

CHAPTER 2

Steam Generators

INTRODUCTION

Man has used the technique of converting water (liquid) into steam (vapor) and using the expansive force of this vapor as far back as 150 B.C. Yet it took human beings more than 1800 years to harness the practical qualities of steam to do useful work.

1711 - first commercial piston-operated mine pump was invented [1], Latter half of the nineteenth century - first steam-operated central power station was introduced.

1882 - Holborn Viaduct power station in London and Pearl Street Station in New York had been put into service using reciprocating steam engines.

The main objective of a steam power station is to generate electrical power. In a steam power station, the electrical energy is produced according to the principle of "external combustion," where the "heat of combustion" of the fuel is transferred to a prime mover by a "working medium." In the steam generator, low-temperature water is the working medium that receives the heat of combustion of fuel and becomes high-energy steam. The heat of steam is converted to mechanical energy in the steam turbine and then to electrical energy in the generator. The sequence of these activities is shown in Figure 2.1. The chemical energy available in fossil fuel (i.e., coal, fuel oil, gas) is converted to heat energy by combustion in a steam generator. The heat thus liberated is absorbed by continuously feeding water in a combination of heat-transfer surfaces, resulting in a continuous generation of steam. The water fed into a steam generator is called feedwater. Steam and feedwater together is called working fluid. The name "steam generator" is also still called "boiler," but modern steam generators in the supercritical class do not involve the "boiling" phenomenon. The fuel-firing equipment of a steam generator should completely burn the fuel used in the furnace to release as much energy as possible. Air is fed into the furnace for combustion of fuel-forming products of combustion or flue gas. The heat released by burning fuel is absorbed in different heat-transfer surfaces to the maximum level possible practically and economically to keep the loss of heat to a minimum. In the heating surfaces, the flue gas transfers its heat to the working fluid. Thus, the feedwater is pre-heated to the saturation temperature and vaporized. The saturated steam thus formed is further superheated. After passing over the heating surfaces at various zones the flue gas is cooled and discharged to the atmosphere through a stack.

The primary function of a steam generator is to generate steam under pressure, but modern steam generators are also required to do the following:

- i. Ensure generation of exceptionally high-purity steam by eliminating all impurities from saturated steam.*
- ii. Raise the degree of superheat of supplied steam as specified and maintain the same temperature over a defined range of load.*
- iii. In large power stations after partial expansion in the turbine steam is returned to the steam generator for further superheating and then transmitted to the turbine for complete expansion. This new degree of superheat is called "reheat" and should also be maintained constant over a defined range of load.*
- iv. While executing the above duties, a steam generator must utilize the heat of combustion of fuel as efficiently as possible.*

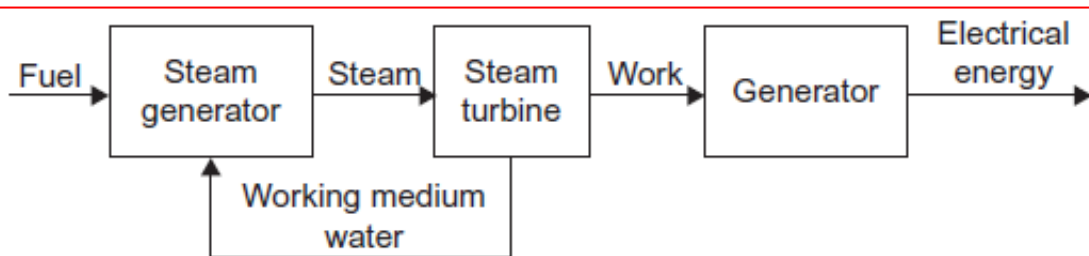


Figure 2.1 Schematic arrangement of a steam power station.

BOILING AND CIRCULATION

Boiling of a liquid refers to a condition where vapor bubbles form on the heating surface of the liquid. Formation of bubbles depends on the fluid properties and the operating and surface conditions. When heat is added to a liquid, its temperature does not increase beyond its saturation temperature corresponding to its pressure, instead, the energy used results in a change of phase from a liquid to a gaseous state, e.g., from water to steam. This process, which takes place at constant pressure and constant temperature, is known as boiling.

The boiling point of a liquid is defined as the temperature at which its vapor pressure is equal to the pressure of the gas above it. This temperature is also called saturation temperature. In an open vessel the boiling point of a liquid refers to a temperature at which its vapor pressure is equal to the external pressure at one atmosphere. In the event that the external pressure becomes less than one atmosphere, the boiling point of the liquid gets reduced. As the external pressure rises above one atmosphere, the boiling point of the liquid also starts rising. Figure 2.2 shows the boiling point of water as a function of the external pressure. Note the boiling temperature of 373 K corresponds to a pressure of 101.3 kPa and temperature of 473 K to a pressure of 1554.0 kPa.

Boiling occurs when heat is added to the liquid at such a rate that its temperature is at least equal to its saturation temperature corresponding to the total pressure over the free surface of the liquid. If the vessel is open to atmosphere, the vapor displaces the air from its surface of the liquid entirely, whereas with evaporation the vapor is removed by diffusion. The heat, which changes water to steam, is called the heat of evaporation or heat of vaporization. The heat transfer that takes place from the wall to the liquid during boiling is given by the convective heat transfer equation

$$Q = hA\Delta T \quad (2.1)$$

where

Q : Rate of heat flow, W

h : Heat transfer coefficient, W/m^2K

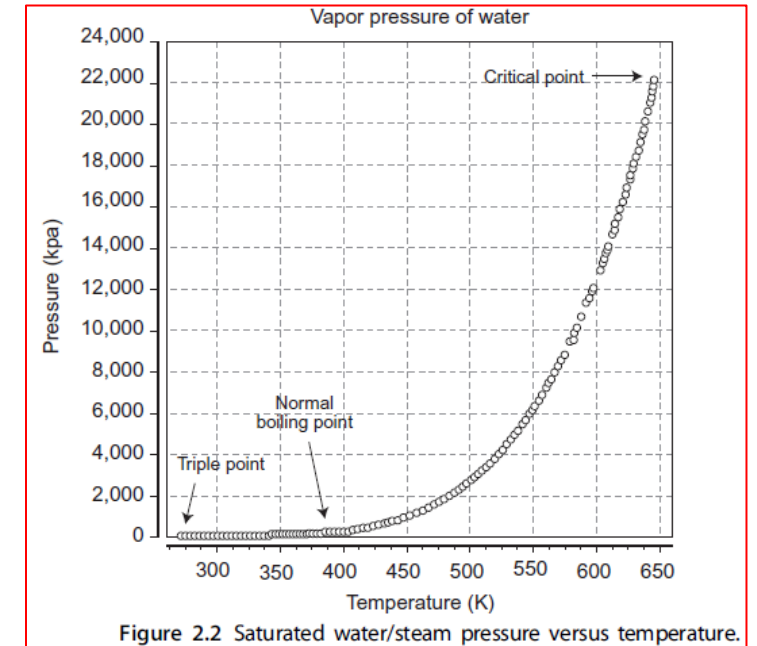
A : Area across which heat flow takes place, m^2

ΔT : Difference in temperature between the heating surface and the temperature of the fluid, K

(Log Mean Temperature Difference (lmtd)) $\longrightarrow \frac{(T_a - T_b)}{\ln(T_a/T_b)}$

The "heat flux" at the heating surface is found from Eq. 2.1 as:

$$q = \frac{Q}{A} = h\Delta T \quad (2.2)$$



• Pool boiling

- When a large volume of liquid is heated by a submerged heating surface and the motion is caused by free convection currents stimulated by agitation of the rising vapor bubbles, it is called pool boiling [2]. During the course of heating it is noted that a general correlation lies between the gradually increasing heat flux, q , from the immersion heater and the corresponding temperature difference between the surface of the heater and the bulk of liquid, ΔT . The result of such correlation is presented in Figure 2.3 [1].
- At point A since the heat flux is very low, no boiling occurs and no bubbles are formed, hence movement of liquid is by natural convection.
- A further increase in heat flux will result in vapor bubble formation at the hot surface, yet the bulk of the liquid may still be below saturation temperature.].
- Further rising of heat flux results in rapid formation of a large number of bubbles on the hot surface.
- Nucleate boiling is incipient boiling, which is characterized by the growth of bubbles on the heated surface that rise from discrete points on a surface, whose temperature is only slightly above the liquid's saturation temperature.

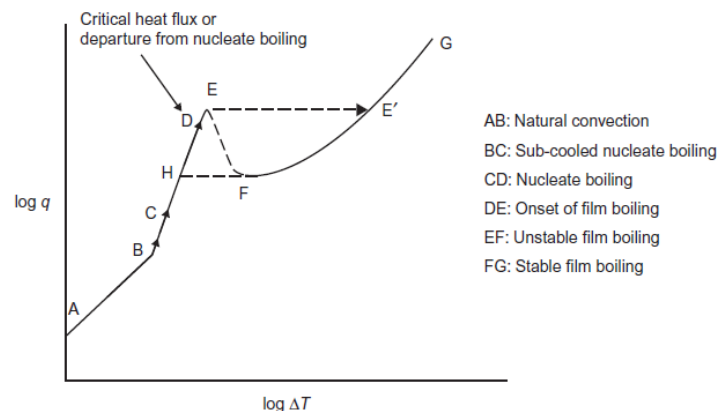


Figure 2.3 Heat flux versus temperature difference.

departure from nucleate boiling (DNB) or the "critical heat flux."

Heat flux is extremely high, as for example with water at atmospheric pressure heat flux is about 1500 kW/m² [2]. The temperature difference, ΔT , between the boiling liquid and the heating surface at which the critical heat flux occurs, is known as the critical temperature difference. During operation of the boiler it is difficult to theoretically predict the heat transfer coefficients of nucleate boiling that would ensure ΔT remains below the critical temperature difference. Nevertheless, the ability to predict the value of the maximum heat flux, q_{\max} , i.e., point E in Figure 2.3, is also useful as this represents an upper limit in the nucleate boiling heat transfer. The following formula, from Roshenow and Griffith, gives the value of q_{\max} in SI units, i.e., W/m² [3].

$$q_{\max}/(\rho_V * \lambda) = 0.0121 * \{(\rho_L - \rho_V)/\rho_V\}^{0.6} \quad (2.3)$$

where

ρ_L : Density of liquid, kg/m³

ρ_V : Density of vapor, kg/m³

λ : Latent heat of the fluid, kJ/kg

Once the critical heat flux data is determined it is compared with the required heat flux data. For the design to be acceptable, the critical heat flux, which is dependent on the following parameters, must always be greater than the heat flux generated in the course of boiling.

