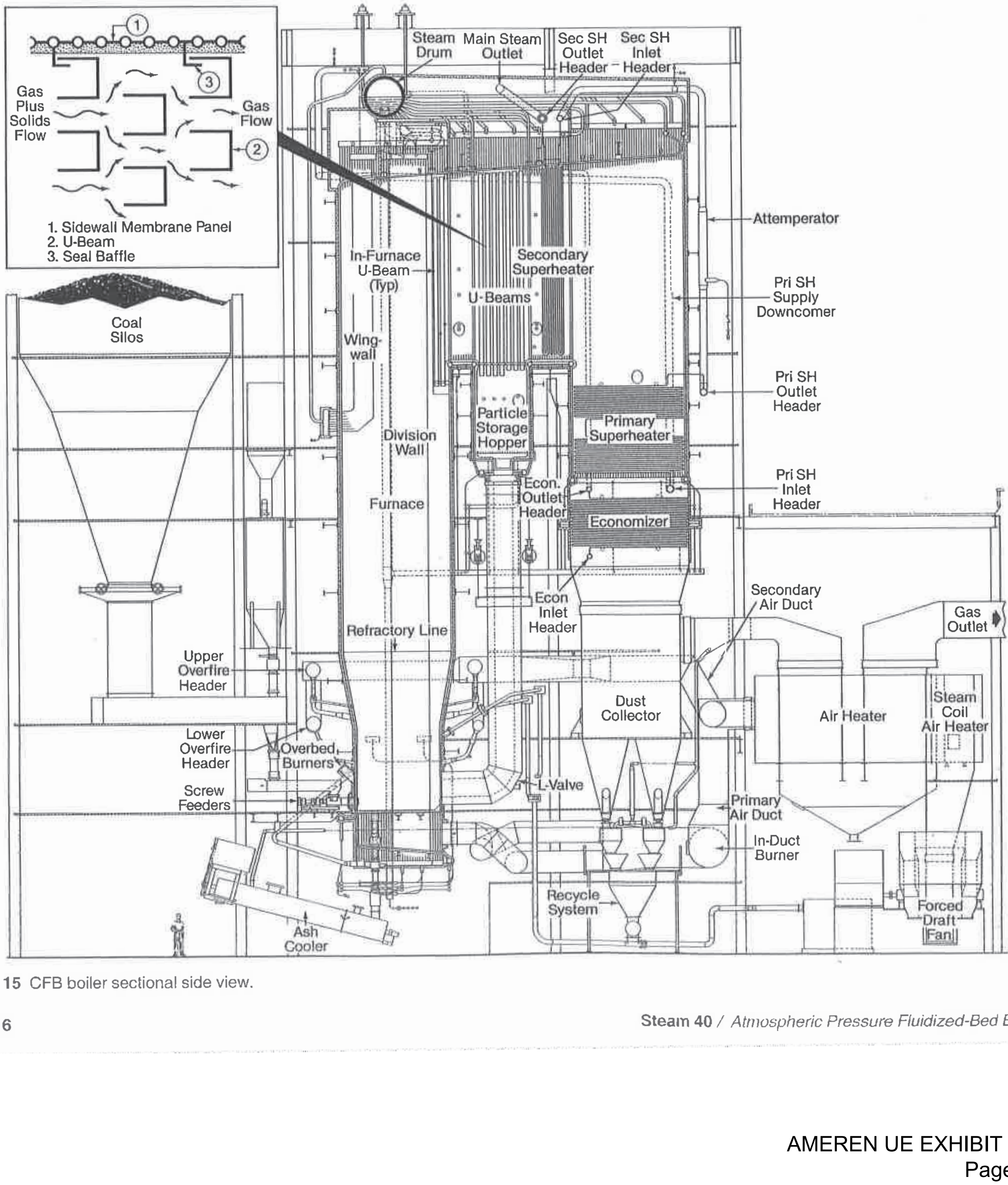

Fig. 15 CFB boiler sectional side view.

ing the U-beams is separated from the gas, flows down the U-channel and discharges from the bottom. The most recent designs use two stages of U-beam collectors which result in better overall collection efficiency than was provided in the first generation of boilers. The first stage is located in the upper furnace and returns the solids directly to the lower furnace. The second stage is located after the furnace and above a particle storage hopper. Bed material collected by the second stage U-beams is recycled to the lower furnace by flow-controlling L-valves.

The L-valve is a nonmechanical device for returning solids to the furnace (Fig. 16). Solids collected in the particle storage hopper flow to the standpipes and serve as a source of inventory for the furnace. Solids flow is induced by injecting a small amount of aeration into the L-valve. With this arrangement, hundreds of thousands of pounds (kg) of solids per hour can be circulated with air flows on the order of 10 ACFM (4.7×10^{-3} m^3/s).

Startup and operation

Startup

Fluidized-bed boilers are brought into operation by first establishing air flow through the unit, heating the bed material to a temperature that is above the fuel's autoignition temperature, and then introducing the fuel into the hot solids and air. The fact that the bed solids are above the fuel's autoignition temperature ensures that combustion is initiated safely.

The heat for raising the bed material to the desired temperature is normally supplied by a burner in the air supply duct that is capable of heating the air and solids to the required temperature, a burner located above the bed that fires into the bed, or a combination of both methods.

The above-bed type of burner is generally used for circulating beds and bubbling beds that do not have in-bed surface. The duct type of burner is used for bubbling beds that contain in-bed cooling surface. Both the above-bed and duct burners must have complete flame safety systems which include independent flame detection and burner safety trip control circuits.

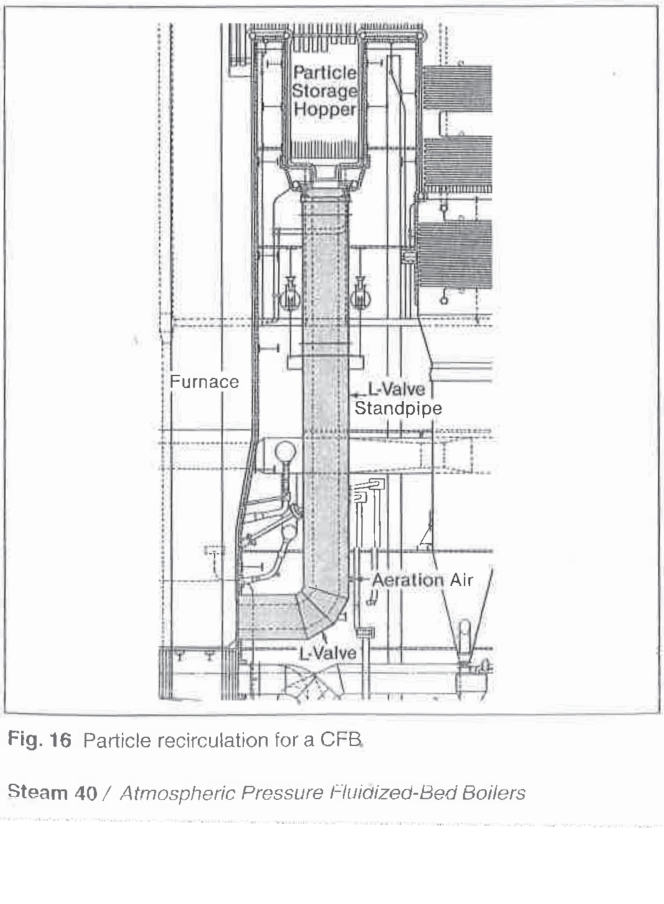

Fig. 16 Particle recirculation for a CFB.

Different fuels have varying ignition temperatures. The following table lists several fuels and the minimum bed temperature that must be attained before the fuel can be fed into the bed.

Fuel	Minimum Bed Temperature, F (C)
Bituminous coal	900 to 950 (482 to 510)
Lignite	900 (482)
Anthracite	1000 to 1050 (538 to 566)
Wet wood	1200 to 1250 (649 to 677)
Oil	1400 (760)
Natural gas	1400 (760)

The actual bed temperature for a particular fuel must be determined by demonstration. Once the minimum safe bed temperature is established, it is only necessary to measure bed temperature to make sure it is above the minimum value whenever feeding fuel to the bed. As fuel is introduced into the bed, it ignites, releases its energy and increases the bed temperature above the minimum value. Further increases in fuel feed will result in higher bed temperatures. This process is continued until the desired boiler load and bed temperature are achieved.

System control

Many of the operating and control features of fluidized-bed and conventional boilers are the same. There are two areas where fluidized-bed and conventional boilers vary significantly.

One is the need for the fluidized-bed boiler control system to monitor and control the transport of large quantities of solids. The other is the need to control the primary and secondary air ratio to achieve minimum emissions and unburned carbon loss. With the ability to control solids and air flows, the unique fluidized-bed parameters of bed temperature and bed inventory can be controlled.

Bed temperature control

Bubbling bed The temperature of a fluidized bed is controlled by manipulating the heat removal process. In a bubbling-bed boiler, the bed itself is limited to the lowermost 4 ft (1.2 m) of the furnace.

Because the heat absorbing surface is located within the bed and solids recycle has only a minor influence on bed temperature, the heat balance around the bed establishes the bed operating temperature. As the load (fuel flow) is changed, it is necessary to change the bed heat balance to achieve the desired bed operating temperature. The heat balance can be altered by the following methods.

For the case of a load reduction, bed depth can be reduced by removing inventory from the system and/or reducing the velocity of the combustion gas through the bed. By decreasing the bed depth, less heat transfer surface will be submerged in the solids. As a result, the amount of heat removed decreases and the bed temperature is maintained at the desired value.

There are limits to the degree of turndown achievable by reducing the bed level and a point is reached where a portion of the bed must be removed from service, or *slumped*. Under these conditions, little heat is removed from the bed in the slumped area. This control requirement dictates that the windbox be designed so that portions of the bed can be isolated and removed from service.

Bed temperatures can also be controlled by excess air and air preheat. By increasing the excess air that flows through the bed, the amount of heat required to heat the air increases and the temperature of the bed decreases. By decreasing the air preheat temperature, the temperature of the bed can also be made to decrease.

Circulating bed In CFB boilers, the solids inventory is distributed more uniformly throughout the furnace than in a bubbling bed. (See Figs. 5 and 7.) The combination of this condition and the large amount of solids that recirculate at high boiler loads results in uniform furnace gas and solids temperatures. In the section on heat transfer, it was pointed out that the heat transfer coefficient in a CFB furnace is greatly dependent on the solids inventory in the furnace. Therefore, by changing furnace inventory, the heat transfer from the gas and solids to the cooling surface can be varied. The net result of this process is the ability to control furnace bed temperature. Therefore, as load changes or as fuel varies, it is possible to increase or decrease furnace temperatures by controlling furnace inventory. In some CFB designs, furnace temperature is moderated with an external heat exchanger. As in bubbling beds, the bed temperature may also be changed by varying excess air and air preheat temperature.

Bed inventory control

The ability to maintain the required bed inventory, as described above, is highly dependent on the properties of the solids that make up the inventory as well as the performance characteristics of the solids collection and recycle system. Furnace inventory is made up of sorbent and/or inert bed material and fuel ash, but each must be properly controlled.

For the system to be under control, the quantity of solids in the furnace must equal or exceed the amount required to support stable combustion and for bed temperature control. If too many solids leave the boiler with the flue gas, furnace inventory will decrease, combustion may become unacceptable and furnace temperatures will increase. This condition must be corrected by modifying the particle size distribution of the solids fed to the boiler or by providing some kind of inventory control system.

Bubbling bed Bubbling-bed inventory is dependent upon the balance between feed solids and the bed drain purge flow, plus the solids that are elutriated and lost from the system through back-end collectors. The amount of fine material leaving the boiler with the flue gas (elutriated fraction) depends on particle size and fluidizing velocity. When solids are recycled from a dust collector, the flow of solids in the freeboard increases significantly. However, bed inventory, which consists mostly of nonentrainable particles, is not affected.

From a practical operating and control standpoint, the minimum solids feed flow rate must provide enough material to maintain the bed drain flow required to remove oversized particles plus make up the back-end loss. With solids input exceeding the minimum value, the rate of bed drain purge is adjusted to maintain the desired bed inventory. In practice, bed inventory is controlled by measuring the pressure drop across the bed and maintaining it at a value corresponding to the desired inventory.

Circulating bed In a circulating fluidized bed, the amount of solids leaving the furnace with the flue gas and the bed inventory are interdependent. Solids leaving the furnace must be collected and returned to the furnace to maintain bed inventory. Overall inventory control is determined by the solids input-output balance similar to a bubbling bed and is controlled in the same way.

In B&W's circulating fluidized-bed system, the inventory consists of solids in the furnace plus those in the particle storage hopper and L-valve standpipes. The inventory distribution between the furnace and storage is established by the external recycle rate which is controlled by nonmechanical L-valves. For a constant total inventory, a reduction in L-valve flow rate transfers inventory from the furnace to the particle storage hopper and L-valve standpipes and vice versa.

Because the solids flow rate circulating through the L-valves is many times larger than the solids input rate, inventory can be exchanged between the furnace and particle storage hopper at the rate of 10 to 20% per minute. This feature allows rapid control response to changes in fuel flow or boiler load.

The inventory split between the dense bed and the upper furnace is dependent upon the particle size distribution of the solids in the furnace and gas velocities in the primary zone and upper furnace. To provide sufficient inventory in the upper furnace and satisfy heat transfer requirements, the total system inventory is controlled by the bed drain purge rate. On the other hand, the bed drain purge rate should be sufficient to prevent accumulation of coarse material in the dense bed.

If the bed drain purge rate required for coarse material

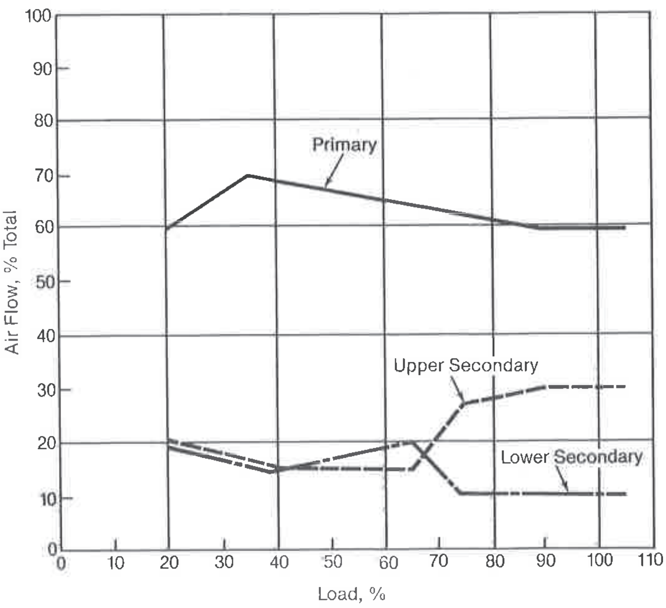

Fig. 17 Air flow distribution.

removal exceeds that needed for total inventory control, a depletion of upper furnace inventory will result. In this case, means must be provided to bring the system back into balance. Furnace inventory size distribution can be corrected by changing the particle size distribution of the input solids or by classifying and recirculating a portion of the bed drain flow. Like the bubbling bed, furnace pressure drop measurements provide the required control signals.

Overfire air control

The split between primary air and overfire air is established, primarily, to optimize fuel burnout and CO and NO_x emissions. Fig. 17 shows a typical curve of air flow versus load for a bituminous coal-fired boiler. As a result of variations among fuels, final air splits are established as a result of unit operation and control system tuning for optimum combustion and furnace inventory distribution.

Summary

By traditional power industry development standards, fluidized-bed boilers represent an emerging technology. However, during the past 20 years the efforts have brought the technology from concept to commercial status. Activities are continuing to improve existing designs and extend unit size to larger capacities.

In the quest to reduce electric power costs and emissions, pressurized fluidized-bed boiler plants are being built and operated. (See Chapter 29.) In addition, investigators are re-examining fluidized-bed gasification to determine if improved power generating systems can be designed. (See Chapter 17.)

Today thousands of fluidized-bed units are in operation and are providing industry with a technique capable of burning a wide variety of fuels in an environmentally improved manner.

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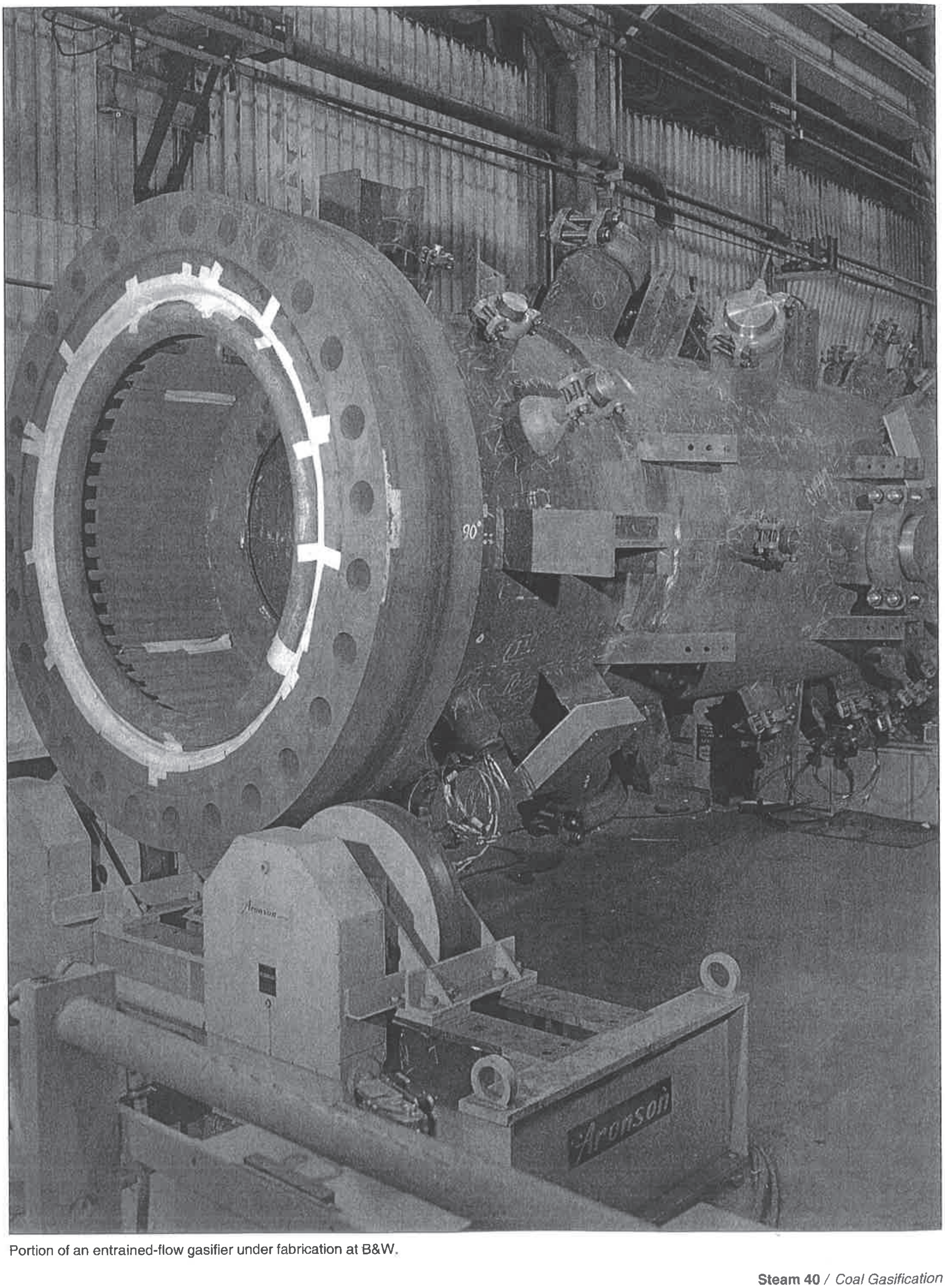


Portion of an entrained-flow gasifier under fabrication at B&W.

Chapter 18
Boilers, Superheaters and Reheaters

In a modern steam generator, various components are arranged to efficiently absorb heat from the products of combustion and provide steam at the rated temperature, pressure and capacity. These components include the boiler, superheater, reheater, economizer and air heater. They are supplemented by systems for steam-water separation (see Chapter 5) and the control of steam outlet temperature. The entire boiler system can be divided into two general sections, the furnace and convection pass. The furnace provides a large open volume with water-cooled enclosure walls where combustion takes place and the combustion products are cooled to an appropriate furnace exit gas temperature (FEGT). The convection pass contains tube bundles which consist of the superheater, reheater, boiler bank and economizer. The convection pass is usually followed by the air heater. The boiler or evaporation-circulation system usually includes the furnace enclosure walls, steam drum and steam-water separation equipment, boiler bank tube bundle, and associated connecting piping (downcomers, supplies and risers as discussed in Chapter 1). This chapter focuses on the boiler, superheater and reheater components plus systems for steam temperature control, steam bypass and unit start-up. Chapter 19 discusses economizers and air heaters.

Boilers

Boiler surface is defined as the tubes, drums and shells which are part of the steam-water circulation system and which are in contact with the hot gases. Although the term *boiler* now frequently refers to the overall steam generating system, the term *boiler surface* excludes the economizer, superheater, reheater or any component other than the steam-water circulation system itself.

While boilers can be broadly classified as shell, fire tube and water tube types, as discussed in the introduction to *Steam*, modern high capacity boilers are of the water tube type. In the water tube boiler, the water and steam flow inside the tubes and the hot gases flow over the outside surfaces. The boiler circulation system is constructed of tubes, headers and drums joined in such a way that water flow is provided to generate steam while cooling all parts. The water tube construction allows greater boiler capacity and higher pressure than shell or fire tube designs. Also, the water tube boiler offers greater versatility in arrangement; this permits the most efficient use of the furnace, superheater, reheater and other heat recovery components.

Boiler configurations

Modern high capacity boilers come in a variety of designs, sizes and configurations to suit a broad range of applications. Sizes range from 1000 to 10,000,000 lb/h (0.13 to 1260 kg/s) and pressures range from one atmosphere to above the critical pressure.

The boiler configuration is largely determined by the combustion system, fuel, ash characteristics, operating pressure and total capacity. The diversity in configurations is illustrated in Figs. 1 through 5.

Typical industrial and small power boilers The Integral Furnace boiler shown in Fig. 1 is an oil- and gas- fired, low pressure two drum package boiler. In small capacities it can be entirely shop assembled and shipped to the site. (See Fig. 1, Chapter 1.) Because it burns a clean fuel, provision has not been made for flyash collection or surface cleaning, and a small furnace volume can be used. A boiler bank, closely spaced tubes between a steam drum and lower drum, provides the heat transfer surface necessary for the rated steaming capacity. Fig. 2 illustrates a two drum Stirling® power boiler (SPB) with a controlled combustion zone (CCZ™) furnace specially designed for effective firing of high moisture wood and biomass. (See also Chapter 28.) A spreader-stoker firing system is supplied and a boiler bank is provided for sufficient steam generating surface.

Additional unique designs include fluidized-bed boilers, process recovery boilers and waste-to-energy boilers discussed in Chapters 16, 26 and 27.

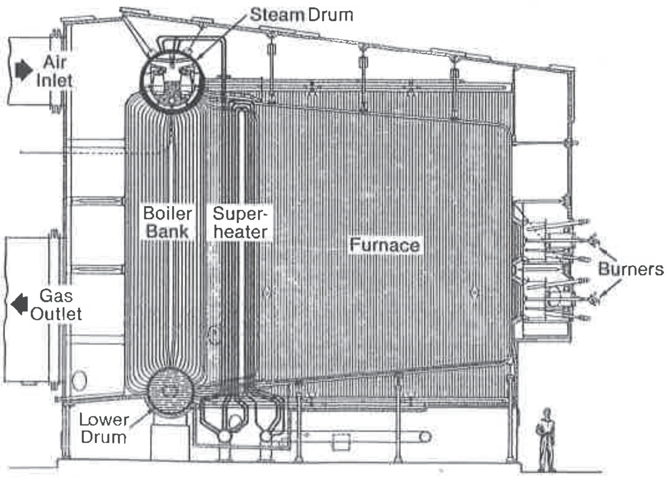


Fig. 1 Integral Furnace industrial boiler for oil and gas firing.

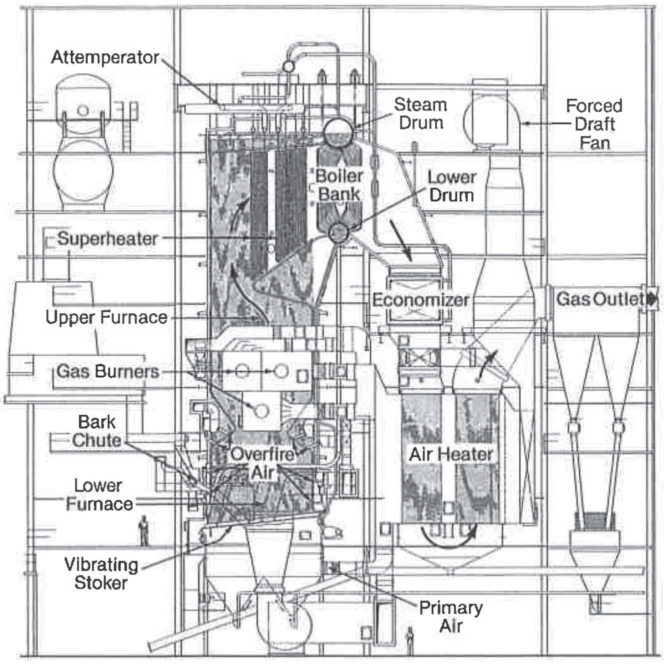


Fig. 2 Two drum Stirling® power boiler — CCZ™ stoker furnace configuration for bark firing.

Large utility boiler designs Figs. 3 and 4 illustrate two variations of the Babcock & Wilcox (B&W) Radiant boiler (RB) for natural circulation drum type steam generating systems. Fig. 5 illustrates one version of the B&W Universal Pressure (UP) boiler designed for once-through flow at supercritical or subcritical pressures. As discussed in Chapter 24, all of these units feature gas-tight, fully water-cooled furnace enclosure walls and floors made of all-welded membrane panel construction. Each design normally includes a single reheat section, although the supercritical once-through boiler has also been supplied as a double reheat unit. Fig. 3 shows an RB unit for coal firing. This is the Carolina-type design (RBC) with downflow convection backpass which minimizes overall steam generator height. Provision is made for sootblower surface cleaning and for flyash collection. A Tower-type (RBT) configuration is also available which features fully drainable convection pass surfaces and a minimum plan area. (See Chapter 24.) Fig. 4 is a Radiant boiler of the El Paso (RBE) configuration for oil and gas firing. This unit is very compact because of the relatively clean fuels being used. The compact design minimizes the boiler footprint (plan area) and support steel. Selected provision may be made for cleaning equipment.

Fig. 5 is a 1300 MW pulverized coal unit for supercritical pressures using vertical furnace wall tubes. As discussed in Chapter 24, the once-through circuitry design eliminates the need for a steam drum. Multiple fluid passes through the furnace minimize the imbalances in steam temperature around the furnace periphery which could result from nonuniform heat input. Other UP designs are available for subcritical pressure operation with spiral circuitry furnaces. (See Fig. 11 of Chapter 24.) Inclined tubes wrapping around the furnace make multiple fluid passes through the furnace enclosure unnecessary.

