



BUDAPEST UNIVERSITY OF TECHNOLOGY
AND ECONOMICS
Faculty of Mechanical Engineering



DEPARTMENT OF ENERGY ENGINEERING

BOILERS

Heat Engines and Boilers

Lecture note for the undergraduate BSc course

Dr. Antal Penninger, Ferenc Lezsovits, János Rohály

BOILERS

1. Theoretical overview

Mankind needs heat energy for making comfortable room conditions, for several industrial processes and for electricity generation. Heat energy gained from combustion more than 90%. (Except for solar, geothermic and nuclear energy.)

Camp fire:

Fire was invented and used in very early age. For a long time camp fire was the only one utilization method, which supplies heat for heating and for cooking or baking.

The simplest way of heat supply when we make fire where the heat demand arises. Examples for this are oven, or fire place at flat heating, or the furnace at industrial metallurgy.

But in most cases it would be very difficult or impossible to make fire where the heat demand arises. That's why an energy transfer medium is needed to deliver generated heat from the fire place to heat demand.

Definition of the boiler:

Boiler means equipment, where fuel burns and heat is transferred from flue gas to energy transfer medium, which is heated up or phase change (boiling) happens.

That equipment which uses heat from other equipment (e.g. gas turbine, or engine) is called heat recovery boiler. This is an exception, because in this boiler there is not combustion.

Heat demands were supplied with ovens and stoves till the mid of 18th century. Boilers were made for the first time only for steam generation. Great development was caused by steam engine invention.

Up to now boilers have gone under long development procedure. There is several type of boiler available on the market. Boilers can be found in large numbers in households. The larger capacity boilers can be found in power stations.

Why we need boilers? Because we need heat.

HEAT DEMANDS

We need heat for several purposes.

These are: - heating of buildings

- hot running water production

- industrial purposes

e.g.: industrial chemistry plants, canning factories, metallurgy factories, etc.

We can characterize heat demands according to:

- temperature demand
- heat power rate
- load changing speed

Characters of heat demands:

	temperature	power rate	load changing speed
Heating	room temp.: 16-24°C	small or medium (e.g. a flat need 10-20 kW in Hungary)	very slow according to weather changing
hot running water	40 - 80 °C	2-20 kW in a flat, 100 kW- 1 MW at central hot running water supply	can be high when lot of people have a bath at the same time
industrial	wide range at every point according to the technological process		
purposes	20°C - 2000°C	1 kW - 100 MW	from very slow to very fast
electricity generation	as high as possible up to 550-600°C (in order to reach high cycle efficiency)	10 MW - 1000 MW	generally slow, but in some cases can be fast

Widely used heat transfer mediums are follows:

Steam:

Temperature: from $t = 100^{\circ}\text{C}$ at ambient pressure
up to $t = 500 - 600^{\circ}\text{C}$ at $p = 100 - 250$ bar

Usage: Steam is the widely used energy transfer medium.

But nowadays for heating is less and less widely used because of less regulability and higher system losses than at water heating.

At industrial systems generally saturated steam is used because large amount of heat (evaporation heat) steam can give at constant temperature (saturated temperature belonging to a constant pressure value), which is ideal for the most of industrial purposes, where generally determinate temperature is needed.

At electricity generation generally superheated steam is used to reach high efficiency, and to avoid water drops in the steam turbine or reciprocating steam engine, because water drops can cause damages in turbines or engines.

Water

Temperature: 90 °C at ambient pressure,
160 °C at higher pressure $p = 8 - 15$ bar
boiling is not allowed at that systems

Usage: Hot water is generally used for heating, usually at low pressure ($t_{\max}=90^{\circ}\text{C}$).
Hot water is also used at industrial plants where the temperature demand is not higher than 90°C (160°C).

Air

Temperature: from 20°C - up to 500°C

Usage: comparatively with steam or water air is rarely used
mainly at air-heating systems
or at industrial plants at drying apparatuses (in wood- and foodstuff industries)

Thermo-oil

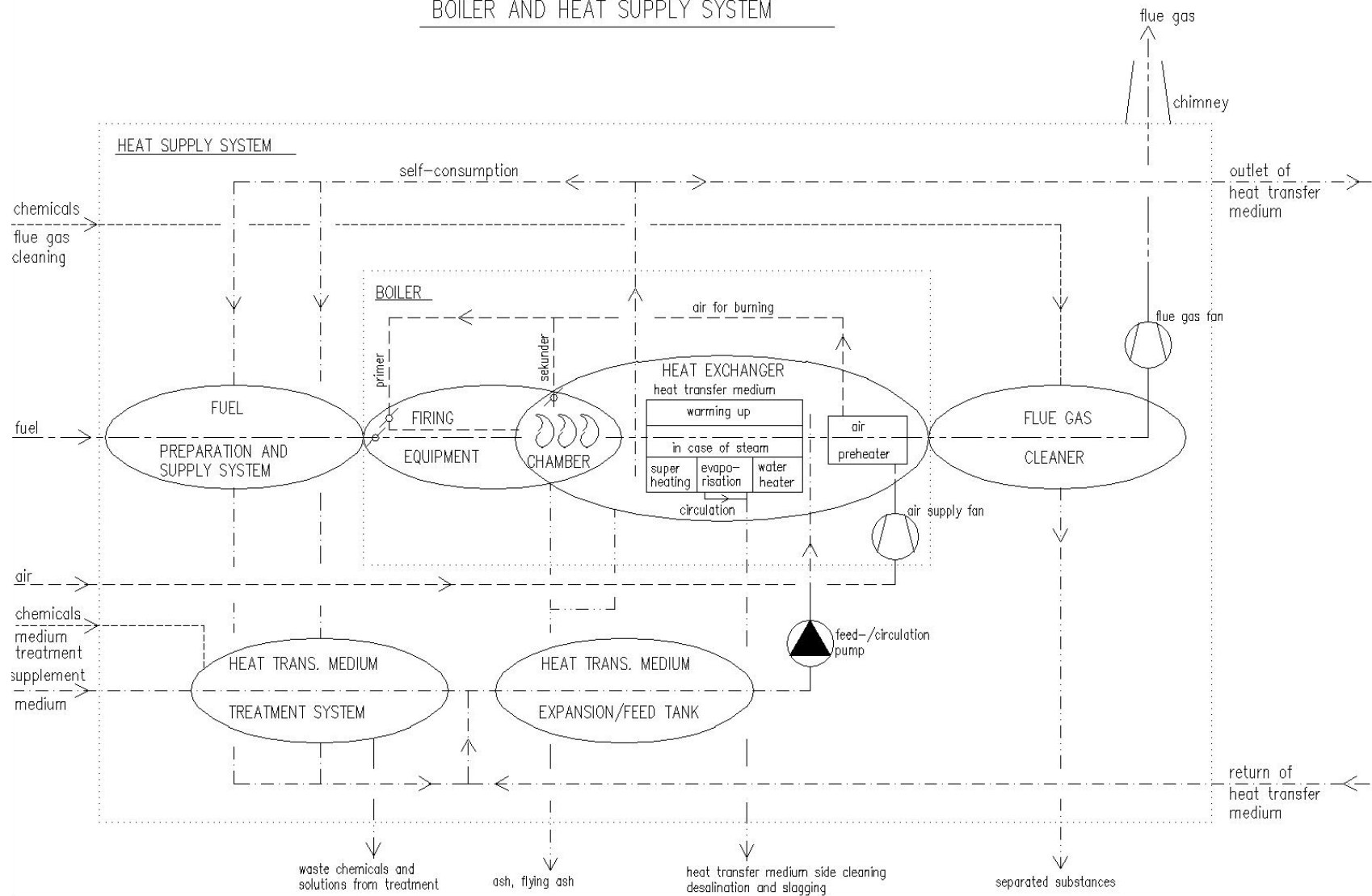
Temperature: $t_{\max} = 300 - 400^{\circ}\text{C}$ at ambient pressure

Usage: Thermo-oil is very rarely used, only in industrial systems, where high temperature is needed at low power rate. It is more economical at low power rate than a steam system with very high pressure. (related saturated steam values
 $p_s = 100$ bar - $t_s = 311^{\circ}\text{C}$, or at critical point $p_c = 220$ bar - $t_c = 374^{\circ}\text{C}$)

When we design a plant, first we have to choose the energy transfer system.

When we have decided what will be the transfer medium, at boiler design or choose a lot of other thing we have to take into account.

BOILER AND HEAT SUPPLY SYSTEM



Elements of boiler and heat supply system

Enclosed Figure shows elements and its connections of the boiler and heat supply system. Inner dotted line shows that scope generally treated as boiler and outer dotted line includes total heat supply system.

Functional parts of boilers

Firing equipment:

Main function is to perform combustion process:

- Tasks:
- Fuel preparation for reaction and feeding into the fire chamber
 - Fuel quantity control according to power demand
 - Feeding of combustion air into the fire chamber and mixing it to fuel
Some times it is divided into primary and secondary air
 - Control of air quantity in order to keep excess air factor at optimal value
 - Starting combustion, ignition
 - Keeping up continuous reaction in order to avoid fluctuation
 - Perform as total and complete combustion as it possible
 - Low level pollutant emission
 - Safe and automatic operation

Firing, or combustion chamber:

Fire chamber is that room where combustion process runs. It is functional part of firing equipment. Firing equipment without combustion chamber is called burner.

It is important that burner has to fit to combustion chamber in order to perform above mentioned tasks. Combustion chamber is generally surrounded by heat exchanger part.

Heat-exchanger part:

Task of heat exchanger part to take heat from flue gas heated up by chemical reaction and transform it to heat transfer medium. Heat can be transferred by means of radiation or convection. At the combustion chamber heat is transferred mainly by radiation, so this part is called radiation heating surface part. At further sections, where flue-gas temperature is lower heat transfer happens mainly by convection, so these heating surfaces are called convective ones. In case there is not phase change at heat transfer medium side at all the parts only heating up of the medium happens. In case of steam generators heating surfaces can be divided into water heater (economizer), evaporation surface and superheater sections. Heat from flue gas also can be used for heating up combustion air. This heating surface is called air preheater.

Auxiliary equipments:*Fuel supply system:*

Task so this system is to prepare fuel for reaction and for feeding into the combustion chamber. Furthermore fuel has to be available when it is needed, so storage or utility connection is also included in this system. In case preparation needs heat it is generally supplied from the boiler, which is called self consumption.

Flue-gas cleaning system

For fulfilling environmental protection rules at a lot of cases flue-gas cleaning system is necessary to apply, in order to remove harmful components from flue-gas. There are several physical and chemical processes which are used depending on harmful components needs to be removed.

Flue-gas removal fan

Its task is to deliver flue gas and giving enough pressure for fluegas side hydraulic resistance. Intensive heat transfer can be gained only with turbulent heat transfer at a certain velocity level, which is caused higher pressure drop. Furthermore fluegas cleaning system also increase fluegas-side resistance.

*Heat transfer medium system:**Circulation or feed pump:*

This pump feeds heat transfer medium into the boiler or keeping up circulation flow in it.

Expansion- / feeding tank

A certain quantity of heat transfer medium has to be available all the time in order to keep up ability of absorption of generated heat by combustion. Otherwise heating surfaces could be overheated and destroyed.

In case of water or thermo-oil systems medium is expanded because of temperature increase. In this case expansion is absorbed by this tank as well, so it is called expansion tank.

Heat transfer medium preparation system

The heat transfer medium is necessary to prepare. Generally it means some kind of filtering. In case of water some softening or desalination process has to be performed. See detailed in feed water preparation section.

BOILER DESIGN

Simple boilers

Steam generation is the process of turning water into steam by application of heat. Essentially, a boiler is a container into which water can be fed and, by the application of heat, evaporated continuously into steam. A steam generator is a complex combination of heat exchangers: economizer, boiler, superheater, reheater, and air preheater. The term *boiler* is often used to mean the whole steam generator in the literature. Boilers are designed to transmit heat from an external combustion source /usually fuel combustion/ to a fluid contained within the boiler itself.

The selection of heat generating equipment is based on the following prime requisites:

1. Quantity of steam or hot water required
2. Pressure, temperature, or steam quality required
3. Anticipation of future needs
4. Location and purpose of the installation
5. Load characteristics

Boiler classification

Boiler may be classified on the basis of any of the following characteristics: Use; pressure; materials; size; tube contents; tube position; firing; heat source; fuel; fluid; circulation; furnace type. There are two general steel boiler classifications, the firetube boiler and the watertube boiler.

Fire tube boilers

The name *firetube* derives from the fact that in boilers of this type all or most of the work is done by heat transfer from hot combustion products flowing inside tubes to the water surrounding them. Combustion takes place in a cylindrical furnace within the shell. Space above the water level serves for steam separation and storage. Firetube boilers are produced in sizes up to about 30 streams per hour. They are restricted to 20 bar steam pressure. The firetube boiler serves its chief use where steam demands are relatively small. It is not used with turbine applications, as it is not well suited for the installation of superheaters.

The firetube boiler is limited in physical size and design adaptability. It has the advantage, however, of large water storage capacity and the ability to dampen the effect of wide and sudden swings /fluctuations/ in steam demand. Because of its large water volume, the time required to arrive at operating pressure from a cold start is considerably longer than for a watertube boiler.