- Operating pressure
- Steam quality, i.e., surface tension, subcooling, etc.
- Type of tube: rifled or plain bore
- Tube diameter
- Flux profile around the tube
- Steam/water flow in the tube
- Angle of inclination of tube

Forced-convection boiling

Besides pool boiling, vapor is also generated by passing liquid through a tube heated either by firing fuel, as in a once-through boiler, or by condensing steam, as is commonly used in process industries. This method of generating vapor is known as forced-convection boiling. The nature of boiling in forced convection is similar to pool boiling in many ways like the regimes of nucleate or film boiling

Both heat transfer and pressure drop are affected by the behavior of the two-phase flow, which keeps on changing along the tube due to gradual evaporation of liquid. In a once-through tube, liquid below or at the saturation point enters at the bottom and gradually evaporates until a dry or superheated vapor leaves at the top. In the process, some or all of the following regions are encountered by the flow system (Figure 2.4):

1. A liquid heating region or a subcooled boiling region: in this region bubbles are formed at the tube surface but subsequently collapse within the liquid. The heat transfer to the liquid will be by natural convection.
2. A saturated nucleate boiling region: where forced convection dominates. This is the region in which the local tube surface temperature is sufficiently superheated corresponding to the local pressure, causing bubbles to form at the surface.
3. An annular region: where the vapor travels along the tube as a continuous stream carrying with it some droplets of liquid, while the remaining liquid flows as a thin film. Heat transfer in this region takes place by conduction and forced convection. Following the annular region tube surface no longer remains wetted resulting in sudden jump in wall temperature; eventually 'dry out' condition or 'critical heat flux' is arrived at.
4. A region of pure vapor: in this region heat transfer takes place by natural convection.

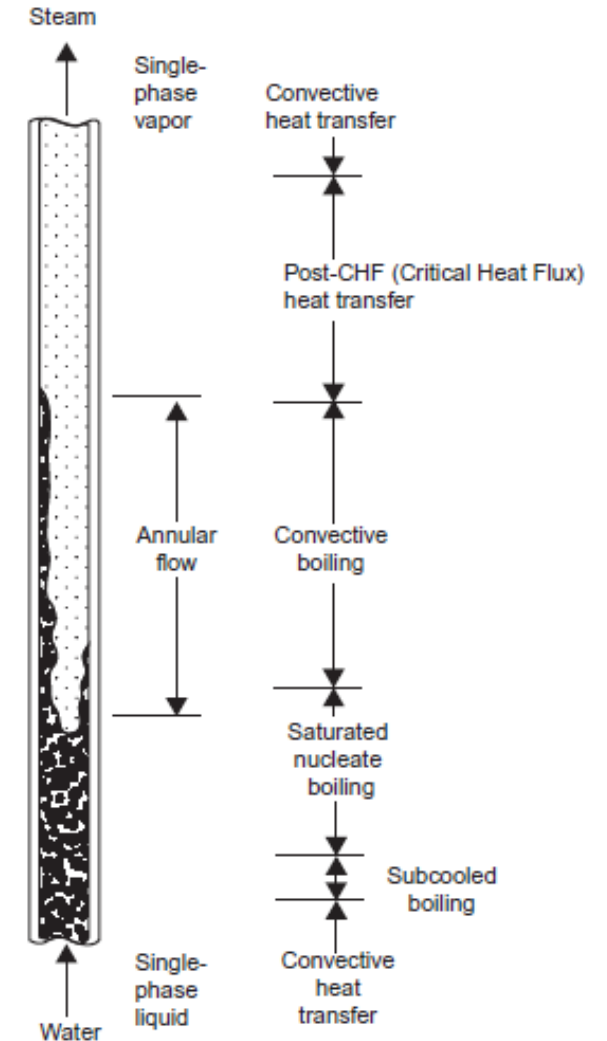


Figure 2.4 Evaporation of liquid in once-through tube.

The first and the fourth regions are extremes consisting of liquid only and vapor only regions, respectively. In these regions in turbulent flow the phenomena of heat transfer to water and to superheated steam in pipes is described by the Dittus-Boelter equation as follows [3]. For turbulent flow through tubes the phenomenon of heat transfer to water is described by;

$$Nu = 0.023 * (Re)^{0.8} (Pr)^{0.4} \quad (2.4)$$

For superheated steam through pipes, the equation is

$$Nu = 0.0133 * (Re)^{0.84} (Pr)^{0.33} \quad (2.5)$$

where

Nu : Nusselt Number = hd/k
 Re : Reynolds Number = $\rho Vd/\mu$
 Pr : Prandtl Number = $c_p\mu/k$
 h : Convective heat transfer coefficient, W/m^2K
 ρ : Density of fluid, kg/m^3
 V : Velocity of fluid, m/s
 d : Internal diameter of pipe, m
 μ : Dynamic viscosity of fluid, $kg/m.s$
 c_p : Specific heat of fluid, $kJ/kg.K$
 k : Thermal conductivity, $W/m.K$

From Eq. 2.4, the heat transfer coefficient of water is found as

$$h_{water} = 1063 * (1 + 2.93 * 10^{-3} * T_f) * (V_f^{0.8}/d^{0.2}) \quad (2.6)$$

From Eq. 2.5, the heat transfer coefficient of superheated steam is

$$h_{steam} = 4.07 * (1 + 7.69 * 10^{-4} * T_g) * \{V_g^{0.79}/(d^{0.16} * l^{0.5})\} \quad (2.7)$$

where

T_f : Absolute temperature of bulk water, K
 T_g : Absolute temperature of superheated steam, K
 V_f : Velocity of water through tube, m/s
 V_g : Velocity of superheated steam through tube, m/s
 d : Internal diameter of pipe, m
 l : Length of pipe, m

Nu= ratio of convective to conductive heat tr.

Pr= ratio of momentum diffusivity to thermal diffusivity.

Re=the ratio of inertial forces to viscous forces.

Example 2.1

In a once-through boiler, feedwater enters through the bottom of tubes, receives heat from combustion of coal, and escapes from the top of the tubes as superheated steam. Calculate the heat transfer coefficients of water and superheated steam using the following data:

Pressure of feedwater at tube inlet = 27.0 MPa

Absolute temperature of feedwater, $T_f = 563$ K

Pressure of superheated steam at tube outlet = 24.5 MPa

Absolute Temperature of superheated steam, $T_g = 838$ K

Velocity of Feedwater through each tube, $V_f = 6$ m/s

Internal diameter of each tube = 63.5 mm

Length of pipe = 30 m

Solution: The density of feedwater at 27.0 MPa and 563 K is

$$\rho_f = 763.942 \text{ kg/m}^3$$

The density of superheated steam at 24.5 MPa and 838 K is

$$\rho_g = 74.532 \text{ kg/m}^3$$

For constant mass flow through each tube,

$$\rho_f * V_f * A = \rho_g * V_g * A$$

Hence, the velocity of superheated steam through each tube is

$$V_g = \rho_f * V_f / \rho_g = 763.942 * 6 / 74.532 = 61.5 \text{ m/s}$$

Applying Eq. 2.6, the heat transfer coefficient of water is

$$h_f = 1063 * (1 + 2.93 * 10^{-3} * 563) * (6^{0.8} / 0.0635^{0.2}) = 20496.43 \text{ W/m}^2 \text{ K}$$

Applying Eq. 2.7, the heat transfer coefficient of superheated steam is

$$h_g = 4.07 * (1 + 7.69 * 10^{-4} * 838) * \{61.5^{0.79} / (0.0635^{0.16} * 30^{0.5})\} = 49.18 \text{ W/m}^2 \text{ K}$$

- **DESIGN**

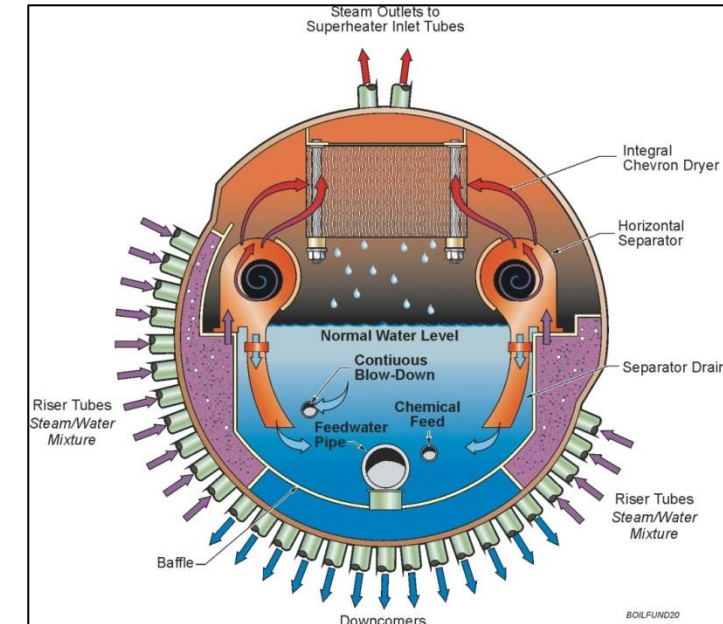
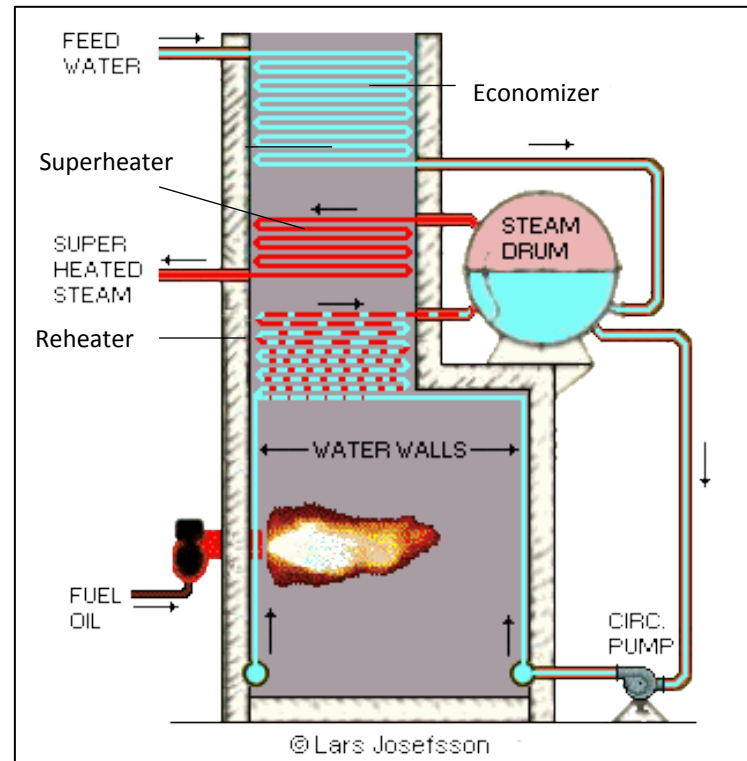
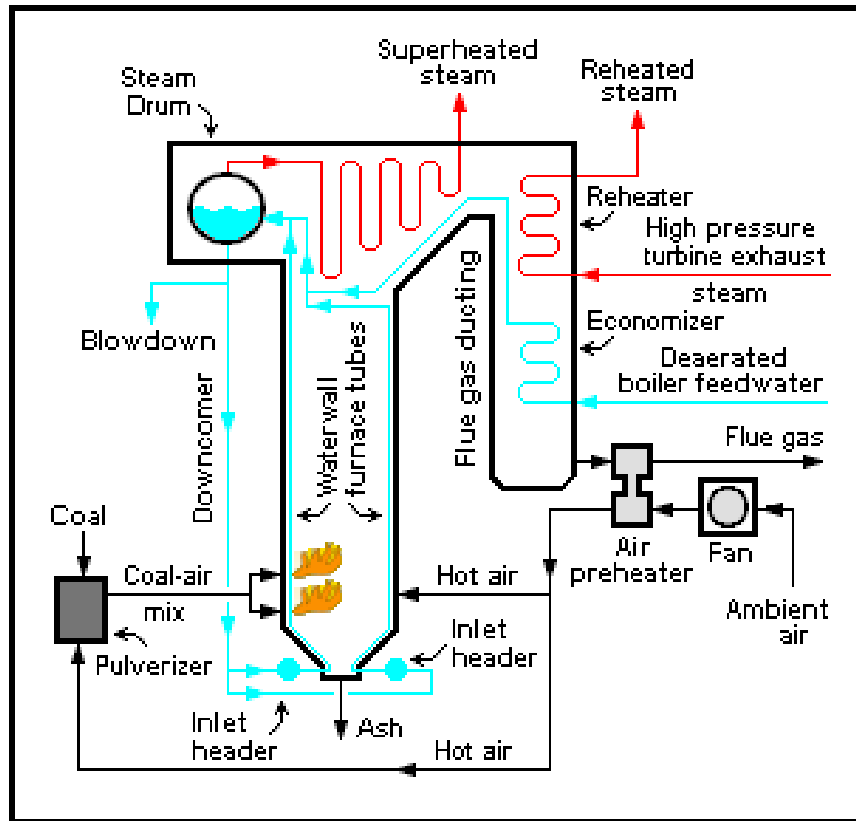
- As discussed earlier, the chemical energy in fuel is converted to heat energy by combustion, and low-temperature water in the steam generator receives this heat of combustion and becomes high-energy steam for use in the steam turbine. The salient features of this heat-transfer process encompasses the following circuits, systems, and/ or areas:
 - a. Water and Steam Circuit
 - b. Furnace
 - c. Fuel-burning System
 - d. Draft System
 - e. Heat Recovery System