Boiler design

Regardless of the size or configuration, modern boiler design remains driven by four key factors: 1) efficiency (boiler and cycle), 2) reliability, 3) capital and operating cost and 4) environmental protection. These factors, combined with specific applications, produce the diversity of designs presented above and discussed at length in Chap-

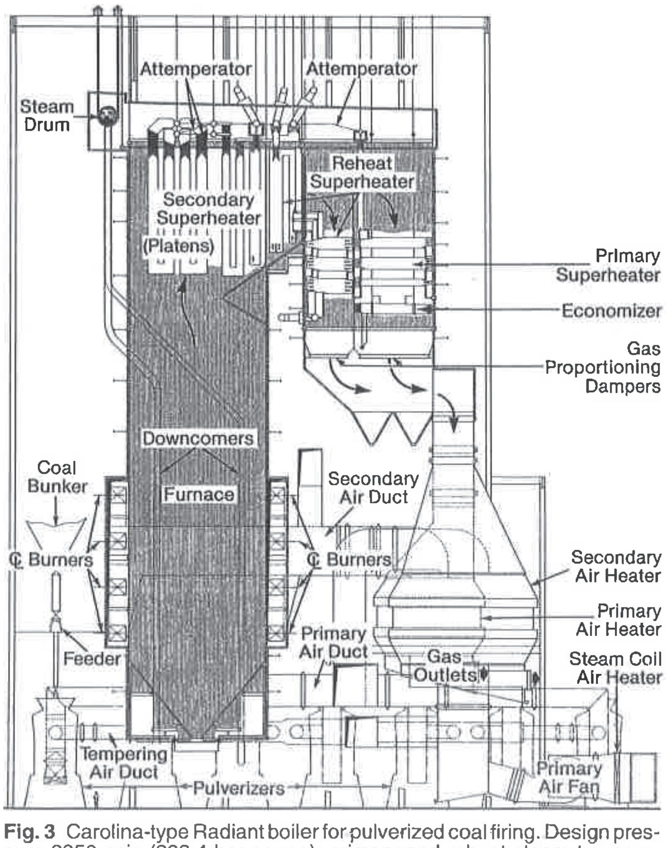


Fig. 3 Carolina-type Radiant boiler for pulverized coal firing. Design pressure 2950 psig (203.4 bar gauge); primary and reheat steam temperatures 1005F (541C); capacity 4,900,000 lb steam/h (617 kg steam/s).

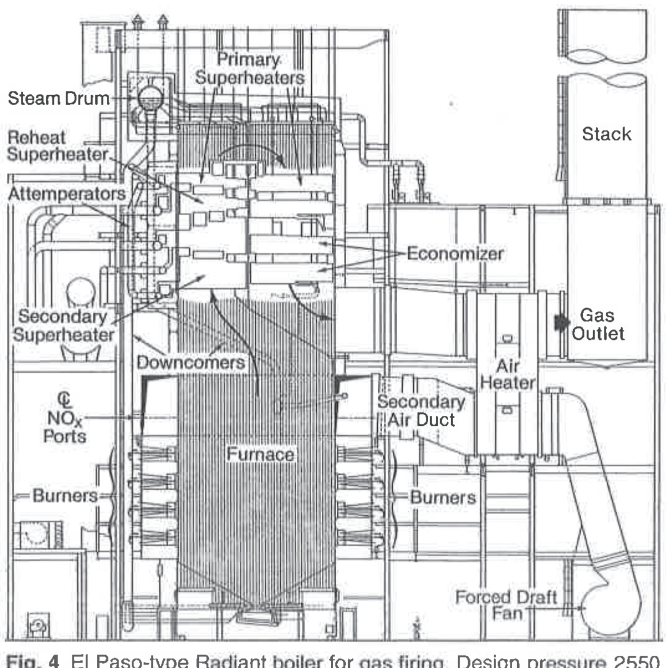


Fig. 4 El Paso-type Radiant boiler for gas firing. Design pressure 2550 psig (175.8 bar gauge); primary and reheat superheat temperatures 955F (513C); capacity 3,825,000 lb steam/h (482 kg steam/s).

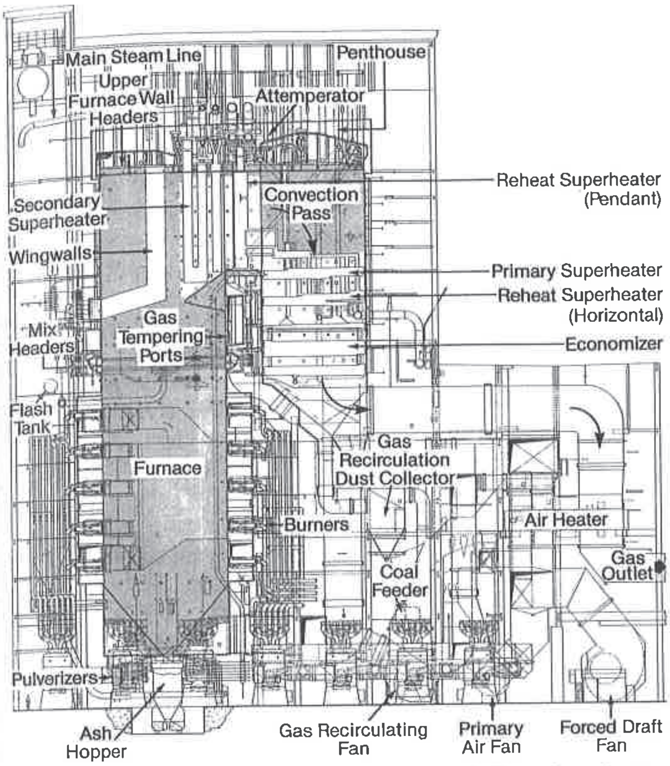


Fig. 5 Universal Pressure boiler for pulverized coal firing. Superheater outlet pressure 3845 psig (265.1 bar gauge); primary and reheat steam temperatures 1010F (543C); capacity 9,775,000 lb steam/h (1232 kg steam/s).

ters 24 through 31. However, all of these units share a number of fundamental elements upon which the site- and application-specific design is based.

The boiler evaluation begins by identifying the overall application requirements specified in Table 1. These are generally selected in an iterative process balancing initial capital cost, operating costs (especially fuel), steam process needs and operating experience. The selection of these parameters can dramatically impact cost and thermal efficiency. These are addressed further in Chapters 1 and 37.

From a boiler evaluation perspective, the temperature-enthalpy diagram shown in Fig. 6 (for a typical high pressure, single reheat unit) provides important design information about the unit configuration. In this example, the relative heat absorption for water preheating, evaporation and superheating are 30%, 32% and 38%, respectively. Reheating the steam increases the total heat absorption by approximately 20%. For cycles at supercritical operating pressures, a second stage of reheat may be added. For process applications, only the preheating and evaporation steps may be required.

Boilers can be designed for subcritical or supercritical pressure operation. At subcritical pressures, the furnace enclosure is cooled by constant-temperature boiling water, and the flow circuits must be designed to accommodate the two-phase steam-water flow and boiling phenomena addressed in Chapter 5. At supercritical pressures, the water acts as a single-phase fluid with a continuous increase in temperature as it passes through the boiler. These designs require special consideration to avoid excessive unbalances in metal temperatures due

Table 1
Use-Derived Specifications for Boiler Design

Specified Parameter	Comments
Steam use	Flow rates, pressures, temperatures — for utility boilers, the particular power cycle and turbine heat balance.
Fuel type and analysis	Combustion characteristics, fouling and slagging characteristics, ash analysis, etc.
Feedwater supply	Water source, analysis and economizer inlet temperatures.
Pressure drop limits	Gas side and steam side.
Government regulations	Including emission control requirements.
Site-specific factors	Geographical and seasonal characteristics.
Steam generator use	Base load, cycling, etc.
Customer preferences	Specific design guidelines such as flow conditions, equipment preferences and steam generator efficiency.

to variations in heat pickup in different flow circuits. In addition, special heat transfer phenomena must also be addressed. (See Chapter 5.)

Two basic fluid circulation systems are used — natural circulation and once-through. In natural circulation systems which operate at subcritical pressures, water is only partially evaporated in the boiler circuits producing a steam-water mixture at the tube outlets. Steam-water separation equipment is provided to separate the steam and water, supply saturated (dry) steam to the superheater and recirculate water back to the boiler circuits. Natural circulation is the result of the density difference between the hot and cold legs of the loop. Indus-

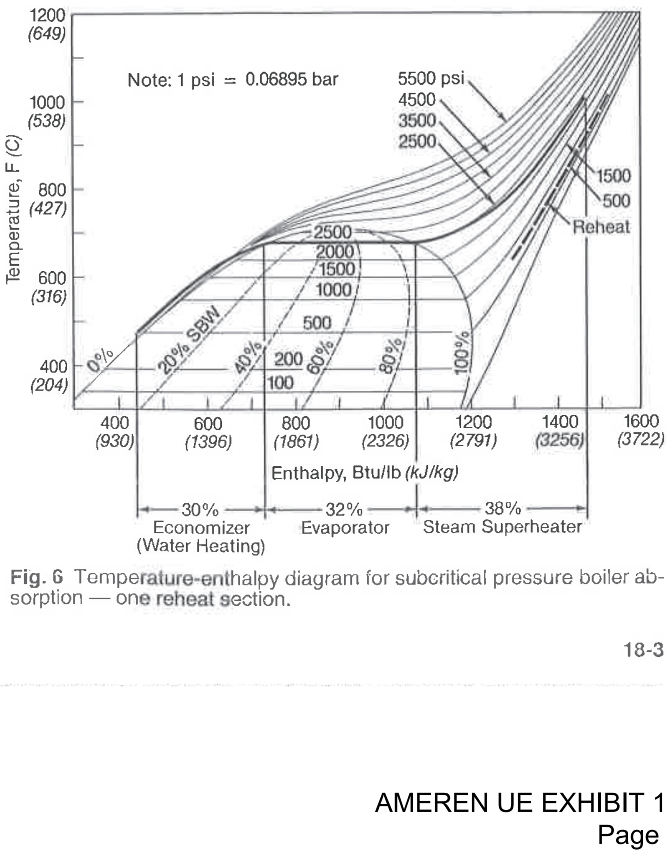


Fig. 6 Temperature-enthalpy diagram for subcritical pressure boiler absorption — one reheat section.

try accepted water chemistry limits are typically less stringent, especially at low pressures, and steam-water pressure drop inside the tubes is less of an issue. In B&W UP once-through designs, the steam drum and internal steam separation equipment are eliminated and a separate startup system added. UP boilers have been designed for both subcritical and supercritical operation. At supercritical pressures, the system can increase overall power cycle efficiency but at a higher initial capital cost. More precise operation is needed (see Chapter 43) and more stringent water treatment is required. (See Chapter 42.) Hybrid recirculating and once-through systems are presented in Chapter 5.

Design criteria Within the preceding framework, the important items which must be accomplished in boiler design are the following:

1. Define the energy input based upon the steam flow requirements, feedwater temperature, and an assumed or specified boiler thermal efficiency.
2. Evaluate the energy absorption needed in the boiler and other heat transfer components.
3. Perform combustion calculations to establish fuel, air and gas flow requirements. (See Chapter 9.)
4. Determine the size and shape of the furnace, considering the location and space requirements of the burners or other combustion system, and incorporating sufficient furnace volume for complete combustion and low emissions. Provision must be made for handling the ash contained in the fuel and cooling the flue gas so that the furnace exit gas temperature (FEGT) meets design requirements.
5. Determine the placement and configuration of convection heating surfaces. The superheater and reheater, when provided, must be placed where the gas temperature is high enough to produce effective heat transfer, yet not so high as to cause excessive tube temperatures or ash fouling. All convection surfaces must be designed to minimize the impact of slag or ash buildup and permit surface cleaning without erosion of the pressure parts.
6. Provide sufficient saturated boiler surface to generate the remainder of the steam not generated in the furnace walls. This can be accomplished with or without an economizer.
7. Design pressure parts in accordance with applicable codes using approved materials.
8. Provide a gas-tight boiler setting or enclosure around the furnace, boiler, superheater, reheater and economizer.
9. Design pressure part supports and the setting for expansion and local conditions, including wind and earthquake loading.

Fuel selection and specification are particularly important. Boiler systems are designed for specific fuels and frequently will encounter combustion, slagging, fouling or ash handling problems if a fuel with characteristics other than those originally specified is fired. All potential boiler fuels must be assessed to determine the most demanding fuel.

Procedures for optimizing steam pressure and temperature, and for evaluating the value of specific auxiliary equipment in a given application are outlined in Chapter 37.

As discussed in Chapter 9, the boiler or combustion efficiency is usually evaluated as 100 minus the sum of the heat losses expressed as a percentage. Chapter 9 provides procedures for calculating the corresponding fuel, air and gas flow rates.

Enclosure surface design

The furnace of a large pulverized coal-, oil- or gas-fired boiler is essentially a large enclosed volume where fuel combustion and cooling of the combustion products take place prior to their entry into the convection pass tube bundles. Excessive gas temperatures entering these tube banks could lead to elevated metal temperatures or unacceptable fouling and slagging. Heat transfer to the furnace enclosure walls is basically controlled by radiation. The walls can be cooled by either boiling water (subcritical pressure) or high velocity supercritical pressure water.

The convection pass enclosure contains the horizontal and vertical downflow gas passes, where most of the superheater, reheater and economizer surfaces are located. (See Fig. 3.) These enclosure surfaces can be water or steam cooled.

The furnace enclosures and convection pass enclosures are usually made of water-cooled tubes in an all-welded membrane construction. These enclosures have also been made from tangent tube construction or closely spaced tubes with an exterior gas-tight seal. For a membrane construction, the tube wall and membrane surfaces are exposed on the furnace side to the combustion process while insulation and lagging (sheet metal) on the outside protect the boiler, minimize heat loss and protect operating personnel. (See Chapter 22.)

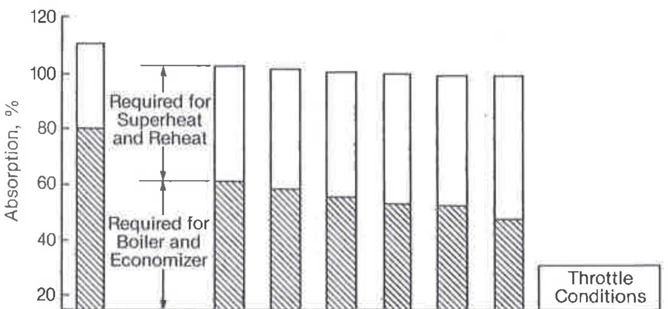
Furnace size versus cycle requirements Besides providing the volume necessary for complete combustion and a means to cool the gas to an acceptable FEGT, the furnace enclosure also provides much of the steam generating surface in a boiler. These roles may not perfectly match one another. In coal-fired units, the minimum furnace volume is usually set to provide the fuel ash specific FEGT. Frequently this results in too much evaporator surface in high pressure boilers and too little surface in low pressure units to meet the thermodynamic requirements for the desired steam exit temperature.

Fig. 7 illustrates the effect of the steam cycle and the operating pressure and temperature on the relative energy absorption between the boiler/economizer and superheater/reheater. As the pressure and temperature increase, the total unit absorption for a given power production progressively declines because of increased cycle efficiency. The boiler and economizer absorption represents the relative amount of heat added to the entering feedwater to produce saturated steam or reach the critical point in a supercritical pressure UP boiler. As the operating pressure increases, the amount of heat required to produce saturated steam declines. Conversely, the amount of heat required for the superheater and reheater increases. The change in required boiler/economizer absorption may not seem significant. However, a 1% shift in absorption is equivalent to approximately 10F (6C) of superheat or reheat temperature.

On a drum unit, the furnace and water-cooled convection pass enclosure walls are the boiler surface. On low pressure units, the amount of heat absorbed in the furnace is usually not adequate to produce all the saturated steam required, and a boiler bank is installed after the superheater. (See Figs. 1 and 2.) On a high pressure unit, the heat absorbed by the furnace and economizer is adequate to produce all the saturated steam required. As the furnace size increases, the economizer can be made smaller to produce the same amount of steam. It can be envisioned that as the furnace is made larger, some point will be reached where an economizer will not be required. However, as the furnace is enlarged to reduce the FEGT, too much steam would be produced, leaving insufficient energy in the flue gas to meet design superheat and reheat temperatures. This situation requires special design features to meet all of the design goals.

Furnace design criteria The furnace is basically a large open volume enclosed by water-cooled walls for combustion. Its shape and volume are established by the selection of fuel and combustion system. For wall firing using circular burners (see Chapter 13), minimum clearances between individual burners, between burners and walls, as well as between burners and the furnace floor are established based upon physical clearances and functional criteria for complete combustion. These clearances prevent fuel stream and flame interaction, assure complete combustion, avoid unacceptable flame impingement on the walls (which could lead to tube overheating or excessive deposits) and minimize the formation of nitrogen oxides (NO_x). The maximum fuel input rate, number of burners and the associated clearances establish: 1) the furnace cross-sectional area, 2) the height of the combustion zone, 3) the height of the overfire air injection zone (if used), and 4) the distance between the burner zone and the furnace floor. Where the fuel is burned on stokers, the furnace cross-sectional area is established by a specified heat release rate per unit bed area. (See Chapter 15.)

As discussed in detail in Chapter 13, combustion system design and its impact on furnace volume and shape have become complex and critical as emissions limits have been reduced. Not only are low NO_x burners such as the B&W DRB-XCL™ used to reduce NO_x emissions, but other techniques, such as in-furnace staging with overfire air (NO_x ports), fuel reburning and reactant injection, can also be considered for NO_x reduction. In addition, some techniques for furnace sorbent injection to reduce sulfur dioxide (SO_2) emissions have also been developed. (See Chapter 35.) Each technique can have an impact on the furnace size and configuration.



								Throttle Conditions
1250 (86.2)		1800 (124.1)	1800 (124.1)	2100 (144.8)	2400 (165.5)	2520 (173.8)	3500 (241.3)	psig (bar gauge)
950 (510)		950 (510)	1000 (538)	1000 (538)	1000 (538)	1000 (538)	1000 (538)	SH deg F (C)
—		950 (510)	1000 (538)	1000 (538)	1000 (538)	1000 (538)	1000 (538)	RH deg F (C)

Fig. 7 Relative heat absorption by operating pressure and steam temperature.

The overall furnace height is established by several criteria. For clean fuels, such as natural gas, the furnace volume and height are generally set to cool the combustion products to an FEGT which will avoid superheater tube overheating. For fuels such as coal and some oils which contain significant levels of ash, the furnace volume and height are established to cool the products of combustion to an FEGT which will prevent excessive fouling of the convection surfaces. The relationships between furnace volume and FEGT are explored in Chapter 4. The specific effects of ash on furnace design are discussed below and in Chapter 20. The furnace height must also be set to provide at least the minimum time to complete combustion and to meet minimum clearance requirements from the burners and NO_x ports to the arch and convective surface.

Ash effects In the case of coal and to a lesser extent with oil, an extremely important consideration is ash in the fuel. If this ash is not properly considered in the unit design and operation, it can deposit on the furnace walls, sloping surfaces and throughout the convection pass tube banks. Ash not only reduces the heat absorbed by the unit, but it increases draft loss, erodes pressure parts and eventually can result in unit outages for cleaning and repairs. (See Chapter 20.)

In coal-fired furnaces the ash problems are the most severe. There are two general approaches to ash handling: the dry ash or dry-bottom furnace and the slag-tap or wet-bottom furnace.

In the dry ash furnace, which is particularly applicable to coals with high ash fusion temperatures, a hopper bottom (Figs. 3 and 5) and sufficient cooling surface are provided so that the ash impinging on furnace walls or the hopper bottom is solid and dry; it can be removed essentially as dry particles. When pulverized coal is burned in a dry ash furnace, about 80% of the ash is carried through the convection banks. The chemistry of the ash can have a dramatic impact on the furnace volume necessary for satisfactory dry-bottom unit operation. This is illustrated in Fig. 8 which compares the furnace volume required for a nominal 500 MW boiler burning a low slagging bituminous or subbituminous coal to that for a high slagging lignite coal. The relationships defining the furnace size requirements are discussed in detail in Chapter 20.

With many coals having low ash fusion temperatures, it is difficult to use a dry-bottom furnace because the slag is molten or sticky; it clings and builds up on the furnace walls and hopper bottom. The slag-tap or wet-bottom furnace has been developed to handle these coals. The most successful form of the slag-tap furnace is that used with Cyclone furnace firing. (See Figs. 7 and 8, Chapter 14.) The furnace comprises a two-stage arrangement. In the lower part of the furnace, sufficient gas temperature is maintained so that the slag drops onto the floor in liquid form. Here, a pool of liquid slag is maintained and tapped into a slag tank containing water. In the upper part of the furnace, the gases are cooled below the ash fusion point, so ash carried over into the convection banks is dry and does not cause excessive fouling. Because of high NO_x emissions, slag-tap designs are used infrequently on new boilers.

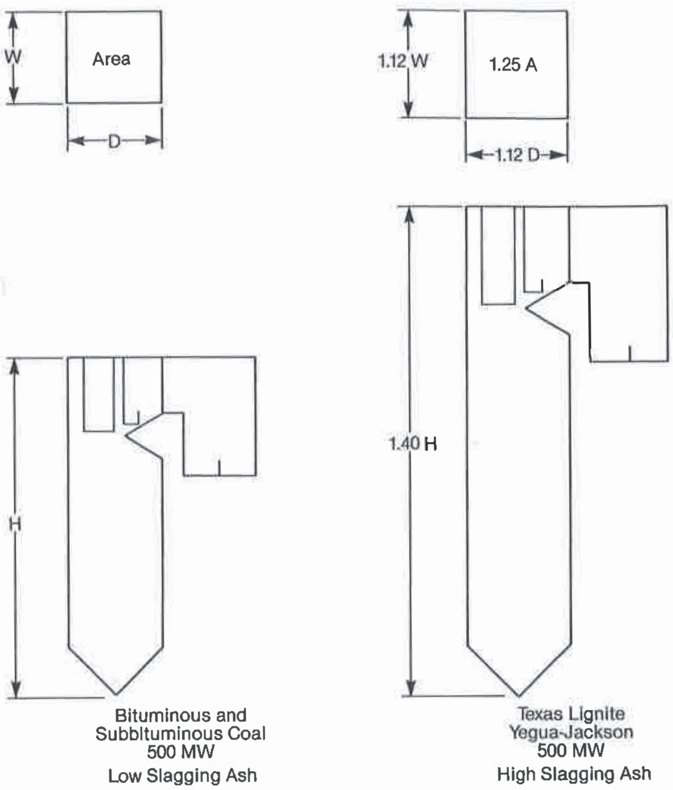


Fig. 8 Boiler size comparison for alternate coal types.

Water-cooled walls Most boiler furnaces have all water-cooled membrane walls. This reduces maintenance on the furnace walls and reduces the temperature of the gas entering the convection bank to the point where slag deposits and superheater corrosion can be controlled by sootblowing.

Furnace wall tubes are spaced on close centers to obtain maximum heat absorption while maintaining tube and membrane temperatures and thermal stresses within limits. The membrane panels (see Fig. 9) are composed of tube rows spaced on centers wider than a tube diameter and joined by a membrane bar securely welded to the adjacent tubes. This results in a continuous wall surface of rugged, pressure-tight construction capable of transferring the required heat from the furnace gas to the steam-water mixture in the tubes. The width and length of individual panels are suitable for economical manufacture and assembly, with bottom and top headers shop-attached prior to shipment for field assembly. Size limitations are frequently governed by shipping clearances and erection constraints. Membrane construction with refractory lining is used in the lower furnace walls of Cyclone-fired, refuse and fluidized-bed units.

Convection boiler surface

Some designs include boiler tubes as the first few rows of tubes in the convection bank. The tubes are spaced to provide gas lanes wide enough to prevent ash and slag pluggage and to facilitate cleaning for dirty fuels. These widely spaced boiler tubes, known as the *slag screen* or *boiler screen*, receive heat by radiation from the furnace and by radiation and convection from the combustion gases passing through them. Another option is the use of water- or steam-cooled wingwalls in the upper furnace. This provides additional radiant boiler surface in the furnace while allowing the furnace size to be optimized.

In the larger high pressure units, superheater surface generally forms the furnace outlet plane. The gas temperature entering the superheater must be high enough to give the desired superheat temperature with a reasonable amount of heating surface and the use of economical materials. The arrangements in Figs. 1 through 5 illustrate various configurations of superheater surface at the furnace outlet. Also, to optimize superheater design, widely spaced steam-cooled platens or wingwalls may be incorpo-

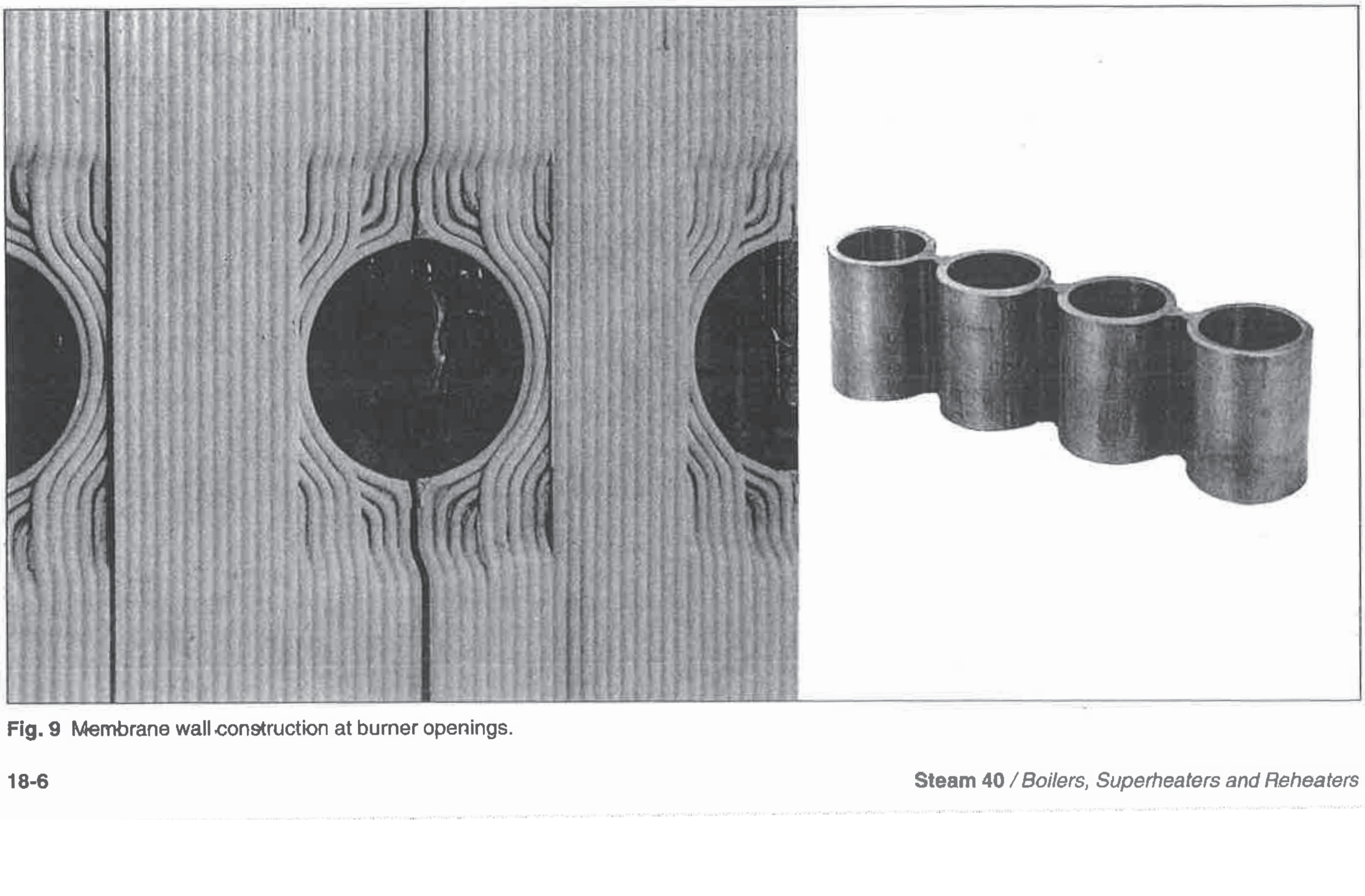


Fig. 9 Membrane wall construction at burner openings.

rated in the upper furnace as shown in Figs. 3 and 5. Fig. 10 shows a plan arrangement of convection surface and the change in average gas temperature.