Its ability to handle overfiring is limited and dependent upon boiler type. With increased output, the gas temperature rises rapidly.

The installed cost of a firetube boiler is relatively low and considerably less than that of a corresponding drum type watertube boiler.

It can be seen that high pressures and large diameters would lead extremely thick shell plates. Hence, there is a rather definite economical limit on pressure and capacity that can be reached with firetube boilers.

Design:

The firetube boiler is designed around its furnace and tube passes. Many arrangements have been developed. Tubes have been placed in horizontal, inclined, and vertical positions, with one or more passes. The boiler is designated as through-tube or return-tube according to the direction of hot gas flow. It may have an internal furnace, or it may be externally fired. The internal furnace may be a straight flue type located within the cylindrical shell and surrounded by waterwalls, or it may be a firebox type with the furnace substantially surrounded by a water-cooled heating surface except at the bottom.

Construction:

The shell of the firetube boiler may be cylindrical or oval. The waterline is usually established at a point not less than 50 mm above the top of the upper row of tubes, or crown sheet. The space above the waterline is called the steam space. The tubes are expanded /rolled or pressure expanded/ or welded into the tube sheet. The tubes may be in vertical banks, or they may be staggered. Tubes are usually 50 to 100 mm in diameter, as dictated by draft loss and type of fuel. Often there is a diameter increase.

The boiler may have one, two or occasionally as many as four passes. A pass being a group of tubes through which the combustion products flow essentially in the same direction. Smoke boxes /also known as hoods or chambers/ collect the combustion gases for rerouting through a second pass or for discharge through the stack or breeching.

2-pass design:

These boilers tend to be simplest in design. A variety of arrangements are employed to extract most of the heat from the combustion gas during the relatively short travel time between the burner and stack. Designers have to give attention to the number and arrangement of the second-pass tubes. While waterside inspection and cleaning are easier when tubes are lined up vertically and horizontally, a staggered layout tends to give a more circuitous flow of water around these tubes, promoting increased heat transfer.

3-pass design:

The 3-pass design firetube boiler offers a popular approach. It adds another boiler length to the travel of the hot gas.

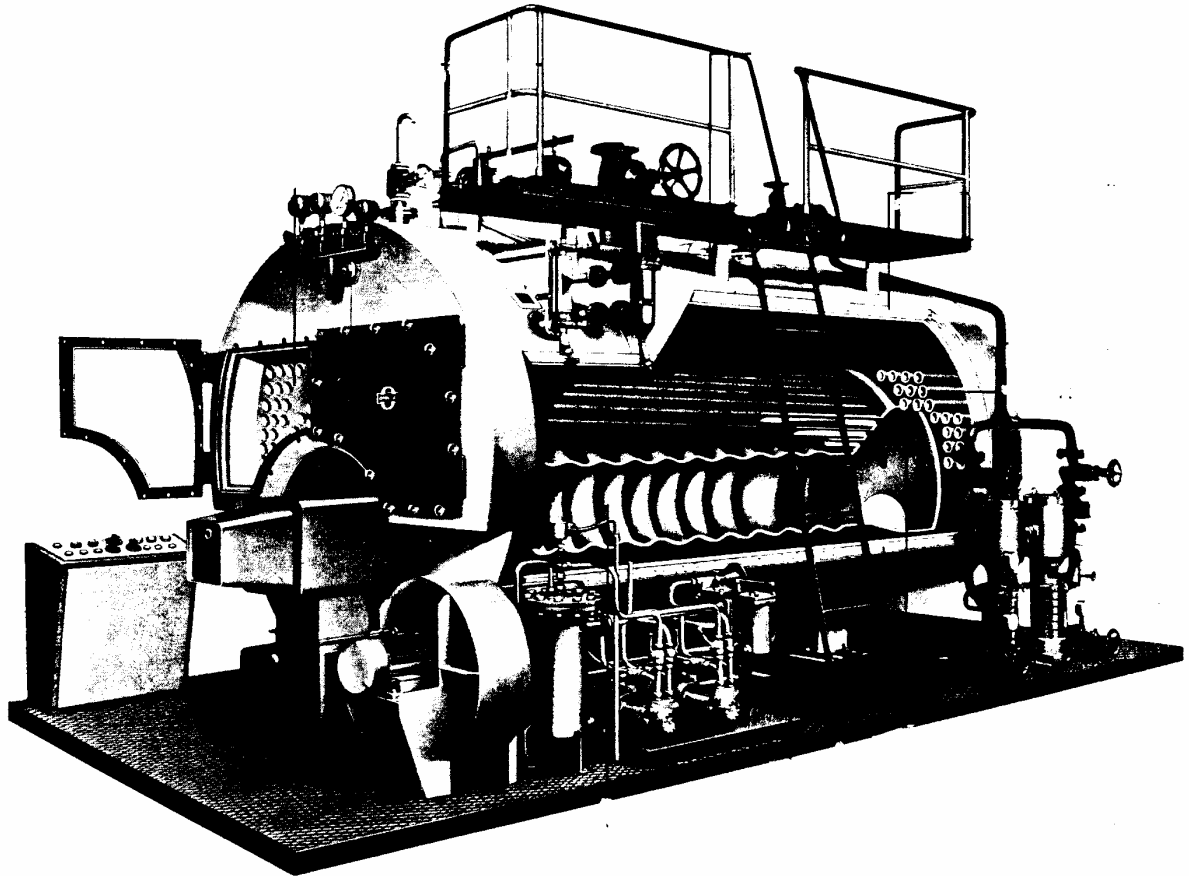


Fig. 2.
Typical up to date firetube boiler with auxiliary equipments

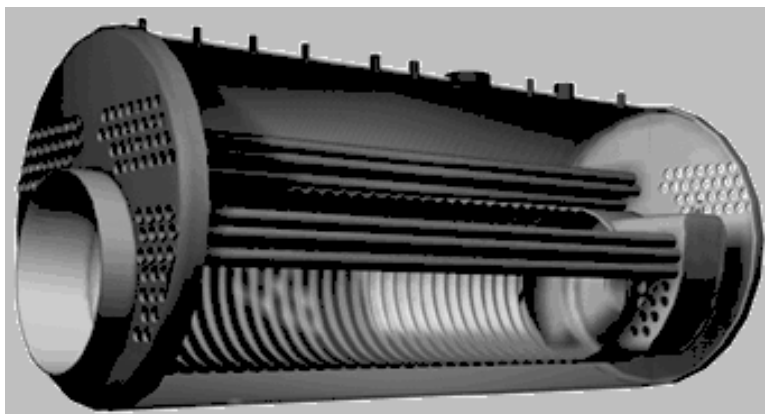


Fig. 3.
Construction and heating surface arrangement of a firetube boiler

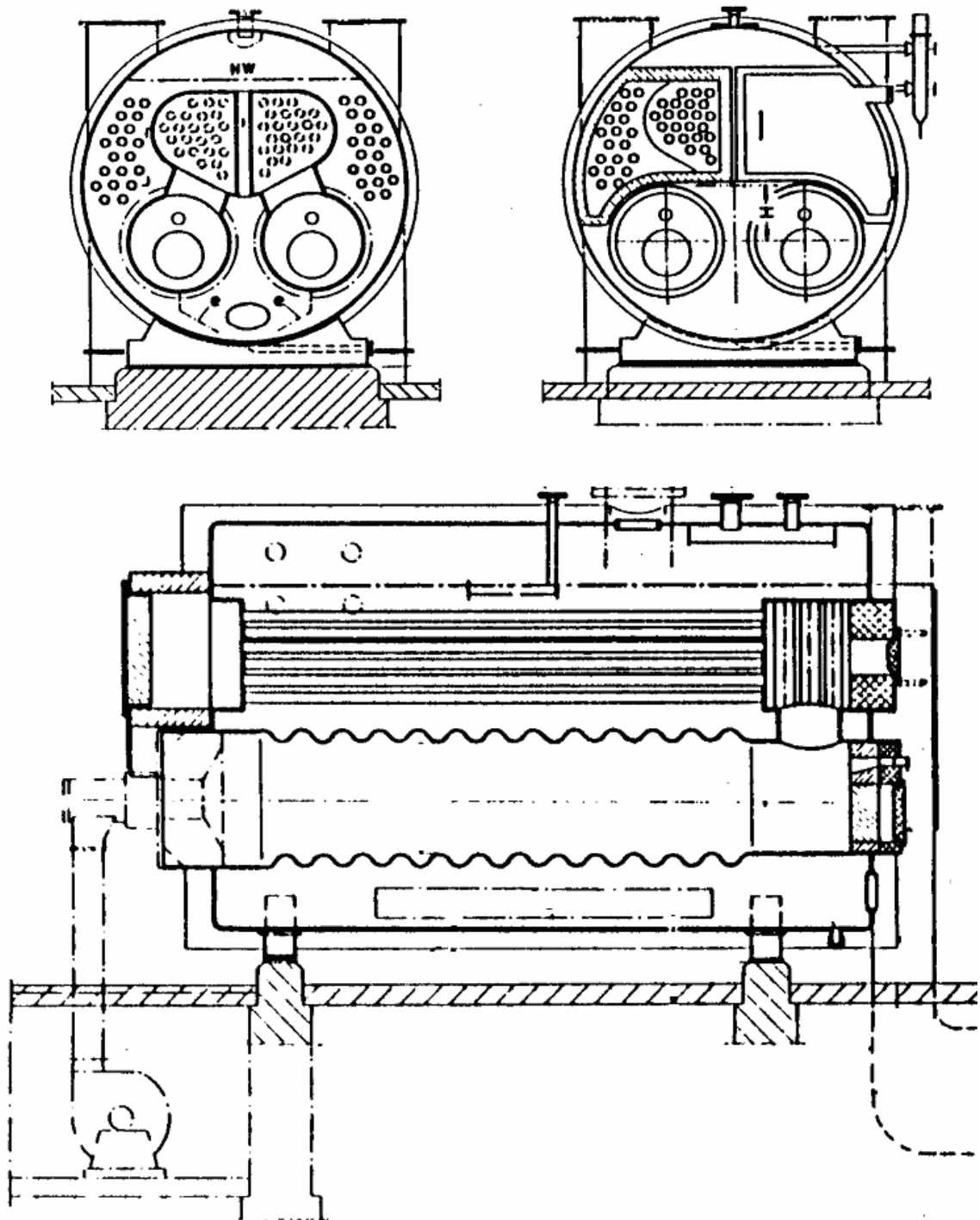


Fig. 4.
Sections of a firetube boiler with two fire chamber.

Watertube boilers:

The firetube boilers become limited as capacity and pressure requirements increase. Larger shell diameters require thicker plates to withstand the pressure and temperature stresses. Temperature differentials in the boiler create high stresses of indeterminate magnitude. These stresses, combined with the effects of precipitates and other deposits, have caused many boiler explosions. Because of its smaller component sizes and ability to accommodate expansion, the steel watertube boiler is much more suitable for large capacities and high pressures through the inherent safety of its design.

The watertube boiler is composed of drums and tubes, the tubes always being external to the drums and serving to interconnect them. The drums are used for storage of water and steam. As they are not required to contain any tubular heating surface, they can be much smaller in diameter than a firetube boiler shell and can therefore be built to resist higher pressures. The tubes contain the entire heating surface. The boiler has natural water circulation except for special positive or controlled /forced/ circulation designs. Watertube boilers may be of the straight-tube or bent-tube type (early developments) or. The various bent-tube boilers, with higher pressure and temperature characteristics, have gradually replaced the straight-tube boiler for high duty service and now monopolized the power generation industry.

The first cost of a small watertube boiler is higher than that of an equivalent firetube boiler. Higher efficiency will eventually offset this higher first cost, however. In utility plant practice, the high first cost of superheaters and economizers is recaptured rapidly.

Recent development of steam generators:

The advent of the water-cooled furnace walls, called water-walls, eventually led to the integration of furnace, economizer, boiler, superheater, reheater and air preheater into the modern steam generator which has higher steam temperatures and pressures which contribute to improved boiler efficiency.

Superheaters and reheaters are heat-absorbing surfaces which raise the temperature of steam above its saturation point. There are many reasons for doing this. First, there is a basic thermodynamic gain in efficiency. Second advantage is that superheating dries the steam before it enters the turbine.

The use of a large number of feedwater heaters (up to seven or eight) means a smaller economizer, and the high pressure means a smaller boiler surface because the latent heat of vaporization decreases rapidly with pressure. Thus a modern high-pressure steam generator requires more superheating and reheating surface and less boiler surface than old units.

Water at 230 to 260 °C from the plant high-pressure feedwater heater enters the economizer and leaves saturated or as a two-phase mixture of low quality. It then enters the steam drum at midpoint. Water from the steam drum flows through insulated downcomers, which are situated outside the furnace, to a header. The header connects to the water tubes that line the furnace walls and act as risers. The water in the tubes receives heat from the combustion gases and boils further. The density differential between the water in the downcomer and that in the water tubes helps circulation. Steam is separated from the bubbling water in the drum and goes to the superheater and the high-pressure section of the turbine. The exhaust from that turbine returns to the reheater, after which it goes to the low-pressure section of the turbine.