- **Water and steam circuit**

- At operating pressure, temperature of the feedwater, prior to entering the steam generator, is less than that at the saturated condition, i.e., the entering feedwater is under the subcooled condition. Within the steam generator, the feedwater temperature is elevated to the near-saturation temperature in the economizer. Thereafter, the sensible heat for bringing the feedwater to saturation and the latent heat for evaporation of saturated feedwater is added in the water-wall for generating steam. This steam is separated from the steam-water mixture coming out from the water-wall and purified in the boiler drum to ensure the supply of saturated steam. The saturated steam is further superheated to attain the desired energy level. The heat-transfer surfaces discussed so far, e.g., economizer, water-wall, superheaters/ reheaters, etc., are located in the steam generator in the flow path of products of combustion so that heat is absorbed efficiently in proper proportions in various heat transfer zones.
- The evaporation zones are comprised of the boiler drum, downcomers, and riser tubes or simply risers. While the boiler drum and downcomers are located outside the heat transfer zone, the riser tubes are exposed to heat transfer. While flowing through the riser tubes the feedwater receives the heat of combustion for evaporation. The riser tubes in modern steam generators are arranged such that they form the enclosure of the combustion chamber or furnace and receive the heat. Hence, riser tubes are usually called water-walls.

The feedwater from the economizer enters the boiler drum, flows down through the downcomers, passes through the pipes to the water-wall bottom header and rises through the riser tubes. In the riser tubes, the feedwater absorbs the heat, part of which is converted to steam and re-enters the boiler drum as a steam-water mixture. From this mixture, steam is separated in the boiler drum and purified. Dry saturated steam from the boiler drum is lead out through the saturated steam pipes to the superheaters. The complete flow path of working fluid is shown in Figures below. The process of separation and purification of steam in the boiler drum is accomplished by drum internals, e.g., cyclones, baffles, etc., chemical and feedwater admission piping, blow-down lines, etc. . The process includes three steps: separation, steam washing, and scrubbing. Separation is the process of removing the bulk mass of water from steam and is accomplished by any one of following means:

- Gravity, Abrupt change in flow direction, Centrifugal action, Impact against a plate, Use of baffles



The arrangement of drum internals must be such as to ensure the following [1,3]:

- i. Provide high-purity steam to prevent salt deposition in superheaters
- ii. Provide steam-free water to the downcomers for maximum circulation
- iii. Assure minimum pressure drop through the separators
- iv. Provide simple design and easy accessibility for drum inspection

The function of the superheaters and reheaters is to raise the total heat content of the steam (superheat the steam). There are two fundamental advantages of providing superheaters and reheaters: first, gain in efficiency of the thermodynamic cycle and second, enhancement of life in the last stage of the turbine by reduction of moisture content of the flowing steam. In low-temperature boilers only convection superheaters are used, but large, high-temperature steam generators use both radiant and convection superheaters along with the reheaters to achieve the required superheat and reheat final steam temperatures. Today, modern steam power plants run at lower load at rated conditions during off-peak hours. Hence, the superheater/reheater elements are designed to achieve anormal operating temperature at about 60% of the boiler maximum continuous rating(BMCR) load for better steam turbine performance. The location of superheaters and reheaters in the flue-gas path are normally split into primary and secondary sections to facilitate reasonably unchanged steam temperature control.

Furnace

In the furnace of a steam generator combustion of fuel takes place with atmospheric air to release heat. Hence, it is essential to ensure complete combustion of fuel under all operating conditions to harness the maximum heat potential of the fuel under combustion.

Fuel-burning system

The primary function of a fuel-burning system is to provide controlled and efficient conversion of chemical energy of the fossil fuel into heat energy. The heat energy thus generated is transferred to the heat-adsorbing surfaces of the steam generator. The main equipment for firing fossil fuel in a furnace is the burner.

An ideal fuel-burning system in a furnace is characterized by the following:

- No excess oxygen or unburned fuel in the product of combustion
- Low rate of auxiliary ignition energy input to initiate and sustain continuity of combustion reaction
- Wide and stable firing range
- Fast response to demand for a change in firing rate
- Uniform distribution of flue-gas weight and temperature in relation to parallel circuits of heat-absorbing surface
- High degree of equipment availability with low maintenance requirement
- Safety must be the main requirement under all operating conditions

Draft system

For sustaining a healthy combustion process in a furnace it needs to receive a steady flow of air and at the same time remove products of combustion from the furnace without any interruption. When only a chimney is used for the release of products of combustion, the system is called a natural draft system. When the chimney is augmented with forced draft (FD) fans and/or induced draft (ID) fans the system is called a mechanical draft system. Small steam generators use natural draft, while large steam generators need mechanical draft to move large volumes of air and gas against flow resistance.

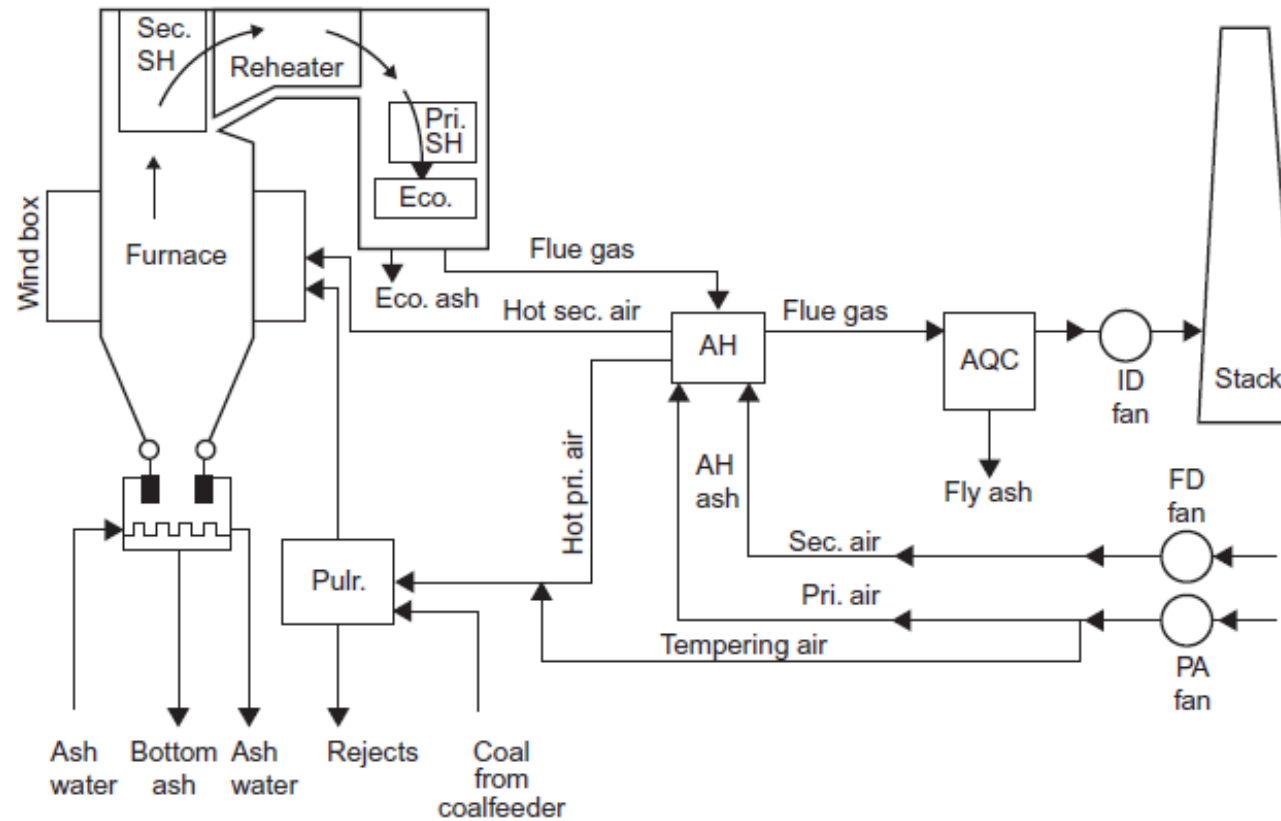


Figure 2.9 Fuel, ash, and draft system.

Heat recovery system

The flue gas or products of combustion leaving the convection heat-adsorbing surfaces in large steam generators still carry considerable heat energy at a temperature much above the steam saturation temperature. This heat, if exhausted to atmosphere without further heat transfer, will result in considerable losses. As a natural consequence of economic operation of a steam generator, a major part of this heat is recovered by placing the economizer thereafter the air heater in the exit flue-gas path. Theoretically it is possible to reduce the flue-gas temperature to that of the incoming air to the air heater, but for technical and economic considerations, as described below, such reduction in exit flue-gas temperature is prohibited.

- i. All fossil fuels are sulfur-bearing fuels. As such, if the exit flue-gas temperature from the air heater falls below the acid dew point of the as-fired fuel, sulfurous or sulfuric acid will form, causing back-end corrosion in the air heater. As a consequence, both maintenance cost and plant outages will increase.
- ii. Capital costs incurred for installing additional heat recovery surfaces may offset the savings obtained by reducing flue-gas temperature.

In *an economizer* the incoming feedwater absorbs heat from the exiting flue gases, raising the temperature of the feedwater as close as possible but lower than its saturation temperature prior to entering into the evaporating circuit. The remaining heat in flue gas downstream of the economizer is recovered in air heater to raise the temperature of combustion air before it enters the furnace. The energy thus recovered returns to the furnace and reduces the amount of heat that must be released by the fuel to maintain the desired furnace temperature, thereby ensuring reduction in the amount of fuel to be burned. The hot air is also used to dry coal in a pulverized coal-fired boiler.