Design of boiler surface after the superheater depends on the type of unit (industrial or utility), desired gas temperature drop and acceptable gas side resistance (draft loss) through the boiler surface. Typical arrangements of boiler and superheater surface for various types of boilers are illustrated in the preceding section on boilers. In designing convection heating surfaces, the objective is to establish the proper combination of several parameters to provide the desired gas temperature drop with allowable flue gas resistance. These parameters are tube diameter and length, tube spacing, number and orientation of tubes, and gas baffling.

In the gas side design of convection surfaces, the quantity of heating surface (ft^2 or m^2) needed for a given load or duty (Btu/h or W) is generally inversely related to gas side flow resistance or pressure loss. Design changes which increase resistance, such as tightening tube spacing perpendicular to flow, result in higher heat transfer rates (Btu/h ft^2 or W/m^2). This in turn reduces the amount of heating surface needed to carry the desired total thermal load. An optimal gas mass flux and design result from balancing the capital cost of the heating surface against the operating cost of fan power needed to overcome the resistance.

For a given gas flow rate, a considerably higher gas film heat transfer coefficient, heat absorption and draft loss result when the gases flow at right angles to the tubes (crossflow) compared to flow parallel to the tubes (longflow). Gas turns between tube banks generally add draft loss and maldistribution with little benefit to heat absorption. Turns leaving the upstream bank and entering the downstream bank of tubes should therefore be designed for minimum resistance and optimum distribution.

Determination of the amount of radiation and convection boiler surface required for a specified heat transfer is illustrated in Chapter 21.

Numerical analysis

When one dimensional analysis and experimental data are insufficient for design alone, numerical computer modeling can be used to determine local heat transfer and flue gas conditions in boilers, superheaters and reheaters. These numerical tools are complex multi-dimensional computer codes which solve conservation equations for mass, momentum, energy and species. (See Chapters 3 and 4.)

A numerical model of the superheater and/or reheater regions generally includes the boiler enclosure, burners and any other geometric aspects which may affect the flow. Downstream and upstream components are sometimes added to provide the best boundary conditions to the area of interest. In applying these computer codes, model size and complexity must be balanced against the associated computer calculation time required. Required numerical inputs include:

1. geometric data — boiler and heat exchanger surface configurations,
2. performance data — heat exchanger pressure drops and heat absorptions,
3. operating data — stoichiometry and furnace exit gas temperature,

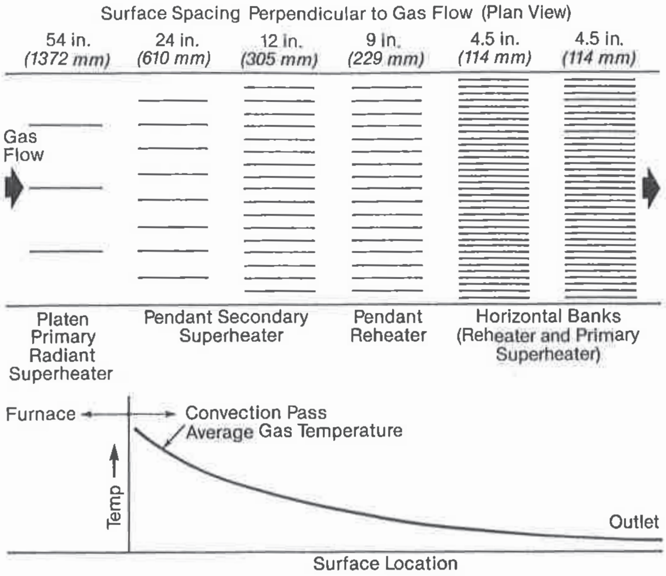


Fig. 10 Schematic plan arrangement of convection surface and change in average gas temperature.

4. burner setup — vane angles and turbulence intensity, and
5. fuel properties — fuel and flue gas quantities, composition and coal fineness, if applicable.

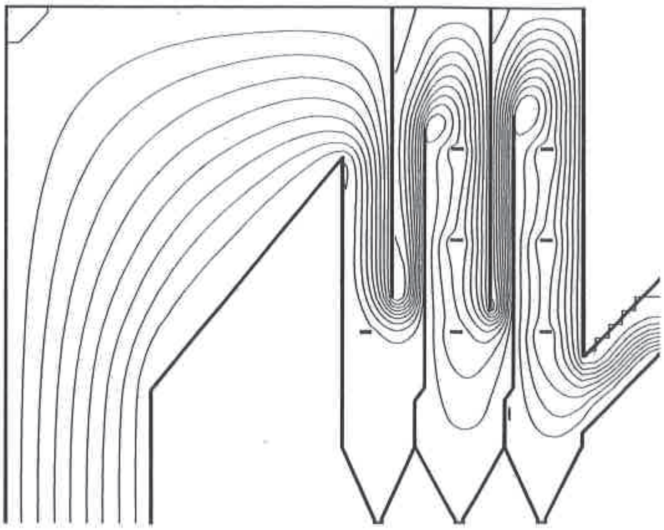
Use of operating data along with predictions based on company standards improves the accuracy of the model by tuning it to actual conditions. This is a necessary step because actual local conditions, e.g., ash thickness and property variation, are generally not known. Often, published data and sound engineering judgment are relied upon to obtain otherwise unavailable information. A numerical model can then be run once the boundary and inlet conditions are calculated and fluid properties are known.

After the model has been successfully run, the results can be postprocessed to determine the dominant physical characteristics. Further postprocessing can be used to obtain derived quantities such as slagging resistance or mixing effectiveness.

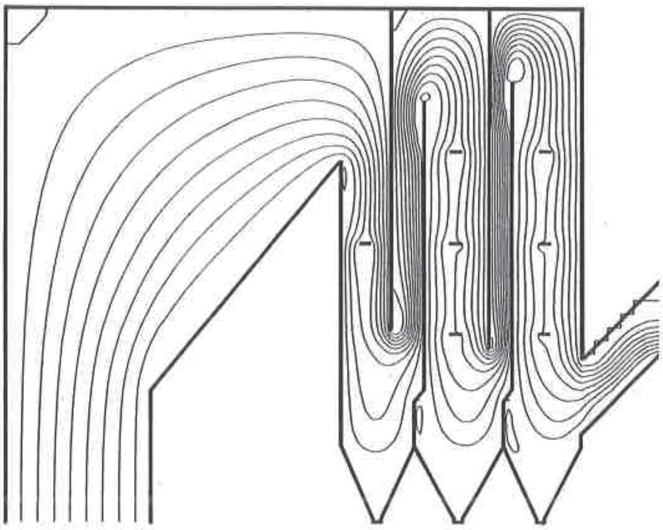
Numerical analyses fall into three general categories: 1) verification of proposed designs, 2) evaluation of design modifications, and 3) investigation of localized problems. Figs. 11, 12 and 13 provide sample numerical computer model results for illustration purposes. Figs. 11a and 11b are typical streamline plots containing curves that are tangent to the gas flow in a proposed boiler design. The design was modified as shown because the amount of heat transfer surface being bypassed by the gas flow lowered the expected heat absorption. Such evaluation of design modifications can be performed prior to fabrication to reduce long term costs and improve unit performance.

A second numerical model application includes an analysis to determine the optimum NO_x port arrangement in a boiler system. Such an analysis can use mixing effectiveness to rank the individual arrangements. Fig. 12 contains plots of fuel-air stoichiometry at an elevation 15 ft (4.6 m) above the NO_x ports before (a) and after (b) NO_x port adjustment. The more uniform distribution with improved mixing shown in (b) results in less NO_x formation.

Numerical modeling is also well suited to investigate boiler problems. The model takes into account the actual boiler geometry and performance characteristics, i.e., heat



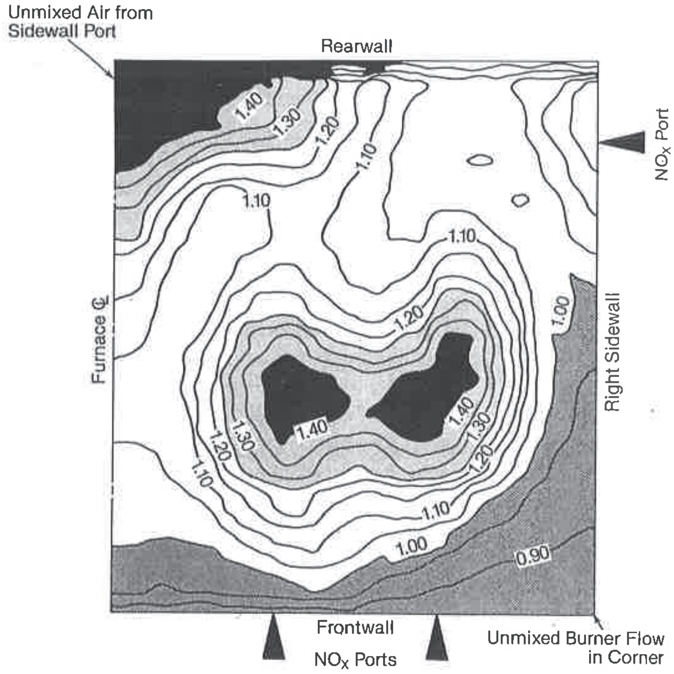
a) Original Convection Pass Baffle Arrangement



b) Improved Convection Pass Baffle Arrangement

Fig. 11 Numerical modeling results — flow streamlines.

a) Initial Mixing Configuration



b) Improved Mixing Configuration

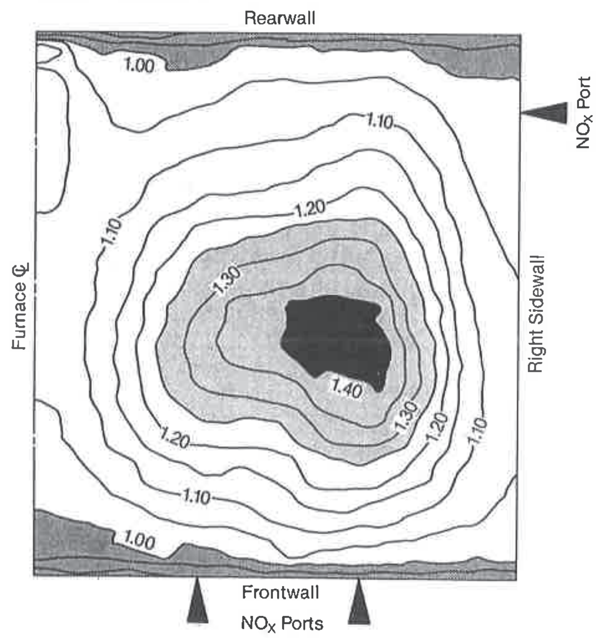


Fig. 12 Numerical modeling results — local combustion fuel-air stoichiometry.

absorption, temperature and pressure drop. The model also provides a more detailed and clearer performance picture than can be attained using analytical, field or laboratory methods. As an example, a unit was experiencing excessive slag accumulation on the lower leading edge of the secondary superheater inlet bank. Numerical modeling was used to determine the cause of the slagging and to evaluate several design modifications to reduce it. High velocities and associated high ash loadings were found to occur on the lower surfaces with a bias toward the sidewalls. Temperature contour plots for the as-built nominal case and for a larger furnace arch case (shown in Fig. 13) indicated that increasing the size of the arch moved the hotter furnace gases away from the secondary superheater problem area, virtually eliminating the hot ash impaction. This reduction of about 100F (56C) in the maximum temperature entering the secondary superheater enhanced the unit's resistance to slagging.

Accurate modeling of boilers and boiler components depends heavily upon the availability of information to determine the proper inlet conditions, boundary conditions and fluid properties. Careful qualification of the model is required and is augmented by known global characteristics such as pressure drop and heat absorption. Correct application of accurate information to develop numerical models will improve designs and determine the cause of localized problems.

Design of pressure parts

Boilers have achieved today's level of safety and reliability through the use of sound materials and safe practices for determining acceptable stresses in drums, headers, tubes and other pressure parts. Boilers must be designed to applicable codes. Stationary boilers in the United States

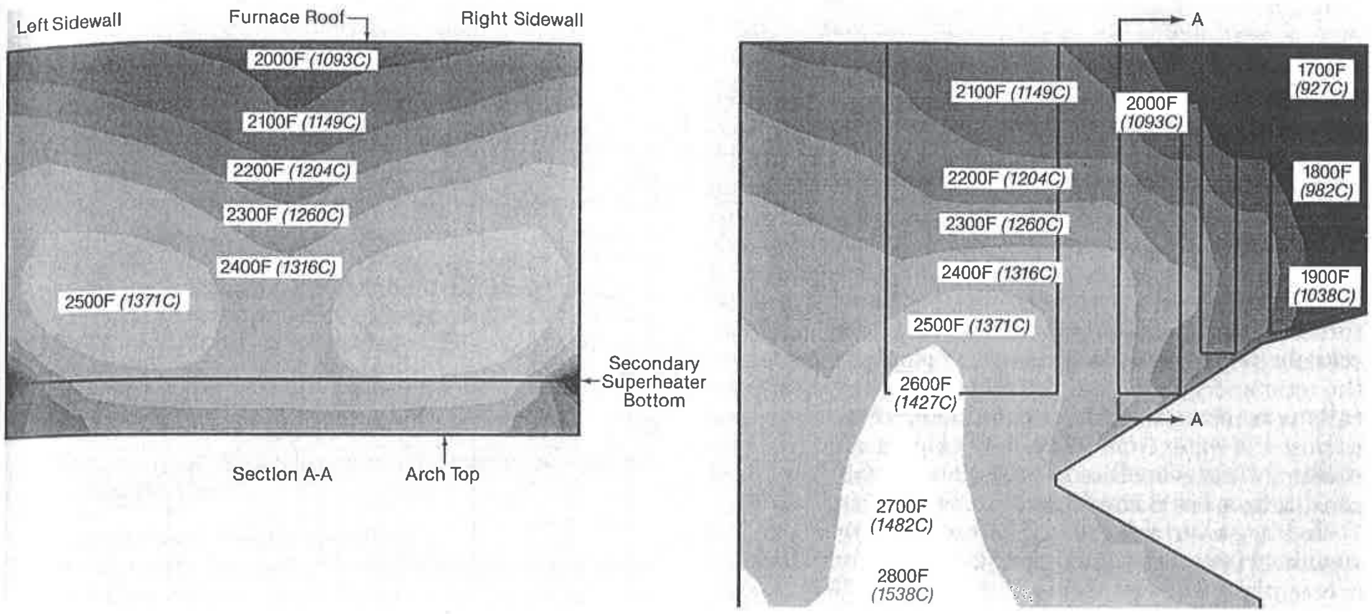


Fig. 13 Numerical modeling results — flue gas temperatures at furnace exit.

(U.S.) are designed to the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code. (See Chapter 7 and Appendix 2.) The allowable design stress depends on the maximum temperature to which the part is subjected and, therefore, it is important that pressure part design temperatures are known and not exceeded in operation. Drum boiler enclosure material temperatures are a function of upset spot heat flux, design pressure, metal conductivity and the saturation temperature corresponding to the maximum boiler operating pressure. These parameters are also used to determine each tube outside diameter and thickness. For boiler tubes, temperatures are maintained at known levels by providing a sufficient flow of water to prevent the occurrence of CHF, or critical heat flux phenomena. (See Chapter 5.) An adequate saturated water velocity must exist for each tube and particular attention must be given to high heat flux zones and sloped tubes with heat on top.

Because steam drums have thick walls, it is necessary to limit the heat flow through them to avoid excessive thermal gradients during startup, shutdown and normal operation. This is particularly important where the drum is exposed to flue gas. Where the drum is penetrated by a number of tube holes, the flow of water through these holes serves to cool the drum wall. Where the heat input through a drum would be too high because of high gas temperature or velocity, insulation may be provided on the outside of the drum or the drum relocated out of the heat.

In a drum-type boiler, steam separation equipment is provided to maintain steam moisture and solids at acceptable levels. (See Chapter 5.) In once-through boilers, all moisture is evaporated in the tubes, so that boiling and superheating occur sequentially. In boilers of this type, steam purity depends on maintaining adequate feedwater purity. (See Chapter 42.)

Boiler safety valves constitute very important protection items. (See Chapter 23.) The ASME Code stipulates that the boiler design pressure must not be less than the high-set safety valve relief pressure. As a practical matter, to avoid unnecessary losses and maintenance from frequent popping of the safety valves, the first valve should be set to relieve at not less than the boiler operating pressure plus 5%. The operating pressure in the steam drum, in turn, depends on the pressure required at the point of use and the intervening pressure drop. As an example, where the steam is used in a turbine, the boiler operating pressure is determined by adding the turbine throttle pressure and the pressure drop through the steam piping, nonreturn valve, superheater and drum internals at maximum unit steam flow.

Boiler enclosure

The methods used to provide a tight boiler setting and a tight enclosure around the superheater, reheater, economizer and air heater are described in Chapter 22.

Boiler supports

Furnace wall tubes are usually supported by the headers to which they are attached, and generating bank tubes and screens are supported by the drum or headers to which they are connected. As discussed in Chapter 7, the following considerations for proper support design are important:

1. The tubes must be arranged and aligned so that they are not subjected to excessive bending-moment stresses in supporting the weight of the tubes, headers, drums, attachments and fluid within. When the unit is bottom supported, the tubes must satisfy column buckling requirements.
2. The holding strength of the tube seats must not be exceeded.
3. Provision must be made to accommodate the expansion of the pressure parts. For a top supported unit, the hanger rods which tie the pressure parts to structural steel must be designed to swing at the proper angle. They must be long enough to withstand the movement without excessive stresses in the rods or the pressure parts. Bottom supported boilers should be anchored only at one point, guided along one line and allowed to expand freely in all other directions. To reduce the frictional forces and resultant stresses in the pressure parts, roller saddles or mountings are desirable for bottom supported heavy loads.

Superheaters and reheaters

Advantages of superheat and reheat

When saturated steam is used in a turbine, the work done results in a loss of energy by the steam and subsequent condensation of a portion of the steam, even though there is a drop in pressure. The amount of work that can be done by the turbine is limited by the amount of moisture that it can handle without excessive turbine blade wear. This is normally between 10 and 15% moisture. It is possible to increase the amount of work done with moisture separation between turbine stages, but this is economical only in special cases. Even with moisture separation, the total energy that can be transformed to work in the turbine is small compared to the amount of heat required to raise the water from feedwater temperature to saturation and then evaporate it. Therefore, moisture content constitutes a basic limitation in turbine design.

Because a turbine generally transforms the energy of superheat into work without forming moisture, this energy is essentially all recoverable in the turbine. This is illustrated in the temperature-entropy diagram of the ideal Rankine cycle (shown in Fig. 6, Chapter 2). While this is not always entirely correct, the Rankine cycle diagrams in Chapter 2 indicate that this is essentially true in practical cycles.

The foregoing discussion is not specifically applicable at steam pressures at or above the critical pressure. In fact, the term *superheat* is not truly accurate in defining the temperature of the working fluid in this region. However, even at pressures exceeding 3208 psi (221 bar), heat added at temperatures above 705F (374C) is essentially all recoverable in a turbine.

The benefit of superheat is indicated by the reduction of cycle heat rate when steam temperatures entering the turbine are raised. For example, in a simple calculation of a 2400 psig (165.5 bar gauge) ideal Rankine cycle with a single reheat stage, an increase in superheat temperature from 900 to 1100F (482 to 593C) reduces the gross heat rate from approximately 7550 to 7200 Btu/kWh. This is more than a 4.5% efficiency improvement attributable to the superheat.

Superheater types

Two basic types of superheaters are available depending upon the mode of heat transfer from the flue gas. The original type was the *convection superheater*, for gas temperatures where the portion of heat transfer by radiation from the flue gas is small. With a unit of this design, the steam temperature leaving the superheater increases with boiler output because of the decreasing percentage of unit heat input that is absorbed in the furnace. This results in more heat available for superheater absorption. Because convection heat transfer rates are almost a direct function of gas flow rate and therefore boiler output, the total absorption in the superheater per pound of steam, and therefore steam temperature, increase with boiler output. (See Fig. 14.) This effect is increasingly pronounced the farther the superheater is located from the furnace and the lower the gas temperature entering the superheater.

A *radiant superheater* receives energy primarily by thermal radiation from the furnace with little energy from convective heat transfer. It usually takes the form of widely spaced [24 in. (609.6 mm) or larger side spacing] steam-

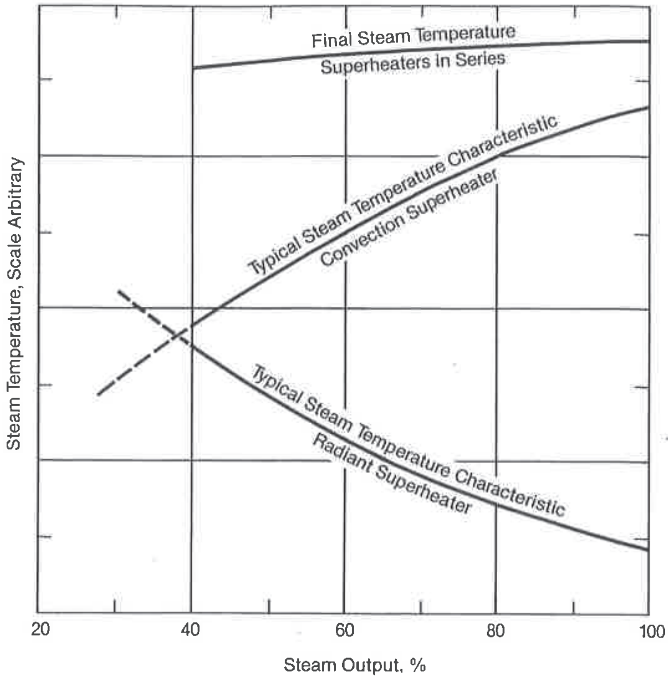


Fig. 14 A substantially uniform final steam temperature over a range of output can be attained by a series arrangement of radiant and convection superheater components.

cooled wingwalls or pendant superheat platens located in the furnace. It is sometimes incorporated into the furnace enclosure curtain walls. Because the heat absorption by furnace surfaces does not increase as rapidly as boiler output, the radiant superheater outlet temperature declines with an increasing boiler output, as shown in Fig. 14.

In certain cases the two opposite sloping curves have been coordinated by the series combination of radiant and convection superheaters to give a flat superheat curve over a wide load range, as indicated in Fig. 14. A separately-fired superheater can also be used to produce a flat superheat curve.

The design of radiant and convective superheaters requires extra care to avoid steam and flue gas distribution differences which could lead to tube overheating. Superheaters generally have steam mass fluxes of 100,000 to 1,000,000 lb/h ft^2 (136 to 1356 kg/m^2 s) or higher. These are set to provide adequate tube cooling while meeting allowable pressure drop limits. The mass flux selected depends upon the steam pressure and temperature as well as superheater thermal duty. In addition, the higher pressure loss associated with higher velocities improves the steam side flow distribution.

The fundamental considerations governing superheater design also apply to reheater design. However, the pressure drop in reheaters is critical because the gain in heat rate with the reheat cycle can be nullified by too much pressure loss through the reheater system. Therefore, steam mass fluxes are generally somewhat lower in the reheater.

Tube sizes

Bare cylindrical tubes of 1.75 to 2.75 in. (44.5 to 69.9 mm) outside diameter are typical in current superheaters and reheaters. Steam pressure drop is higher and alignment is more difficult with the smaller diameters, while larger diameters result in higher pressure stresses.

Recent designs have called for greater spans between supports for horizontal superheater tubes and for wider tube spacing or fewer tubes per row to avoid slag accumulation. The 2.5 in. (63.5 mm) tube has met these new conditions with minimum sacrifice of the smaller tube advantages; 2.75 or 3 in. (69.9 or 76.2 mm) tubes are used to advantage in some cases. When steam temperatures increase, the allowable stresses may force a return to the smaller diameter, thinner-walled tubes.

Bare tubes are used almost exclusively in superheaters. Extended surface on superheater tubes in the form of fins, rings or studs makes gas side cleaning difficult, and the added thickness can increase metal temperature and thermal stress beyond tolerable limits.

Relationships in superheater design

Effective superheater design must consider several parameters including:

1. the steam temperature specified,
2. the range of boiler load over which steam temperature is to be controlled,
3. the superheater surface required to give this steam temperature,
4. the gas temperature zone in which the surface is to be located,
5. the type of steel, alloy or other material best suited for the surface and supports,
6. the rate of steam flow through the tubes (mass flux or velocity), which is limited by the permissible steam pressure drop but which, in turn, exerts a dominant control over tube metal temperatures,
7. the arrangement of surface to meet the characteristics of the anticipated fuels, with particular reference to the spacing of the tubes to prevent accumulations of ash and slag or to provide for easy removal of these formations in their early stages, and
8. the physical design and type of superheater as a structure or component.

A change in any of these items may require a counterbalancing change in some or all of the other items.

The steam temperature desired in advanced power station design is typically the maximum for which the superheater designer can produce an economical component. Economics in this case requires the assessment of two interrelated costs — initial investment and the subsequent cost of upkeep to minimize operating problems, outages and replacements. The steam temperature desired is, therefore, based upon an iterative evaluation of variations in factors 3 through 7, past operating experience and the requirements of the particular project. Operating experience in recent years has resulted in the use of 1000F (538C) or 1050F (566C) steam temperatures for the superheat and reheat in nearly all new large utility boilers.

After the steam temperature is specified, the amount of surface necessary to provide this superheat must be established. This is dependent on parameters 5 through 8. Because there is no single correlation, this quantity must be determined by trial and error, locating the superheater in a zone of gas temperature that satisfies the design criteria. In standard boilers, the zone is fairly well established by unit arrangement and by the space designated for superheater surface.

After the amount of surface is established for the optimum location and tube spacing, the steam mass flux or velocity, steam pressure drop, and superheater tube metal temperatures are calculated considering material options and thickness requirements. The material determination is then made based on economic and functional optimization of the tubes, headers and other components. It may be necessary to compare several arrangements to obtain an optimum combination that:

1. requires an alloy of less cost,
2. gives a more reasonable steam pressure drop without jeopardizing the tube temperatures,
3. gives a higher steam mass flux or velocity to lower tube temperatures,
4. gives the tube spacings that minimize ash accumulations with various types of fuel,
5. permits closer spacing of the tubes, thereby making a more economical arrangement for a favorable fuel supply,
6. gives an arrangement of tubes which reduces the draft loss for an installation where this parameter evaluation is crucial, and
7. permits the superheater surface to be located in a zone of higher gas temperature, with a subsequent saving in surface.