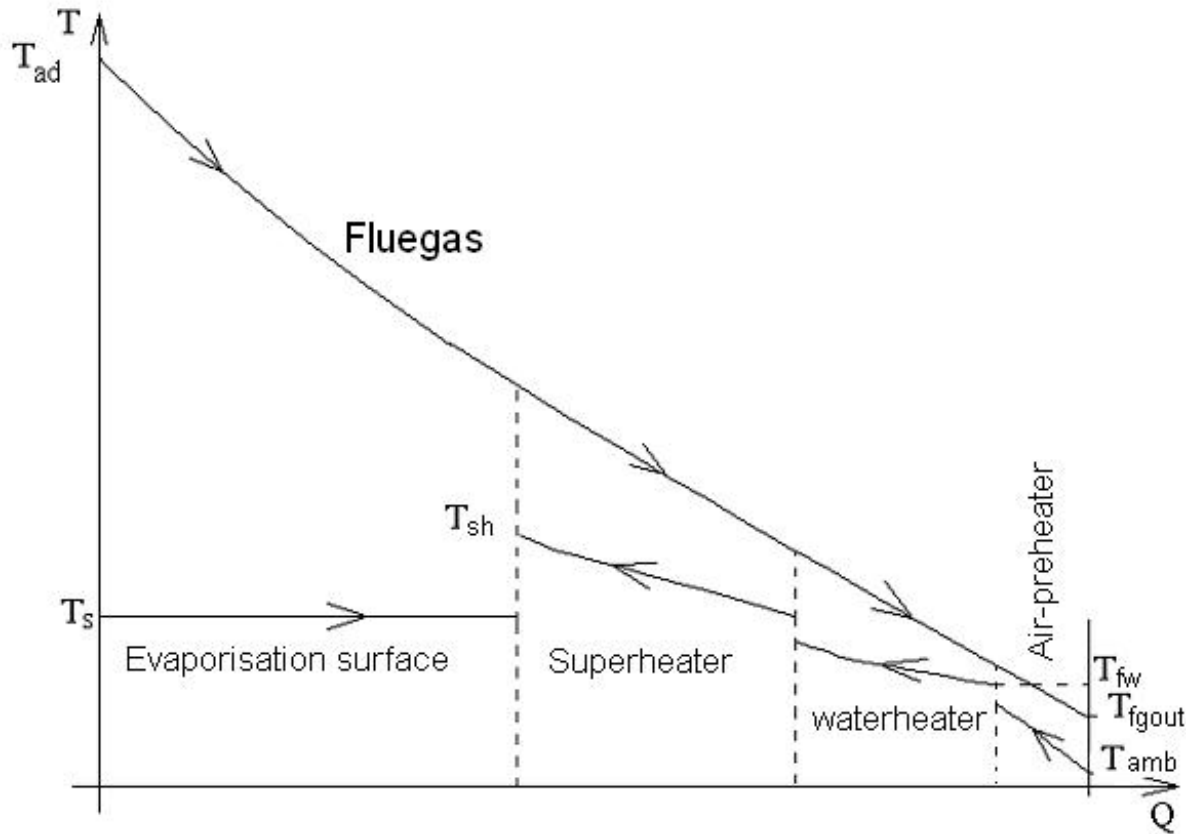


Fig.5.
Q-T diagram of fire tube boilers in general case

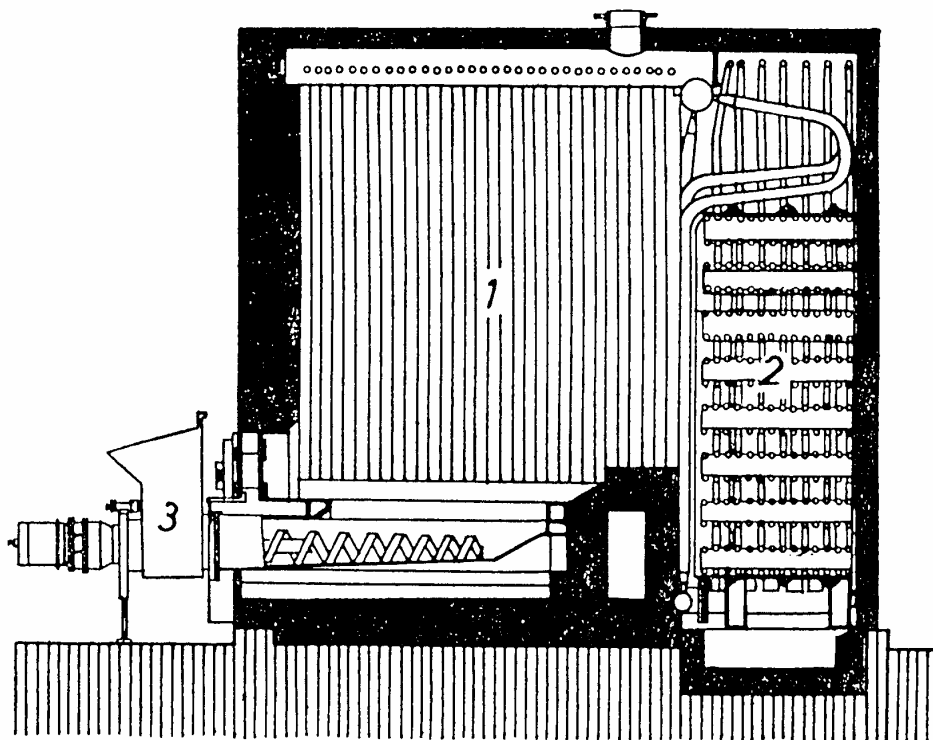


Fig.6.
Cross-section of a waste wood fired boiler with stoker firing.

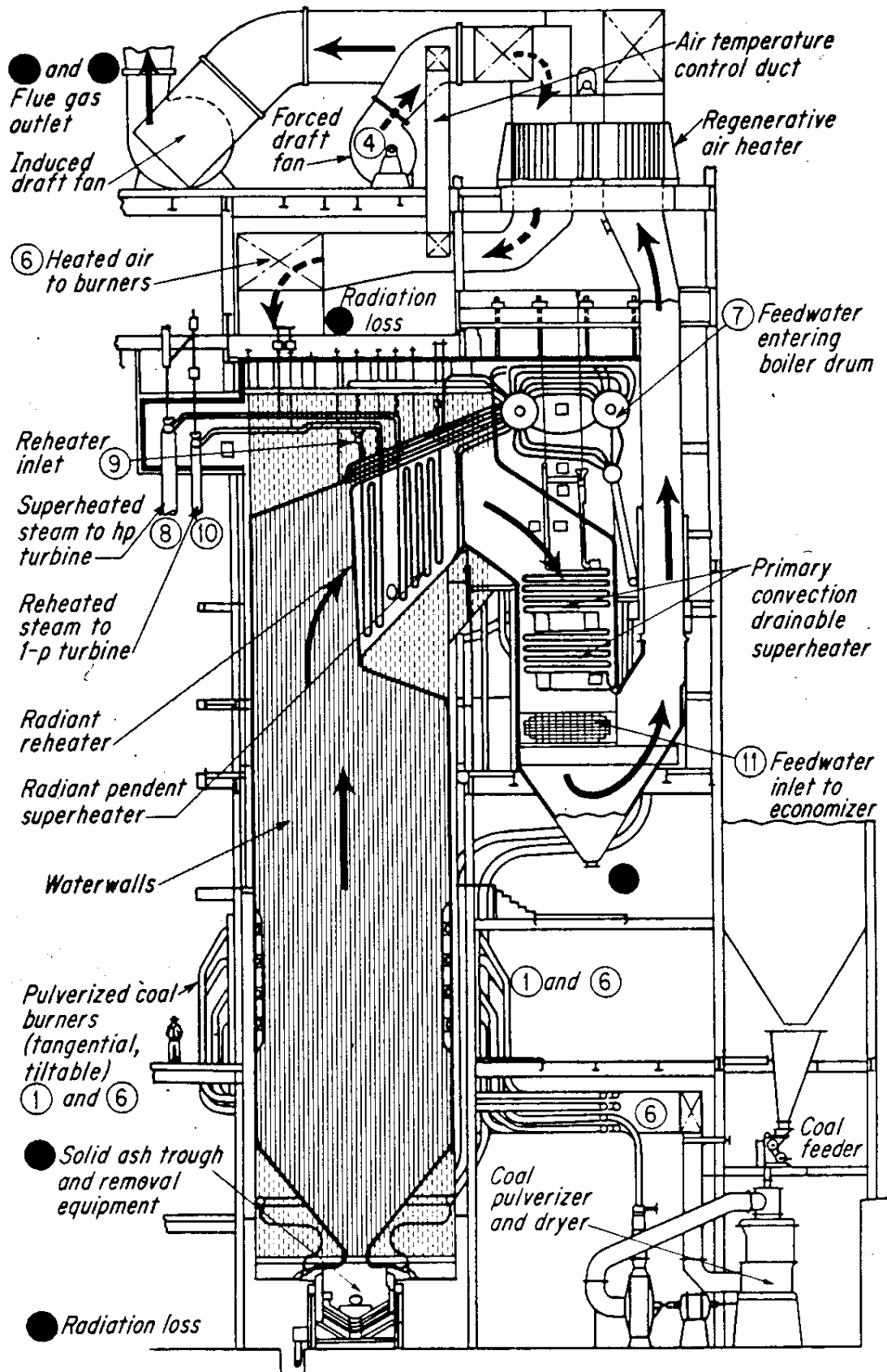


Fig.7.
Pulverised coal fired water tube boiler

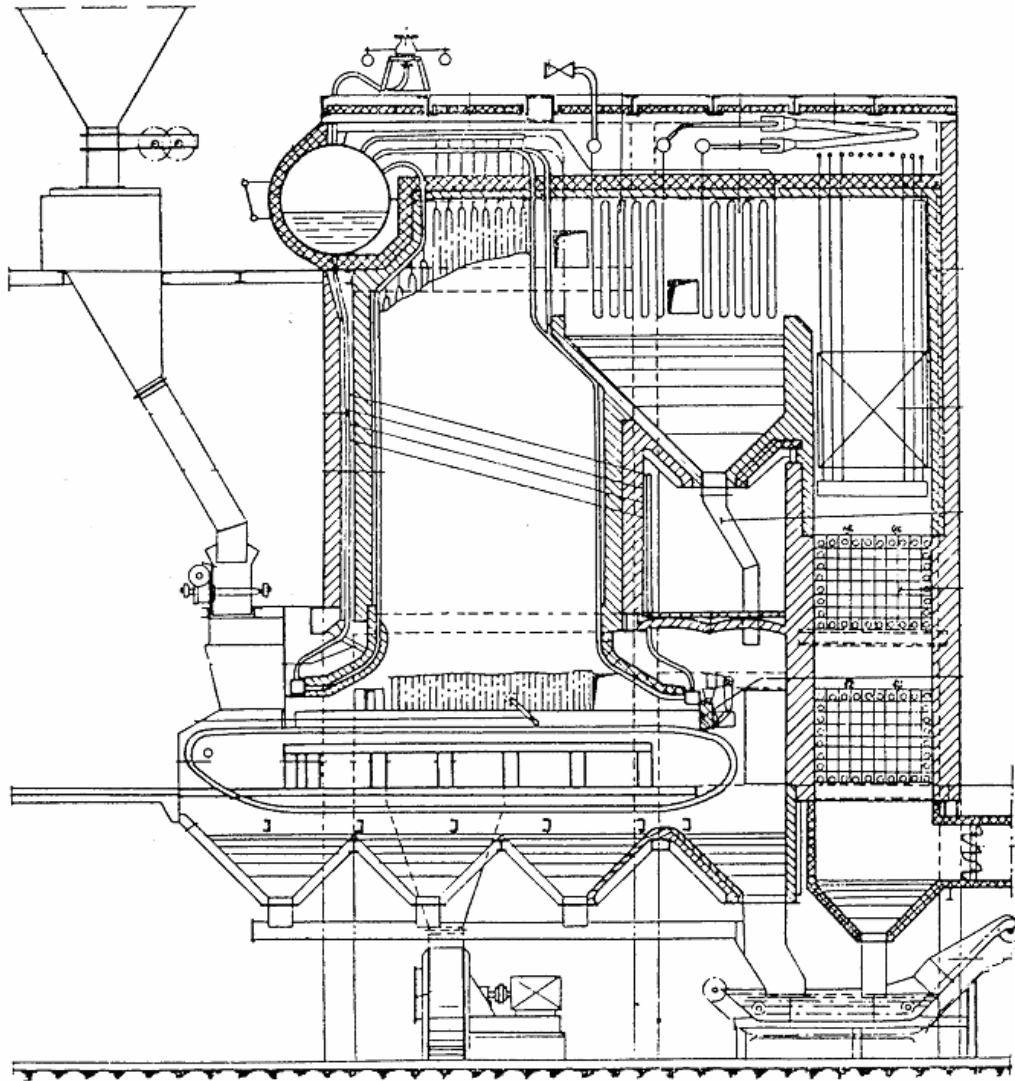


Fig.8.
Block coal fired boiler with traveling grate
which produce 20 t/h steam at 41 bar, 450 °C

Control and regulation of boiler operation

Fuel-air ratio control

It is necessary to keep excess air factor at optimal value. Earlier fix adjustment has been used for different fuel flow. Nowadays this task is performed mainly from signals given by oxygen sensors installed into exhaust fluegas section.