CLASSIFICATION AND UTILIZATION

The function of a steam generator is to provide controlled release of heat in the fuel and to transfer this heat safely, reliably, and efficiently to the feedwater and steam. In a boiler no work is done on or by the fluid as it passes through the system. Hence, based on its use, location of fire and water spaces, type of circulation, arrangement of steam and water spaces, etc., steam generators are classified as follows [7]:

a. As per location of fire and water spaces

- Fire Tube Boiler
- Water Tube Boiler

b. Based on circulation

- Natural Circulation Boiler
- Forced/Assisted Circulation Boiler

c. Based on pressure requirement

- Subcritical Boiler
- Supercritical Boiler

d. As per arrangement of steam and water spaces

- Drum Type Boiler
- Once-through Boiler

e. Based on type of firing/heat transfer

- Stoker-fired Boiler
- Fossil Fuel (Gas/Oil/Coal)-fired Boiler
- Fluidized-bed Boiler
- Waste Heat Recovery Boiler

Fire-tube boiler

In this type of boiler hot gases pass through the tubes and water is contained in the shell. As the name suggests, its general construction is as a tank of water perforated by tubes that carry hot flue gases from the fire. The tank is usually cylindrical in shape to realize the maximum strength from simple structural geometry. The tank may be installed either horizontally or vertically. Typically the furnace and the grate, on which fuel is burnt, are located underneath the front end of the shell (Figure 2.10).

The gases pass horizontally to the rear, then either release through the stack at the rear or reverse directions and pass through the horizontal tubes to stack at the front. The fire tubes can be placed horizontally or vertically or at an inclined plane in a furnace. In this type of boiler boiling takes place in the same compartment where water is stored. As a result, only saturated steam used to be produced in older designs of this type of boiler, but today, a fire-tube boiler may generate superheated steam as well. It has a cylindrical barrel containing the fire tubes, but also has an extension at one end to house the "firebox." This firebox has an open base to provide a large grate area and often extends beyond the cylindrical barrel to form a rectangular or tapered enclosure.

Today, they are used extensively in the stationary engineering field, typically for low-pressure steam use such as for heating a building. However, the steam-generating capacity and outlet steam pressure of these boilers are limited, therefore, they are unable to meet the needs of larger units. Another disadvantage of these boilers is that they are susceptible to explosions.

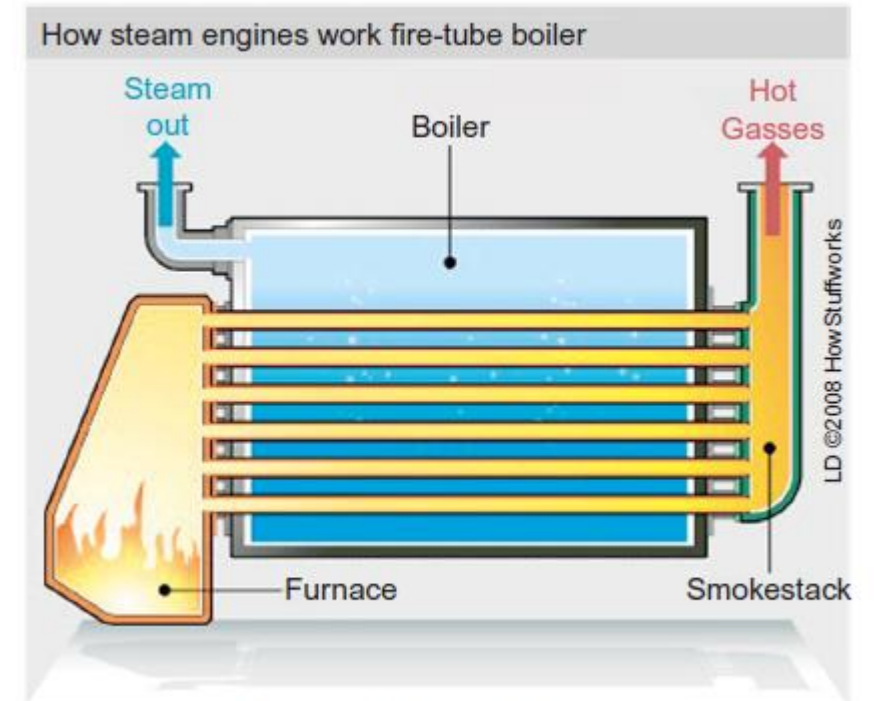


Figure 2.10 Fire-Tube Boiler.

Water-Tube Boiler (Figure 2.11)

To generate high evaporation rate accompanied by high steam pressure, the fire-tube boiler becomes exorbitantly heavy; therefore, the size and weight become extremely difficult to manage. In regards to the size of the shell, these shortcomings are circumvented by passing flue gases outside the tubes, instead of inside, and water is circulated through the tubes for evaporation. Baffles are installed across the tubes to allow cross flow of flue gases to ensure maximum exposure of the tubes. On the basis of configuration of tubes inside the furnace, this boiler is further classified as "straight-tube boiler" and "bent-tube boiler."

In a water-tube boiler water circulates in tubes heated externally by the hot flue gas. Fuel is burned inside the furnace, creating hot gas that heats up the water in the steam generating tubes. Cool water at the bottom of the steam drum returns to the feedwater drum of small boilers via large-bore "downcomer tubes," where it helps pre-heat the feedwater supply. In large utility boilers, feedwater is supplied to the steam drum and the downcomers supply water to the bottom of the water-walls. The heated water then rises into the steam drum. Here, saturated steam is drawn off the top of the drum. In large utility boilers water-filled tubes form the walls of the furnace to generate steam and saturated steam coming out of the boiler drum re-enter the furnace through a superheater to become superheated. The superheated steam is used in driving turbines.

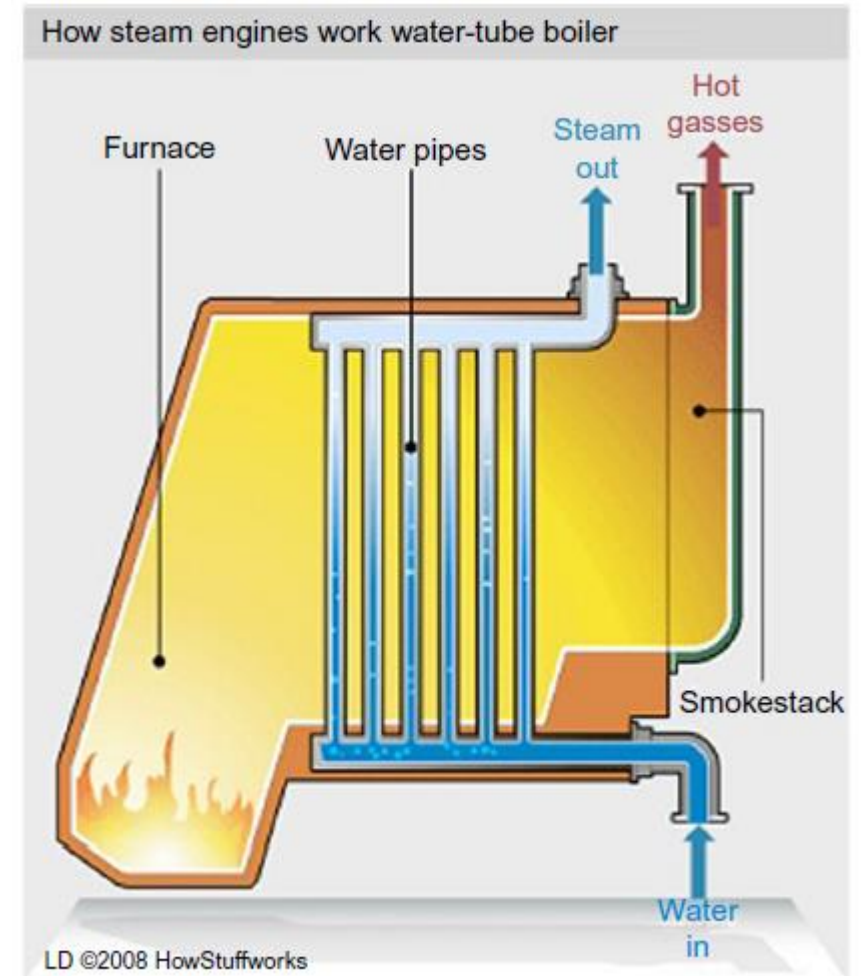


Figure 2.11 Water-tube boiler.

Natural circulation boiler

Boilers in which motion of the working fluid in the evaporator is caused by thermosiphon effect on heating the tubes are called "natural circulation boilers." As discussed in [section 2.2](#) circulation of water in natural circulation boilers depends on the difference between the density of a descending body of relatively cool and steam-free water and an ascending mixture of hot water and steam. The difference in density occurs

because the water expands as it is heated, and thus, becomes less dense. All natural circulation boilers are drum-type boilers.

Forced/assisted circulation boiler

As noted in [section 2.2](#) the density difference between the saturated liquid and saturated vapor starts diminishing at 18 MPa or higher fluid pressure, thus it is difficult to maintain natural circulation of fluid flow in boiler tubes. In such cases fluid flow is ensured with the help of forced/assisted circulation using pumps. The forced/assisted circulation principle applies equally in both supercritical and subcritical ranges.

Subcritical boiler and supercritical boiler

The critical pressure is the vapor pressure of a fluid at the critical temperature above which distinct liquid and gas phases do not exist. As the critical temperature is approached, the properties of the gas and liquid phases become the same, resulting in only one phase. The point at which the critical temperature and critical pressure is met is called the critical point. The critical pressure and critical temperature of water and steam are 22.12 MPa and 647.14 K, respectively. Any boiler that operates below the critical point is called a subcritical boiler, and one that operates above the critical point is known as a supercritical boiler. (Supercritical boilers are covered in [section 2.10](#).)

Drum-type boiler

In a drum-type boiler the drum acts as a reservoir for the working fluid. These boilers have one or more water drums, depending on the size and steam-generating capacity. The drum is connected to cold downcomers and hot riser tubes through which circulation of working media takes place. The lower portion of the drum with feedwater is called the water space and the upper portion of the drum occupied by steam is called the steam space. A drum-type boiler can be either the natural circulation type or forced/assisted circulation type. Drum-type boilers are essentially subcritical boilers; they operate below the critical pressure of the working fluid. The economic design pressure limit of fluid in a drum-type boiler is around 18 MPa.

- **Once-through boiler**

This type of boiler does not have a drum. Simply put, a once-through boiler is merely a length of tube through which water is pumped, heat is applied, and the water is converted into steam. In actual practice, the single tube is replaced by numerous small tubes arranged to provide effective heat transfer, similar to the arrangement in a drumtype boiler. The fundamental difference lies in the heat-absorbing circuit. Feedwater in this type of boiler enters the bottom of each tube and discharges as steam from the top of the tube. The working fluid passes through each tube only once and water is continuously converted to steam. As a result there is no distinct boundary between the economizing, evaporating, and superheating zones. The circulation ratio of this type boiler is unity [6]. These boilers can be operated either at subcritical or at supercritical pressures.