It is possible to achieve a practical design with optimum economic and operational characteristics and with all criteria reasonably satisfied, but a large measure of experience and the application of sound physical principles are required for best results. The calculation methods for superheater performance are given in Chapter 21.

Relationships in reheater design

Superheaters and reheaters are similar in design, but the reheater is limited by the permissible steam pressure drop. Steam mass fluxes or velocities in reheater tubes should be sufficient to keep the difference between the bulk steam and metal surface temperature below 150F (83C). Ordinarily this is done with a pressure drop through the reheater tubes that is 4 to 5% of reheater inlet pressure. This allows another 4 to 5% pressure drop for the reheat piping and valves without exceeding the usual 8 to 10% total allowable pressure loss. The pressure drop allocated for piping is usually distributed with one-third to the cold (inlet) reheat piping and two-thirds to the hot (outlet) reheat piping.

Tube metals

Oxidation resistance, allowable stress and economics determine the materials used for superheater and reheater tubes. The use of carbon steel is maximized. However, carefully selected alloy steels are used where required. Additional information on these materials including maximum metal temperatures are given in Chapter 6.

Variations in basic superheater heat transfer surface arrangements are shown in Fig. 15. These variations in surface arrangement permit the economic tradeoff between material unit costs and differences in the quantity of surface required due to thermal-hydraulic considerations. The main purpose of the counterflow versus parallel flow comparisons is to show basic relationships between heat transfer surface requirements and the corresponding tube metal temperatures and requirements.

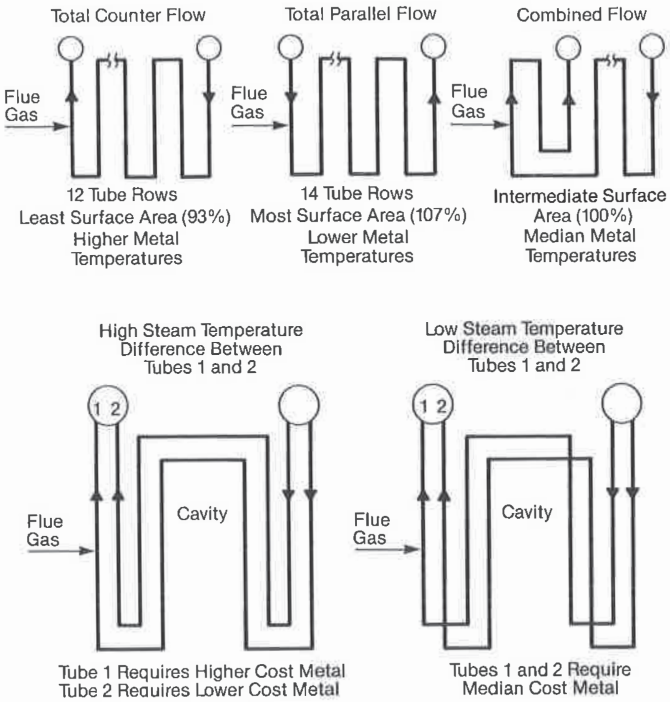


Fig. 15 Typical superheater heat transfer surface arrangements.

Supports for superheaters and reheaters

Because superheaters and reheaters are located in zones of relatively high gas temperature, it is preferable to have the major support loads carried by the tubes themselves. For pendant superheaters, the major support points are located outside of the gas stream, with the pendant loops supporting themselves in simple tension. Figs. 16a and 16b illustrate standard support arrangements for a pendant superheater outlet section with major section supports above the roof line. Where adequate side spacing is available and the ash cleaning is not abrasive, steam-cooled wraparound guides are used. Where abrasive ash cleaning is anticipated (coal firing), high chromium-nickel alloy ring-type guides are used. In addition, in the higher gas temperature zones, steam-cooled side to side ties are used to maintain side spacings. For closer side spaced elements, steam-cooled wraparounds are not practical and mechanical ties, such as D-links, are used to maintain alignment as shown in Fig. 17. In this case the clear backspacing between tubes and the size of attachments have been kept to a minimum. This serves to reduce the thermal stresses imposed on the tube wall. Fig. 17 also shows a typical arrangement of a pendant reheater section and illustrates the support of a separated bank by a special loop of reheater, which permits all major supports to be kept above the roof and out of the gas stream.

In horizontal superheaters, the support load is usually transferred to boiler or steam-cooled enclosure tubes or economizer stringer tubes by lugs, one welded to the support tubes and the other to the superheater tubes as indicated in Fig. 18. These lugs are made of carbon steel through high chromium-nickel alloy. Depending upon design considerations, they must slide on one another to provide relative movement between the boiler tubes and the superheater tubes. Saddle-type supports, also shown in Fig. 18, provide for relative movement between adjacent tubes of the superheater.

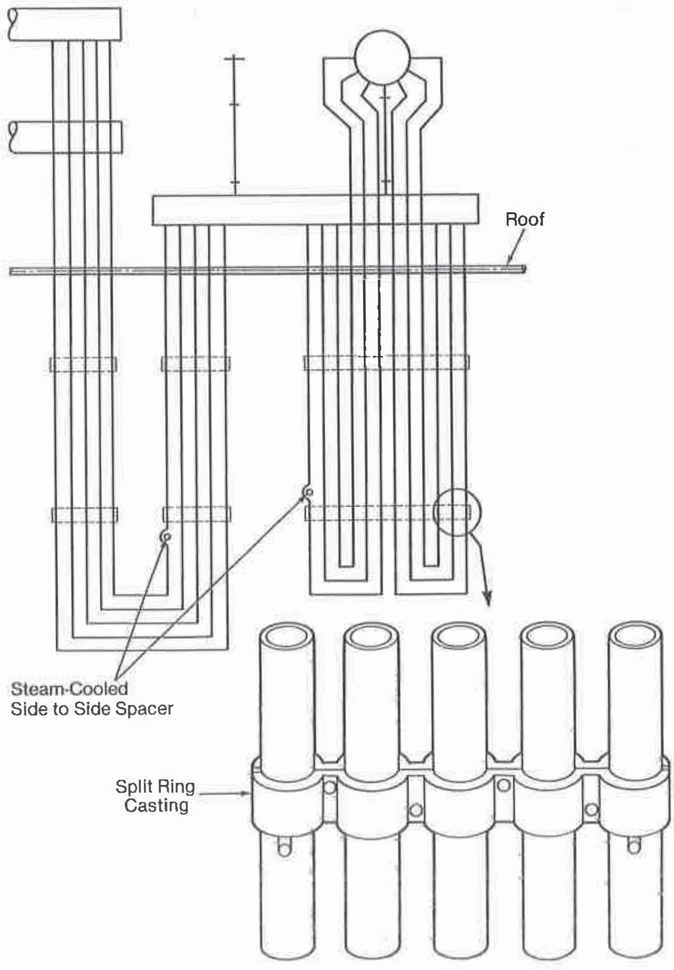


Fig. 16a Pendant superheater section with split ring casting supports.

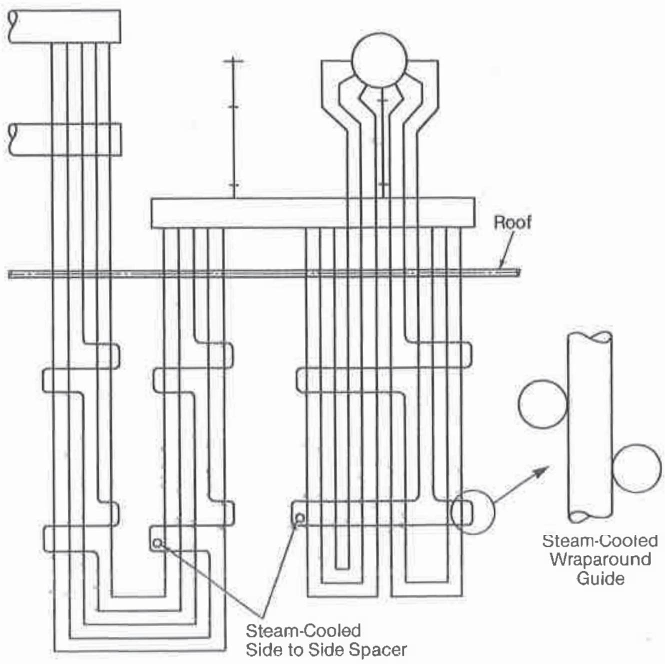


Fig. 16b Pendant superheater section with steam-cooled wraparound guide.

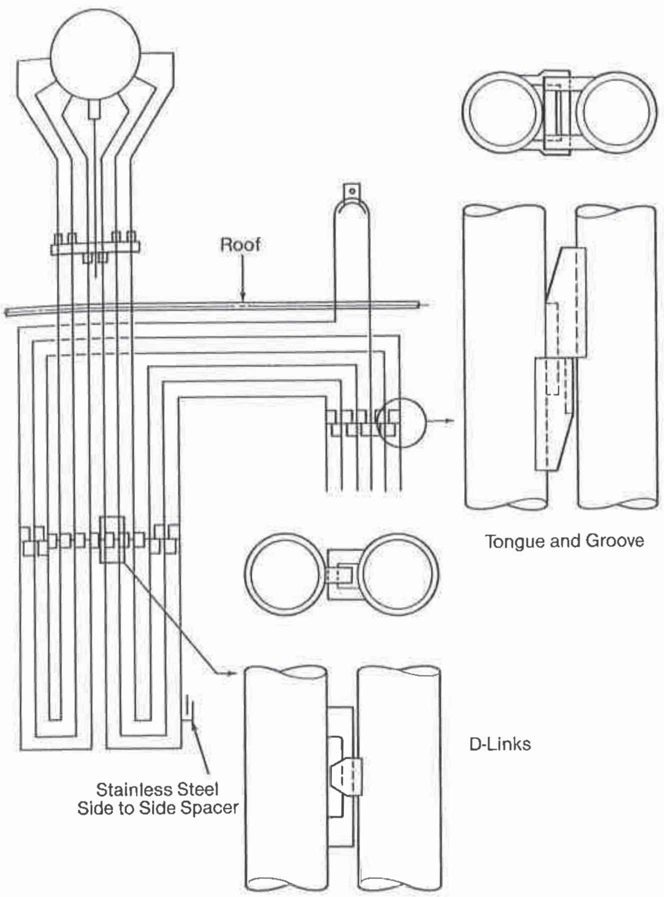


Fig. 17 Pendant reheater section with supports.

As units grow in size, the span of the superheater tubes may become so great that it is impossible to end-support these tubes. Most of the larger units use stringer tubes, generally hung from the economizer outlet, to support the superheater tubes. Tube spacing within the section is maintained by saddle-type supports.

Internal cleaning

The internal cleaning of superheater and/or reheater surfaces is not normally required; however, under certain circumstances these surfaces have been chemically cleaned. This is discussed in depth in Chapter 42. During startup, steam line blowing is used to remove residual scale, oils and residual debris.

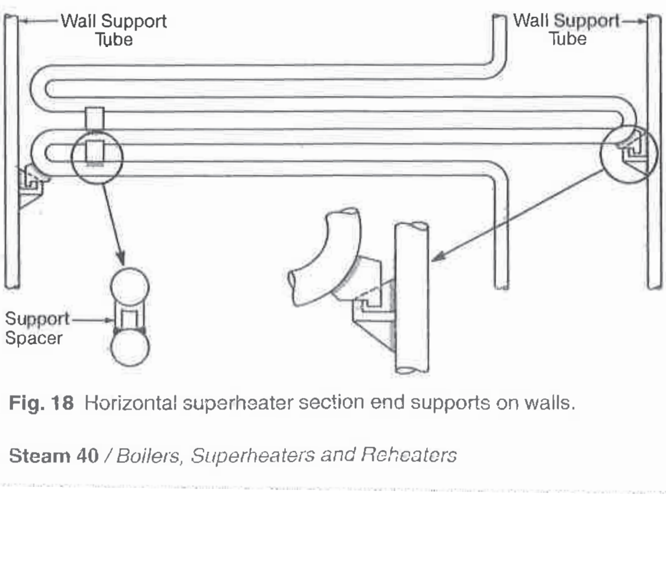


Fig. 18 Horizontal superheater section end supports on walls.

External cleaning and surface spacing

Units purchased today are designed for continuous operation, in some cases for 18 to 24 months between outages, so gas side cleanability is critical. Usually one year between outages is considered acceptable. To enhance cleanability, the superheater sections (platens) of modern utility boilers are spaced according to the gas temperature and the fuel fired. Fig. 10 illustrates the spacing for pulverized coal-fired units. The backspacing in the direction of gas flow is usually set at 0.50 to 0.75 in. (12.7 to 19.1 mm) clear space between tubes in the high temperature zones, with increased spacing allowable in the horizontal surface zones of less than 1500F (816C) entering gas temperature. These spacings are empirical; they are based on tube fouling and erosion experience and on manufacturing requirements. Surface arrangement and cleanability are discussed in Chapter 23 under *Sootblowers*.

Steam temperature adjustment and control

Improvement in the heat rate of modern boiler units and turbines results in large part from the high cycle efficiency possible with high steam temperatures. The importance of regulating steam temperatures within narrow limits is illustrated by the fact that a change of 35 to 40F (19 to 22C) corresponds to a change of about 1% in heat rate at pressures above 1800 psi (124.1 bar).

Other important reasons for accurate steam temperature regulation are to prevent failures due to excessive metal temperatures in the superheater, reheater or turbine; to prevent thermal expansion from dangerously reducing turbine clearances; and to avoid erosion from excessive moisture in the last stages of the turbine.

The control of temperature fluctuations from variables of operation, such as slag or ash accumulation, is important. However, superheater and reheat steam temperatures in steam generation are mainly affected by changes in steam output.

With drum-type boilers, steam output and pressure are maintained constant by firing rate, while the resulting superheat and reheat steam temperatures depend on basic design factors, such as total surface quantity and the ratio of convection to radiant heat absorbing surface. Steam temperatures are also affected by other important operating variables such as excess air, feedwater temperature, changes in fuel that affect burning characteristics and ash deposits on the heating surfaces, and the specific burner combination in service. In the Universal Pressure once-through boiler, which has a variable steam-water transition zone, steam output, pressure and temperature are controlled by the coordination of the firing rate and the boiler feedwater flow rate, leaving reheat steam temperature as a dependent variable. (See Chapter 41.) Standard performance practice for steam generating equipment usually permits a tolerance of ± 10F (6C) in a specific steam outlet temperature.

Definitions of terms

Adjustment is a change in the arrangement of heating surface which affects steam temperature but can not be used to vary steam temperature during operation. This could be the addition or deletion of component surface or boiler surface ahead of the superheater and/or reheater.

Control is the regulation of steam temperature during operation without changing the arrangement of surface. Examples are operation of attemperators, gas proportioning dampers and gas recirculation fans.

The *attemperator* is an apparatus for reducing and controlling the temperature of a superheated fluid passing through it. This is accomplished by spraying high purity water into an interconnecting steam pipe usually between superheater stages or upstream of a reheater inlet.

Effect of operating variables

Many operating variables affect steam temperatures in drum-type units. To maintain constant steam temperature, means must be provided to compensate for the effect of these variables.

Load As load increases, the quantity and temperature of the combustion gases increase. In a convection superheater (see *Superheater Types* above), steam temperature increases with load, the rate of increase being less the closer the superheater surface is to the furnace. In a radiant superheater (see *Superheater Types* above), steam temperature decreases as load increases. Normally a proportioned combination of radiant and convection superheater surface is installed in series in a steam generating unit to maintain substantially constant steam temperature over the control range of the unit. (See Fig. 14.)

Excess air For a change in the amount of excess air entering the burner zone there is a corresponding change in the quantity of gas flowing over the convection superheater; therefore an increase in excess air generally raises the steam temperature.

Feedwater temperature An increase in feedwater temperature causes a reduction in superheat because, for a given steam flow, less fuel is fired, less gas flows over the superheater and the gas temperatures are lower.

Heating surface cleanliness Removal of ash deposits from heat absorbing surfaces ahead of the superheater reduces the temperature of the gas entering the superheater and, subsequently, the steam temperature. Removal of deposits from the superheater surface increases superheater absorption and raises steam temperature.

Use of saturated steam If saturated steam from the boiler is used for sootblowers or auxiliaries, such as pump or fan drives, an increased firing rate is required to maintain constant main steam output; this raises the steam temperature.

Blowdown The effect of blowdown is similar to the use of saturated steam but is to a lesser degree because of the low enthalpy of water as compared to steam.

Burner operation The distribution of heat input among burners at different locations or a change in burner adjustment usually has an effect on steam temperature due to changes in furnace heat absorption rate.

Fuel Variations in steam temperature may result from changing the type of fuel burned or from day to day changes in the characteristics of a given fuel.

Adjustment

A power generating unit represents a large capital investment, and means should be provided at a reasonable cost for adjusting steam temperature to meet changing conditions. For instance, if a long term fuel change will have a considerable effect on steam temperature, it is good engineering practice to design for compensating physical alterations of the equipment.

Adjustment for regulating steam temperature is required when the operating conditions depart from the conditions on which the design is based. To provide the ultimate adjustment to meet such variations in operation, the design should accommodate anticipated changes at minimum expense.

The basic method of adjustment for regulating steam temperature is the addition or removal of superheater and/or reheater surface. A good design will provide an economical way of doing so.

On certain unit designs, adjustment is also possible by a reduction or increase in the amount of saturated surface ahead of the superheater surface. Such alterations modify the gas temperature at the inlet to these superheaters. If saturated surface is removed to increase steam temperature, this type of adjustment is relatively simple and, in general, costs less than the addition of superheater and/or reheater surface. However, the addition of saturated surface to decrease steam temperature can be difficult and expensive, or even impractical.

Limited use of a refractory coating on selected areas of water-cooled furnace surfaces is also permissible to increase gas temperatures entering the superheater surfaces. This can have a favorable effect on combustion and carbon loss, but refractory should not be added in areas where undesirable ash would deposit. This may also add to maintenance costs.

One of the simplest, least expensive and most effective means of adjustment in regulating steam temperature is to change the mass velocity of the gas flowing over the superheater elements by baffle modifications, if the unit design permits. Several standardized boilers, especially in the smaller sizes, have an adjustable baffle suitable for steam temperature regulation. This feature permits varying the steam temperature control range as much as 20%. The limit of variation is the effect on draft loss and efficiency. A 10F (6C) increase in the temperature of the gas at the boiler outlet reduces efficiency by about 0.25%.

Control

Control is necessary to regulate steam temperature within required limits in order to correct fluctuations caused by operating variables — particularly boiler load. Besides boiler load variation, ash deposition on heat transfer surfaces is the most frequent cause of steam temperature fluctuations. This condition can usually be corrected by changes in the sequence or frequency of sootblower operation. Selective operation of furnace wall blowers or implementation of unit load reductions to induce slag shedding from the furnace walls can reduce gas temperatures entering the superheater surfaces.

The time in which a turbine may be brought to full load is established by its manufacturer in accordance with a safe steam temperature-time curve. Because the temperature of the steam is directly related to the degree of expansion of the turbine elements and consequently the maintenance of safe clearances, this temperature must be regulated within permissible limits by an accurate control device.

The removal of feedwater heaters or pulverizers from service can impact steam temperature and require steam temperature control.

Among the means of control for regulating steam temperature are: attemperation, gas proportioning dampers, gas recirculation, excess air, burner selection, movable burners, divided furnace with differential firing, and separately-fired superheaters. With attemperation, steam temperature is regulated by diluting high temperature steam with low temperature water or by removing heat from the steam. By comparison, the other methods of control are based on varying the amount of heat absorbed by the steam superheating surfaces.

Attemperation Attemperators may be classified as two types — direct contact and surface. The direct contact design is exemplified by the spray type, where the steam and the cooling medium (water and saturated steam) are mixed. In the surface design, which includes the shell type and the drum type, the steam is isolated from the cooling medium by the heat exchanger surface. Surface-type attemperators are rarely used on current utility boiler designs, while spray attemperators are generally used on all units which have attemperation requirements.

The superheater attemperator may be located in one of two places: at some intermediate point between two sections of the superheater or at the superheater outlet. The ideal location for the superheater attemperator for *process control* would be at the superheater outlet. Control would be direct and there would be no time lag. However, problems with this location include that: 1) water may carry over into the turbine, and 2) the spray does not protect the superheater metals from overheating. The superheater attemperator, located between superheater stages, addresses these concerns and is therefore generally preferred. The steam temperature leaving the superheater does not exceed the maximum temperature desired. In addition, steam from the elements of the first stage superheater is so thoroughly mixed that it enters the second stage superheater at a uniform temperature. For reheat superheater applications, the attemperator is located at the reheater inlet.

The superheater spray attemperator, illustrated in Fig. 19, has proven most satisfactory for regulating steam temperature. High purity water is introduced into the superheated steam line through a spray nozzle at the throat of a venturi section within the line. Because of the spray action at the nozzle and the high velocity of the steam passing through the venturi throat, the water vaporizes, mixes with and cools the superheated steam. An important construction feature is the continuation of the venturi section into a thermal sleeve downstream from the spray nozzle; this protects the high temperature piping from thermal shock. This shock could result from nonevaporated water droplets striking the hot surface of the piping. The reheat attemperator is similar but does not have a venturi section.

The spray attemperator provides a quick acting and sensitive control for regulating steam temperature. It is important that the spray water be of highest purity, because solids entrained in the water enter the steam and may cause troublesome deposits on superheater tubes, piping or turbine blades. High pressure heater drains are a source of extremely pure water but require a separate high pressure corrosion resistant pump if used for attemperator supply. Normally, boiler feedwater is satisfactory, provided condenser leakage and makeup do not introduce too much contamination. The total solids concentration in the spray water should not exceed 2.5 ppm and the spray water should not add more than 40 ppb solids to the steam flow.

Three attemperator arrangements are possible depending on the boiler performance requirements:

1. *Single-stage attemperator* A single attemperator may be installed in each of the connecting pipes between two stages of superheat.
2. *Tandem attemperator* A single-stage attemperator with two spray water nozzles may be installed in series in the connecting pipe between two stages of superheat. This arrangement is used where the spray quantity exceeds the capacity of a single nozzle or where the required turndown can not be achieved with a single spray water nozzle. The usual application requires a spray control valve for each spray nozzle. The operation is sequential with the control valve for the downstream spray nozzle opening first and closing last.
3. *Two-stage attemperator* Two single-stage attemperators are used. The first unit is located in the connecting pipes between the first and second stages of superheat and the second is between the second and third stages of superheat. A spray control valve is required for each stage of spray. The first stage spray attemperator is used first, with the maximum spray flow based on a minimum allowable difference between the temperature of the steam leaving the attemperator and the saturation temperature. The second stage attemperator is used after the flow limit is reached on the first stage unit.

In most instances, the pressure loss from the boiler feed pump through the feedwater heaters, piping and boiler to the attemperator location in the superheater results in sufficient boiler feed pump discharge pressure to provide the required pressure differential in the attemperator system. In certain cases, however, the feed pump discharge pressure is not sufficient due to the low pressure loss in the boiler and feedwater system or due to the high pressure differential the spray systems require to suit particular boiler characteristics. In these cases, a booster stage in the boiler feed pump is desirable to raise the spray water to the required pressure. Other options are a separate spray water booster pump or an additional feed line valve to increase boiler side resistance at the required loads.

Over the past years there has been an increasing frequency of problems in industrial steam power cycles, involving deposits in the superheater and on turbine blading, which has resulted in failures. Many of these problems are due to impurities in the attemperator spray water. To as-

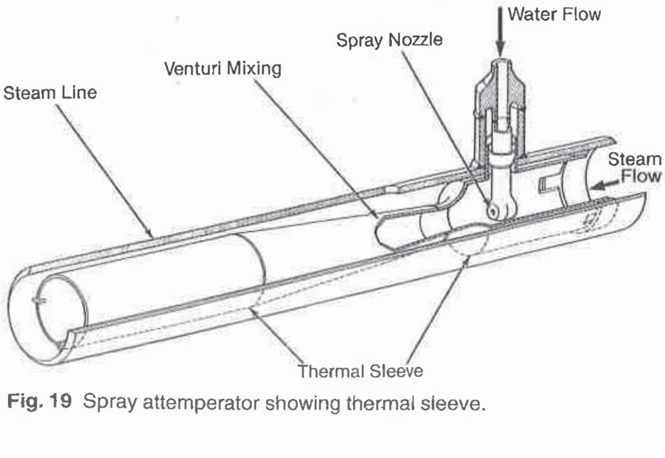


Fig. 19 Spray attemperator showing thermal sleeve.

sure high quality spray water, additional cleanup equipment can be installed on the condensate return and feedwater makeup.

For low pressure units with feedwater purity less than that required for spray attemperation, the B&W condenser attemperator system is an economical and reliable system used to produce high quality spray water. (See Chapter 25.)

Steam attemperation may be used on drum and once-through boilers as described in the *Bypass and Startup* section of this chapter.

UP boiler attemperator applications UP boilers are supplied without superheater spray attemperators. However, they can be supplied when specified to reduce variations in superheater outlet temperatures during transients. Spray attemperators are installed between the primary superheater outlet and the secondary superheater inlet, upstream of the high pressure superheater stop valve. Spray water is supplied from the UP boiler economizer inlet. Spray attemperation corrects main steam temperature deviations only on a temporary basis and must not be the primary means of steam temperature control. Under steady-state conditions, steam temperature is determined by the ratio of firing rate to feedwater flow. Special rules apply to surface and boiler design for spray attemperators supplied with a UP boiler.

Gas proportioning dampers As shown in Fig. 3, the horizontal convection tube banks in the back end of a boiler can be divided into two or more separate gas passes separated by a baffle wall. The use of dampers in these gas passes then permits proportioning of the gas over the heat transfer surfaces and the control of reheat and superheat temperatures.

Design considerations for such systems include the following:

1. Dampers must be placed in a cool gas zone to assure maximum reliability (typically downstream of all boiler heat transfer surfaces).
2. Draft loss through the unit could increase for some designs, particularly with alternate fuels, so this parameter must be optimized.
3. Control system design and tuning are critical because damper control response is slower than with spray attemperators. Therefore, spray attemperators are used for transient control.
4. Under maximum bias conditions, the gas temperatures at the dampers and the heat transfer surfaces nearest the dampers will be at their highest. These temperatures then set the metal design requirements.

Gas proportioning dampers are combined with spray attemperation for overall optimal steam temperature control systems. Spray attemperators provide for short term transient temperature control. The gas proportioning dampers provide longer term control and adjustment between superheat and reheat temperatures with a minimum impact on overall unit efficiency.

Gas recirculation Another method of controlling superheat or reheat is gas recirculation. As the name implies, gas from the boiler, economizer or air heater outlet is reintroduced to the furnace by fans and flues. For the sake of clarity, recirculated gas introduced in the immediate vicinity of the initial burning zone of the furnace and used for steam temperature control is referred to as *gas recirculation* and recirculated gas introduced near the furnace outlet and used for control of gas temperature is referred to as *gas tempering*. Fig. 20 shows an application of gas recirculation through the hopper bottom and gas tempering in the upper furnace of a Radiant boiler. In most instances the gas is obtained from the economizer outlet. The recirculated gas must be introduced into the furnace in a manner that avoids interference with the fuel combustion. The amount of recirculated gas is expressed as a percentage of the gas that remains downstream of its withdrawal point.