Hot- and warm water boiler operation control

In these type of systems there is not phase change only heating up of heat transfer medium happens. Useful heat can be expressed according to following expression.

$$Q_{\text{useful}} = m_{\text{medium}} \cdot c \cdot (t_{\text{out}} - t_{\text{in}}) \quad [\text{kW}]$$

Heat supply is controlled mainly by two ways:

- In case of quantity control inlet and outlet temperature level constant and mass flow rate of heat transfer medium is modified
- In case of quality control inlet and outlet temperature level modified and mass flow rate of heat transfer medium is constant

- Power level control happens generally from outlet temperature of heat transfer medium

Safety limits:

- Maximal outlet temperature
- Minimal inlet temperature
- Overpressure limitation
- Heat transfer medium absence limitation

Steam generator operation control

In case of steam generator useful heat can be calculated according to following expression:

$$Q_{\text{useful}} = m_{\text{steam}} \cdot (h_{\text{out}} - h_{\text{in}})$$

Heat demand variation results variation of steam mass flow rate. Mass flow rate variation at constant power level and at constant feed water parameters results outlet enthalpy and outlet pressure variation.

- Power level control happens by means of outlet pressure control, because of previously detailed phenomena.
- Outlet temperature control
When only saturated steam is generated outlet temperature is not needs to be controlled. But in case of superheated steam generation superheat temperature is affected by load conditions, and several operation parameters, so it needs to be controlled. Earlier several methods were used for superheat temperature control, but nowadays only spray type superheat control is used.

Spray type superheat control

Of all the direct contact types, the spray attenuator has proved most satisfactory. Water of high purity is injected directly into the superheated steam main or into a mingling chamber. The water is sprayed in through a series of nozzles at the throat of a venturi section, forming a fine mist. It is quickly vaporized and first mixes with and then cools the superheated steam. The design must include a thermal sleeve to protect the piping from thermal shock caused by water droplets striking the hot surface.

The spray attenuator may be used as an interstage or aftercooler desuperheater.

It is relatively inexpensive, and when water conditions permit, it is very satisfactory.

The cooling water may be as much as 5 % to 10 % of the total evaporation by weight. It is quick acting and sensitive as a control means. The steam pressure drop is comparable to the temperature reduction. Spray attenuators may be arranged to provide steam at several different pressures and temperatures.

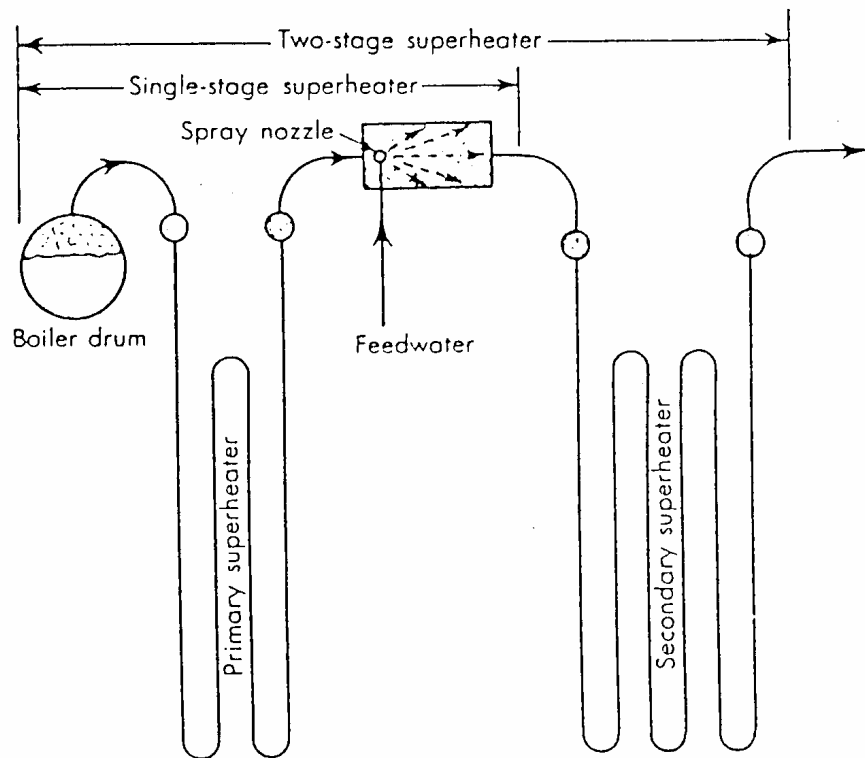


Fig. 9.

Installation of a spray type attenuator

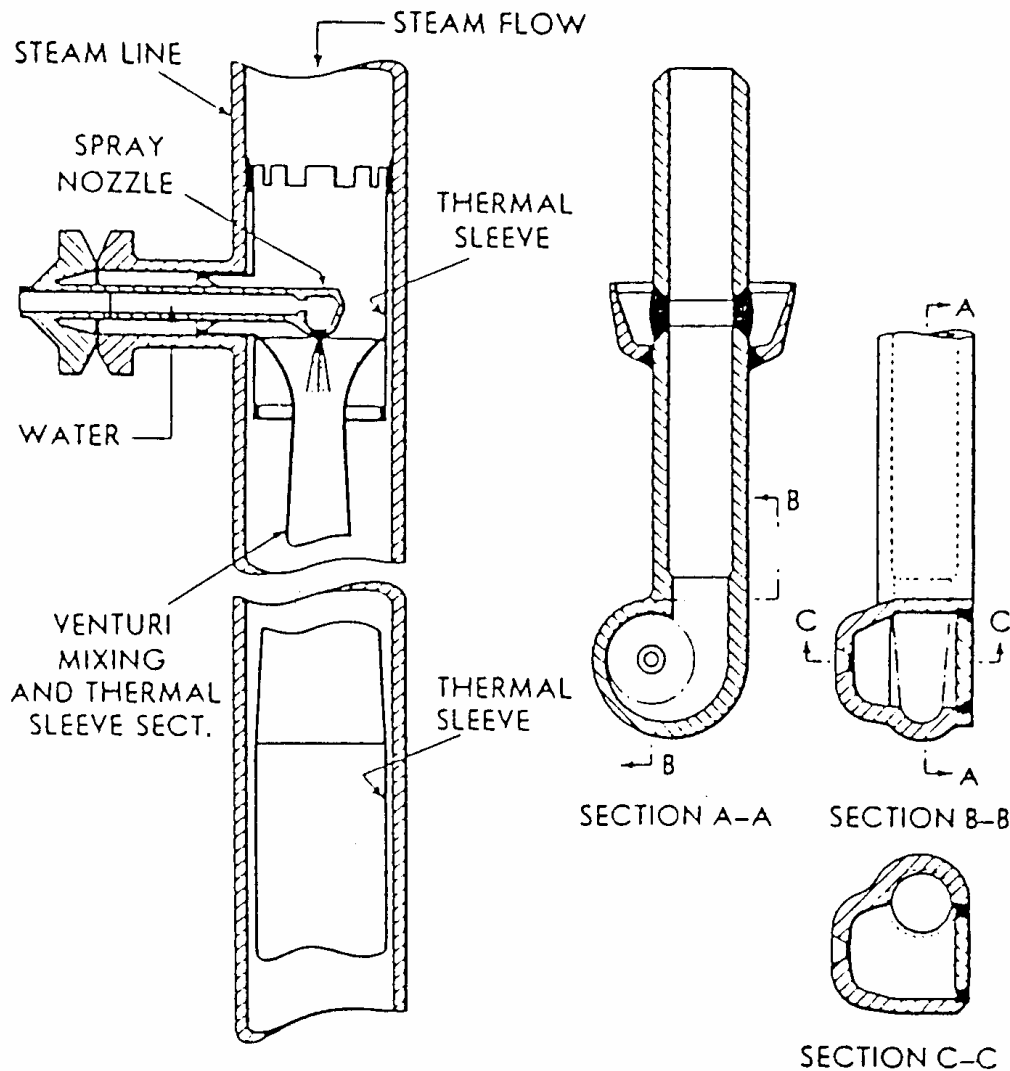


Fig.10.

Typical injector construction of spray type attemperators

Cooler water

For surface type attemperators the cooling medium is feedwater or water taken from the boiler drums. When boiler water is used, the piping is designed to provide natural circulation, thereby assuring a positive flow of water at all times. Any steam incidentally generated passes into the boiler steam space and is discharged behind a separator in the steam and water drum.

For spray attemperators the spray water must be of the highest purity, for entrained solids can leave troublesome deposits on superheater tubes, piping, and turbine blades or cause turbine blade erosion. Improper adjustment or control of the sprays, moreover, may permit the passage of water as well as spray. Condensate or boiler feedwater proves satisfactory if condenser leakage and boiler water make-up do not produce too much contamination. A source of extremely pure water is the condensate from high-pressure heaters. A high-temperature, high-pressure, corrosion-resistant pump is used. The total solids in the spray water should not exceed 2.5 ppm.

- Feed water quantity control
According to actual evaporated water quantity the same amount needs to be feed into the boiler. Generally water level in the drum is used for this task. In case of once-through type boilers it has to be connected with power level control.
- Water quality control
At any type of water treatment a certain amount of salt is delivered to the boiler with feed water. But the steam delivers nearly any salt out from the boiler. So salt is accumulated in boiler water. Over a certain concentration it can cause operation troubles e.g. foaming of water, etc. So salt concentration needs to be held below a certain concentration level. This is solved by letting out some boiler water quantity, which is called slugging or desalination. Fresh feed water will reduce salt concentration.

Safety limits:

- Maximal outlet pressure is generally limited by both electronic and mechanical way with safety valves
- Maximal outlet temperature
- Water level limitation
- Boiler water quality limitation

Boiler operation

In general case boiler operation is potential danger, so it needs continuous attendance to supervise. With fulfilling special requirements boiler can be operated without permanent attendance. But it does not mean without attendance, it means only that supervisor visit needs to be held regularly, and not continuously. Special requirement means that boiler has to be protected automatically from all the possible dangerous operation condition.

Heat balance on boilers and boiler efficiency

Heat balance on boilers means that inlet and outlet heat power is equal at equilibrium state.

$$Q_{in} = Q_{out}$$

In case of instacioner procedures storage capacity of different boiler parts has to be taken into account.

Possible components of input heat:

- Input heat in chemical bound of fuel.

$$Q_{fuel} = \Sigma B \cdot H_i \quad [kW]$$

- Input physical heat of fuel:

$$Q_{fuelphysical} = \Sigma B \cdot c_{pfuel} \cdot (t_{in} - t_{amb}) \quad [kW]$$

- Input heat of hot air:

$$Q_{air} = \Sigma B \cdot \lambda \cdot \mu_{Lo'} \cdot c_{pair} \cdot (t_{in} - t_{amb}) \quad [kW]$$

where:

λ excess air factor

$\mu_{Lo'}$ specific theoretical air demand

(see definition in stoichiometric calculations)

Input heat: $Q_{in} = Q_{fuel} + Q_{fuelphysical} + Q_{in} + Q_{other}$

$$\dot{Q}_{in} = B \cdot (H_i + \lambda \cdot \mu_{Lo'} \cdot c_{pair} \cdot (t_{air} - t_{amb}) + c_{fuel} \cdot (t_{fuel} - t_{amb})) + \dot{Q}_{other} \quad [kW]$$

Some part of input heat is converted to useful heat remaining part become loss.

$$Q_{in} = Q_{useful} + Q_{loss}$$

Efficiency of the boiler means ratio of useful and inlet heat power. When useful power replaced with the difference of inlet and loss power, $Q_{useful} = Q_{in} - Q_{loss}$, two form of boiler efficiency determination can be gained.

$$\eta_{boiler} = \frac{\dot{Q}_{useful}}{\dot{Q}_{in}} = 1 - \frac{\dot{Q}_{loss}}{\dot{Q}_{in}}$$

Determination of boiler efficiency originated from useful heat power is called direct efficiency. Efficiency determination from losses is called indirect boiler efficiency.

Useful heat power can be determined from mass flow rate of heat transfer medium and from inlet and outlet enthalpy:

$$Q_{\text{useful}} = m \cdot (h_{\text{out}} - h_{\text{in}}) \quad [\text{kW}]$$

For determination of direct boiler efficiency fuel and heat transfer medium flow rate needs to be measured in addition to inlet and outlet medium pressure measurement.

Direct efficiency does not give information about reasons of boiler efficiency variation.

Determination of indirect efficiency:

Different types of loss can be separated into two groups: - Firing type losses
- Heat exchanger type losses

Firing type losses:

Firing types losses are originated from not total or not complete combustion of the fuel, which means that unburnt combustible parts remaining after combustion end.