- **Stoker-fired boiler**

Prior to the commercial use of "fluidized-bed boilers" , stoker-fired boilers provided the most economical method for burning coal in almost all industrial boilers rated less than 28 kg/s (100 tph) of steam. This type of boiler was capable of burning a wide range of coals, from bituminous to lignite, as well as byproducts of waste fuels. However, over the years this type of boiler has become less popular due to more efficient technological advancement. In stoker-fired boilers coal is pushed, dropped, or thrown on to a grate to form a fuel-bed. Stokers are divided into two general classes: overfeed, in which fuel is fed from above, and underfeed, wherein fuel is fed from below. Under the active fuel-bed there is a layer of fuel ash, which along with air flow through the grate keeps metal parts at allowable operating temperatures. [Figure 2.12](#) and [Figure 2.13](#) show a chain-grate overfeed stoker and a traveling grate overfeed stoker, respectively.

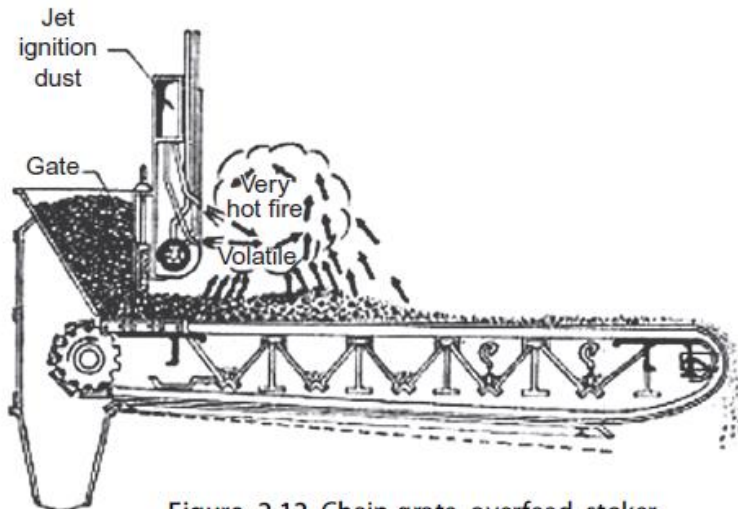


Figure 2.12 Chain-grate overfeed stoker.

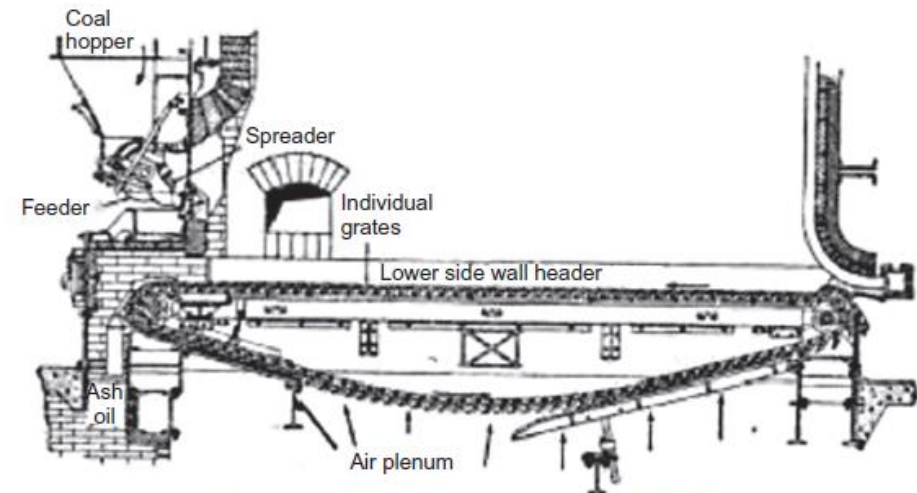


Figure 2.13 Traveling-grate overfeed stoker.

Fossil fuel-fired boiler

Coal, fuel oil, and natural gas are the main types of fossil fuel [8]. These fuels can generate a substantial quantity of heat by reacting with oxygen. These fuels consist of a large number of complex compounds comprised of five principal elements: carbon (C), hydrogen (H), oxygen (O), sulfur (S), and nitrogen (N). Any of these three fuels, i.e., coal, fuel oil, and natural gas, can be used in steam power stations, but coal plays a large role in power generation because of the enormous reserves of coal in many countries around the world, e.g., the United States, China, Mongolia, India, Indonesia, Russia, Poland, South Africa, Australia, etc. Most of the large steam generating stations in these countries are coal based, and coal is expected to remain a dominant fuel for this purpose for many years. Coal-fired power plants at present account for about 41% of power produced globally [9].

Coal-based steam-generating stations also dominate in those countries where natural gas and fuel oil are abundantly available, since the cost of natural gas and fuel oil is exorbitantly high compared to the cost of coal. Additionally, natural gas and fuel oil are more economical in other industries. In modern, large-capacity, coal-fired steam generators coal is burnt in suspension.

Fluidized-bed boiler

Fluidized-bed combustion ensures burning of solid fuel in suspension, in a hot inert solid-bed material of sand, limestone, refractory, or ash, with high heat transfer to the furnace and low combustion temperatures (1075-1225 K). The combustor-bed material consists of only 35% coal. Fluidized-bed combustion is comprised of a mixture of particles suspended in an upwardly flowing gas stream, the combination of which exhibits fluid-like properties. Fluidized-bed combustors are capable of firing a wide range of solid fuels with varying heating value, ash content, and moisture content. In this type of boiler, pollutants in products of combustion are reduced concurrently with combustion much of the ash and hence the particulate matter as well as sulfur is removed during the combustion process. Further, lower temperature combustion in the fluidized bed results in lower production of NOX and obviates any slagging problem.

Waste-heat recovery boiler

A waste-heat recovery boiler (WHRB) or a heat-recovery steam generator (HRSG) is a heat exchanger that recovers heat from a gas stream and in turn produces steam that can be used in a process or to drive steam turbines. A common application for HRSGs is in a combined cycle power plant (CCPP), where hot exhaust gas from a gas turbine is fed to the HRSG to generate steam.

BOILER MOUNTING AND ACCESSORIES

Equipment that are directly attached to, or within, the boiler are generally called boiler mountings. They are essential for safety, economics, and convenience. These mountings include water-level gauges, safety or relief valves, drain and blow-down valves, vent valves, water and steam sample connections, stop-check valves, soot blowers, etc. Accessories for measuring boiler-operating conditions include pressure gauges, water-level gauges, thermometers, thermocouples, water and steam flow meters, alarm arms, etc. There are also combustion control equipment and measuring devices. Interlock and protection devices protect the boiler from abnormal operating conditions such as low drum water level, high steam pressure, high steam temperature, and other off-normal conditions [10].

In many countries of the world it is mandatory that each and every boiler be provided with at least the following:

- i. Two safety valves on the boiler drum.
- ii. In case of boilers fitted with integral superheaters, an additional safety valve at the end of superheater outlet header.
- iii. Two means of indicating water level in boiler drum. These indicators should be fitted with shut-off valves and drain cocks.
- iv. A steam-pressure gauge.
- v. A steam-stop valve connecting the boiler or superheater to the steam delivery pipe. This valve should be located as near the outlet from the final superheater as is convenient and practical. In the case of boilers without superheaters the valve location should be as close as practical to the drum.
- vi. A non-return type of feed-check valve.
- vii. Two independent feed apparatus; each such apparatus should have a capacity of no less than the maximum continuous rating of the boiler. In case of battery of boilers two independent feed apparatus should be considered, provided a total supply of feedwater, no less than the combined maximum continuous rating of all active boilers, can be maintained even if any one of the two sources of power supply should fail.
- viii. A blow-down cock or valve placed at or as near as practical to the lowest point of the drum. The cock or valve should be fitted with a device that indicates its open and closed positions. (Blow-down is required to reduce the concentration of soluble salts in the boiler water, which results from the hardness of feedwater.)
- ix. Low-water alarms, equipped with automatic tripping device, to disconnect fuel supply and start the feed pump simultaneously in the event of low water in the boilers.
- x. Every boiler should be fitted with a valve or cock with a receiving screw for attachment of the Boiler Inspector's pressure gauge.
- xi. A manhole for inspection and effective cleaning of drum internals.

Every boiler that is provided with automatic water-level and/or firing control should also comply with the following additional requirements:

- 1. In the event of automatic-control failure, the boiler should be capable of being brought under immediate manual control.
- 2. Means of manual resetting should be provided to turn off the fuel and air supplies to the boilers should there be a failure of electricity supply or any fault in electrical circuits to water-level and/or firing control equipment.

Where the evaporating capacity of the boiler is greater than 1.03 kg/s (3.7 tph), it should have automatic water-level alarms and firing controls and can be tested regularly without altering the level of water in the boiler.

The control equipment for automatic water-level controls should be such as to regulate the feedwater supply to the boiler to effectively maintain the level of water in the boiler drum according to the pre-determined limits.

Automatic firing controls should effectively control the supply of fuel and air to the combustion equipment at all times. It is necessary to maintain complete control over the combustion process based on the quantity of CO, CO₂, or O₂ in the flue gases. The combustion equipment should shut off the fuel supply to burners or the fuel and air supply to the furnace under the following conditions [11]:

- a. Flame failure or pilot flame failure in cases of gas, oil, or pulverized fuel-fired boilers
- b. Failure to ignite fuel at the burner within a pre-determined time
- c. Fall in drum water level below a pre-determined safe level
- d. Failure of forced draft or induced draft fans or any automatic flue damper
- e. Increase or decrease in furnace pressure to a pre-determined value

Each boiler is also provided with steam/water/air soot blowers to dislodge soot and fly ash from the fire side passages of the boiler.

SUPERHEATERS AND REHEATERS

The superheater is one of the most critical elements of a boiler plant. Since it is located in the high-temperature zone it acquires close to the highest value of steam temperature as allowed by the metallurgical limit. The average temperature at which heat is supplied is increased by the superheating and hence the ideal cycle efficiency is increased and accordingly specific steam consumption (kg/kW.s) decreases. The efficiency of an un-superheated cycle does not increase continuously with boiler pressure up to the critical pressure, but reaches the maximum when the pressure is about 16 MPa; thereafter, it drops with an increase in pressure. This is due to the fact that the latent heat and the quantity of heat transferred at higher temperature becomes smaller as the pressure increases. Eventually, the effective average temperature at which heat is added starts to decrease and the efficiency falls. In contrast to the efficiency of an un-superheated cycle, the efficiency of the superheated cycle increases continuously with pressure. Furthermore, the dryness fraction at the steam turbine exhaust of an un-superheated cycle reduces as the boiler pressure is increased, resulting in the formation of droplets in the steam that erode the turbine blades and also reduce the turbine isentropic efficiency. To avoid these problems, the dryness fraction of steam is not allowed to fall below about 88% when passing the steam through superheater [12].