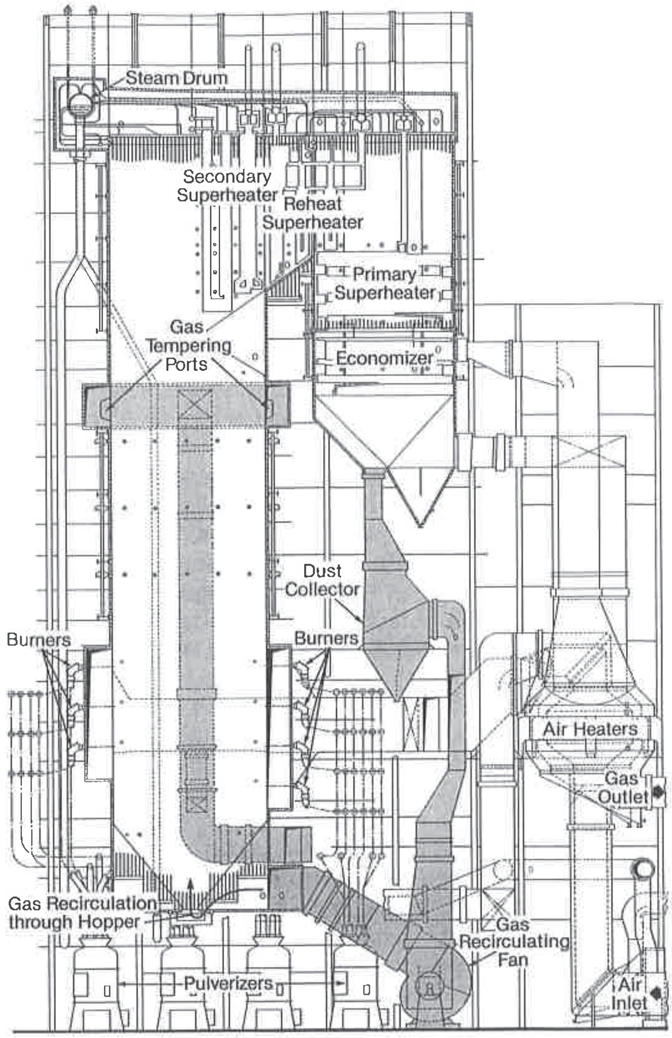


Fig. 20 Radiant boiler with gas tempering for gas temperature control and gas recirculation for control of furnace absorption and reheat temperature.

While recirculated gas may be used for several purposes, its basic function is to alter the heat absorption pattern within a steam generating unit. Recirculated gas has the special advantage of providing heat absorption adjustment that may be used as a design factor in initial surface arrangement and as a method of controlling the heat absorption pattern under varying operating conditions.

An important feature of recirculated gas is that its use changes only the pattern of heat absorption through a boiler; it has a negligible effect on the total boiler heat absorption and the weight of the gas sent up the stack. The thermal effect of recirculated gas depends on the amount of gas recirculated, the location of gas introduction and the furnace heat release rate.

Fig. 21 shows the variation in heat absorption with gas recirculation into the hopper. Introduction of gas at this

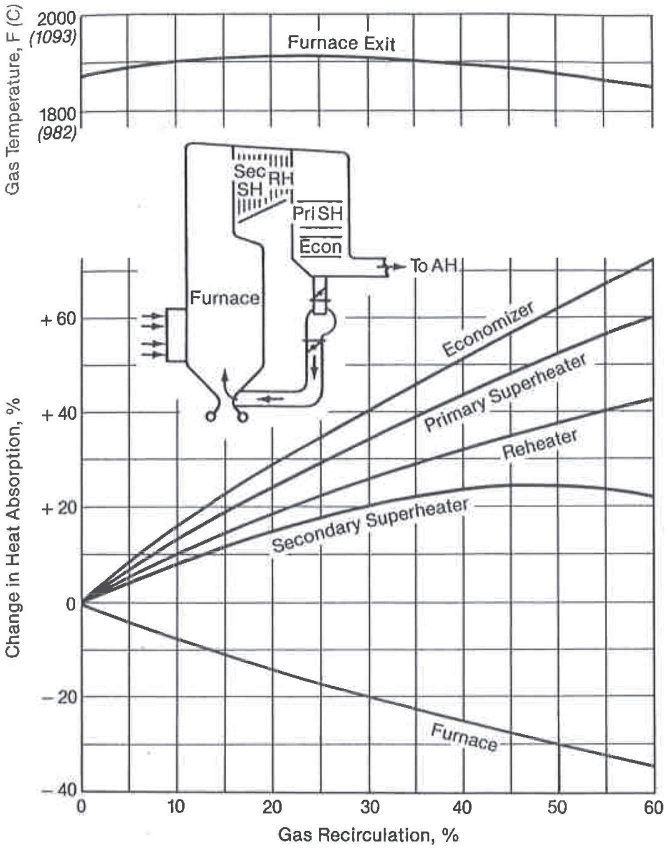


Fig. 21 Effect of gas recirculation on heat absorption pattern at a constant firing rate.

location produces a marked reduction in furnace absorption and increases the absorption of the convection section. Furnace heat absorption is primarily a function of the gas temperatures and gas temperature patterns throughout the furnace, because the heat is mainly transferred by radiation. Therefore, the introduction of gas recirculation into the furnace hopper reduces furnace absorption by altering the gas temperature pattern.

The major portion of the heat absorbed in the superheater, reheater and economizer is transferred by convection, which depends on gas temperature and gas mass flow rate. Both parameters are affected by gas recirculation. Therefore, when the mass velocity of gas flowing through a convection bank is increased by gas recirculation, the amount of heat transferred may increase, decrease or remain unchanged depending on changes in the relationship between the temperature and weight of the gas entering the bank. Fig. 21 illustrates a condition in which the gas temperature entering the secondary superheater (FEGT) is relatively unchanged by gas recirculation. Increasing the amount of gas recirculation, therefore, increases the heat absorption in the secondary superheater. The heat absorption in the reheater, primary superheater and economizer is also increased, with the greatest increase occurring at the cold end of the unit. This is a typical example of the variation in convection pass heat absorption pattern by gas recirculation.

While gas recirculation into the hopper always reduces furnace heat absorption, its effect on the FEGT depends mainly on furnace rating. This exit gas temperature may increase, decrease or, as shown in Fig. 21, be essentially unchanged by gas recirculation. In general, recirculated gas introduced in the hopper decreases the FEGT of a unit operating at high furnace loading and increases this gas temperature at low loading.

Fig. 22 illustrates the effect of introducing tempering gas at a point near the furnace exit. Because the portion of the furnace in which the bulk of heat absorption occurs is unaffected by the recirculated gas, the furnace heat absorption is decreased only slightly. There is, however, a large decrease in FEGT caused by dilution of the hot combustion gases with cooler recirculated gas.

In the case of tempering gas introduced near the furnace outlet, the reduction in FEGT is usually sufficient to overbalance the effect of gas weight increase, and the heat absorption in the secondary superheater is decreased. The effect of gas tempering on the primary superheater and economizer follows the pattern that was shown in Fig. 21, with the greatest change in heat absorption again occurring at the cold end. Because of the location of the reheater in Fig. 22, its absorption remains constant regardless of the percentage of gas tempering.

Figs. 21 and 22 illustrate the effect of introducing recirculated gas into the hopper or at a point adjacent to the furnace exit. Introduction of gas at intermediate points results in heat absorptions and gas temperatures between those shown. To show the effect of recirculated gas only, Figs. 21 and 22 have been based on constant firing rates.

Excess air Boiler operators have long known that the steam outlet temperature of a convection superheater on a drum type or separator type unit can be increased at fractional loads by decreasing the furnace heat absorption through an increase in the amount of excess combustion

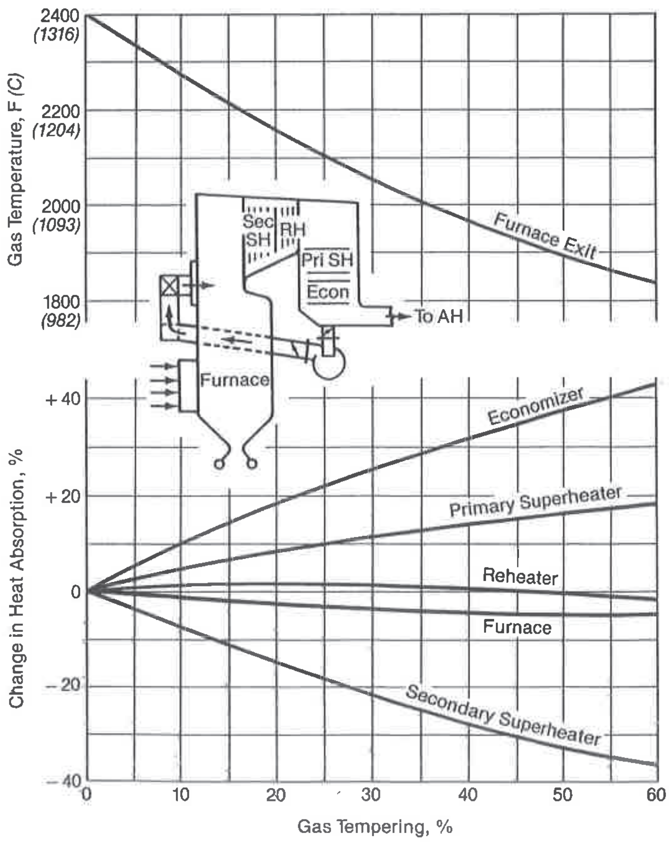


Fig. 22 Effect of gas tempering on heat absorption pattern at a constant firing rate.

air. The resulting greater weight of gas sent to the stack increases the stack loss. However, the drop in boiler efficiency can be offset by the increase in turbine efficiency.

Burner selection It is often possible to regulate steam temperature by selective burner operation. Higher steam temperatures may be obtained at less than full load by operating only the burners giving the highest furnace outlet temperature. When steam temperature reduction is required, firing may be shifted to the lower burners. This method of control can be improved by distributing the burners over an extended height of the burner wall or by installing a special burner near the furnace outlet.

Movable burners Regulation of steam temperature by changing the furnace absorption pattern can also be effected by using movable burners to raise or lower the main combustion zone in the furnace. Tilting burners are used for this purpose.

Differentially-fired divided furnaces In some divided furnaces, the superheater receives heat from one section of the furnace only, while the other section of the furnace generates only saturated steam or may include a reheater. Steam temperature is regulated by changing the proportion of fuel input between the two furnaces. This arrangement, similar in principle to the separately-fired superheater, was formerly widely used in marine practice. The differentially-fired divided furnace method of steam temperature control is no longer used today.

Separately-fired superheaters A superheater that is completely separate from the steam generating unit and independently fired may serve one or several saturated steam boilers. This arrangement is not generally economical for power generation, where a large quantity of high temperature steam is needed.

Reheat steam temperature

The need for regulating the temperature of reheated steam and the methods of adjustment and control to do so are, in general, the same as for superheated steam. However, the designer does not have the same freedom of action as when only a superheater is used. For instance, in a drum-type unit, the removal of boiler heating surface ahead of the superheater to increase superheat temperature or the removal of superheat surface itself to reduce superheat steam temperature results in increased reheat steam temperature, which may be undesirable. Furthermore, to reduce the gas temperature below slagging limits for convection tube banks and to give the desired steam temperature, such a large proportion of the total input must be absorbed in the furnace and in the superheater and reheater, so there is no boiler surface ahead of the superheater available for adjustment purposes. Some boilers have furnace division walls or water-cooled wingwalls, which may provide adjustment surface if furnace exit gas temperature permits.

Most of the control methods described for regulating steam temperature also affect the reheat temperature if a reheater is incorporated.

Bypass and startup systems

High pressure drum and once-through Universal Pressure boilers must respond to the change in required operating modes of large fossil fueled generating plants. This requires rapid, frequent and reliable unit startups and load changes to meet demand with economical electrical production. During startup or low load (less than 20%) conditions, regular water attemperator steam temperature control systems are usually ineffective as the outlet steam temperature tends to follow the flue gas temperature because the gas flow is substantially higher than the steam flow. However, provision must still be made to match the differing flow, pressure and temperature needs of the steam turbine and boiler. To address these special requirements, three bypass and startup systems have been developed for drum and once-through UP boilers.

Drum boiler bypass system

The B&W Radiant boiler bypass system minimizes startup time, controls shutdowns in anticipation of restarts, provides control of steam temperature to match turbine metal temperature, and allows dual pressure operation of boiler and turbine for better load response. These features reduce stresses in the turbine for improved turbine availability and reduced maintenance costs. The drum bypass system consists of a control system and carefully engineered steam valves and piping as shown in Fig. 23. Probes monitor gas temperatures at the superheater and reheater outlet tubes to permit control of firing rates and gas temperatures during startup.

Operational benefits

Decreased unit startup time The bypass system substantially reduces the time required for a cold startup because it controls the temperature differences of the saturated boiler surface, superheater surface and turbine.

This is accomplished by providing direct control of the steam temperature by mixing saturated steam with superheat and reheat outlet steam as shown in Fig. 23. This arrangement provides the desired steam temperature for the turbine without restricting the startup firing rate of the boiler.

Rapid load changing The system has a set of superheater stop and bypass control valves, which allow dual pressure operation with the throttle pressure controlled separately from drum pressure. This control system permits constant pressure operation of the major boiler components and variable pressure operation of the turbine during load changes. Dual pressure operation minimizes thermal stresses in the boiler and turbine. A dual pressure shutdown keeps the boiler near full pressure and the turbine metal near maximum temperature in preparation for a quick restart. In addition, it allows more rapid load changes than full variable boiler pressure operation.

A superheater bypass diverts excess steam from the boiler to the condenser, thereby separating firing rate from drum pressure control during shutdown and startup. This feature is used during shutdown and next morning restart to keep the turbine and steam piping near full temperature for quick starts and minimum stresses.

Operation The flexible bypass system can conform to most operating changes. For example, it can adopt to a turbine malfunction by adjusting the steam temperature and flow through any load range from synchronization to load points. Although the bypass system enhances unit operation, the boiler can be operated as a conventional nonbypass unit at any time. The following valves (see Fig. 23), with their associated functions, provide the required system flexibility:

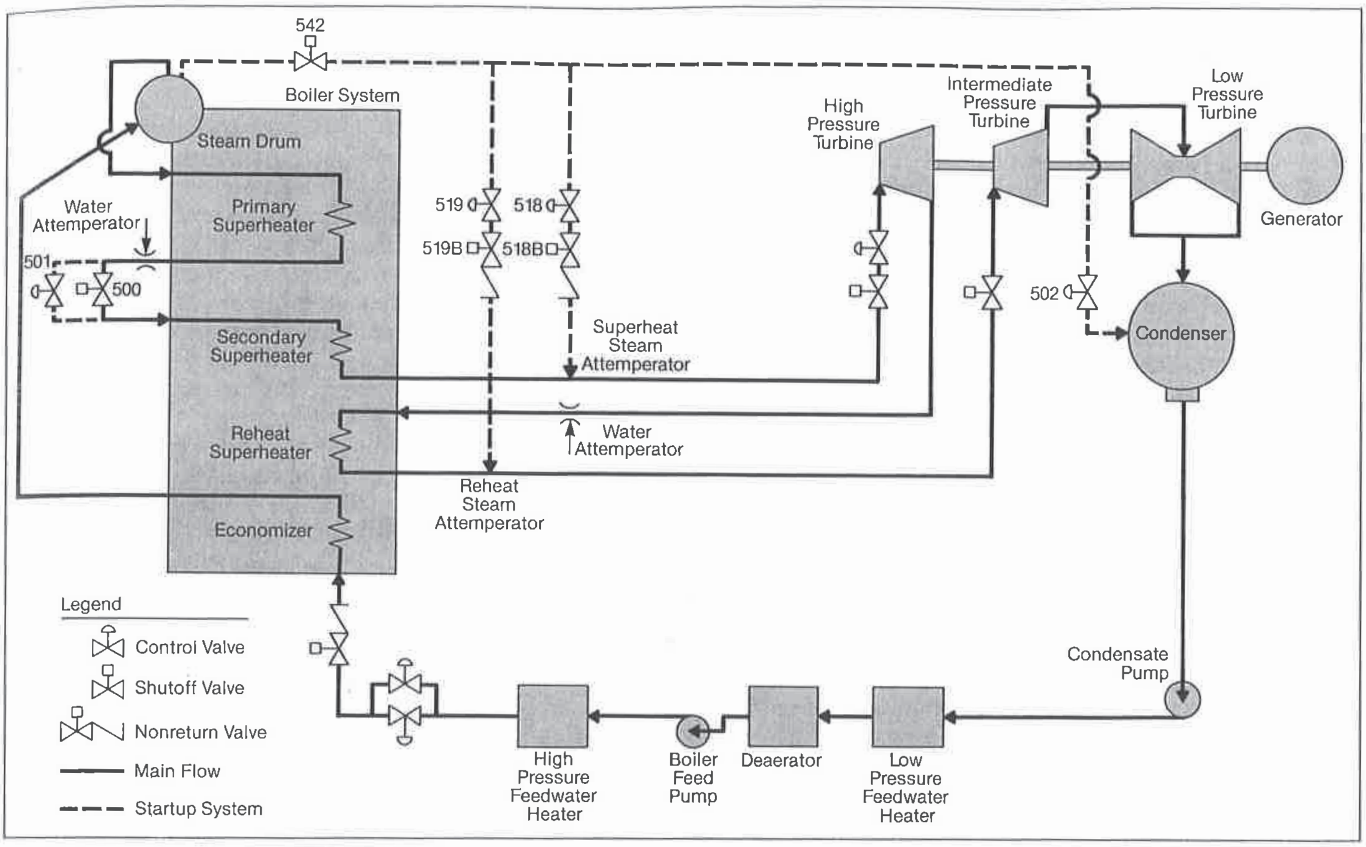


Fig. 23 Drum boiler bypass system schematic.

Valve No.	Function
500	*Secondary superheater stop valve* Separates the secondary superheater from the rest of the steam generator. Closed when using bypass system. Usually designed to open above 70% load.
501	*Secondary superheater stop valve bypass valve* Controls steam pressure leaving the secondary superheater below drum pressure for startup, variable pressure operation and steam attemperation. Design capacity is usually 70% of boiler design flow.
502	*Primary superheater bypass to condenser control valve* Permits firing for control of gas temperature entering the superheater to achieve high steam temperature during startup. Used primarily during hot starts, hot restarts and shutdowns. Closed at loads above 20%.
518	*Main steam outlet attemperator control valve* Reduces main steam temperature to match first-stage turbine metal temperature during cold starts and starts following weekend outages. Closed at loads above approximately 20%. The 518B shutoff valve provides isolation when the 518 valve is closed.
519	*Reheat outlet steam attemperator control valve* Reduces reheat steam temperature to match reheat bowl temperature during cold starts and postweekend starts. Closed at loads above approximately 20%. The 519B shutoff valve provides isolation when the 519 valve is closed.
542	*Primary superheater bypass system shutoff valve* Used to isolate bypass system during normal operation. Closed when the 518, 519 and 502 valves are closed.

Design considerations

Drum pressure and superheater outlet pressure control On restarts following overnight shutdowns, the gas temperature leaving the furnace has to be kept high. This maintains high main steam and reheat temperatures to match high turbine metal temperatures. The turbine must be rolled at a low pressure to prevent a large adiabatic throttling temperature drop when steam is admitted.

The saturation surface can be isolated from the secondary superheater surface by means of the secondary superheater stop valve (500) and bypass valve (501). The boiler can then be fired at the desired rate to raise steam temperatures or drum pressure while maintaining low pressures in the secondary superheater and entering the high pressure turbine. If the drum pressure increases too rapidly or reaches its limit, the primary superheater bypass valve (502) relieves pressure from the drum to the condenser, avoiding condensate waste. A better turbine metal temperature match is obtained with maximum heat input to the boiler and the firing limit dictated by metal protection of the convection pass surface.

Reheat steam temperature control Control of reheat steam temperature is maintained by the steam attemperator valve (519) during startups and shutdowns. Before steam is taken to the turbine, the reheater is without flow. The reheat metal absorbs heat from the flue gas and eventually reaches the flue gas temperature, which can approach

1000F (538C). When steam is first admitted to the turbine and passes through the reheater, the reheater outlet steam temperature rises very rapidly to the gas temperature level, resulting in a poor match with reheat bowl temperatures in the intermediate pressure turbine for cold starts and starts after a weekend shutdown. Water attemperators are not effective at low loads where reheater steam temperatures are controlled by flue gas temperatures in the reheater banks.

Reheat steam attemperation with saturated steam from the drum limits the rise of reheat outlet steam temperature when steam is first admitted to the turbine and offers positive reheat steam temperature control up to about 20% of full load.

Main steam temperature control The main steam attemperator valve (518) controls main steam temperature. Water attemperators are not effective during low boiler loads, where superheater steam temperatures are determined by the flue gas temperatures in the superheater banks. Therefore, the main steam temperature is controlled by a steam attemperator valve, using saturated steam from the drum to lower temperatures.

Design conditions

Cold start A cold start is defined as a unit startup from no drum pressure or from ambient furnace gas temperature. Turbine metal temperatures are less than 300F (149C) and prewarming is required.

Warm start A warm start is defined as a unit startup after a two day shutdown, such as a weekend shutdown. Turbine metal temperatures are at least 300F (149C). Remaining drum pressure can be as high as 500 psig (34.5 bar gauge).

Hot start A hot start is defined as a unit startup following a six to eight hour outage. The boiler is closed up to retain the maximum internal energy. Boiler drum pressure is quite high and vacuum will have normally been maintained. Turbine metal temperatures of 900 to 925F (482 to 496C) may exist following a controlled shutdown and a decay in turbine metal temperatures of 100F (56C) may occur during the shutdown. Therefore, minimum steam temperatures of 800 to 825F (427 to 441C) are needed during the restart. The steam temperature is usually less than the temperature limit for gas entering the superheater during a no-flow condition.

Hot restart If the unit is tripped but ready for a restart very quickly, turbine metal temperatures may be 1000F (538C) and drum pressure may be near the pressure existing at the time of the trip. To achieve the high gas temperatures needed, it may be necessary to continue unit firing and dump excess steam to the condenser.

Controlled shutdown When shutting the unit down, the bypass system may be used to facilitate the operation that follows the shutdown. If maintenance on the unit is scheduled, the turbine metal temperature and boiler pressure should be kept as low as possible while still carrying load. The throttle pressure is 501 valve controlled and necessary sprays to reduce steam and metal temperatures per the cooldown curves are controlled by the 518 and 519 valves. After the unit is tripped, the fans may remain on to cool the boiler.

Variable pressure operation Under most circumstances, the unit heat rate can be increased at partial loads if boiler pressure is reduced with load. This mode of operation is normally referred to as *variable pressure operation*.

The secondary superheater stop valve (500) and the stop valve bypass valve (501), used with the B&W bypass system, permit operating with the drum and primary superheater pressures at constant levels while the secondary superheater outlet pressure is varied with load. Operating in this mode maintains turbine steam temperatures at the design level over a greater load range than is possible under constant throttle pressure operation.

Maintaining drum and primary superheater pressure relatively constant at reduced loads permits rapid load pickup. The savings in pump power, which could be achieved by permitting drum pressure to vary with load, are eliminated when the rapid load pickup feature is used.

Startup without bypass system The unit can be started and operated without use of the bypass system. Fully opening the 500 valve allows starting up the unit like a conventional drum boiler.

UP boiler startup system

A key requirement of UP startup and bypass systems is the need for minimum design circulation flows in high heat absorbing circuits for cooling before the unit can be fired. Additional important features include providing a turbine bypass until steam pressure and temperatures are matched, reducing the bypass flow pressure and temperature before condenser and auxiliary equipment steam admission, recovering heat during startup, providing clean water for full startup, accelerating the startup processes, and providing greater unit flexibility through dual pressure boiler operation. UP boiler plants, while originally designed for base load operation, have been adapted to load cycling, including daily operation on the bypass system.

Two different UP systems are available — one for constant furnace pressure operation over the load range and a second for variable furnace pressure operation over the load range.

Constant furnace pressure startup system For constant furnace pressure the startup system has the steam separator (*flash tank*) located in a bypass that can be isolated from the boiler during normal operation. The general arrangement is shown in Fig. 24.

The boiler feed pump supplies the minimum required flow of feedwater during startup and low load operation to protect the furnace circuitry. Included in the startup circuitry are the economizer, furnace convection pass enclosure and primary superheater.

The fluid leaving the primary superheater (at full pressure) bypasses the secondary superheater through the pressure reducing valve (207) to the flash tank, where the steam-water mixture is separated during startup.

Water level in the flash tank is controlled by drain valves 230 and 241, with the 230 valve controlling the flow to the deaerator for maximum heat recovery. Excess water (above the capability of the deaerator) is discharged to the condenser through valve 241. If the drains are not within water quality limits (see Chapter 42), all of the flow is through valve 241 to the condenser and polishing system.

The 242 block valve remains closed until a level is established in the flash tank to assure that water does not enter the steam lines. Once a level is established, the 242 valve is opened so the deaerator steam line from the flash tank can be used to hold pressure in the deaerator (controlled by the 231 valve). This permits returning all of the drains to the condenser through the 241 valve during a hot cleanup without using an auxiliary steam source for maintaining deaerator pressure, and also serves to recover the heat in the flash tank steam during cleanup.

The low pressure steam line, the low pressure superheater nonreturn valve (205), and the downstream high pressure steam line connect the flash tank to the secondary superheater inlet downstream of the high pressure superheater block valve (200) and stop/control valve (401). After a water level is established in the flash tank, 205 valves normally open at 300 psig (20.7 bar gauge) and dry steam flows to the secondary superheater for warming steam lines. The turbine bypass valve (210) normally opens at 300 psig (20.7 bar gauge) main steam pressure to assist with warming and boiling out the superheater during the initial stage of startup. When sufficient steam is available, the turbine is rolled and placed on line.

Steam separated in the flash tank in excess of that required is relieved through the 240 valve to the condenser. This valve also acts as an overpressure relief valve to avoid tripping spring loaded safety valves on the flash tank. The 240 valve has an adjustable set point, which can be set to hold various flash tank pressures at particular load points during startup.

The entire bypass system is sized to handle minimum required flow during startup and to permit operating at minimum load on the flash tank.

The transition from operation on the flash tank to once-through flow is made at minimum load. As the steam entering and leaving the flash tank at this time is dry and superheated, the transition from flow through the bypass to once-through flow is accomplished, with minimum fluctuation in steam temperature, by opening the 200 block valve and the 401 combination stop/control valve and by closing the 207 and 205 bypass valves.

Variable throttle pressure Above minimum load the unit can be operated at constant or variable throttle pressure, with the 401 valve controlling the throttle pressure from minimum to 100% load, while maintaining full pressure in the upstream circuits.

The variable throttle pressure feature permits operating the unit with the throttle valves essentially wide open. This eliminates turbine metal temperature changes resulting from valve throttling and permits rapid load changes without being limited by turbine heating or cooling rates. Shutdown with variable throttle pressure maintains high temperatures in the turbine metals and permits rapid hot restarting.

Steam temperature control The means for controlling main steam and reheat temperatures at normal operating loads are not effective during startup or at very low loads.

The startup system shown in Fig. 24 includes provision for steam attemperation from the flash tank to the main and reheat steam outlet headers for precise control of steam conditions during startup to meet the turbine metal temperature requirements.