Different forms of firing losses:

ξ_{gas} - unburnt gas ($\text{CO}, \text{C}_x\text{H}_y$)
 ξ_{soot} - soot
 ξ_{coke} - coke
 ξ_{flyash} - combustible part of flying ash
 ξ_{ash} - combustible part of bottom ash

Considering above mentioned losses can be calculated the firing efficiency:

$$\eta_F = 1 - (\xi_{\text{gas}} + \xi_{\text{soot}} + \xi_{\text{coke}} + \xi_{\text{flyash}} + \xi_{\text{ash}})$$

In case of oil and gas firing, when it fulfils environmental protection requirements, firing loss is neglectable. In case of solid fuel firing generally it is worth to take into account. In this case it is necessary to distinguish inlet fuel flow from actually burning, fluegas-developing fuel flow.

$$B_{\text{fg}} = \eta_F \cdot B \quad [\text{kg/s}]$$

Loss quantity can be determined from operational measurement results.

$$Q_{\text{loss}} = \text{massflow} \cdot \text{burnable content} \cdot \text{heating value of burnable part} \quad [\text{kW}]$$

Loss factor is given by the ratio of loss heat power and input power.

$$\xi = Q_{\text{loss}} / Q_{\text{in}}$$

Heat exchanger type losses:

Heat exchanger type loss is the common name of heat produced by combustion, but going another direction than heat transfer medium, which is actually loss.

Different forms of heat exchanger type losses:

ξ_{fg}	– fluegas heat loss
ξ_{rad}	– radiation heat loss
$\xi_{ashheat}$	– ash physical heat loss

Fluegas heat loss:

Heat delivered to the ambient air because flue gas has higher temperature than initial or ambient one. In all of the cases this is the largest loss, which determines mainly the boiler efficiency.

At an up to date boiler it is generally in between $\xi_{fg} = 5 - 10 \%$

At earlier constructions it is in between $\xi_{fg} = 10 - 15 \%$

When fluegas is cooled below water vapor dew-point temperature (which is generally in between 40-60°C) extra heat can be gained. It can cause that overall boiler efficiency can be above 100 % in case when input heat is calculated from lower heating value.

Calculation of fluegas loss factor:

$$\xi_{fg} = Q_{fg} / Q_{in}$$

$$Q_{fg} = m_{fg} \cdot (h_{fgout} - h_{fgamb}) = B \cdot (\mu_{Vo'} + (\lambda - 1) \cdot \mu_{Lo'}) \cdot c_{pfg} \cdot (t_{fgout} - t_{amb}) \quad [kW]$$

$$\xi_{fg} = \frac{B \cdot (\mu_{Vo'} + (\lambda - 1) \cdot \mu_{Lo'}) \cdot c_{pfg} \cdot (t_{fgout} - t_{amb})}{B \cdot (H_i + \lambda \cdot \mu_{Lo'} \cdot c_{pair} \cdot (t_{airin} - t_{amb}) + c_{fuel} \cdot (t_{fuel} - t_{amb}))} =$$

$$= \frac{(\mu_{Vo'} + (\lambda - 1) \cdot \mu_{Lo'}) \cdot c_{pfg} \cdot (t_{fgout} - t_{amb})}{H_i + \lambda \cdot \mu_{Lo'} \cdot c_{pair} \cdot (t_{airin} - t_{amb}) + c_{fuel} \cdot (t_{fuel} - t_{amb})}$$

As it can be seen from this formula, there is not needed to know fuel flow. There are only two main parameter which is determined by operation conditions, which are follows:

λ	- excess air factor
t_{fgout}	- outlet temperature of fluegas

Fuel and air inlet temperature also can be changed, but generally these values are quite constant at a certain installation. Other parameters depend only on fuel composition.

Outlet fluegas temperature can be measured easily by thermometer. Excess air factor can be calculated from results of fluegas component measurements (O_2 , or CO_2 (RO_2) see stoichiometric calculations). In this way the largest loss component can be determined in a not too expensive and quite fast way.

Radiation type loss:

Radiation type loss is called the heat transferred to the ambient air by outer surface of the boiler. The name originates from ancient boiler construction, where brick works actually radiated heat to the ambient. Nowadays this heat is transferred mainly by convection, but the name remains the same. Actual value can be calculated according to heat transfer rules considering actual insulation solution. This loss factor varies in between $\xi_{\text{rad}} = 0.5 - 1.0 \%$ referring to maximal load. But the heat loss power is independent from load level, it is constant. ($Q_{\text{rad}} = \text{const.}$). This cause that loss factor is in inverse proportionality with load. (1% loss at nominal load increases up to 5% at 20% part load)

Ash physical heat loss:

It is only in case of solid fuel firing, where bottom ash removed from fire chamber in hot condition. For loss factor determination bottom ash quantity and temperature needs to be measured.

Comparison of direct and indirect boiler efficiency

Both methods shall give the same value. But in real some difference can be experienced because of measurement inaccuracies.

Generally determination by indirect method is simpler, because fuel and heat transfer medium measurement is not needed.

Furthermore indirect method gives information on waste heat distribution and can be information base of efficiency increment. Direct method cannot be used for this purpose, but it can be good control of indirect method.

CONTROL CALCULATION

There are several types of control calculation:

- heat transfer calculation
- fluid flow and pressure drop calculation
- mechanical stress calculation
- heat stress calculation

Generally you have to make control calculation:

- at the maximum load (full load) and
- at the minimum load of the boiler.

But you always have to take into account if there is other critical situation at operation. You have to control every possibility of critical situations to avoid crash and industrial accident. It is necessary to take into account every situation during the work of the boiler, from start till stop.

The exact calculation you can make if you know the exact construction.

At the design you can make approximate calculations to determine sizes, but at a design process a lot of things have to be taken into consideration.

- heat transfer
- flue gas flow
- water\steam flow
- pressure drop
- mechanical stress
- heat stress
- soot/ash coating
- etc.

After the determination of the construction you have to make control calculations. If the result of the control calculation is not acceptable, you can change the construction and you can make the control calculations again.

HEAT BALANCE ON COMBUSTION CHAMBER

Walls made from tubes surrounded the fire chamber in the boiler to absorb heat.

Different types of construction (development) :

- a; bare tubes tangent to the refractory
- b; bare tubes embedded in the refractory
- c; studded tubes
- d; membrane wall

The membranes act as FINS to increase the heat tr. as well as to afford a continuous rigid and pressure tight construction for the furnace.

Nowadays membrane wall is generally used.

Types of heat received by water walls:

a; radiation (most heat is received by radiation)

Luminous flame cause radiation:

- emit all wavelengths and strong visible radiation
- particulates such as soot particles during combustion (combustion process and fuel type have influence on soot formation; eg.coal and oil + fuel rich burning zones)
- particular triatomic components of flue gas CO_2 , H_2O , SO_2 , are selective radiators: emit and absorb radiation in certain wave lengths.

Radiation intensity of black body according to Planck law with Wien type simplicity.

$$I_{\lambda 0} = \frac{c_1}{\lambda^5} \cdot e^{-\frac{c_2}{\lambda T}}$$

Radiation energy density at a certain temperature can be gained with integration from zero to endless this function. This is called Stefan-Boltzman rule.

$$E_s = \int_0^{\infty} I_{\lambda} d\lambda = \sigma \cdot T^4 \quad [\text{W/m}^2]$$

ε emissivity factor is used for feature of real body comparing with black body.

$$\varepsilon = I_{\text{real}} / I_{\lambda 0}$$

When ε is constant, independent from wave length it is called grey body radiation. This is typical solid particulate radiation, which can be experienced nearly every flame. This gives yellow and red color of flame, caused by soot, coke or solid particulate combustion.

$$\varepsilon = \text{constant} = E_{\text{real}} / E_{\text{black}}$$

When ε is the function of wavelength it is called color body radiation. This is the case of gas radiation, mainly three atomic gas components radiation significant. Monoatomic and diatomic gases are poor radiators.

$$\varepsilon = f(\lambda)$$

Emitted heat by radiation (Q_r) depends on

- gas temperature; (\overline{T}^4)
- partial pressures of the individual constituent radiating gases; $X_i[\text{v.}\%] \sim P_i$
- shape and size of the gases
- proximity of the absorbing body
- temperature of that body; (\overline{T}_w^4)

$$Q_r = \varepsilon_{fw} \cdot \sigma \cdot A \cdot (\overline{T}_f^4 - \overline{T}_w^4) [kW]$$

ε_{fw} : view factors between the flame and the water walls

$$\varepsilon_{fw} = \frac{1}{\frac{1}{\varepsilon_f} + \frac{1}{\varepsilon_w} - 1}$$

Where: ε_w : emissivity of water walls $\varepsilon_w = 0,6$

ε_f : emissivity of flame $\varepsilon_f = 0,35 \dots 0,85$

$\sigma = 5.6787 \cdot 10^{-8} \text{ W / m}^2 \text{ K}$: Stefan-Boltzman constant

A: effective water wall surfaces subject to radiation; [m²]

$$A = \overline{\Psi}_k \cdot A_{\text{furnace}}$$

where:

\bar{Y}_k - surface effectiveness

$$\bar{Y}_k = \frac{\sum A_i \cdot Y_i}{\sum A_i}$$

$$Y_i = \frac{\text{absorbed heat by the water walls}}{\text{emitted heat by the flame}} = k \cdot x$$

where : k - radiation coefficient

$$(k = 1 \text{ in case of membrane wall})$$

x - dirtiness factor of the water walls

in case of gas fuel: $\xi = 0.65-0.7$

in case of oil fuel: $\xi = 0.55-0.6$

in case of coal fuel: $\xi = 0.45-0.55$

Flying ash and soot can accumulate on the surfaces and reduces the heat transfer coefficient

T_f : average gas temperature in the furnace

T_w : average water wall surface the temperature surrounding the flame.

T_o : adiabatic flame temperature ($\sim 2000^\circ\text{C}$ in coal fueled furnace)

$$\bar{T}_w \gg \bar{T}_f \text{ and } \bar{T}_f = \sqrt{T_o \cdot T_{fgout}}$$

Useful, practical assumptions: where: $T_{fgout} \approx 850^\circ\text{C}$ flue gas temperature at the outlet of the furnace

Heat balance equation of the combustion chamber: $Q_{in} = Q_r + Q_{fgout}$

Q_{in} : heat flow into the combustion chamber

(fuel, preheated air, ... e.g. steam in steam assisted oil atomizers)

Q_r : heat transfer to the water walls by radiation

Q_{fgout} : outflowing heat towards superheater economizer and air preheater

Heat flow into the combustion chamber:

$$Q_{in} = B \cdot [H_i + \lambda \cdot \mu_{Lo} \cdot c_{pair} \cdot t_{hotair} + c_{fuel} \cdot t_{fuel}]$$

Heat flow out from the combustion chamber

$$Q_{fgout} = B \cdot \mu_v \cdot c_{pfg} \cdot t_{fgout}$$

where: - B : mass flow rate of fuel [kg/s]

- μ_v : specific flue gas amount, [kg fluegas\ kg fuel]

- c_{pfg} : mean specific heat of flue gas [kJ/kg K]

- t_{fgout} outlet flue gas temperature [$^\circ\text{C}$] or ($T_{fg} - 273,15$) [K]

The heat balance on the combustion chamber gives:

$$Q_r = Q_{in} - Q_{fgout}$$

Now we can substitute the outflowing flue gas heat in ADIBATIC burning, where there is no heat flow to the surroundings ($Q_r=0$ [kW])

In adiabatic case...

$$Q_{in} = Q_{fgout}$$

$$Q_{fgout/adiabatic} = B \cdot \mu_v \cdot c_{pfg} \cdot t_0$$

where: t_0 is the adiabatic flame temperature, [°C] substituting this to the equation at the top

$$\varepsilon_{fw} \cdot \sigma \cdot A \cdot (\bar{T}_f^4 - \bar{T}_w^4) = B \cdot \mu_v \cdot c_{pfg} \cdot (T_0 - T_{fgout})$$

Now, applying the assumptions on temperatures

$$\bar{T}_w \gg \bar{T}_f \text{ and } \bar{T}_f = \sqrt{T_0 \cdot T_{fgout}}$$

$$\varepsilon_{fw} \cdot \sigma \cdot A \cdot \bar{T}_0^2 \cdot \bar{T}_{fgout}^2 = B \cdot \mu_v \cdot c_{pfg} \cdot (T_0 - T_{fgout}) \quad /: T_0^4$$

$$\left(\frac{T_{fgout}}{T_0} \right)^2 = \frac{1}{\varepsilon_{fw}} \cdot \frac{B \cdot \mu_v \cdot c_{pfg}}{\sigma \cdot A \cdot \bar{T}_0^3} \cdot \left(1 - \frac{T_{fgout}}{T_0} \right)$$

$$\text{where: } \frac{B \cdot \mu_v \cdot c_{pfg}}{\sigma \cdot A \cdot \bar{T}_0^3} = \text{const.} = Ko$$

Ko - Konakov number ~ conv. heat flow/rad. heat flow

$$\text{Introducing: } \Theta_{out} = \frac{T_{fgout}}{T_0}$$

$$\Theta^2 + \frac{Ko}{\varepsilon_{fw}} \cdot \Theta - \frac{Ko}{\varepsilon_{fw}} = 0$$

$$\Theta_{1,2} = \frac{-\frac{Ko}{\varepsilon_{fw}} \pm \sqrt{\left(\frac{Ko}{\varepsilon_{fw}}\right)^2 + 4 \cdot \frac{Ko}{\varepsilon_{fw}}}}{2}$$

$$\Theta_{out} = \frac{1}{2} \cdot \frac{Ko}{\varepsilon_{fw}} \cdot \left[\sqrt{1 + 4 \cdot \frac{\varepsilon_{fw}}{Ko}} - 1 \right]$$

Or according to the another semi empirical method which was developed by "Gurvics":

$$T_1 = \frac{T_0}{E \cdot \left(\frac{\varepsilon}{Bo}\right)^{0.6} + 1}$$

where: E - flame position coefficient
 ε - flame emissivity
 Bo - Boltzmann number

$$Bo = \frac{Q_s}{\sigma \cdot \chi \cdot A \cdot T_0^3 \cdot (T_0 - T_1)}$$

where: Q_s - heat transferred by radiation in the fire chamber
 σ - Stefan-Boiltzmann constant
 χ - Coefficient of surface utilization
 A - Whole surface of the fire chamber

CONVECTIVE HEATING SURFACES

Convective superheater, economiser and air preheater and control calculations of these constructions

We call “Convective Heating Surfaces” surfaces which are built in the boiler after the burning chamber until the boiler exhaust.