Even though superheating increases the dryness fraction, with the current operating steam temperature of about 810 K or above it is sometimes difficult to maintain the dryness fraction at or above 88%, particularly in large units. This problem is circumvented by adding another heat transfer surface called a "reheater" in the fluegas path. With a combined superheat-reheat arrangement, the steam expands in the turbine at two different pressure stages. The steam coming from the superheater expands in the high-pressure turbine to some intermediate pressure. The lowtemperature exhaust steam from the high-pressure turbine is then passed through the reheater in the steam generator to raise the steam temperature at a constant pressure to a level equal or higher than the superheat steam temperature. The reheated steam is then fed to the intermediate pressure turbine. The superheater and reheater heat transfer surfaces absorb heat from the products of combustion or flue gases either through radiation, i.e., they receive heat through radiation from the furnace or through convection, which means these surfaces receive heat from the hot flue gas from the furnace by convection heat transfer [13]. Accordingly, these heat transfer surfaces are called "radiant type" or "convection type."

ECONOMIZERS

In a subcritical boiler, the temperature of flue gases that leave steam-generating surfaces is determined by the saturation temperature of boiling water. The higher the steam pressure the higher the saturation temperature. Therefore, to permit the exchange of heat from flue gases to steam-generating surfaces, the temperature of flue gases must be above the saturation temperature of the water and is generally maintained at around 648-818 K. At this high temperature if the flue gases are exhausted to atmosphere, the loss in boiler efficiency will be exorbitantly high and unacceptable.

Hence, there is an opportunity to absorb the residual heat of flue gases leaving the superheater and reheater by the incoming subcooled feedwater in a separate heat transfer surface known as the "economizer." With this type of additional heat absorption the working fluid efficiency of the boiler also is enhanced substantially. For each 5.5 K increase in the feedwater temperature, the boiler efficiency rises by about 1%. For each 22K drop in flue gas temperature, there is an increase in boiler efficiency by 1% in a conventional boiler [1].

When flue gases travel across water-walls and superheaters only the latent heat and superheat of steam are supplied by the flue gases. The primary state of this latent heat is accomplished in the economizer, which is placed in series with the feedwater heaters of the regenerative system. The function of the economizer is, therefore, to raise the temperature of feedwater downstream of the highest pressure feedwater heater to the saturation temperature corresponding to the boiler pressure. The economizer, comprised of a bank of tubes over which flue gases pass, is a forced flow convection heat exchanger through which feedwater is supplied at a pressure higher than the pressure at the evaporating section. The size of the economizer is dictated by a comparative analysis of its costs and savings of fuel due to improvement in thermal performance. Economizer size is also governed by the temperature of the feedwater at the economizer inlet and the temperature of the exit flue gas from the economizer. With very low feedwater temperature, there is a possibility of external corrosion of economizer tubes due to the condensation of flue-gas constituents passing over them if the surface temperature of the tubes falls below the acid or water dew point of the flue-gas stream. In the event that the feedwater contains a significant amount of dissolved oxygen (>7 ppb (parts per billion) in high-pressure boilers) pitting and corrosion may take place inside the tubes. This problem is overcome by deaerating the feedwater and maintaining a pH of around 9.

An economizer may be designed as follows [5]:

a. According to geometrical arrangement

- Horizontal Tubes (Figure 2.15)
- Vertical Tubes (Figure 2.16)

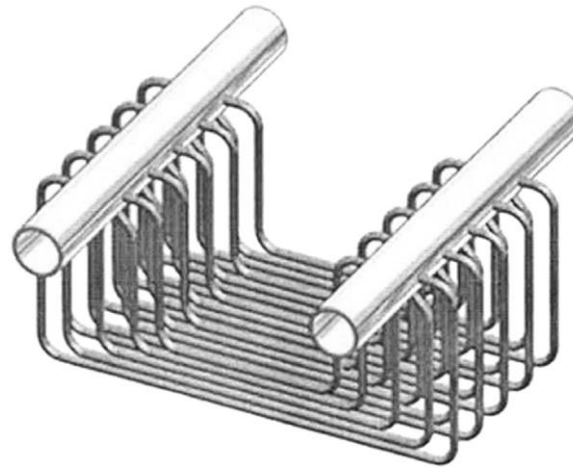


Figure 2.15 Horizontal tube economizer

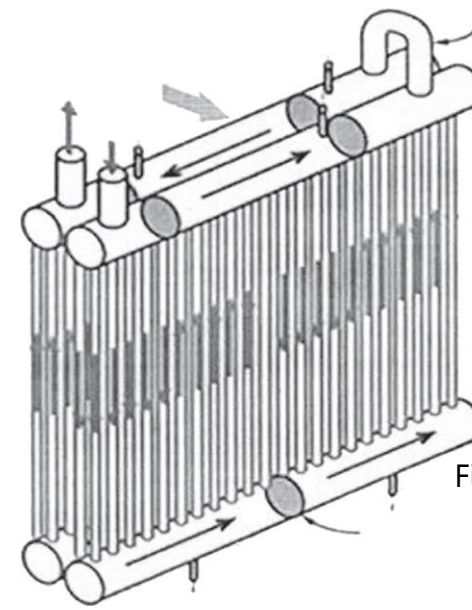


Figure 2.16 Vertical tube economizer.

b. Based on the direction of flue gases with respect to tubes in the bank

- Longitudinal Flow
- Cross Flow (Figure 2.17)

c. According to the relative direction of flue-gases flow and feedwater flow

- Parallel Flow
- Counter Flow (Figure 2.18)

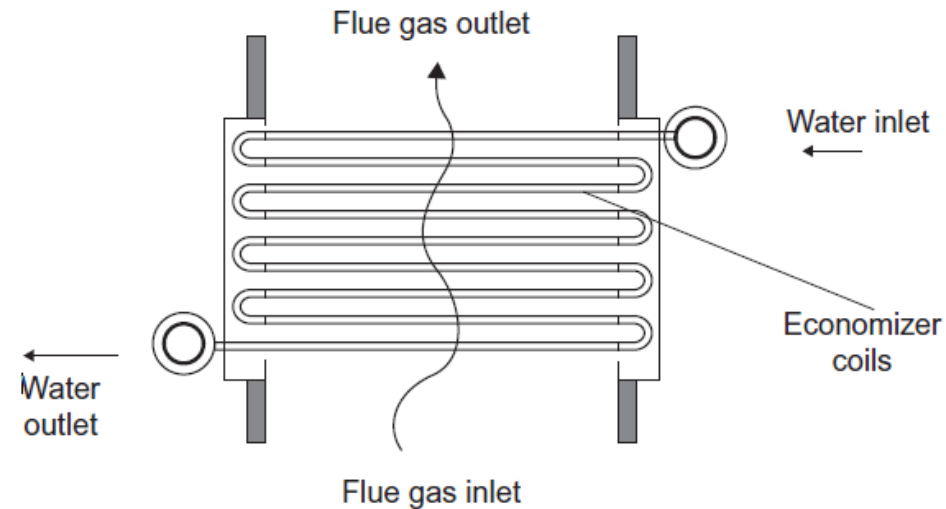


Figure 2.17 Crossflow economizer.

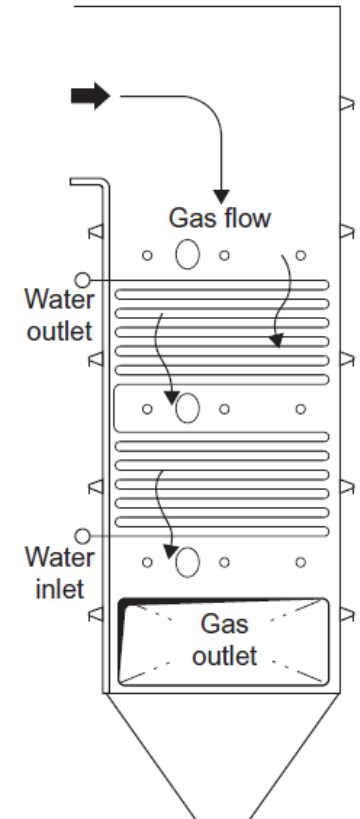


Figure 2.18 Counter flow economizer.

d. With regard to thermal performance

- Steaming
- Non-steaming

e. Based on the details of design and form of heating surface

- Plain Tube
- Finned Tube (Figure 2.19)
- Return-bend Tube
- Continuous Tube

f. According to the spacing of tubes and pattern of tubes (Figure 2.20)

- Staggered Tube
- In-line Arrangement

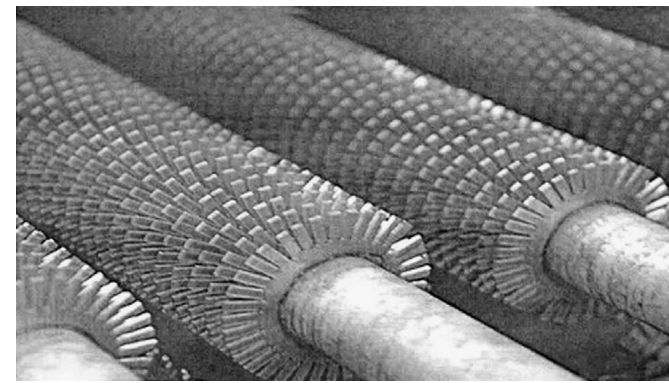


Figure 2.19 Finned-tube economizer

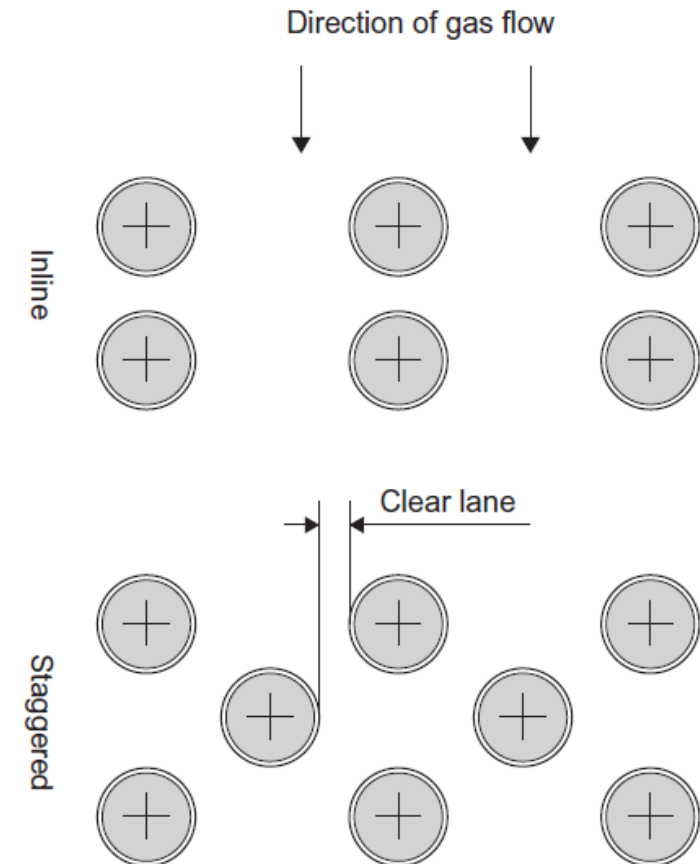


Figure 2.20 Inline and staggered tube economizer.

Counter-flow economizer (Figure 2.18)

- It is always best for flue gases from the boiler to flow down across the economizer tubes and for water to enter at the bottom and flow up through the tubes, thus reducing both surface and draft losses. Up-flow of water eliminates unstable water flow, gives the most uniform gas distribution, and makes the proper use of a steaming economizer possible.