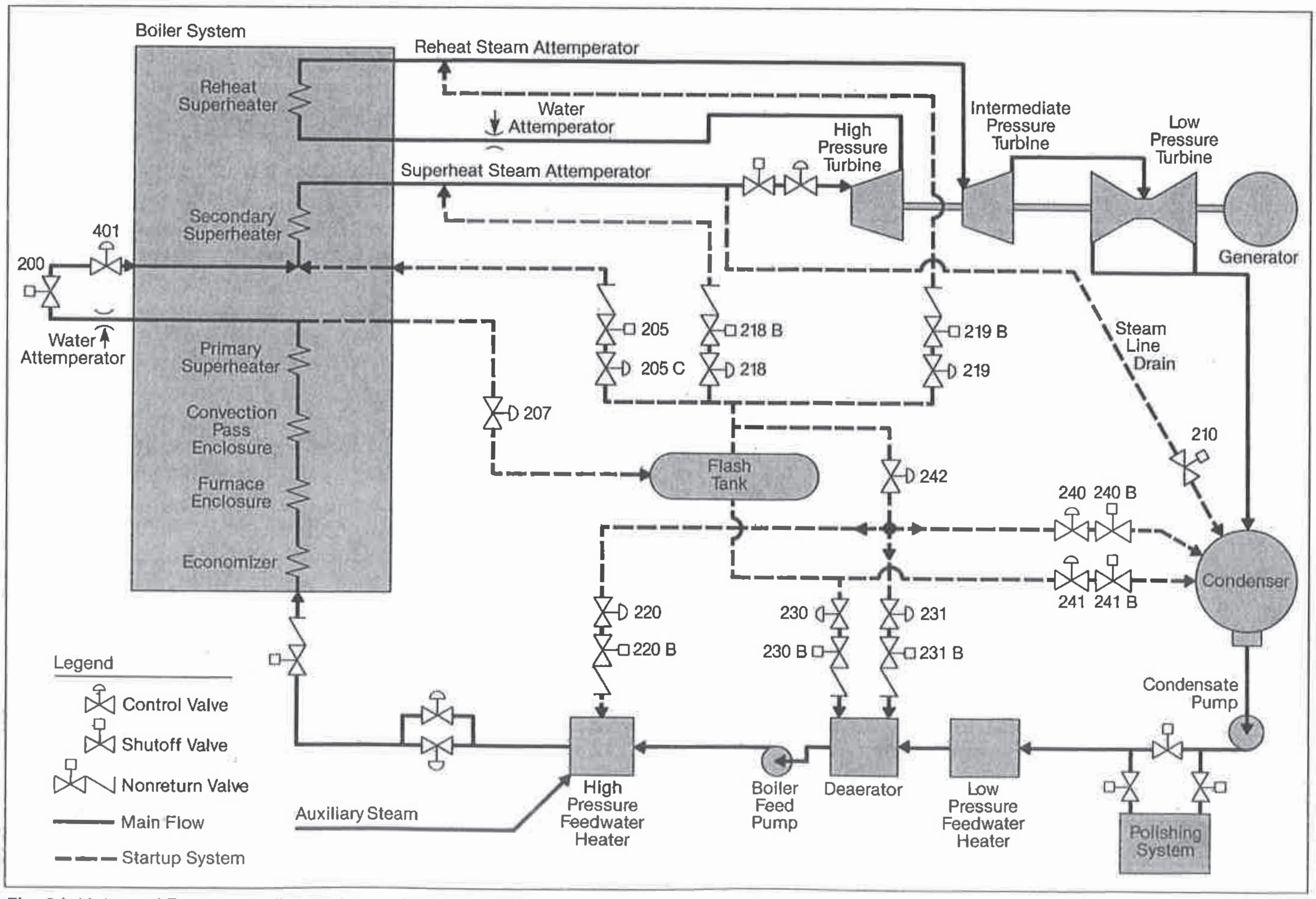


Fig. 24 Universal Pressure boiler startup system — constant pressure furnace operation.

The superheater outlet steam attemperator valve (218) is used at loads less than 20% to introduce saturated steam from the flash tank to the superheater outlet header. Initial rolling of the turbine, for a cold start, may be done with saturated steam passing from the flash tank through the 218 valve. This steam may be mixed with a limited quantity of steam passing through the 205 valves and the secondary superheater to control the high pressure turbine inlet temperature down to about 550F (288C). The 205C control valve achieves the necessary pressure drop between the flash tank and the secondary outlet header for attemperation.

The reheat outlet steam attemperator valve (219) is used at loads below about 20% of full load to introduce flash tank steam to the reheat outlet header. The ratio of flow through the attemperator valve to that through the high pressure turbine is limited mainly for turbine and turbine control considerations.

Overpressure relief The bypass system is also used to relieve excessive pressure in the boiler during a load trip. This is accomplished by the use of the 207 valve, which sends excess steam to the flash tank.

Variable furnace pressure startup system For units capable of operating with variable furnace pressure over the load range, the startup system has the steam separator located in the main flow path upstream of the primary superheater. The general arrangement is shown in Fig. 25.

The boiler feed pump supplies the minimum required flow of feedwater during startup and low load operation to protect the furnace circuits. Included in the startup circuitry are the economizer, the furnace enclosure, the wingwall pass (on units where the wingwalls are upstream of the primary superheater), and the steam separator.

The steam-water mixture is separated in the vertical steam separator during startup and low load operation. The unit is started with low furnace pressures to obtain the maximum amount of steam early in the startup; the 330 and 341 valves control the drains from the separator. The 330 valve also controls the flow to the deaerator for maximum heat recovery.

The 342 block valve remains closed until there is a steam-water mixture entering the separator, and it is closed any time a high level is indicated in the separator to prevent water from entering the superheater and reheater through the steam attemperators. Once the 342 valve is opened, the deaerator steam line holds pressure in the deaerator (controlled by the 331 valve). This permits returning all of the drains to the condenser through the 341 valve during a hot cleanup without using an auxiliary steam source for maintaining deaerator pressure; it also serves to recover the heat in the separated steam during cleanup.

During the initial stage of startup, the steam flow from the superheater is through the main steam line drains and 310 valve to warm the steam lines.

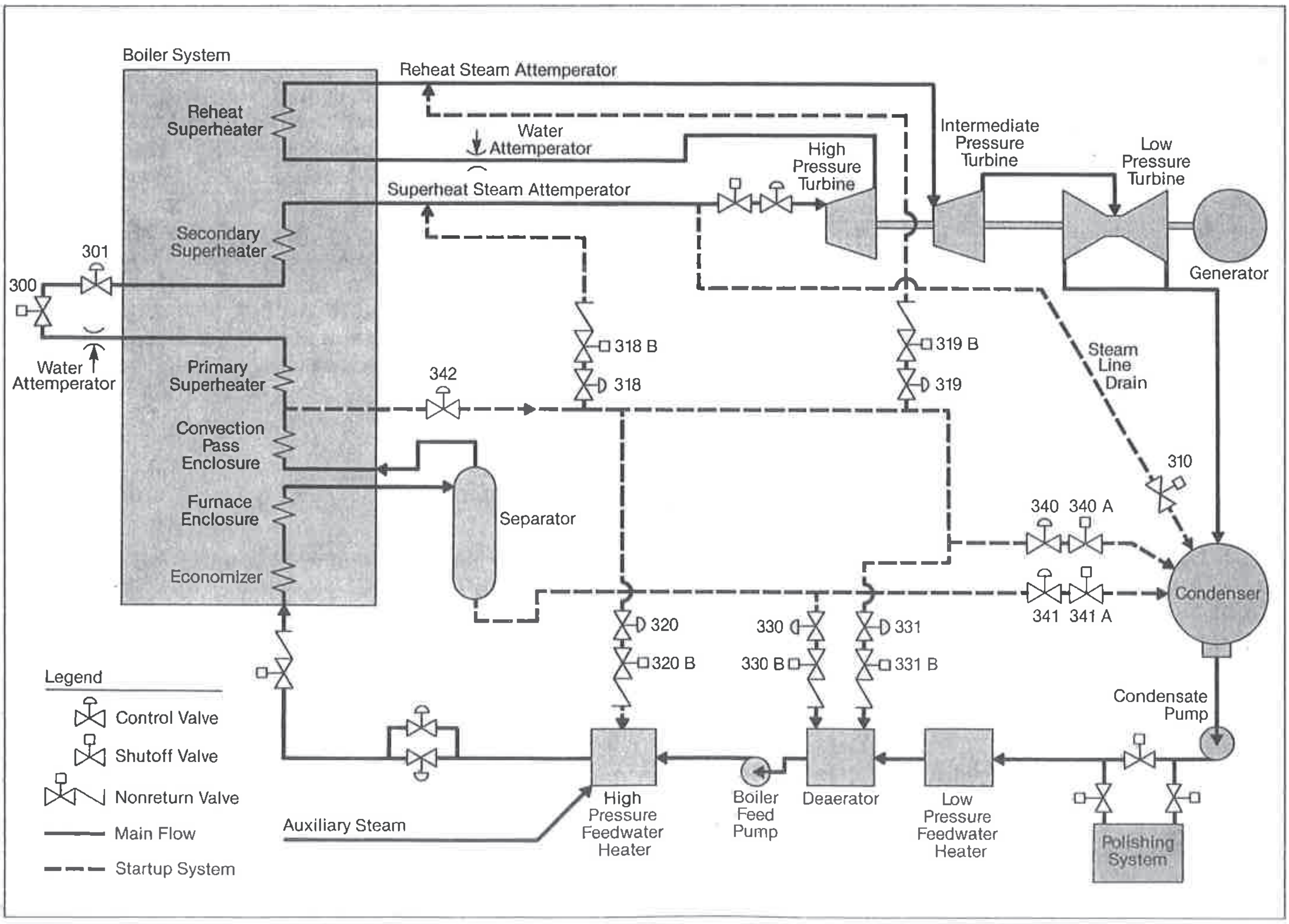


Fig. 25 Universal Pressure boiler startup system — variable pressure furnace operation.

As the enthalpy of the fluid entering the separator increases, the drains diminish until there is dry steam entering the separator. The drain valves are closed at this time, the deaerator is being controlled with turbine extraction steam, and the unit is on once-through operation with all the flow going to the turbine.

Division valves (dual pressure capability) On hot starts, including starts following overnight shutdowns, the gas temperature leaving the furnace has to be kept high to maintain high main steam and reheat temperatures. This generally results in an excessively rapid rise in throttle pressure, which is undesirable because of the resulting large throttling temperature drop when admitting steam to the turbine.

By means of the superheater combination stop and control valve (301), the boiler surface can be isolated from the secondary superheater surface. The overfiring required to raise and maintain steam temperatures can be allowed to raise saturation temperature or boiler pressure while maintaining the desirable low pressures in the secondary superheater and entering the high pressure turbine. Dual pressure operation also permits more rapid rates of load change. As the boiler temperature and pressure can be maintained at the higher level, the unit also responds more quickly to a change in load demand.

Superheater bypass to condenser When reaching the maximum desired boiler pressure or when starting up with the superheater stop valve open, the superheater bypass to the condenser valve (340) provides a means to control boiler pressure during hot start conditions. During the transient loading period for the unit following a hot restart, the superheater bypass valve (340) may be opened to permit higher firing rates and therefore sustain raised steam temperature until the boiler control load is reached.

The superheater bypass to the condenser is used to relieve excessive pressure in the boiler during a load trip. It can also be used as an overpressure relief valve to supplement and avoid popping spring loaded safety valves.

Steam temperature control The startup system shown in Fig. 25 includes provision for attemperation with saturated steam. This steam is taken from downstream of the separator and injected into the main and reheat steam outlet headers for precise control of steam conditions during startup in order to meet the turbine metal temperature requirements.

The superheater outlet steam attemperator valve (318) is used in a manner similar to the 218 valve discussed above under *Constant Furnace Pressure, Steam Temperature Control.*

The division valve (301) is used to achieve the necessary pressure drop between the separator and the secondary outlet header for attemperation.

The reheat outlet steam attemperator valve (319) is used at loads below 20% of full load to introduce separator steam to the reheat outlet header. The ratio of flow through the attemperator valve (319) to the flow through the high pressure turbine is limited due to turbine and turbine control considerations.

Permissible rates of load change

The permissible rates of load change, based on the allowable rates of temperature change in the boiler components for various modes of operation, are shown in Fig. 26.

The longer time required for load change with variable pressure operation is due to the greater change in furnace enclosure temperature and the need to restrict the rate of temperature change to avoid tearing casing attachments from the enclosure walls. With dual pressure operation, the furnace enclosure remains at constant pressure and practically constant temperature; however, the secondary superheater inlet header goes through a large temperature change with variable pressure operation, and the time required for load change is based on the limiting rate of temperature change for the outlet header.

If the entire unit is operated at constant pressure, there is virtually no boiler imposed limit on the rate of load change except for the rate of change possible from the firing equipment. With a large load change the expected rate would be a maximum of about 5% of full load per minute. The rate of load change is also restricted by the turbine metals which go through a large temperature swing due to the throttling at low loads and are limited by the design number of cycles.

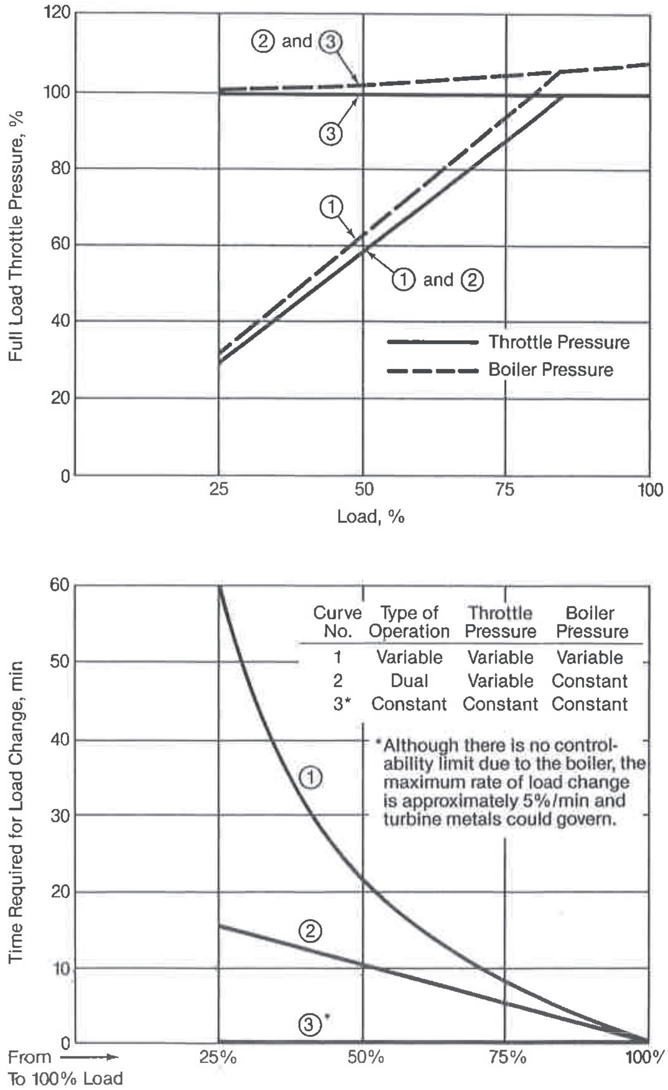
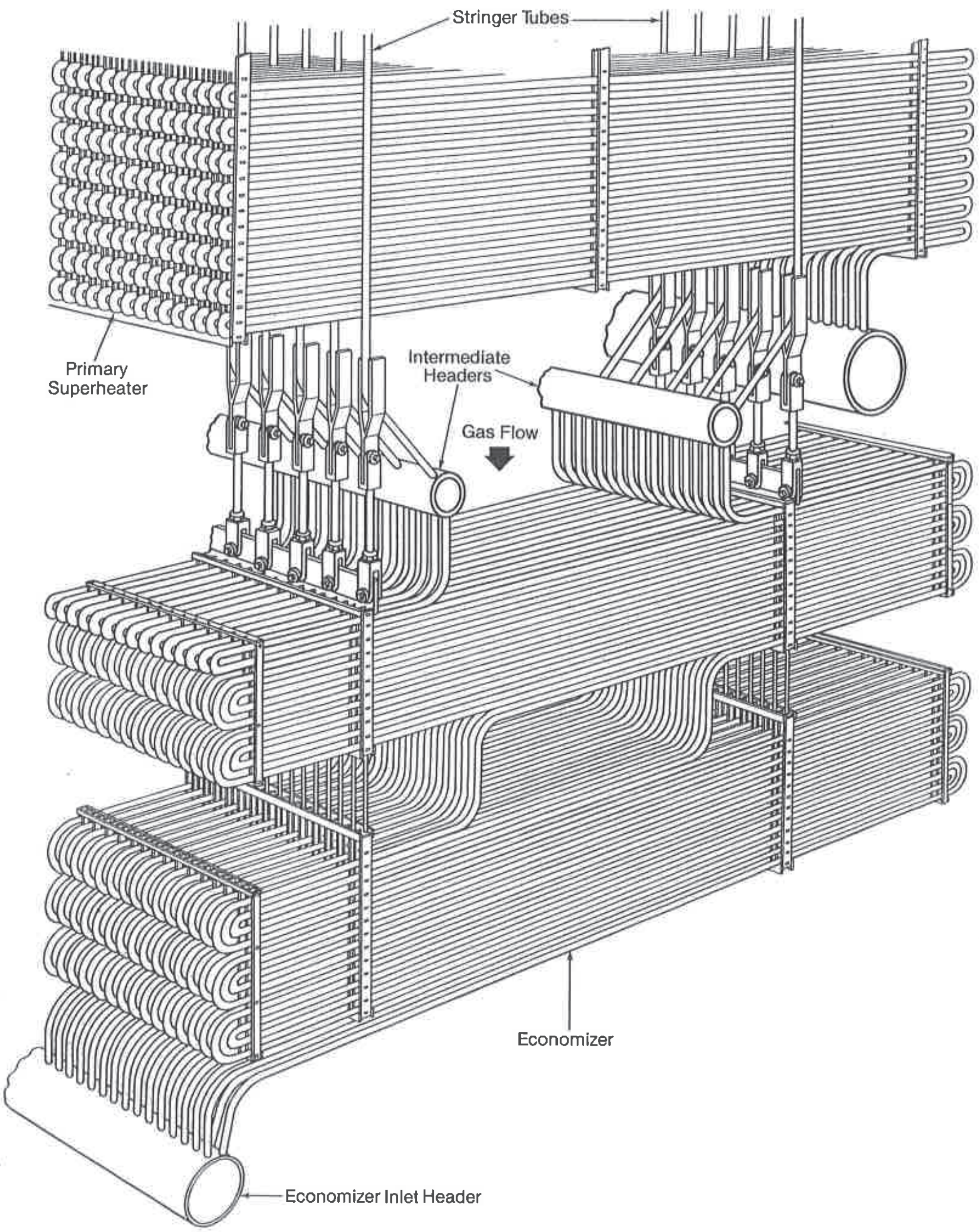


Fig. 26 Permissible rates of load change for three boiler operating configurations.



Typical utility boiler economizer.

Chapter 19
Economizers and Air Heaters

Economizers and air heaters perform a key function in providing high overall boiler thermal efficiency by recovering the low level, i.e., low temperature, energy from the flue gas before it is exhausted to the atmosphere. For each 40F (22C) that the flue gas is cooled by an economizer or air heater, the overall boiler efficiency increases by approximately 1% (Fig. 1). Economizers recover the energy by heating the boiler feedwater while air heaters heat the combustion air. Air heating also enhances the combustion of many fuels and is critical for pulverized coal firing for drying the coal and ensuring stable ignition.

In comparison to the furnace waterwalls, superheater and reheater, economizers and air heaters require a large amount of heat transfer surface per unit of heat recovered. This is because of the relatively small difference between the temperature of the flue gas and the temperature of either the feedwater or the combustion air. Use and arrangement of the economizer and/or air heater depend upon the particular fuel, application, boiler operating pressure, power cycle and overall minimum cost configuration.

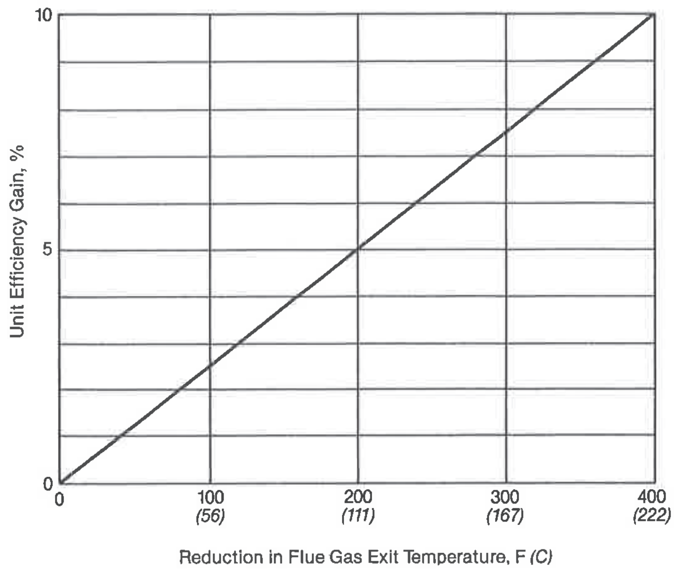


Fig. 1 Approximate unit efficiency increase due to an economizer and air heater.

Economizers

Economizers are basically tubular heat transfer surfaces used to preheat boiler feedwater before it enters the drum (recirculating units) or furnace surfaces (once-through units). The term economizer comes from early use of such heat exchangers to reduce operating costs or economize on fuel by recovering extra energy from the flue gas. Economizers also reduce the potential of thermal shock and strong water temperature fluctuations as the feedwater enters the boiler drum or waterwalls. Fig. 2 shows an economizer location on a coal-fired boiler. The economizer is typically the last water-cooled heat transfer surface upstream of the air heater. (See facing page.)

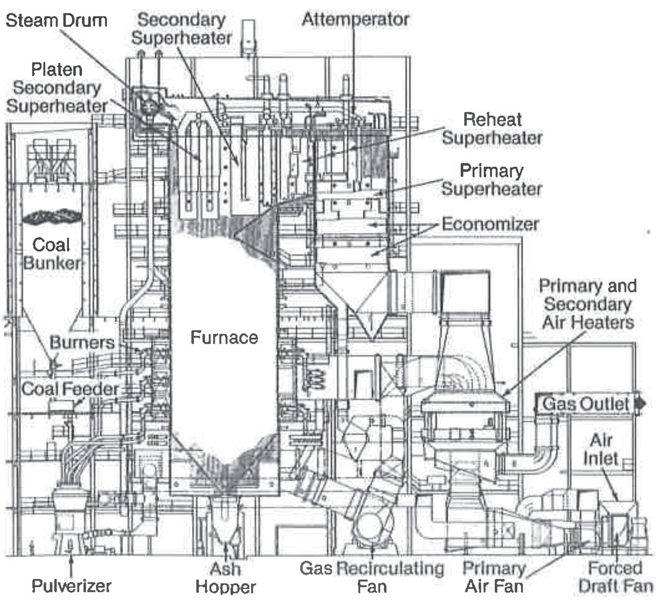


Fig. 2 Economizer and air heater locations in a typical coal-fired boiler.

Economizer surface types

Bare tube

The most common and reliable economizer design is the bare tube, in-line, crossflow type. (See Fig. 3a.) When coal is fired, the flyash creates a high fouling and erosive environment. The bare tube, in-line arrangement minimizes the likelihood of erosion and trapping the ash as compared to a staggered arrangement shown in Fig. 3b.

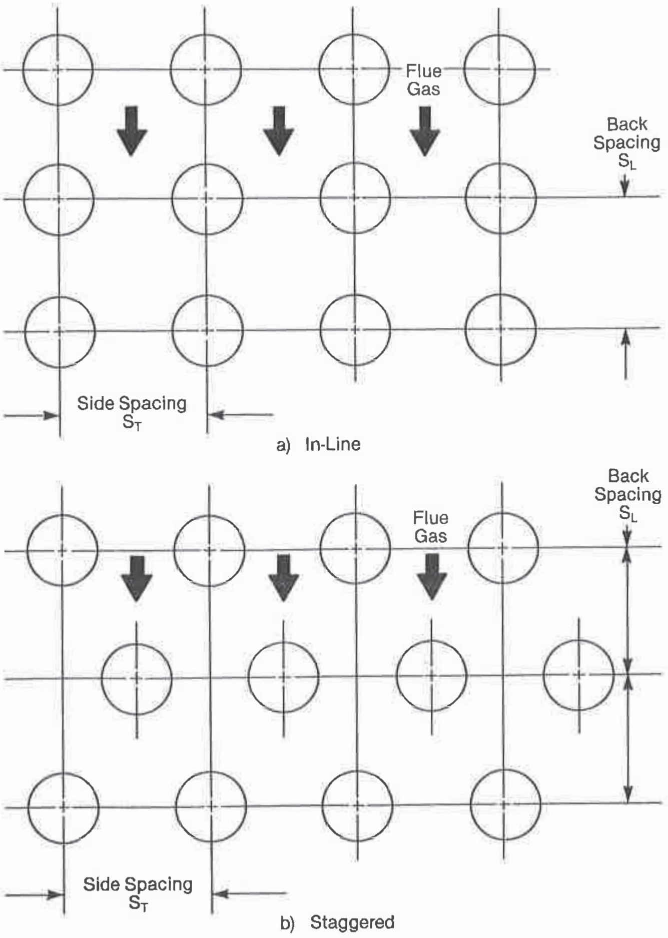


Fig. 3 Bare tube economizer arrangements.

It is also the easiest geometry to be kept clean by sootblowers. However, these benefits must be evaluated against the possible larger weight, volume and cost of this arrangement.

Extended surfaces

To reduce capital costs, most boiler manufacturers have built economizers with a variety of fin types to enhance the controlling gas side heat transfer rate. Fins are inexpensive nonpressure parts which can reduce the overall size and cost of an economizer. However, successful application is very sensitive to the flue gas environment. Surface cleanability is a key concern. In selected boilers, such as Cyclone furnace units (see Chapter 14), extended surface economizers are not recommended because of the coarser flyash characteristics.

Stud fins Stud fins have worked reasonably well in gas-fired boilers. However, stud finned economizers can have higher pressure loss than a comparable unit with helically-finned tubes. Studded fins have performed poorly in coal-fired boilers because of high erosion, loss of heat transfer, increased pressure loss and plugging resulting from flyash deposits.

Longitudinal fins Longitudinally-finned tubes in staggered crossflow arrangements, shown in Fig. 4, have also not performed well over long operating periods. Excessive plugging and erosion in coal-fired boilers have resulted in the replacement of many of these economizers. In oil- and gas-fired boilers, cracks have occurred at the points where the fins terminate. These cracks have propagated into the tube wall and caused tube failures in some applications.