These are:

- superheater
- evaporator
- economizer
- air preheater

Each heating surface can not be found in every steam generator.

In industrial systems where saturated steam needed, you don't find superheaters.

Where the pressure is low ($p < 12$ bar) generally economizer and air preheater can't be find.

Superheaters are built when superheated steam needed.

Mainly at electricity generation only superheated steam is used, to reach high efficiency and avoid droplets in the steamturbine.

The higher the steam pressure, the higher the saturated steam temperature. So you can cool down flue gas close to the dew point ($t = 150-200$ C), to reach high efficiency, you have to built in economizer and/or air preheater, because in that case this temperature is under saturated temperature.

THE ORDER OF THE HEATING SURFACES

According to the second law of thermodynamic heat doesn't go from the lower temperature level to a higher one. That's why the flue gas has to be at higher temperature than the heat absorption fluid.

Temperature level of different types of heat absorption fluid:

-superheated steam	$t = \text{from } t_s(200-370^\circ\text{C}) \text{ to } 300-500^\circ\text{C}$
-saturated steam	$t_s = 200-370^\circ\text{C}$
-feedwater	$t = \text{from } 105^\circ\text{C} \text{ to } t_s(200-370^\circ\text{C})$
-air	$t = 15-200^\circ\text{C}$

You can see that the superheated steam is at the highest and the air is at the lowest temperature level. Evaporator heating surfaces generally to the fire chamber are built, but at lower pressure level it can be found at convective areas too. Going along by the flue gas way in the boiler you can find heating surfaces in the following sequence:

	Temp.range	Flue gas temp. (after the fire Chamber $t \sim 1000^\circ\text{C}$)
-superheaters	$300 \rightarrow 500^\circ\text{C}$	$1000^\circ\text{C} \rightarrow 600^\circ\text{C}$
-economizer	$105 \rightarrow 300^\circ\text{C}$	$600^\circ\text{C} \rightarrow 300^\circ\text{C}$
-airpreheater	$20 \rightarrow 200^\circ\text{C}$	$300^\circ\text{C} \rightarrow 150^\circ\text{C}$

These temperatures are only examples. They can be at various levels at different types of boiler, but principals are the same.

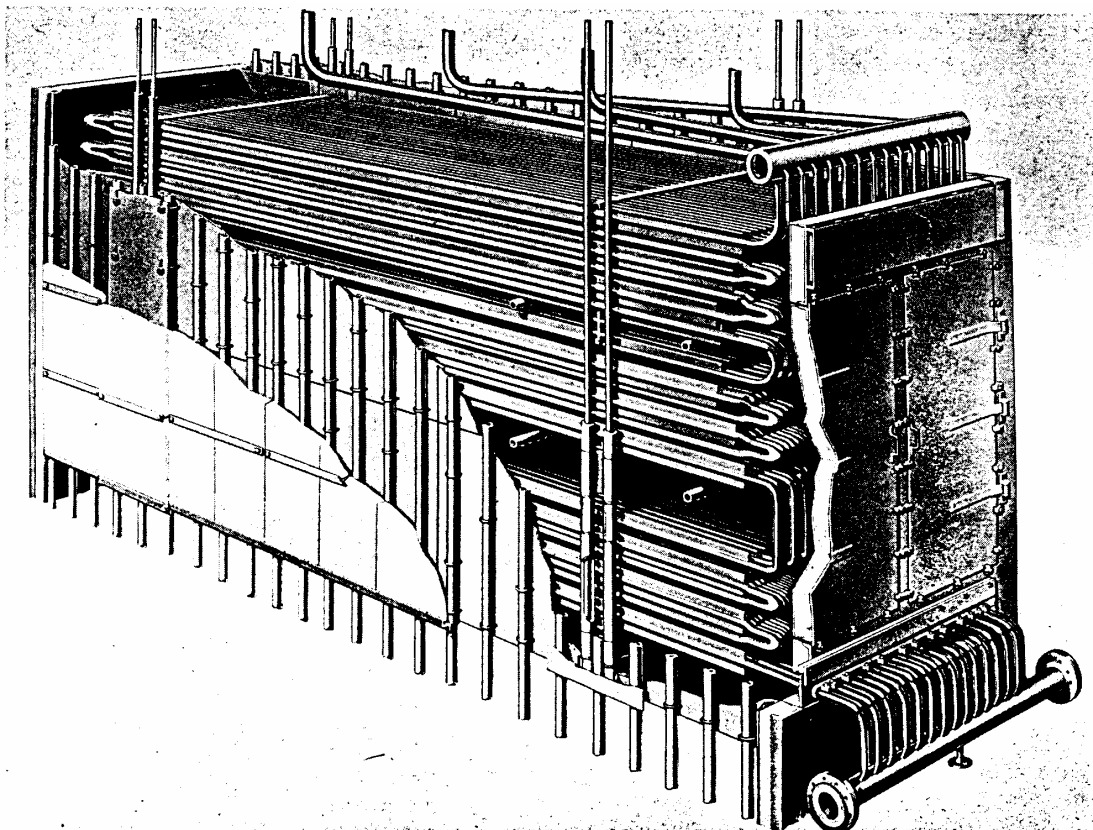


Fig.11.
Continuous tube type economizer

Heat transfer calculation

- Input data:
- sizes of the heating surface
 - construction of the heating surface
 - built in materials
 - flue gas - inlet temperature
 - inlet pressure
 - mass flow rate
 - heat absorp.fluid - inlet temperature
 - (water/steam/air) - inlet pressure
 - mass flow rate

We don't know the outlet temperatures.

The calculation process is an iteration process. First you have to estimate the outlet temperature of the flue gas and the heat absorption fluid. Then you can calculate the average temperatures.

$$\text{flue gas: } \quad \bar{g}_{fg} = \frac{g_{fgin} + g_{fgout}}{2}$$

$$\text{heat abs. fluid } \quad \bar{t}_w = \frac{t_{win} + t_{wout}}{2}$$

Knowing the average temperatures you can determine the characteristic features belonging to the temperature and pressure both of the flue gas and the heat abs. fluid, which is needed to the calculation.

Firing happens generally at ambient pressure in boilers. Flue gas properties can calculate according to the ideal gas rules. $\left(\frac{p}{\rho} = \frac{m}{M} \cdot R_u \cdot T \right)$

These are:

- density ρ
- thermal conductivity λ
- Prandtl number Pr
- specific heat c_p
- kinematic viscosity ν
- etc.

Then according to the construction you can determine the heat transfer coefficients.

Heat transfer coefficient calculation.

There are several semi empirical equation to determine heat transfer coefficient.

For this dimensionless numbers are used.

Most commonly used dimensionless numbers:

- Nusselt number: $Nu = \frac{\alpha \cdot L}{\lambda}$

- Reynolds number $Re = \frac{w \cdot L}{\nu}$

- Prandtl number $Pr = \frac{\nu}{a}$

Where: α - heat transfer coefficient

L - specific size

λ - thermal conductivity

w - fluid flow velocity

ν - kinematic viscosity

a - temperature conductivity $a = \frac{\lambda}{\rho \cdot c_p}$

where: ρ - density of the fluid

c_p - specific heat at constant pressure

In the boiler fluid flow is generally turbulent, at both side, water and flue gas.

Fluid flow inside the tube:

$$Nu = 0.021 \cdot Re^{0.8} \cdot Pr^{0.43} \cdot \left(\frac{Pr}{Pr_w} \right)^{0.25} \cdot \varepsilon_l$$

valid for: $10^4 < Re < 5 \cdot 10^5$ and $0.6 < Pr < 2500$

where: - L specific size

- t standard temperature

- Pr_w

- ε_l

- inside tube diameter

- fluid average temperature

- Prandtl number at the wall temperature

- coefficient against long/diameter ratio

$$\frac{l}{d} = 1 \Rightarrow \varepsilon_l = 1.5$$

$$\frac{l}{d} = 50 \Rightarrow \varepsilon_l = 1.0$$

Fluid flow around (between) tubes:

Tubes in series arrangement:

$$Nu = 0.23 \cdot Re^{0.65} \cdot Pr^{0.33} \cdot \left(\frac{Pr}{Pr_w} \right)^{0.25} \cdot \varepsilon_\psi$$

Tubes in staggered (chequerred) arrangement:

$$Nu = 0.41 \cdot Re^{0.6} \cdot Pr^{0.33} \cdot \left(\frac{Pr}{Pr_w} \right)^{0.25} \cdot \varepsilon_\psi$$

valid for: $2 \cdot 10^2 < Re < 2 \cdot 10^5$

where: - w specific velocity - fluid flow velocity in the narrowest cross-section

- ε_ψ

- coefficient according to the angle including between the fluid flow and tubes

$$\alpha = 90^\circ - \varepsilon_\psi = 1.0$$

$$\alpha = 10^\circ - \varepsilon_\psi = 0.56$$

Heat transfer coefficient in case of water boiling:

$$\alpha = 2.8 \cdot p^{0.176} \cdot q^{0.7} \quad [\text{W} / \text{m}^2 \cdot \text{K}] \quad \text{valid at: } 0.2 \text{ bar} < p < 98 \text{ bar}$$

$$\alpha = 1.27 \cdot q^{0.75} \cdot e^{\frac{p}{62}} \quad [\text{W} / \text{m}^2 \cdot \text{K}] \quad \text{valid at: } 6.0 \text{ bar} < p < 173 \text{ bar}$$

where - p - saturated pressure [bar]

- q - heat flux [W/m²]

These are only examples.

According to the surface arrangement you can find several cases in the literature.

Heat transfer coefficient has different value range at different types of fluid:

In case of: water boiling: $5000 < \alpha < 20000 \text{ W/m}^2\text{K}$

In case of water flow: $500 < \alpha < 2000 \text{ W/m}^2\text{K}$

In case of steam flow: $100 < \alpha < 1000 \text{ W/m}^2\text{K}$

In case of air or flue gas: $10 < \alpha < 200 \text{ W/m}^2\text{K}$

Heat transmission coefficient

$$k = \frac{1}{\frac{1}{\alpha_{fg}} + \sum \frac{\delta_i}{\lambda_i} + \frac{1}{\alpha_w}} \quad [W/m^2 K]$$

where: α_{fg} - flue gas heat transfer coefficient

α_w - water/steam side heat transfer coefficient

δ - thickness of the tube or other surface

(In case of soot or scale coating possibility coating also has to be taken into consideration.)

λ - thermal conductivity

Transferred heat:

$$Q_{transferred} = k \cdot F \cdot \Delta t_{ln}$$

where: k - heat transmission coefficient

F - heating surface area

Δt_{ln} - logarithmical temperature difference $\Delta t_{ln} = \frac{\Delta t_{greatest} - \Delta t_{smallest}}{\ln \frac{\Delta t_{greatest}}{\Delta t_{smallest}}}$

Heat quantities

Three types of heat quantities have to be equal: $Q_{fg} = Q_{transferred} = Q_{water/steam}$

Flue gas heat:

$$Q_{fg} = B \cdot \mu'_v \cdot c_{pfg} \cdot (t_{fgin} - t_{fgout}) \quad [kW]$$

where : B - mass flow rate of fuel [kg/s]

μ'_v - specific mass flow rate of fuel [kg/s]

c_{pfg} - specific heat of flue gas [kJ/kg°K]

t_{fg} - flue gas temperature [°C]

Water/steam:

$$\dot{Q}_{water/steam} = \dot{m}_w \cdot (h_{wout} - h_{win}) \quad [kW]$$

where : \dot{m}_w - mass flow rate of steam or water [kg/s]

h_w - water/steam enthalpy [kJ/kg]

If the three heat quantity don't equal, you can correct flue gas out and water/steam out temperatures according to the difference, and make the control calculation again, until three types of heat quantity will be equal.

Hydraulic fluid flow calculations

Hydraulic fluid flow calculations can be done after heat transfer calculation, because results of it are needed, e.g. velocity, temperature, density etc.