Steaming economizer

- A steaming economizer may be defined as an economizer in which the temperature rise in the economizer is equal to or more than two-thirds of the difference between the economizer inlet feedwater temperature and the saturation temperature of the fluid at that pressure. This may be expressed mathematically as $(T_o - T_i) = (2/3) * (T_s - T_i)$ where T_o is the economizer outlet feedwater temperature, T_i the economizer inlet feedwater temperature, and T_s is the saturation temperature of feed water at its prevailing pressure. The steam content of water at the economizer outlet should not exceed 20% of the feedwater flow at full load, and should be lower at part loads. When feedwater make-up is low, it is advantageous to produce part of the steam in an economizer rather than in the evaporating section. From commercial point of view steaming is not allowed in the economizer, since there is the possibility of tubefailure due to two phase flows through the tubes.

Plain-tube economizer

- In some designs of boilers a continuous loop-type economizer is used. Tubes of this type of economizer have relatively small diameter to escalate the magnitude of heat transfer and to minimize fouling.

- **Finned-tube Economizer (Figure 2.19)**

- Heat absorption in the economizer tubes may also be intensified by welding fins to the plain tubes on the flue-gas side. This type of arrangement helps make the economizer compact, and it also increases the use of metal per unit heating surface area, providing much more gain in heat absorption and requiring smaller space. These types of economizers are more suited to gaseous and liquid fuel-fired boilers and in combined cycle plants. In coal-fired boilers, finned tubes often get plugged with ash and are usually not used.

Return-bend tube economizer

When it is essential to clean the inside of the tubes mechanically this design is best. This type of economizer is basically a horizontal steel-tube economizer, one end of which is bent 180 and the other end is equipped with a special flange bolted with a return-bend fitting.

Staggered-tube economizer (Figure 2.20)

A staggered arrangement of economizer tubes is preferable for gas and oil-fired boilers. This arrangement provides ascending motion of the feedwater in the economizer, resulting in free release of flue gases entering from the top of the economizer tubes and flowing down across the staggered tubes leaving at the bottom. This economizer is not suitable for coal-fired boilers. In a coal-fired boiler tubes must be in line to allow free flow of ash chunks through the bank of tubes.

AIR HEATERS

The temperature of flue gases downstream of the economizer is still quite high (588-698 K). The remaining heat in these flue gases is trapped in another heat absorbing surface called the air heater. The air heater uses a colder heat-receiving fluid (ambient air) and thus the outlet flue-gas temperature from the air heater is reduced to about 393-433 K, while the temperature of ambient air is raised from atmospheric to about 523-613 K prior to using this hot air in combustion. The heat recovered is recycled to the furnace by the combustion air and is added to the energy released from the fuel. This higher heat energy is further absorbed in the evaporating and/or the superheating surfaces, resulting in gain in boiler efficiency and less fuel consumption for the same output. The fuel savings are almost directly proportional to the rise in air-heater outlet air temperature. For a 110 K air temperature rise, typical fuel savings are about 4% and for a 280 K air temperature rise, fuel savings are about 11% [14]. Beyond fuel savings, air pre-heating is also required for drying coal in the pulverizer. The typical air temperature for drying coal ranges from 423-588 K.

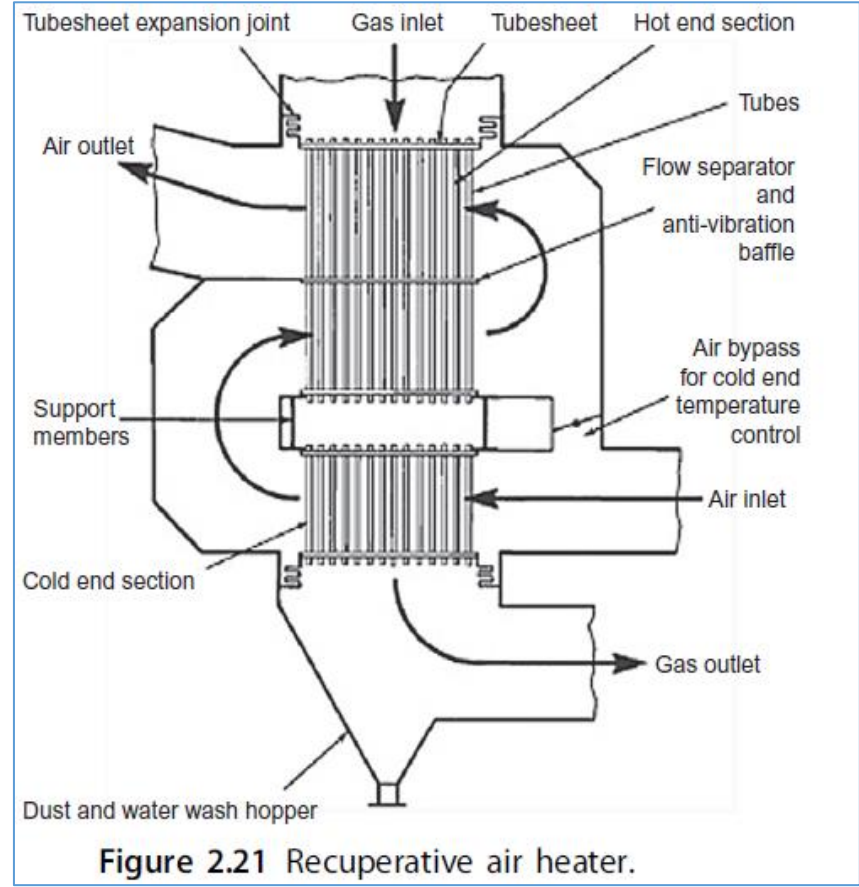
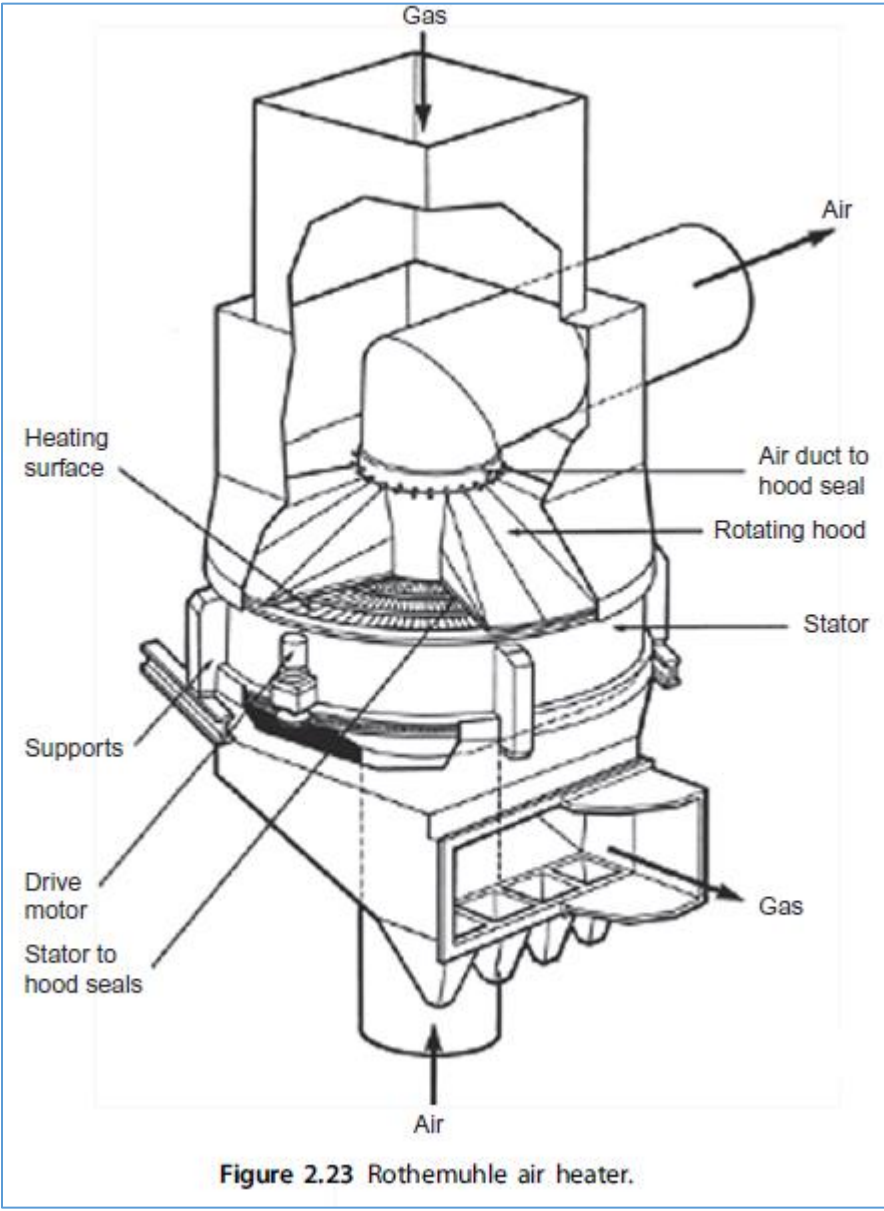
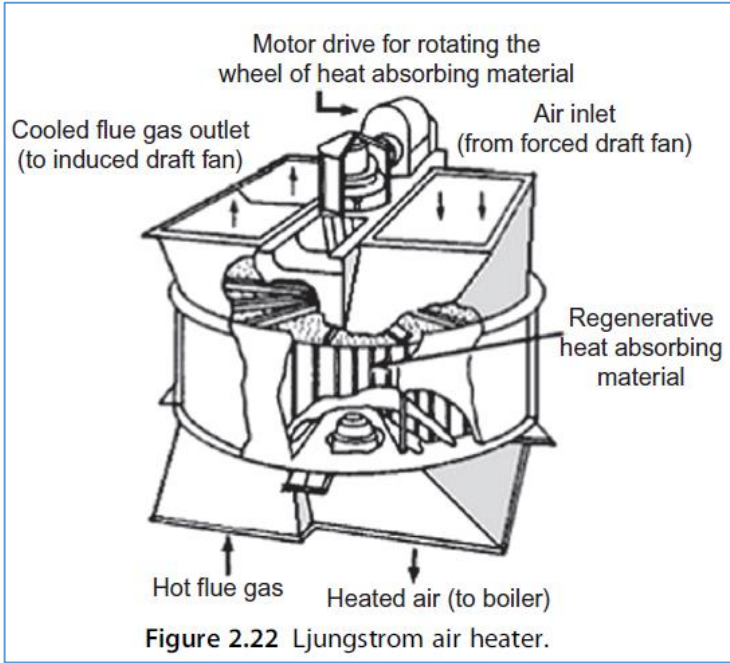
When considering the principle of operation there are two types of air heaters: **recuperative** or **regenerative**. Both recuperative and regenerative air heaters may be arranged in vertical or horizontal configuration. The heat transfer efficiency of regenerative air heaters is high compared to the efficiency of recuperative air heaters. However, the disadvantages of regenerative air heaters are leakage of air into the gas space and transport of fly ash into the combustion air system.