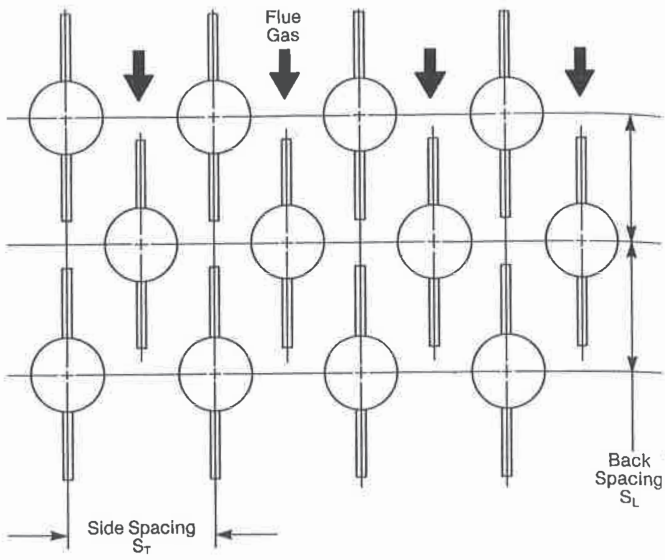


Fig. 4 Longitudinal fins, staggered tube arrangement. (Fin width exaggerated for clarity.)

Helical fins Helically-finned tubes (Fig. 5) have been successfully applied to some coal-, oil- and gas-fired units. The fins can be tightly spaced in the case of gas firing due to the absence of coal flyash or oil ash. Four fins per inch (1 fin per 6.4 mm), a fin thickness of 0.06 to 0.075 in. (1.5 to 1.9 mm) and a height of 0.75 in. (19.1 mm) are typical. For 2 in. (51 mm) outside diameter tubes, these fins provide ten times the effective area of bare tubes per unit tube length. If heavy fuel oil or coal is fired, a wider fin spacing must be used and adequate measures taken to keep the heating surface as clean as possible. Economizers in units fired with heavy fuel oil can be designed with helical fins, spaced at 0.5 in. (13 mm) intervals. Smaller fin spacings promote plugging with oil ash, while greater spacings reduce the amount of heating surface per unit length. Sootblowers are required and the maximum bank height should not exceed 4 to 5 ft (1.2 to 1.5 m) to assure reasonable cleanability of the heating surface. An in-line arrangement also facilitates cleaning and provides a lower gas side resistance.

Rectangular fins The square or rectangular fins, arranged perpendicular to the tube axis on in-line tubes as shown in Fig. 6, have had some success in retrofits. The fin spacing typically varies between 0.5 and 1 in. (13 and 25 mm) and the fins are usually 0.125 in. (3.18 mm) thick. There is a vertical slot down the middle because the two halves of the fin are welded to either side of the tube. Most designs are for gas velocities below 50 ft/s (15.2 m/s).

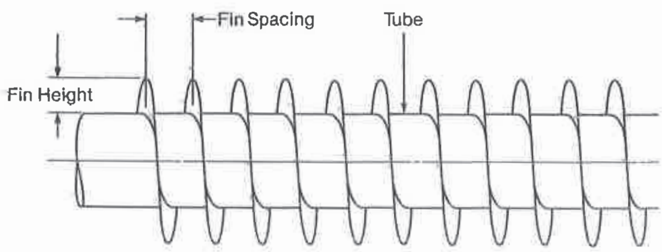


Fig. 5 Helically-finned tube.

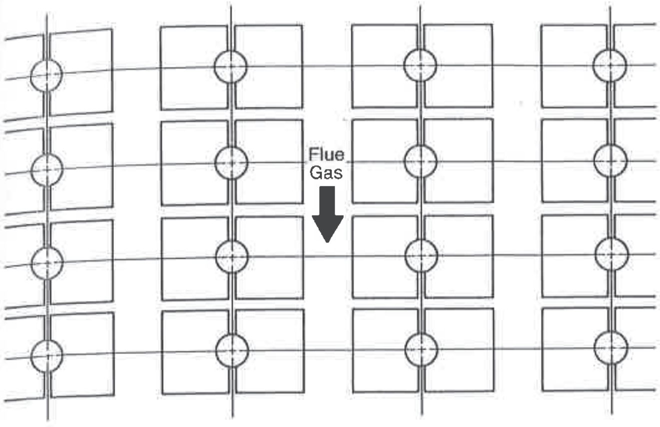


Fig. 6 Rectangular fins, in-line tube arrangement.

Baffles The tube ends should be fully baffled (Fig. 7) to minimize flue gas bypass in finned bundles. Such bypass flow can reduce heat transfer, produce excessive casing temperatures and with coal firing, can lead to tube bend erosion because of very high gas velocities. Baffling is also used with bare tube bundles but is not as important as for finned tube bundles.

Velocity limits

The ultimate goal of economizer design is to achieve the necessary heat transfer at minimum cost. A key design criterion for economizers is the maximum allowable gas velocity (defined at the minimum cross-sectional free flow area in the tube bundle). Higher velocities provide better heat transfer and reduce capital cost. For clean burning fuels, such as gas and low ash oil, velocities are typically set by the maximum economical pressure loss. For high ash oil and coal, gas side velocities are limited by the erosion potential of the flyash. This erosion potential is primarily determined by the percentage of Al_2O_3 and SiO_2 in the ash, the total ash in the fuel and the gas maximum velocity. Experience dictates acceptable velocities. Fig. 8 provides sample base velocity limits as a function of ash characteristics.

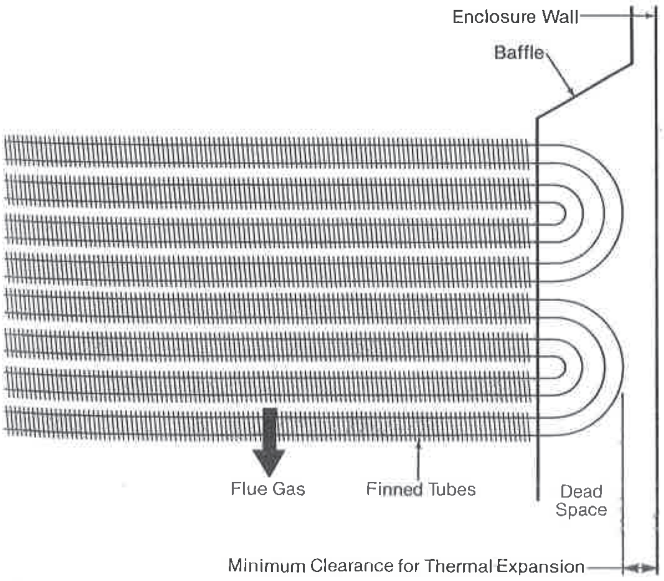


Fig. 7 Baffling of bare tube return bends for finned tube bundles.

Further criteria may also be needed. For example, a 5 ft/s (1.5 m/s) reduction in the base velocity limit is recommended when firing coals with less than 20% volatile matter. In other cases, such as Cyclone boilers, high flue gas velocities can be used because much less flyash is carried into the convection pass, as much of the ash (> 50%) is collected in the bottom of the boiler as slag. Particles that carry over are also less erosive. (See Chapter 14.)

For a given tube arrangement and boiler load, the gas velocity depends on the specific volume of flue gas which falls as the flue gas is cooled in the economizer. To maintain the gas velocity, it can be economical to decrease the free flow gas side cross-section by selecting a larger tube size in the lower bank of a multiple bank design. This achieves better heat transfer and reduces the total heating surface.

Other types of economizers

Fig. 9 depicts an industrial boiler with a long flow economizer, often used in chemical recovery boilers. Such heating surfaces consist of vertical, longitudinally-finned (membraned) tubes through which the feedwater flows upward. The gas flows downward in pure counterflow, outside the tubes and fins. While the heat transfer is less efficient than crossflow banks of tubes, there is minimal gas side resistance and fouling products are removed through hoppers at the bottom of the enclosure.

Steaming economizers

Steaming economizers are defined as meeting the following enthalpy relationships:

$$H_2 - H_1 \geq {}^2/_3 \left(H_f - H_1\right) \tag{1}$$

where

H_2 = enthalpy of fluid leaving economizer (to drum)
H_1 = enthalpy of fluid (water) entering economizer
H_f = enthalpy of saturated water at economizer outlet pressure

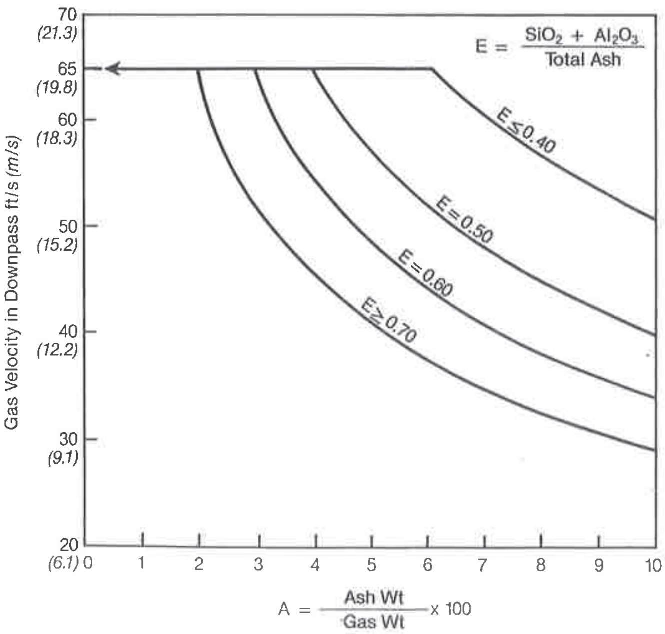


Fig. 8 Base maximum allowable velocity for pulverized coal-fired boiler economizers.

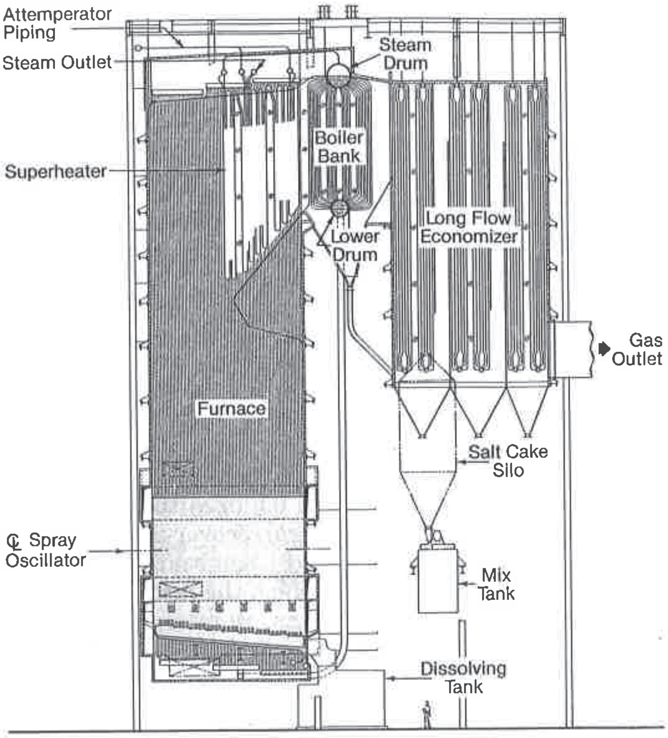


Fig. 9 Long flow economizer for a chemical recovery boiler.

These economizers can be economical in certain boilers. They require careful design and must be oriented so the water flows upward and the outlet is below the level of the drum. This avoids water hammer and excessive flow instabilities. The enthalpy equation accounts for possible steaming due to flow imbalances and differences in individual circuit heat absorptions.

High pressure, drum type units are usually sensitive to feedwater temperatures close to saturation. To enhance circulation, feedwater temperature to the drum should normally be at least 50F (28C) below saturation temperature.

Performance

Heat transfer

Bare tubes The equations discussed in Chapter 4 can be used to evaluate the quantity of surface for an economizer. For the economizer shown in Fig. 2 with the upflow of water and downflow of gas and nonsteaming conditions, the bundle can be treated as an ideal counterflow heat exchanger with the following characteristics:

1. bundle log mean temperature difference correction factor = 1.0,
2. heat absorbed by the tube wall enclosures and heat radiated into the tube banks from various cavities can generally be neglected,
3. all of the energy lost by the flue gas is absorbed by the water, i.e., no casing heat loss,
4. the water side heat transfer coefficient is typically in the range of 2000 Btu/h ft^2 F (11,357 W/m^2 K) and has only a small overall impact on the economizer performance, and
5. the effect of gas side ash deposition can be accounted for by a cleanliness factor based upon experience.

In general the heat transfer rate is primarily limited by the gas side heat transfer for in-line bare tube bundles. In this case, the overall heat transfer coefficient (flue gas to feedwater) used in the heat exchanger calculation can be approximated by the following relationship:

$$U = 0.98\ (h_c + h_r)\ k_f \qquad \textbf{(2)}$$

where

U = overall heat transfer coefficient, Btu/h ft^2 F (W/m^2 K)
h_c = gas side heat transfer coefficient for a bare tube bundle, Btu/h ft^2 F (W/m^2 K) (Chapter 4, Equations 54 and 55)
h_r = inter-tube radiation heat transfer coefficient, Btu/h ft^2 F (W/m^2 K) $\approx$ 1.0 for coal firing
k_f = surface effectiveness factor = 0.7 for coal, 0.8 for oil and 1.0 for gas

Finned tubes Heat transfer performance of finned tube economizers can be evaluated in a similar fashion except that appropriate relationships for extended surfaces should be used. In addition, because the gas side heat transfer has been enhanced, the water side heat transfer coefficient and tube wall thermal resistance are more significant and must be included in the evaluation. (See Chapter 4.) As a general guideline, the overall heat transfer coefficient can be approximated by the following relationship for most types of economizer fins:

$$U = 0.95\ (h_g k_f) \qquad \textbf{(3)}$$

where h_g is the gas side heat transfer coefficient for the heat transfer across finned tube bundles evaluated with the procedures defined in Chapter 4. A calculated example is provided in Chapter 21 for a bare tube economizer tube bundle.

Gas side resistance

The gas side pressure loss across the economizer tube bank can be evaluated using the crossflow correlations presented in Chapter 3. The pressure loss should be adjusted for the number of tube rows using the correction factors provided. The gas side resistance across the in-line finned tube banks is approximately 1.5 times the resistance of the underlying bare tubes.

Water side pressure drop

The water side pressure loss can be evaluated using the procedures in Chapter 3 where the total pressure loss ΔP_T is calculated:

$$\Delta P_T = \Delta P_f + \Delta P_l + \Delta P_z \qquad \textbf{(4)}$$

where

ΔP_f	= friction pressure loss	Ch. 3, Eq. 42
ΔP_l	= sum of the local losses (entrance, bends and exits)	Ch. 3, Eq. 47
ΔP_z	= static head loss	Ch. 5, Eq. 10

The design pressure is then evaluated by the sum of the drum design pressure and the total pressure loss ΔP_T rounded up to the nearest 25 psig (1.7 bar gauge).

If the calculated water side pressure drop is excessive, the number of parallel flow paths must be increased. If the gas velocity can be increased, the water side pressure drop can also be reduced by increasing the tube size, usually in increments of 0.125 in. (3 mm). As indicated in Chapter 3, the dynamic pressure drop is inversely proportional to the fifth power of the inside tube diameter. This is significant and may be advantageous for retrofit applications. Sometimes, a material upgrade (from SA-210A1 to SA-210C, for example) can permit use of the optimum tube wall thickness.

Economizer support systems

Economizers are located within tube wall enclosures or within casing walls, depending on gas temperatures. In general, casing enclosures are used at or below 850F (454C) and inexpensive carbon steel can be used. If a casing enclosure is used, it must not support the economizer. However, tube wall enclosures may be used as supports.

The number of support points is determined by analyzing the allowable deflection in the tubes and tube assemblies. Deflection is important for tube drainability. Figs. 10 through 12 show typical support arrangements for bare tube economizers.

Wall or end supports are usually chosen for relatively short spans and require bridge castings or individual lugs welded or attached to the tube wall enclosures. (See Fig. 10.) Another possibility exists if enclosure wall (usually primary superheater circuitry) headers are present above the economizer (for example, Fig. 11).

Quarter point stringer supports are used for spans exceeding the limits for end supports (Fig. 12). The stringers are mechanically connected to the economizer sections, which are held up by ladder type supports. The supports exposed to hot inlet gases may be made of stainless steel, while lower grade material is normally used to support the lower bank which is exposed to reduced gas temperatures. In Babcock & Wilcox (B&W) designs, stringer tubes also usually support other horizontal convection surfaces above the economizer. (See frontispiece.) Bottom support is sometimes used if the gas temperature leaving the lowest economizer bank is low enough.

Bank size

The bank size is limited by the following constraints:

1. type of fuel,
2. fabrication limits,
3. sootblower range,
4. maximum shipping dimensions,
5. construction considerations, especially for retrofits, and
6. maintenance.

Bank depths greater than 6 ft (1.82 m) are rare in new boilers, while larger banks can be tolerated in retrofits.

Access requirements

Cavities around the banks are needed for field welding, tube leg maintenance and sootblower clearance. A sufficient number of access doors must be arranged in the enclosure walls to access these cavities. Cavity access can be provided from the outside through individual doors or from the inside through special openings across stringers or collector frames. The minimum cavity height should be 2 ft (0.6 m) of crawl space.

Headers

B&W economizer header designs are based on American Society of Mechanical Engineers (ASME) Code requirements. Inlet headers are frequently located inside the gas stream and may receive feedwater through one or both ends. Regardless of design, it is necessary to properly seal the inlet pipe where it penetrates the enclosure by using brackets and flexible seals. The seal becomes especially important in pressurized (forced draft) units. Other important considerations are tube leg flexibility, differential expansion and potential gas temperature imbalances and upsets.

The outlet headers, not to be confused with intermediate headers seen in larger boilers and stringer supports, receive the heated feedwater and convey it to the drum

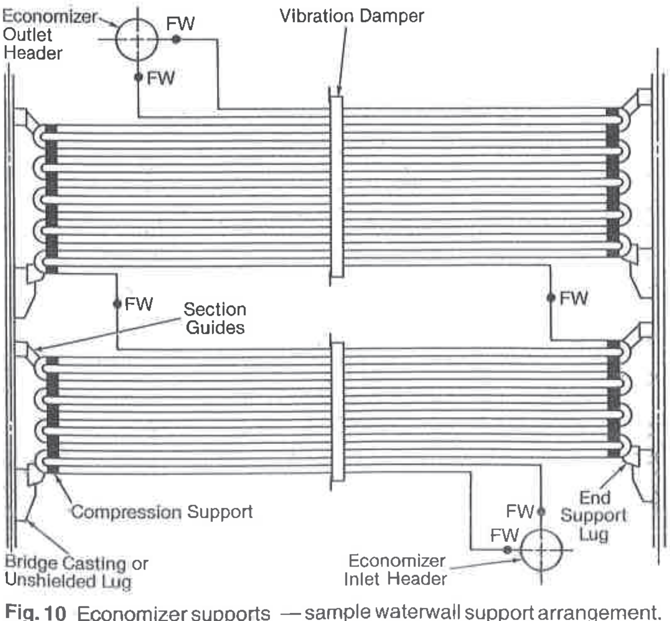


Fig. 10 Economizer supports — sample waterwall support arrangement. (FW = field weld.)

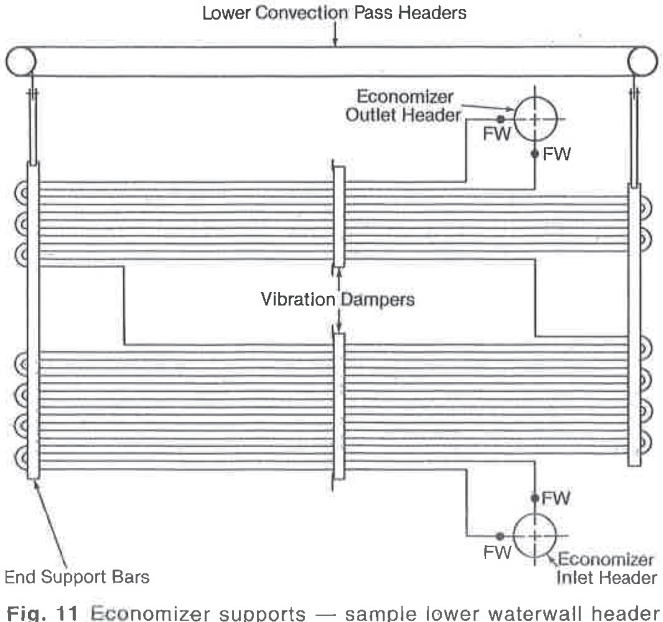


Fig. 11 Economizer supports — sample lower waterwall header arrangement. (FW = field weld.)

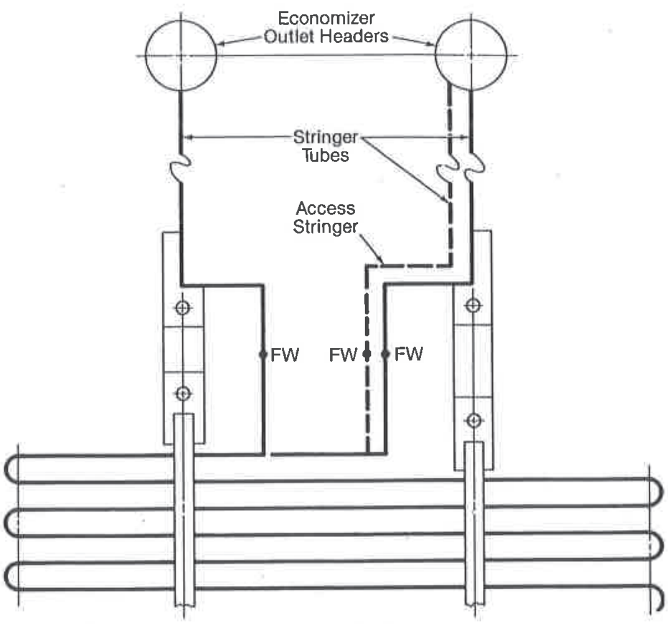


Fig. 12 Economizer supports — sample stringer support arrangement.

or, in the case of once-through boilers, to the downcomer supplying the furnace circuitry. Inlet and outlet headers must be large enough to assure reasonable water flow distribution in the economizer banks. Flow velocities are typically less than 20 ft/s (6 m/s).

Vibration ties

Vibration ties or tube guides are required on some end-supported tube sections. These ties may be needed if the natural frequencies within the boiler load range are in or near resonance with the vortex shedding frequency.

Stringer tubes are also subject to vibration. This vibration is magnified by long unsupported stringer tube lengths near the large cavity below the convection pass roof.

Tube geometry, materials and code requirements

Economizer tube diameters typically range between 1.75 and 2.5 in. (44.5 and 63.5 mm). Tubes outside this range are sometimes used in retrofits. Smaller tubes are normally used in once-through, supercritical boilers where water side pressure drop is less of a consideration. In these units, tube wall thicknesses are minimized.

The ASME Code requires that the design temperature for internal boiler pressure parts is at least 700F (371C). The calculated mean tube wall temperature in economizers seldom reaches this temperature. It usually lies 10 to 20F (6 to 12C) above the fluid temperature, which seldom exceeds 650F (343C) along any economizer circuit.

The minimum tube wall thickness is determined in accordance with procedures outlined in Chapter 7.

In coal-fired boilers, the side spacing is usually determined by the maximum allowable gas velocity and gas side resistance, which are functions of a given tube size. If fins are used, the side and back tube spacings should permit the fin tips to be at least 0.5 in. (13 mm) apart. For bare tubes, a minimum clear spacing of 0.75 in. (19 mm) is desirable.

The minimum back (vertical) spacing of the tubes should be no less than 1.25 times the tube outside diameter. Smaller ratios can reduce heat transfer by as much as 30%. Ratios larger than 1.25 have relatively little effect on heat transfer but increase the gas side resistance and bank depth.

Air heaters

Air heaters are used in most steam generating plants to heat the combustion air and enhance the combustion process. Most frequently, the flue gas is the source of energy and the air heater serves as a heat trap to collect and use waste heat from the flue gas stream. This can increase the overall boiler efficiency by 5 to 10%. Air heaters can also use extraction steam or other sources of energy depending upon the particular application. These units are usually employed to control air and gas temperatures by preheating air entering the main gas-air heaters.

Air heaters are typically located directly behind the boiler, as depicted in Fig. 2, where they receive hot flue gas from the economizer and cold combustion air from the forced draft fan. The hot air produced by air heaters enhances combustion of all fuels and is needed for drying and transporting the fuel in pulverized coal-fired units.

Classification of air heaters

Air heaters are classified according to their principle of operation as recuperative or regenerative.

Recuperative

In a recuperative heat exchanger, heat is transferred continuously through stationary solid heat transfer surfaces which separate the hot flow stream from the cold flow stream. The most common heat transfer surfaces are tubes and parallel plates. Recuperative heat exchangers function with little cross-contamination, or leakage, between streams.

Tubular air heaters In a typical tubular air heater, energy is transferred from the hot flue gas flowing inside many thin walled tubes to the cold combustion air flowing outside the tubes. The unit consists of a nest of straight tubes that are roll expanded or welded into tubesheets and enclosed in a steel casing. The casing serves as the enclosure for the air or gas passing outside of the tubes and has both air and gas inlet and outlet openings. In the vertical type (Fig. 13), tubes are supported from either the upper or lower tubesheet while the other (floating) tubesheet is free to move as tubes expand within the casing. An expansion joint between the floating tubesheet and casing provides an air/gas seal. Intermediate baffle plates parallel to the tubesheets are frequently used to separate the flow paths and eliminate tube damaging flow induced vibration.

Carbon steel or low alloy corrosion resistant tube materials are used in the tubes which range from 1.5 to 4 in. (38 to 100 mm) in diameter and have wall thicknesses of 18 to 11 gauge [0.049 to 0.120 in. (1.24 to 3.05 mm)]. Larger diameter, heavier gauge tubes are used when the potential for tube plugging and corrosion exists. Tube arrangement may be in-line or staggered with the latter being more thermally efficient.

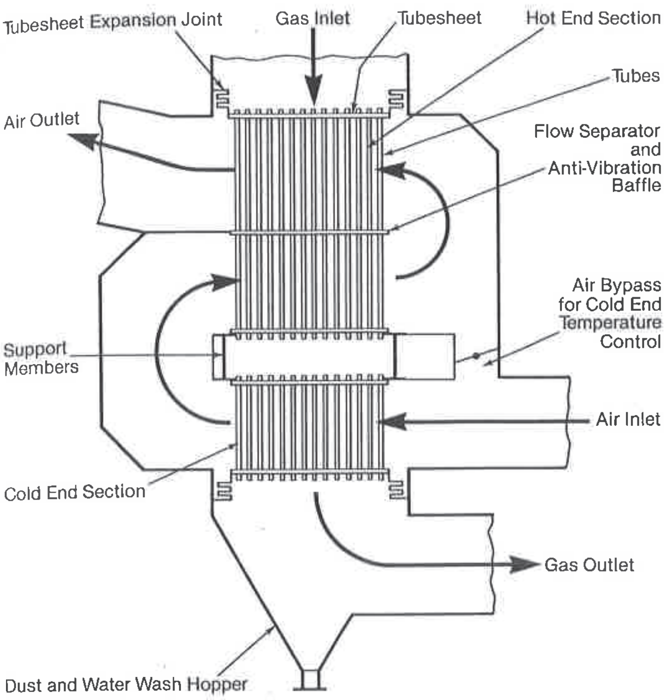


Fig. 13 Vertical type tubular air heater.

The most common flow arrangement is counterflow with gas passing vertically through the tubes and air passing horizontally in one or more passes outside the tubes. A variety of single and multiple gas and air path arrangements are used to accommodate plant layouts. Designs frequently include provisions for cold air bypass or hot air recirculation to control cold end corrosion and ash fouling. Modern tubular air heaters are shop assembled into large, transportable modules. Several arrangements are shown in Fig. 14.