Pressure drop calculation:

Pressure drop calculation can be done according to *Bernoulli* equation:

$$\Delta p = \frac{\rho}{2} w^2 \left(\frac{l}{d} \cdot \lambda + \sum \xi \right) \text{ [Pa]}$$

where: ρ	density	[kg/m ³]
w	velocity	[m/s]
l	length	[m]
d	equivalent diameter	[m]
λ	resistance factor	[-]

In case of turbulent flow *Blasius rule can be used:*

$$\lambda = 0.316 / \text{Re}^{0.25}$$

$$\text{Re} = w \cdot d / \nu \text{ [-] Reynolds number)}$$

BOILER CIRCULATION

To remove heat from the boiler surfaces, it is necessary that adequate and positive water and steam circulation be provided (in a predetermined direction) throughout the boiler circuits. The flow of water, steam, or other fluid within the boiler is called circulation.

When heated water decreases in density and tends to raise to the top of the vessel, conversely, cooler water tends to drop to the bottom. When water is heated to the boiling point, small steam bubbles form on the heated surface. These bubbles cling to the metal (because of surface tension) until they become large enough to overcome the tension or until they are swept away by water circulation.

Steam is much lighter than water and rises rapidly. The movement of steam through the water creates turbulence and circulation.

A few high-pressure steam boilers and most hot water boilers depend on positive (forced) circulation, which is induced by pumps. Nearly all other boilers depend on natural circulation. The only function of water flow in a boiler system is to insure that sufficient water is fed to each tube receiving heat to keep its surface thoroughly wetted. Any additional flow is meant to provide an ample working margin and to remove salts and contaminants to the blowdown point.

The water flow velocity in a well designed natural circulation boiler is as high as in a positive circulation boiler.

In the past, watertube sizes were determined by empirical rule or factors, including the possibility of fouling because of scaling, ease of cleaning, and sighting the interior. Today the tube area merely represents that area which will pass the required quantity of water at the available entering velocity. With positive circulation, this area is fixed by the pumping capacity. With natural circulation, system characteristics (including the proportion of riser to downcomer tubes at various steaming rates) will govern.

Proper circulation is a necessity. The rate of steam release is directly related to the circulation rate of the water. Sluggish or stagnant circulation permits large bubbles to form, resulting in blistered or burned-out tubes. Restriction on free circulation must be kept to a minimum so that the water does not have to be forced through the tubes. Water movement in the tubes or water passages must be in one direction only, if at all possible. Water movement both upward and downward in the same tube will cause trouble.

Natural circulation

Water can circulate through boiler circuits solely because of the difference in density between the contents of the generating tubes and the downtakes. Natural circulation is the movement of the circulating fluid in conformance with this available head.

In any given natural circulation system the movement of the steam and water will increase to a maximum value, at which the rate of evaporation in the fireside generating tubes will exceed the rate of water flow in the downcomers and recirculators. After this, any further increase in heat absorption will result in a decrease in circulation and a lowering of the water level in the tubes.

Decreased circulation occurs as soon as the two opposing forces, - upflow and downflow - are balanced. An increase in density differential in itself tends to increase the flow, but is opposed by losses due to shock, fluid friction, turbulence and acceleration, all of which tend to reduce the flow. Watertube boilers are particularly sensitive to a drop-off in flow rate. To minimize losses, it is necessary to use large tubes.

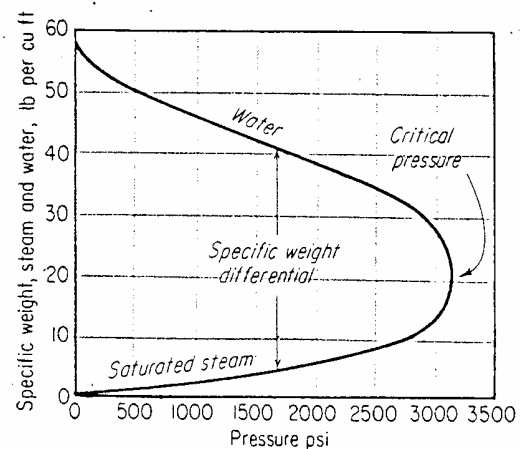
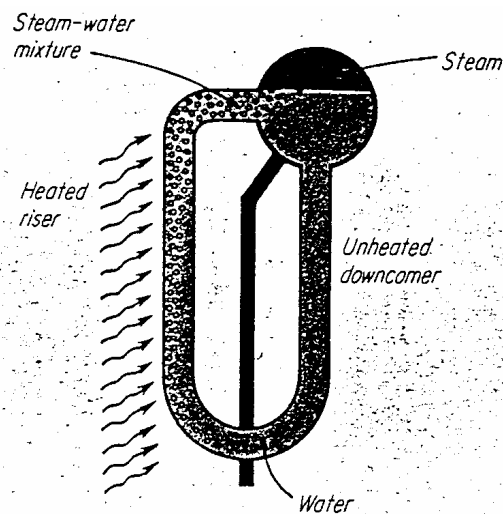
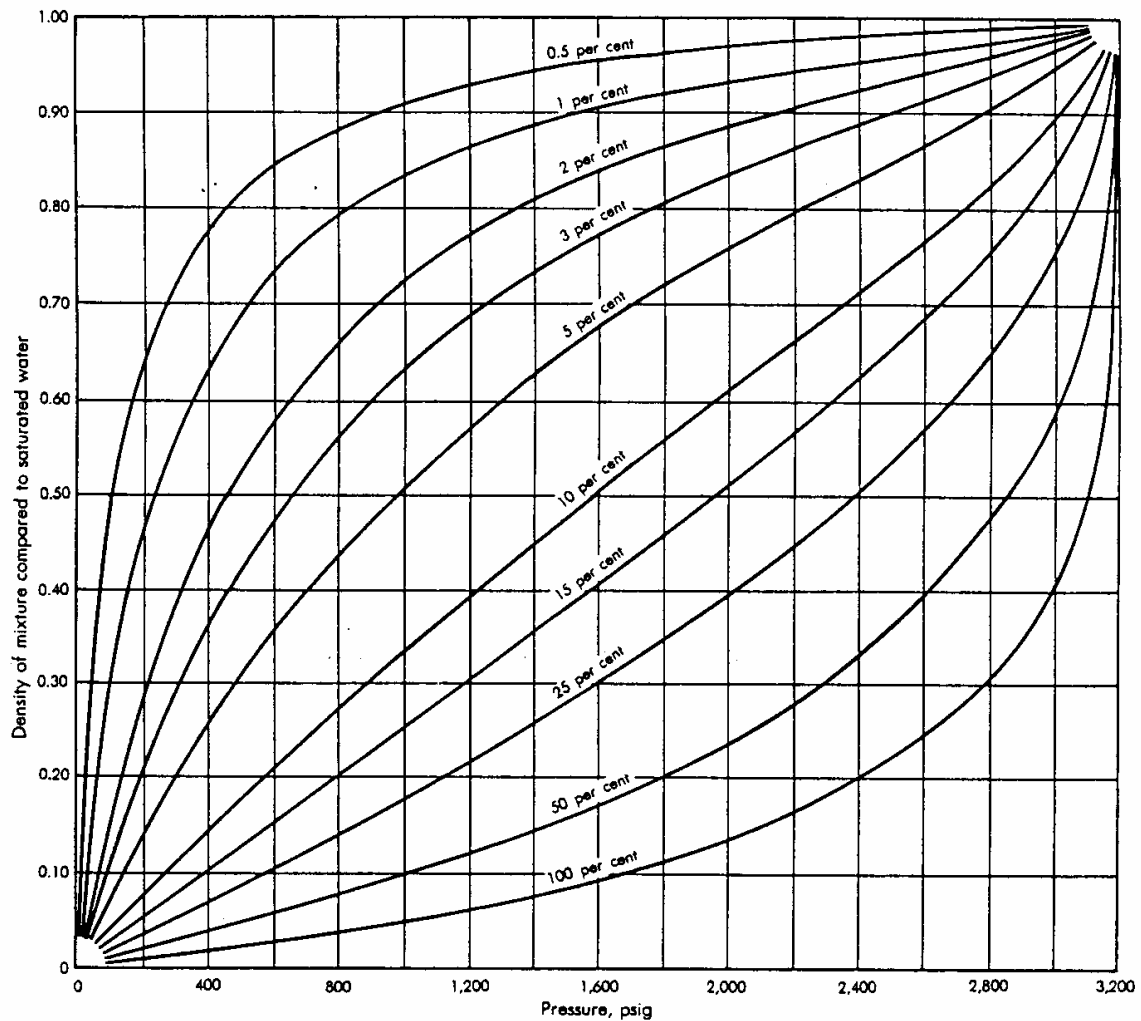


Fig.12.

Natural circulation curve and specific weight variation

Watertubes vary widely in circuit velocity and water to steam ratio, depending on circuit location, heat transfer rate, tube length, and tube diameter. Variable firing rates make it quite difficult to design for adequate circulation.

As the pressure increases, the density differential between the downcomer and the riser becomes smaller. To maintain a given circulation rate, it becomes necessary to increase the vertical height of the circulatory system and eliminate all steam from the circulating water before it enters the downcomer.



Density relationship of steam and water mixture

Fig. 13.

A natural circulation boiler tends to be self-compensating for the usual heat-absorption variations, including those caused by sudden overloads, soot and the like, non uniform fuel bed or burner conditions, and inability to forecast actual conditions over the operating lifetime.

Self-compensation may be illustrated as follows. The heat-absorption rate may be somewhat higher than predicted because of miscalculation of friction loss, localized hot spots, or other unforeseen circumstances. These reduce the average density of the circuit, and the flow to the circuit increases.

The phenomena of natural circulation make up a most complex study, which must rely largely on experience and experimental data rather than theoretical computations.

The broad principles of natural boiler circulation are the following:

1. Limitation on the percentage of steam by volume in the steam-water mixture (determined by pressure and type of circuit) at maximum capacity
2. Minimum water velocity entering heated portion of tubes
3. Separation of steam and water in drum to give steam free water in downcomer supply
4. Segregation of circuits having different absorption rates.

There are as many factors affecting the circulation of the water within the boiler as there are boiler types.

1. Firetube type boilers

The water travels down the drum sides, then up between the tubes. Because of large water volume and lack of channeling, circulation tends to become sluggish although a definite pattern exist.

2. Watertube boilers

In watertube boilers the convection heating surface is arranged so that some of the tubes act as downcomers and the balance act as risers, and the balance act as risers, depending upon the relative amount of heat to which the tubes are exposed.

At high capacity boilers external downcomers are used.

The flow areas of the downcomers depend upon the amount of wall exposed, the effective head of water on the wall, and the amount of heat absorbed. There are several arrangements of downcomer and riser systems that tie the wall surface into the boiler circuit.

3. Once-through boilers

In contrast to natural circulation and forced circulation designs - in which more water is circulated than steam is generated and a drum or drums serve as a collecting and steam-releasing point - the once-through design consists, in theory, of a single tube (no drum) into which goes feedwater and out of which comes saturated or superheated steam. In actual units, of course, the theoretical single circuit becomes a number of parallel circuits. At pressures below critical, a once-through unit may have a separator to deliver saturated steam to the superheater and to return collected moisture to the feed pump suction. The once-through cycle is, of course, ideally suited for pressures above the critical point where water turns to steam without actually boiling.

Circulation ratio

It would be highly desirable for a boiler to evaporate all the water circulating through the generating tubes. If this happened however the amount of solids deposited on the tubes would be greater than could be tolerated. It is customary, therefore to circulate from 3 to 20 times more water than is evaporated. This mixture of steam and water washes the precipitates from the tubes.

Total evaporation also presents other problems, including maintaining a constant water level, and the foaming caused by inadequate bubble release surface.

The ratio by weight of the water fed to the steam-generating tubes to the steam actually generated is called the circulation ratio. In a once-through positive circulation boiler the circulation ratio is 1:1. In a recirculation boiler, the circulation ratio can be from 3:1 up to 20:1. The typical circulating rate is 5:1.

Mechanical- and thermalstress calculations

Mechanical- and thermalstress calculations are very important from safety reasons.

Materials

For boiler constructions only that material can be used which has proofed strength at elevated temperature.

Calculation method

There are standardized calculation method is available for general cases.

But in complex cases further calculations could be needed, e.g. with some finite element method.

Calculation procedure

- From heat transfer and hydraulic calculations there shall be available wall temperature and pressure values.
- Than taking into account operation temperature and strength at this temperature allowable stress needs to be calculated.
- After it follows stress analysis of different parts. Actual stress has to be lower than allowable in each case

$$\sigma_{\text{real}} < \sigma_{\text{allowd}}$$

Stress types:

Primary stress:

Primary stress is caused by main load, e.g. inner pressure, bending, etc. considering regular shape of different parts.

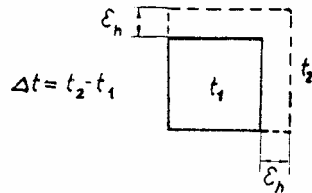
Secondary stress:

Secondary stress is caused by cuttings, outings, supports, etc., which modify normal stress distribution.

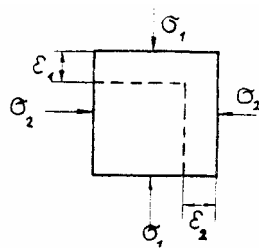
Thermal stress

Because of operation at high temperature different parts of the boiler would like to expand in different level. When this expansion is restricted it cause stress in the material it is called thermal stress. Calculation happens according to follows. For the first time dimensions of free expansion needs to be calculated.