Recuperative air heater (Figure 2.21)

In a recuperative air heater, heat from the flue gases is transferred continuously to air through a heating surface. In this type of air heater the metal parts are stationary and heat is transferred by conduction through the metal wall. A commonly used recuperative air heater is the tubular type, but for lower air and flue-gas pressures the air heater can also be the plate type. Tubular air heaters are normally the counter-flow shell-and-tube type where hot flue gases pass through the inside of the tubes and air flows outside. Baffles are provided to ensure maximum air contact with hot tubes. These heaters, however, consume much metal and occupy a large space.

Regenerative air heater

In a regenerative air heater, the heating surface is swept alternately by the flue gas and the air undergoes alternate heating and cooling cycles and transferring of heat by thermal storage capacity of the heat transfer surfaces. The gas and air counter-current flows move through the rotor separately and continuously. Thus, when half of the heating elements are exposed to flue gases at any instant, the other half is exposed to air. This air heater may include either rotating heat transfer surfaces with stationary air/gas distribution hoods (Ljungstrom design: [Figure 2.22](#)) or rotating air/gas distribution hoods, where the heat transfer surfaces are stationary (Rothemuhle design: [Figure 2.23](#)).



Leakage

- Leakage is a normal phenomenon in all air heaters. While leakage in a new recuperative air heater is essentially zero, leakage occurs with the passage of time and due to accumulation of thermal cycles. Nevertheless, leakage in a recuperative air heater can be restricted to 3% with regular maintenance.
- With a rotary regenerative air heater air leakage is inherent and its design value ranges from 5-10%, which increases further over time as seals wear. When the higher-pressure air side passes to the lower-pressure gas side leakage occurs through the gaps between rotating and stationary parts.

The rate of such leakage is given by the following expression [1]:

$$W_L = KA(2g\Delta P\rho)^{0.5} \quad (2.8)$$

where

W_L = Leakage flow rate, kg/s

K = Discharge coefficient (generally 0.4–1.0)

A = Flow area, m²

g = Acceleration due to gravity (1 kgm/Ns²)

ΔP = Pressure differential across gap, kg/m²

ρ = Density of leaking air, kg/m³

Ambient air infiltration through holes, doors, gaskets, expansion joints, etc., into lower-pressure gas streams is another form of air-heater leakage. The air-heater leakage can be determined by analyzing the %O₂ content of flue gases entering and leaving the air heater (dry basis) and is given by

$$\%Leakage = \frac{\%O_2Leaving - \%O_2Entering}{21 - \%O_2Leaving} * 0.9 * 100 [1] \quad (2.9)$$

Example 2.2

While evaluating the performance of a boiler it is observed from flue-gas analysis that %O₂ content of flue gases entering and leaving a rotary air heater are 5.0% and 7.5%, respectively. Determine the percentage infiltration of ambient air.

Solution: Applying Eq. 2.9, we get

$$\%Leakage = (7.5 - 5.0)/(21.0 - 7.5) * 90 = 16.67\%$$

Hence, the percentage of infiltration of ambient air in the boiler is 16.67%.

(NOTE: In a rotary air heater, whose seals are in good condition, leakage of air should not exceed 10%. The above result reveals that the seals of the air heater are in bad shape and replacement and proper adjustment of seals should be done.)

Insulation of Steam Generators

All hot surfaces at temperatures above 333 K must be insulated for personal protection. The thickness of insulation is chosen to ensure a maximum external surface temperature of 328 K for metal surfaces and 333 K for non-metallic surfaces. In addition to personal protection, insulation is also applied to flue-gas paths and flue gas ducts to ensure the flue-gas temperature entering the chimney remains above its acid dew point to prevent low-temperature gas corrosion. Additionally, thermal insulation is applied to a steam generator to ensure the steam is supplied to its point of application at the correct temperature without any drop. Various heat-traced piping and vessels are also applied with insulation to ensure maximum transfer of heat and minimum heat loss from surfaces.

Thermal insulation materials: There are four types of thermal insulation materials: granular, fibrous, cellular, and reflective. Granular materials, such as calcium silicate, contain air entrained in the matrix. Fibrous materials, such as mineral wool, contain air between fibers. Cellular materials, e.g., cellular glass and foamed plastics, contain small air or gas cells sealed or partly sealed from each other. Reflective insulation materials consist of numerous layers of spaced thin-sheet material of low emissivity, such as aluminum foil, stainless steel foil, etc. In practice, a combination of two or more of the above four types are used.

Properties of thermal insulation materials: Heat from a hot surface is transferred or lost through an insulation material to the surrounding area by radiation and convection. Thermal insulation should limit the heat loss to a minimum. The main requirement of thermal insulation materials is "low thermal conductivity" since heat is transferred through insulation material by conduction (at 373 K, thermal conductivity of mineral wool is 0.047 W/m.K, that of glass fiber is 0.038 W/m.K, of calcium silicate is 0.057 W/m.K and of magnesia is 0.062 W/m.K). Thermal insulation material should have the following properties to ensure satisfactory performance of the material for the life of a plant:

- Material must be suitable for continuous use at maximum operating temperatures without degradation of its physical properties
- It must be non-corrosive to plant and pipe work if wet by rain or leakage of water/steam
- It should not be permanently damaged if contaminated with water
- It should have adequate compressive strength to resist local loads, such as foot traffic, ladders, etc.
- It must have adequate flexural strength and impact resistance to permit transportation and application without breakages
- Material must be non-combustible
- It should not cause any discomfort or health hazards to operating personnel
- It should be tight, free of voids, and well anchored
- Removal and replacement should be required only during maintenance or modification of the plant

Design of thermal insulation

From basic heat transfer theory [13] it is known that under steady state conditions the heat loss from a hot flat surface through thermal insulation to the ambient air is expressed as

$$Q = (T_h - T_c)/(L/k) \quad (2.10)$$

or

$$Q = (T_c - T_a) * f \quad (2.11)$$

Combining Eq. 2.10 and Eq. 2.11 we get

$$Q = (T_h - T_a)/(L/k + 1/f) \quad (2.12)$$

where

Q = Heat loss, W/m^2

T_h = Hot surface temperature, K

T_c = Cold surface temperature, K

T_a = Ambient air temperature, K

L = Thickness of insulation, m

k = Thermal conductivity of insulation material, $W/m.K$

f = Surface coefficient of insulation material, $W/m^2.K$

(combined effect of radiation and convection heat transfer)

For steady flow of heat through composite walls, e.g., wall of a vessel lined with thermal insulation, Eq. 2.12 takes the shape of

$$Q = (T_h - T_a)/(L_1/k_1 + L_2/k_2 + 1/f) \quad (2.14)$$

where

Q = Heat loss, W/m^2

T_h = Hot surface temperature, K

T_a = Ambient air temperature, K

L_1 = Thickness of wall, m

k_1 = Thermal conductivity of material of wall, $W/m.K$

L_2 = Thickness of insulation, m

k_2 = Thermal conductivity of insulation material, $W/m.K$

f = Surface coefficient of insulation material, $W/m^2.K$

The temperature of a cold surface or the surface of an insulation material can then be calculated from Eq. 2.11 as

$$T_c = (Q/f) + T_a \quad (2.13)$$

The expression of cold-surface temperature, T_c , remains the same as Eq. 2.13.

- The heat loss from insulated cylindrical surfaces (e.g., pipes, tubes, small diameter vessels) can be written as

$$Q = (T_h - T_a) / [(r_i * \ln(r_o/r_i) * (1/k)) (r_i/r_o) * 1/f] \quad (2.15)$$

where

Q = Heat loss, W/m²

T_h = Hot surface temperature, K

T_a = Ambient air temperature, K

r_i = Inner radius of insulation, m

r_o = Outer radius of insulation, m

k = Thermal conductivity of insulation material, W/m.K

f = Surface coefficient of insulation material, W/m².K

The temperature of cold surface or the surface of insulation material can then be calculated from

$$T_c = (Q/f) * (r_i/r_o) + T_a \quad (2.16)$$

For steady flow of heat through composite cylindrical surfaces, Eq. 2.15 changes to

$$Q = (T_h - T_a) / [\{ (r_i/k_1) * \ln(r_c/r_i) \} + \{ (r_c/k_2) * \ln(r_o/r_c) \} + (r_i/r_o) * 1/f] \quad (2.17)$$

where

Q = Heat loss, W/m²

T_h = Hot surface temperature, K

T_a = Ambient air temperature, K

r_i = Inner radius of wall, m

r_c = Outer radius of wall/Inner radius of insulation, m

r_o = Outer radius of insulation, m

k_1 = Thermal conductivity of material of wall, W/m.K

k_2 = Thermal conductivity of insulation material, W/m.K

f = Surface coefficient, W/m².K

The expression of the cold-surface temperature remains the same as Eq. 2.16.

Example 2.3

Hot-surface temperature of a 150-mm thick fire brick wall is maintained at 1088 K. If the average thermal conductivity of the wall material is 0.196 W/m.K determine the maximum heat loss through the wall to ensure its cold-surface temperature does not exceed 333 K. For an ambient air temperature of 294 K, what is the surface coefficient of the wall material?

Solution: Applying Eq. 2.10, the maximum heat loss through the brick wall may be determined as

$$Q = (1088 - 333) / (0.15/0.196) = 986.53 \text{ W/m}^2$$

The surface coefficient of the wall material can be found from Eq. 2.11 as

$$f = 986.53 / (333 - 294) = 25.296 \text{ W/m}^2.\text{K}$$

Example 2.4

Steam at a temperature of 623 K is flowing through a pipe with an outer diameter of 100 mm. The pipe is insulated with 20-mm thick mineral wool, the average thermal conductivity of which is 0.055 W/m.K. Determine the outside surface temperature of mineral wool if the ambient air temperature is 300 K. Assume the hot-surface temperature of the mineral wool is the same as that of the flowing steam and the value of the surface coefficient of the insulating material is 30 W/m².K.

Solution: The inside diameter of the insulating material may be assumed to be the same as the outside diameter of the pipe. Hence, $r_i = 50$ mm and $r_o = 70$ mm. Applying Eq. 2.15 we get

$$\begin{aligned} Q &= (623 - 300) / \left[\left\{ \frac{0.05}{0.055} \times \ln\left(\frac{0.07}{0.05}\right) \right\} + \left(\frac{0.05}{0.07} \right) \times \frac{1}{30} \right] \\ &= 323 / (0.9091 \times 0.3365 + 0.0238) \\ &= 979.64 \text{ W/m}^2 \end{aligned}$$

The cold-surface temperature of insulating material is calculated from Eq. 2.16 as

$$\begin{aligned} T_c &= (979.64 / 30 \times 0.05 / 0.07) + 300 \\ &= 323.32 \text{ K} \end{aligned}$$

Example 2.5

One vessel having a carbon-steel wall of thickness 10 mm is carrying saturated steam and water at 441 K. The vessel is insulated with magnesia of thickness 50 mm. If the ambient air temperature is 303 K, determine the heat loss from the vessel.

Given:

- i. thermal conductivity of carbon steel is 52 W/m.K
- ii. thermal conductivity of magnesia is 0.05 W/m.K
- iii. surface coefficient of insulation surface is 10 W/m².K

Solution: Assuming no temperature drop across the metal and applying Eq. 2.14 we get

$$\begin{aligned} Q &= (441 - 303) / (0.01 / 52 + 0.05 / 0.05 + 1 / 10) \\ &= 151.83 \text{ W/m}^2 \end{aligned}$$