Cast iron air heaters Cast iron tubular air heaters are heavy, large and durable. Their use is mainly limited to the petrochemical industry, but some are used on electric utility units. Cast iron is used because of its superior corrosion resistance. Rectangular, longitudinally split tubes are assembled from two cast iron plates and individual tubes are assembled into air heater sections. Air heaters are usually arranged for a single gas pass and multiple air passes with air flow inside the tubes. Heat transfer is maximized by fins cast into inside and outside tube surfaces.

Plate air heaters This type of air heater transfers heat from hot gas flowing on one side of a plate to cold air flowing on the opposite side, usually in crossflow. Heaters consist of stacks of parallel plates. Sealing between air and gas streams at plate edges is accomplished by weld-

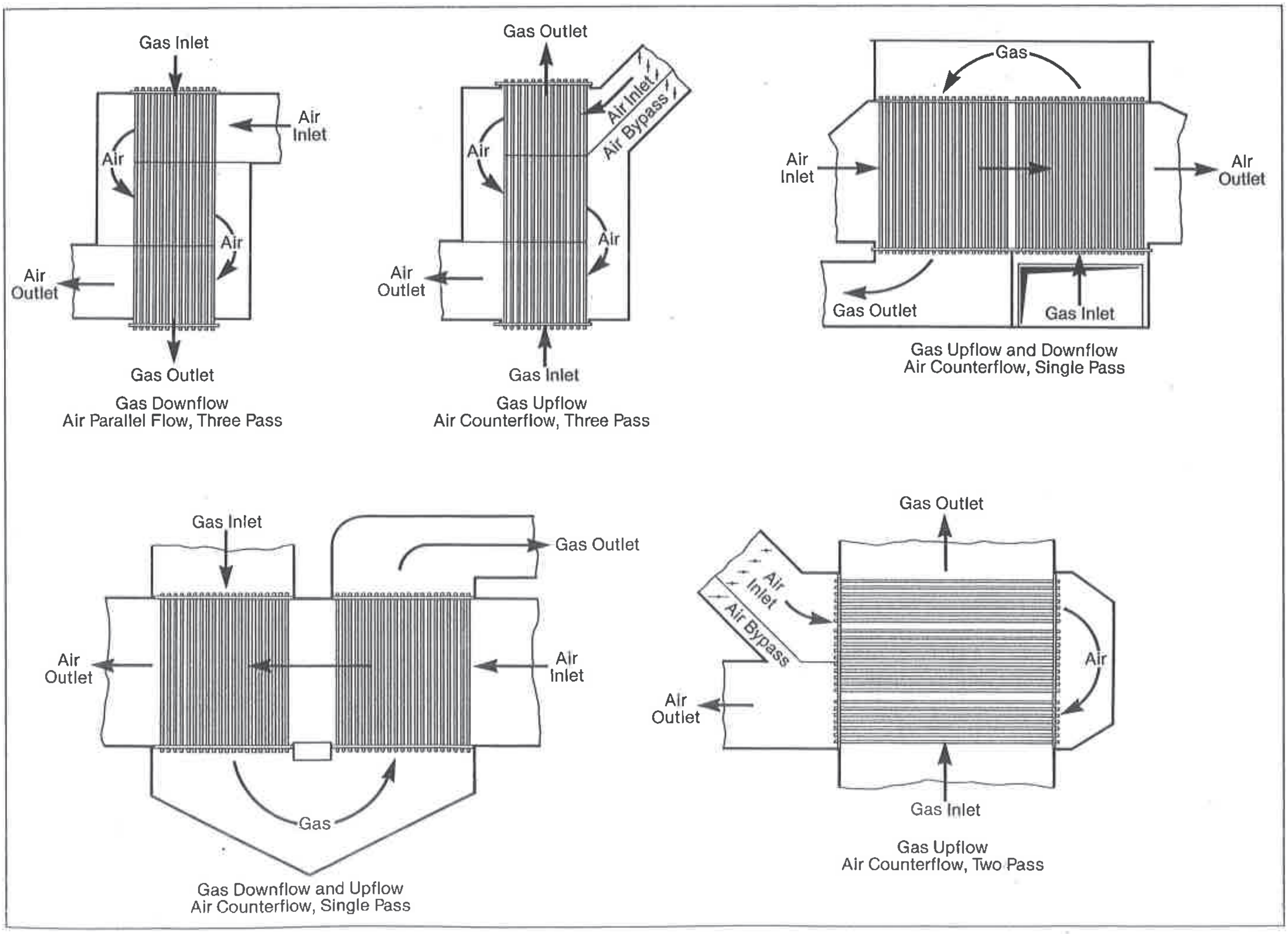


Fig. 14 Various tubular air heater arrangements.

ing or by a combination of gaskets, springs and external compression of the plate stack. Plate materials and spacing can be varied to accommodate operating requirements and fuel types.

Steel plate air heaters were some of the earliest types used, but their use declined due to plate to plate sealing problems. However, recent sealing developments have prompted increased use in industrial and small utility applications. Plate modules may be combined to make different size air heaters with a variety of flow path arrangements. A single gas pass, two air pass plate air heater is shown in Fig. 15. Modern plate units are somewhat smaller than tubular units for a given capacity and exhibit minimal air to gas leakage.

Steam coil air heaters Steam coil and water coil recuperative air heaters are widely used in utility steam generating plants to preheat combustion air. Air preheating reduces the corrosion and plugging potential in the cold end of the main air heater. Occasionally, they serve as the only source of preheated combustion air. These heaters consist of banks of small diameter, externally finned tubes arranged horizontally or vertically in air ducts between the combustion air fan and main air heater. Combustion air, passing in crossflow outside the tubes, is heated by turbine extraction steam or feedwater flowing inside the tubes. Ethylene glycol is sometimes used as the hot fluid to prevent out of service freezing damage.

Heat pipe A heat pipe is a simple, highly efficient device for transporting thermal energy. The basic thermosyphon type heat pipe used in steam generation unit air heaters consists of an evacuated sealed pipe which has been partially filled with a heat transfer fluid (Fig. 16). The evaporator end of the pipe is exposed to a heat source (hot flue gas) and the other end, the condenser, is placed in a heat sink (cold combustion air). Heat absorbed from the flue gas evaporates the fluid which travels to the combustion air end where heat is released as the fluid condenses. The condensed fluid returns by capillary action and gravity to the evaporator end. Fluid circulation within the tube is continuous as long as there is a temperature difference between evaporator and condenser ends; fluid temperature is nearly isothermal at approximately the average of air and gas temperatures. Large quantities of heat are transferred as the heat of condensation and vaporization are used. Heat pipes operate with the evaporator end lower than the condenser end; they are inclined from the horizontal. Internal surfaces of heat pipes are roughened or grooved to assist fluid circulation and external surfaces are usually finned to increase heat transfer surface area.

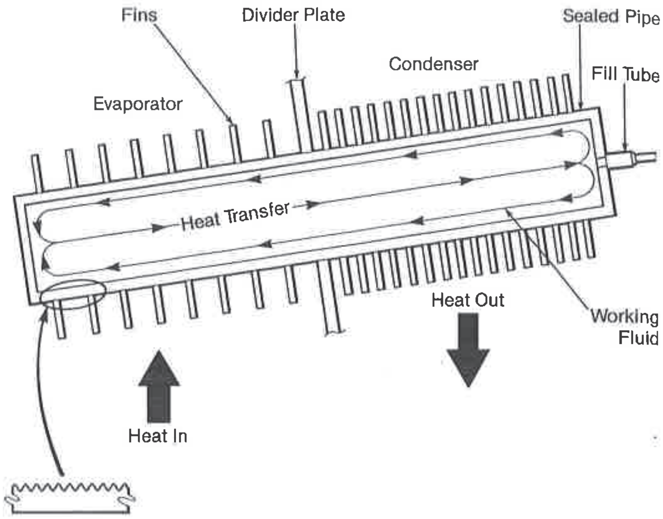


Fig. 16 Heat pipe schematic.

Heat pipe air heaters consist of bundles of parallel heat pipe tubes. About half of the tube length is exposed to flue gas flow and the remaining length is exposed to air flow. A central divider plate separates air and gas and supports the tube bundle. Heat pipe bundles or modules can be combined and enclosed in casing to build air heaters capable of accepting a variety of flow configurations.

Tube bundle configuration is usually in-line for dirty gas applications such as coal and heavy oil and staggered for natural gas and light oil. Typically, 2 in. (51 mm) diameter carbon steel tubes up to 40 ft (12.2 m) long are used. They feature three steel fins per in. (1 per 8.5 mm) on the gas side and up to ten fins per in. (1 per 2.5 mm) on the air side. Corrosion resistant alloy materials can increase cold end corrosion life.

Heat pipe air heaters are smaller than tubular air heaters and air to gas leakage is minimal as in other types of recuperative air heaters. Due to the isothermal behavior of each tube, these units can operate at a lower gas outlet temperature for a specific minimum metal temperature (MMT) compared to a tubular or regenerative air heater. This can permit operation at higher boiler efficiency and can reduce the potential for air heater cold end corrosion.

Of utmost importance is the heat transfer fluid's long term compatibility with tube wall material. Incompatibility can result in internal corrosion which produces noncon-

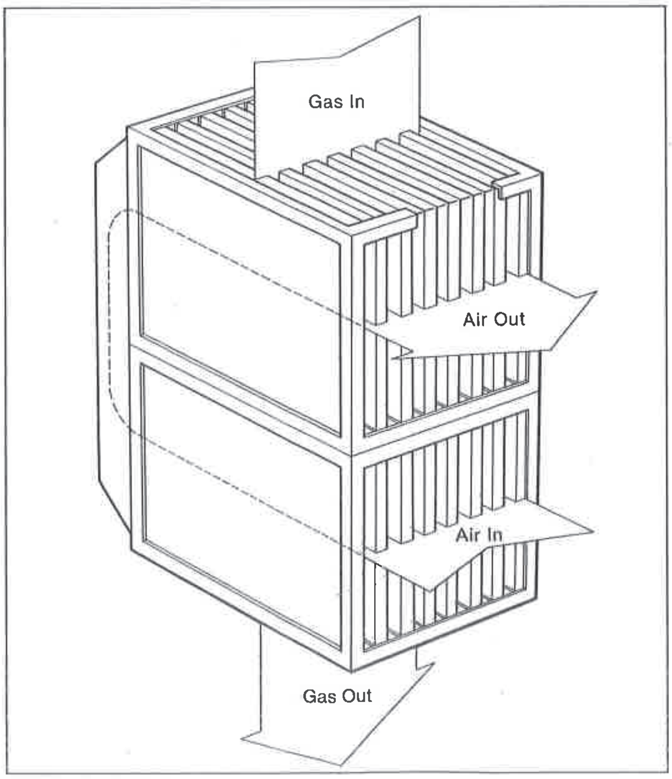


Fig. 15 Single gas pass, two air pass plate air heater.

densable gases, reduces heat transfer rates and jeopardizes tube pressure integrity. Hydrocarbons and water based fluids, commonly used in carbon steel tubes, are subject to temperature limits of 400 to 800F (204 to 427C). Water based fluids can not be allowed to freeze. High alloy steel or nonferrous tube materials may be used to extend temperature limits or to allow more fluid choices, but their expense precludes use in large heaters.

Heat pipe air heaters have been used successfully in the petrochemical industry. A limited number have also been applied to electric utility steam generation units. More widespread adoption is expected when long term reliability is proven.

Regenerative

In a regenerative air heater, heat is transferred indirectly as a heat storage medium is alternately exposed to hot and cold flow streams. A variety of materials can be used as the medium and periodic exposure to hot and cold flow streams can be accomplished by rotary or valve switching devices. In steam generating plants, tightly packed bundles of corrugated steel plates serve as the storage medium. In these units either the steel plates, or surface elements, rotate through air and gas streams or rotating ducts direct air and gas streams through stationary surface elements.

Regenerative air heaters are relatively small and are the most widely used type for combustion air preheating in electric utility steam generating plants. Their most notable operating feature is that a small but significant amount of air leaks into the gas stream due to the rotary operation.

Ljungström The most prevalent regenerative air heater is the Ljungström type (Fig. 17), which features a cylindrical shell plus a rotor which is packed with bundles of heating surface elements and is rotated through counterflowing air and gas streams. The rotor is enclosed by a stationary housing which has ducts at both ends. Air flows through one half of the rotor and gas flows through the other half. Metallic leaf-type seals minimize air to gas leakage and flow bypass around the rotor. Bearings in upper and lower beam assemblies support and guide the rotor at the central shaft. A rotor speed of one to three rpm is provided by a motor driven pinion engaging a rotor encircling pinrack. Both vertical and horizontal shaft designs are used to accommodate various plant air and gas flow schemes. The vertical shaft design is more common.

Rothemühle The Rothemühle-type regenerative air heater uses stationary surface elements and rotating ducts (Fig. 18). The surface elements are supported and contained within a stationary cylindrical shell called the stator. On both sides of the stator, a double wing symmetrical hood rotates synchronously on a common vertical shaft. The central shaft is supported by bearings within the stator and the hoods are driven slowly by a pinion which engages a pinrack encircling the lower hood. Stationary housings surround the hoods. Heat is transferred as flow streams are directed through the heating surface in counterflow fashion, one flow stream inside the hoods and the other outside. Either air or gas may pass through the hoods. However, air is more common because it requires less fan power. Special spring mounted sealing systems employing cast iron seals are used at hood rotating to stationary interfaces to minimize air to gas leakage.

Several features distinguish the Rothemühle from the Ljungström heater. Its relatively low rotating weight (20% of total weight) contributes to high reliability. The stationary stator permits air heater loads to be distributed equally to a number of surrounding points, permitting transfer of significant duct loads through the air

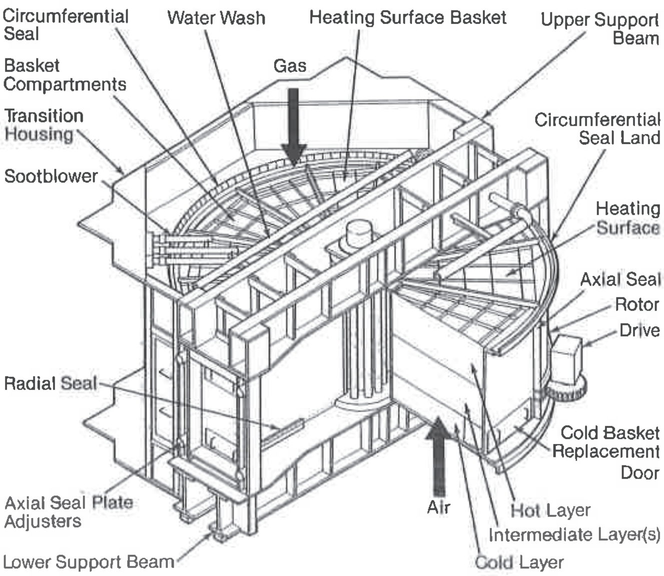


Fig. 17 Ljungström-type air heater.

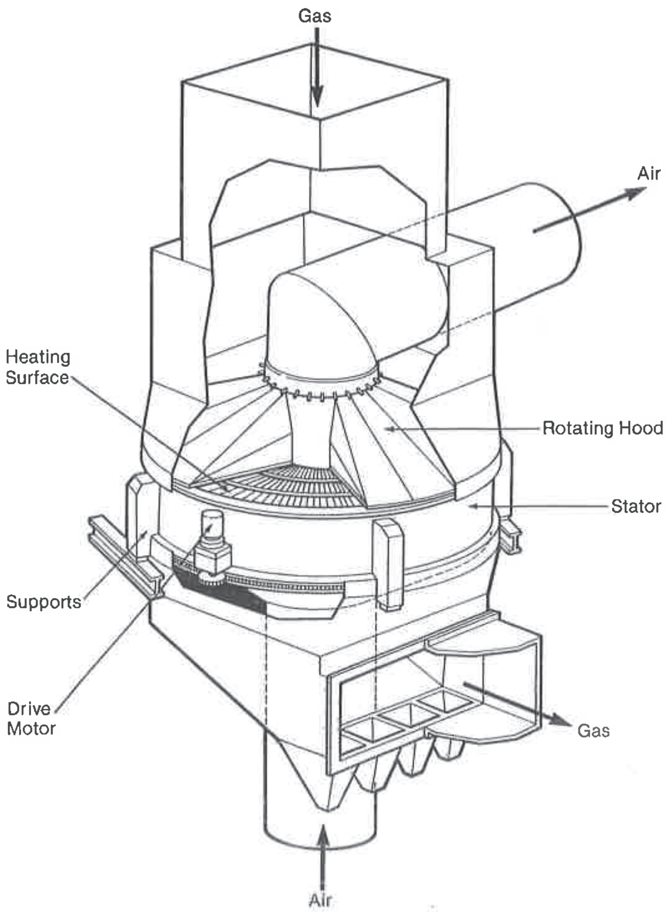


Fig. 18 Rothemühle-type air heater.

heater into structural steel. The spring mounted hood sealing system, which adapts to the curvature of the stator during operation, allows hot starts without overloading the drive motor. A simple, permanent early warning fire detection system can be embedded in the stator.

Regenerative heating surface Regenerative air heater surface elements are a compact arrangement of two specially formed metal plates. Each element pair consists of a combination of flat, corrugated or undulated plate profiles. The roll formed corrugations and undulations serve to separate the plates to maintain flow paths, increase heating surface area and maximize heat transfer by creating flow turbulence. The steel plates, 26 to 18 gauge thick, are typically spaced 0.2 to 0.4 in. (5 to 10 mm) apart. Closely spaced, highly profiled element pairs exhibit a high heat transfer rate, pressure drop and fouling potential while widely spaced element combinations, where one plate is flat, exhibit a low heat transfer rate, low pressure drop and reduced fouling potential. The combination of plate profile, material and thickness is selected for maximum heat transfer, minimum pressure drop, good cleanability and high corrosion resistance.

Surface elements are stacked and bundled into self-contained baskets and are installed into air heater rotors and stators in two or more layers. The surface layer at the air inlet side, designated the cold layer, is distinguished from other layers by design. Cold layers, which are subject to corrosion and ash fouling, are typically 12 in. (300 mm) deep for economical replacement. Heavy gauge, open profile elements are used for corrosion resistance and cleanability. Practically all cold layer elements are low alloy corrosion resistant steel or, when high corrosion potential exists, porcelain enamel coated steel. Hot and intermediate surface layers are more compact than cold layers and use thinner plates. Figs. 17 and 19 illustrate several heating surface element profiles and air heater surface arrangements.

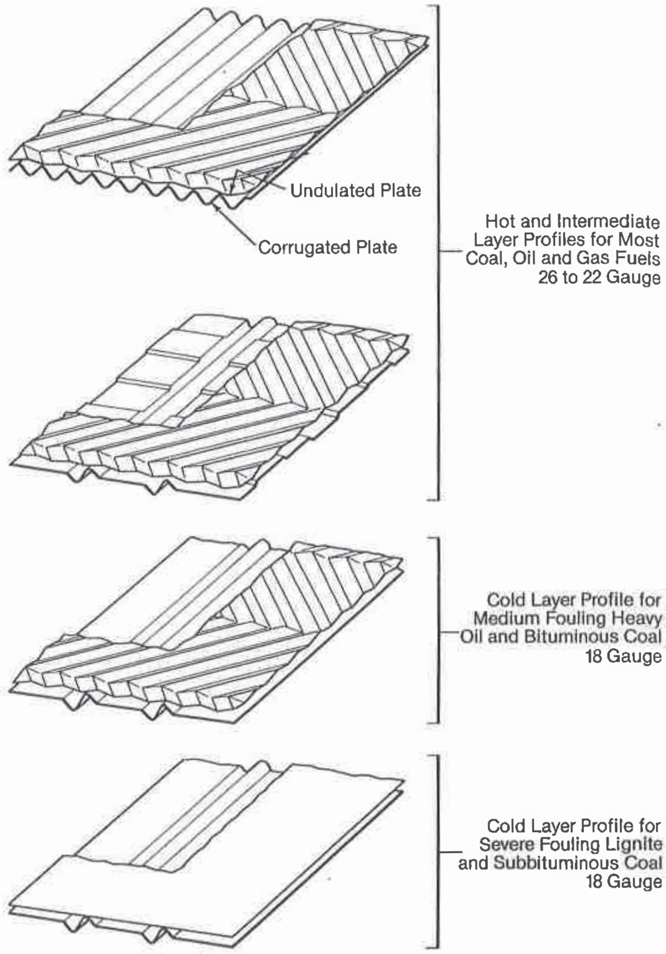


Fig. 19 Regenerative air heater surface element profiles.

Advantages and disadvantages

Many subtle differences exist between air heater designs within a particular type. However, there are some general advantages and disadvantages associated with each type which are listed in Table 1. Note that the recuperative heat pipe air heater is listed separately.

Performance and testing

Air heaters are designed to meet performance requirements in three areas: thermal, leakage and pressure drop. Low performance in any area increases boiler operating costs and may cause unit load curtailment.

Thermal performance

The thermal performance and surface area (A) of a recuperative air heater can be evaluated by:

$$A = Q / (U \quad LMTD \quad F) \qquad \textbf{(5)}$$

where Q is the total thermal load [Btu/h (W)], U is the overall heat transfer coefficient, $LMTD$ is the log mean temperature difference and F is the corresponding geometry correction factor. The performance, U, $LMTD$ and F can be evaluated using the correlations and methodology presented in Chapter 4. The overall U should include convection and radiation components as well as the appropriate gas and air side fouling factors. U typically ranges from 3 to 10 Btu/h ft^2 F (17 to 57 W/m^2 K).

Performance verification Thermal performance is measured by comparing the test gas outlet temperature to its design value. The true outlet temperature is obtained by correcting the measured temperature for air heater leakage and deviations from design conditions.

The ASME Performance Test Code, Section 4.3 (PTC 4.3), provides the following equation which is based on an air heater mass flow heat balance and assumes that the source of all leakage is from the entering air:

Table 1
Advantages and Disadvantages of Air Heater Types

Type	Advantage	Disadvantage
Recuperative	Low leakage No moving parts	Large and heavy Difficult to replace surface
Heat pipe	Low leakage High minimum metal temperatures No moving parts	Difficult to clean Temperature restrictions
Regenerative	Compact Easy to replace surface	Leakage High maintenance Fire hazard

$$T_2 = T_{2m} + \left(\frac{\% \, lkg}{100}\right)\frac{c_{pa}}{c_{pg}}(T_{2m} - T_1) \qquad \textbf{(6)}$$

where

T_2 = air heater gas outlet temperature corrected for leakage, F (C)
T_{2m} = measured gas temperature leaving air heater, F (C)
$\% \, lkg$ = percent air leakage with respect to inlet gas flow
c_{pa}, c_{pg} = specific heat of air and gas respectively, Btu/lb F (J/kg C)
T_1 = air inlet temperature, F (C)

The measured gas outlet temperature must also be corrected for deviations in various operating parameters such as mass flow rates and operating temperatures in order to accurately assess performance. Suppliers and the ASME Performance Test Codes provide various correction curves and factors for this purpose.

Leakage

Air flow passing from the air side to the gas side is called leakage. It is quantified in pounds per hour (kg/s) but is frequently expressed as a percentage of the gas inlet flow. Leakage is undesirable primarily because it represents fan power wasted in conveying air which bypasses the boiler combustion zone. Leakage can also reduce an air heater's thermal performance.

All air heaters leak. Recuperative units may begin operation with essentially zero leakage, but leakage occurs as time and thermal cycles accumulate. With regular maintenance, leakage can be kept below 3%.

Air heater leakage is inherent with the rotary regenerative design. There are two types of leakage, gap and carryover. Gap leakage occurs as higher pressure air passes to the lower pressure gas side through gaps between rotating and stationary parts. Its rate is given by the following general expression:

$$w_l = KA \,(2g_c \,\Delta P \rho)^{1/2} \qquad \textbf{(7)}$$

where

w_l = leakage flow rate, lb/h (kg/s)
K = discharge coefficient, dimensionless (generally 0.4 to 1.0)
A = flow area, ft^2 (m^2)
g_c = 32.17 lbm ft/lbf $s^2 \times (3600 \text{ s/h})^2$
= 4.17×10^8 lbm ft/lbf h^2 (1 kg m/N s^2)
ΔP = pressure differential across gap, lb/ft^2 (kg/m^2)
ρ = density of leaking air, lb/ft^3 (kg/m^3)

Carryover leakage is the air carried into the gas stream from each rotor (stator) heating surface compartment as the surface passes from the air stream to the gas stream. This leakage is directly proportional to the void volume of the rotor and the rotation speed.

Regenerative air heater design leakage ranges from 5 to 15% but increases over time as seals wear. During recent years effective automatic sealing systems, which nearly eliminate leakage rise due to seal wear, have been applied. These systems monitor and adjust rotating to stationary seals on-line.

Another source of air to gas flow, which appears as air heater leakage, is outside air infiltration into lower pressure gas streams. Infiltration may occur at casing cracks or holes, flue expansion joints and access doors or gaskets. This sometimes neglected source can be significant and difficult to detect if leaks occur under lagging and insulation.

Air heater leakage can be obtained directly as the difference between air or gas side inlet and outlet flows based on velocity measurements. However, because velocity measurements are difficult to obtain accurately in large duct cross-sections, air heater leakage is more accurately based on calculated gas weights using gas analysis, boiler efficiency and fuel analysis data. (See Chapter 9.) Approximate air heater leakage can be determined by the following formulas based on gas inlet and outlet analysis (dry basis).

$$\% \text{ Leakage} = \frac{\% \ O_2 \text{ Leaving} - \% \ O_2 \text{ Entering}}{21 - \% \ O_2 \text{ Leaving}} \times 90 \qquad \textbf{(8)}$$

Test air heater leakage should be corrected for deviations from design cold end air to gas differential pressure and inlet air temperature before comparison to design leakage.

Pressure drop

In recuperative air heaters, gas or air side pressure drop arises from frictional resistance to flow, inlet and exit shock losses and losses in return bends between flow passes. In regenerative air heaters, the main cause is heating surface frictional flow resistance. In both cases, pressure drop is proportional to the square of the mass flow rate. Typical values at full load flows are 2 to 7 in. wg (0.5 to 1.7 kPa).

Air and gas side pressure drop values are the differences between terminal inlet and outlet static gauge pressures. Correction of measured pressure drops for deviations from design flows and temperatures is necessary before comparison to design values.

Operational concerns

There are several operating conditions and maintenance concerns common to most air heaters. These include corrosion, plugging and cleaning, erosion and fires. Air heaters used with high ash and/or high sulfur content fuels require more attention and maintenance than those firing clean fuels such as natural gas.

Corrosion

Air heaters used on units firing sulfur bearing fuels are subject to cold end corrosion of heating elements and nearby structures. In a boiler, a portion of the sulfur dioxide (SO_2) produced is converted to sulfur trioxide (SO_3) which combines with moisture to form sulfuric acid vapor. This vapor condenses on surfaces at temperatures below its dew point of 250 to 300F (120 to 150C). Because normal air heater cold end metal temperatures are frequently as low as 200F (93C), acid dew point corrosion potential exists. The obvious solution would be to operate at metal temperatures above the acid dew point but this results in unacceptable overall boiler heat losses. Most air heaters are designed to operate at MMTs somewhat below the acid dew point, where the efficiency