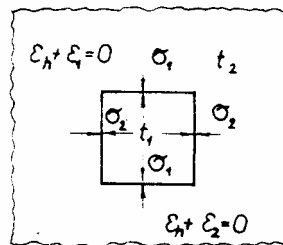
$$\varepsilon_h = \beta_{le} \cdot \Delta t$$



Then this part is pressed back to original dimension. In this way stress and reaction forces can be calculated.



In real these processes run parallel



Mechanical stress in plate fixed in two direction:

$$\sigma_{1,2} = -\frac{E}{1-\nu} \cdot \beta_{le} \cdot \Delta t$$

Thermal stress can be formed at different parts operated with different temperature level, or in the same part when temperature distribution varies in wide range.

Fatigue:

Boiler parts can have different stress level at different operation conditions. Stress fluctuation can cause fatigue for that part. Amplitude of stress can be calculated as follows.

$$\sigma_a = \frac{\sigma_{R \max} - \sigma_{R \min}}{2}$$

For a certain stress amplitude cycle number limit can be determined.

$$n_{\max} = f(\sigma_{\max})$$

When more than one type fatigue amplitude exist cumulative degradation method can be used. For each amplitude maximal cycle number needs to be determined. Than with following expression can be determined if actual part has reached its fatigue limit or not.

$$\frac{N_1}{n_{1 \max}} + \frac{N_2}{n_{2 \max}} + \dots + \frac{N_m}{n_{m \max}} \leq 1$$

where: N – actual cycle at a certain amplitude
n – maximal cycle of the same amplitude

Creep-rupture strength:

When certain parts operated at high temperature for a long time, creeping phenomena can be experienced. In order to avoid this operation time has to be limited.

Creep-rupture strength value can be gained from material data: $\tau_{\max}(\sigma, t)$

When more than one type of stress state exists cumulative degradation method can be used. For each stress maximal operation time needs to be determined. Than with following expression can be determined if actual part has reached its limit or not.

$$\frac{\Delta \tau_1}{\tau_{1 \max}} + \frac{\Delta \tau_2}{\tau_{2 \max}} + \dots + \frac{\Delta \tau_m}{\tau_{m \max}} \leq 1$$

Feed water preparation and quality

Dependable and economic operation of steam power plants requires high-purity feed water to prevent deposition of solids on the walls of the heating surfaces and their intensive corrosion, and to generate high-purity steam.

Boiler units of electric stations are fed with turbine condensate. However, a part of it is lost. In high-power condensing power plants the losses of the condensate amount to 1 to 3 per cent, and in heating and power plants, where it is impossible to return the condensate from processing apparatus, the losses reach tens of per cent. The losses have to be compensated by adding water.

Natural water cannot be added to the turbine condensate without special treatment, since it contains mechanical impurities and various dissolved salts and gases. Getting into the boiler, the solids contained in the feed water form scale and sludge while the dissolved gases /oxygen and carbon dioxide/ bring about corrosion of the boiler walls.

The scale-forming salts include silicates, carbonates and sulphates of magnesium and calcium. The total hardness of water is determined by the total concentration in it of calcium and magnesium ions. Water hardness is subdivided into carbonate /temporary/ hardness, traced to the presence of calcium and magnesium bicarbonates in it, and non-carbonate hardness, caused by other salts of calcium and magnesium. Hardness is expressed in milligram-equivalents or microgram-equivalents per litre.

Scale is deposited in a strong layer on the walls of the heating surfaces. Sludge separates from the boiler water in fine suspended particles. Secondary scale formation takes place in a number of cases, resulting from adhesion of sludge to the heating surfaces. Of all salts, the most active in scale formation are calcium silicate CaSiO_3 and calcium sulphate CaSO_4 yielding hard and dense scale.

Deposition of scale and adherence of sludge to the heating surfaces deteriorate dependable operation of boiler units and reduce their efficiency, due to a low heat conductivity of scale and sludge. Deposition of scale and sludge on the walls of the tubes that are exposed to the high-temperature flue gases may cause inadmissible overheating of these walls followed by a reduction in the strength of metal, deformation and failure of the tubes. By creating a large additional thermal resistance, any internal contamination of the heating temperature and a drop in the boiler efficiency.

The higher the pressure of the steam generated, the higher and stricter are the requirements imposed on the boiler feed water and especially on the feed water intended for once-through boilers generating steam in supercritical conditions. At higher pressures it becomes more difficult to generate pure steam, since it may be contaminated not only with the salts carried over by the water bubbles, but also with some impurities, silica acid and sodium compounds which can be dissolved in the higher pressure steam. The impurities contained in the steam deposit in the pipeline fittings, valves and also in the turbine valves and blading. This results in leaking of the valves and fittings and in a drop of the turbine efficiency and capacity.

Formation of scale in a steam boiler can be prevented either by removing the dissolved salts from the water or by converting them into easily soluble compounds that do not precipitate from the solution even at high salt concentration.

Water may be treated with the purpose of softening it, i.e. reducing the content of calcium and magnesium salts which make the water hard, or demineralization of water may be the aim. There is a number of water softening methods: hot process, precipitation, ion exchange and combined processes.

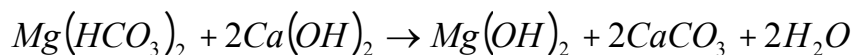
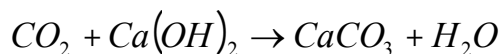
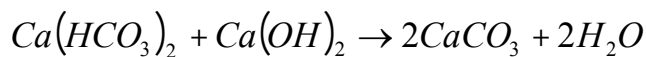
The hot-process method of water softening consists in heating the water to decompose calcium bicarbonate into water, carbon dioxide and precipitating calcium carbonate:



If water contains magnesium bicarbonate, it yields upon boiling a precipitate of poorly soluble magnesium hydroxide. It is impossible to ensure deep softening of water with this method, and it is not suitable for power boilers.

The precipitation or soda-lime methods of water softening are based on precipitation of calcium carbonate and magnesium hydroxide caused by adding certain chemicals to the water. Used as chemicals are lime, caustic soda and soda employed both separately and in various combinations.

At present liming is used as a preliminary treatment, to be followed by ion-exchange softening. Liming results in a sharp drop in carbonate hardness of the water, removal of dissolved carbon dioxide from it, elimination of magnesium hardness, and also in a reduction in the total solids of the water:



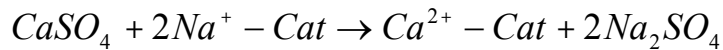
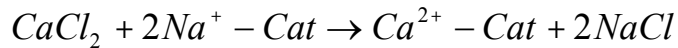
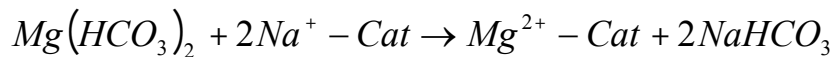
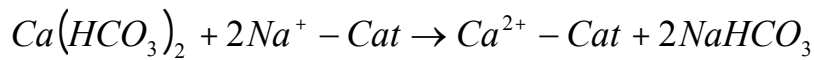
The soda-lime process was widely practiced in the past. The residual hardness of the soda-lime treated water is however still high. This method of water softening is not applied any more, owing to insufficient softening of the treated water, bulky devices and high costs.

The ion-exchange method of water softening has found now wide application due to its simplicity, economic advantages, small capital costs and high purity of the treated water. Substances employed in cation-exchange softening are known as cation-exchange resins or cation exchangers. In this country sulphonated coal is used as a cation exchanger which is produced by treating crushed coal with sulphuric acid.

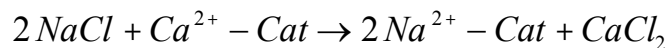
The cation-exchange methods of softening consists in the replacement of the cations in the ion exchanger by the cations contained in the water. Ion exchangers are insoluble in water. They may contain sodium (Na^+), hydrogen (H^+), or ammonium (NH_4^+) ions attached to the sulphonated coal matrix (carrier) and therefore the following cation exchange cycles are distinguished: Na^+ -exchange cycle; H^+ -exchange cycle; NH_4^+ -exchange cycle; and also combined cation-exchange cycles. The main characteristic of sulphonated coal is its nominal capacity, equal on the average to 300 g-eq/m³.

The process of cation-exchange water softening is very simple and consists in passing the water through a column packed with the cation exchanger.

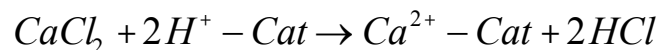
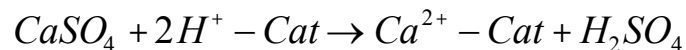
The Na^+ -exchange cycle has found the widest application in water treatment, owing to its simplicity, low costs of the common salt used to regenerate the cation exchanger, and deep softening of water. In the Na^+ -exchange cycle bicarbonates, chlorides and sulphates of calcium and magnesium, contained in the water, are converted into the corresponding salts of sodium that yield no scale



Since the sodium ions in the exchanger are gradually replaced by the calcium and magnesium ions contained in the passing water, the rate of exchange becomes slower with time. Finally the process stops, and the cation exchanger does not soften the water. The exchange capacity of Na⁺-exchangers is restored by passing a solution of common salt through it:



When handling water of a high initial carbonate hardness, it is of an advantage to practise combined softening, involving preliminary soda-lime treatment of water. The drawback of the Na⁺-exchange cycle is a high alkalinity and high total salt content of softened water. In the H⁺-exchange cycle calcium and magnesium salts are removed from water, and as a result of demineralisation it becomes acidic, for instance:



H⁺-exchangers are regenerated with sulphuric acid. The H⁺-exchange cycle is employed in particular as the first stage of water treatment for super high and supercritical boiler units, following the method of mixed bed demineralisation. The shortcomings of the H⁺-exchange cycle include high costs of the corrosion-resistant equipment, pipelines and fittings, and also intricate attendance and operation.

In industrial boiler plants wide use has found the NH₄⁺-exchange cycle combined with the Na⁺-exchange cycle.

Complete demineralization of water can be attained by treating it first with a cation exchanger, then with an anion exchanger. Anion exchange is the name given to a process involving exchange of anions between water and exchanger. Anion exchangers consist of an insoluble matrix and anions which are attached to it and may be replaced by other anions contained in water. Anion exchange is usually preceded by the H⁺-exchange treatment of water.

Anion exchangers are divided into two groups: weak basic, absorbing only the anions of strong acids (SO₄²⁻, Cl⁻, NO₃⁻) and strong basic, absorbing the anions of strong acids (HCO₃⁻, HSiO₃⁻).

Reverse osmosis is a new up to date method of water treating.

Osmosis effect is well known for a long time. If two solutions are bordered with a semi-permeable layer, from lean solution to thick solution go solvent molecules to reduce concentration difference, between two solutions. Because of diffusion pressure of thick solution increases. This process flows until the pressure became as high as it forms an osmotic equilibrium state. 1 mol/l concentration causes 22.4 bar osmotic pressure.

When the process is reversed, higher than osmotic pressure is applied on thick solution, water molecules go through semi-permeable layer, and salt and other pollutant molecules (which molecules are greater than water molecules) remain in thick solution.

This way we can get appropriate clean water for boiler operation as well.

This method is widely and widely used nowadays, because it's very plain, and it does not need any chemicals for operation, but it needs a very special semi-permeable layer, which is nowadays a bit expensive.

In deep water demineralization plants cheaper weak anion exchangers are used in the first stage and strong anion exchangers in the second and third stage.

Water can also be dematerialized by distillation in apparatus referred to as evaporators. They are heated with steam and fed with preliminary softened water. Distillation is the most expensive method of water treatment.

The following data can give an idea of the high requirements imposed on the dematerialized feed water intended for once-through boilers; its salt content should not exceed 0.05 mg/liter, hardness 3 mg/kg.

Dissolved corrosive-aggressive gases, oxygen and carbon dioxide, are removed from water by thermal deaeration. Depending on the pressure at which deaeration of water takes place, thermal deaerators are classified as vacuum, atmospheric and high-pressure deaerators. To ensure a more complete removal of dissolved oxygen from the feed water intended for super high and supercritical pressure boiler units, it is expedient to practice additional thermal deaeration of water by treating it with hydrazine hydrate.

Environmental pollution

Environmental pollution needs to be minimized in order to save our globe.

Pollutants can be different.

– Carbon dioxide (CO_2) is the result of complete combustion, but it is considered as green house gas. Reduction ways are reduction of energy consumption and usage of renewables.

– Unburnt gas components (CO , C_xH_z).

With an up to date firing equipment and with correct adjustment emission can be reduced to minimal level.

– Solid particulates (flying ash, soot, coke)

One way of reduction can be at firing side. But in case of solid fuel firing generally some kind of filter needs to be installed to meet with emission rules.

– Sulphur oxides (SO_2 , SO_3)

Determined by sulphur content of the fuel. In case of large quantity desulphurization unit needs to be installed.

– Nitrogen oxides (NO_x)

Generated from Nitrogen content of the fuel and partially from air. Quantity is generally proportional with combustion chamber temperature. General reduction method is to keep combustion chamber temperature at acceptable low level.

BOILERS and INDUSTRIAL FURNACES (problems and their solutions)¹

BOILER EFFICIENCY, LOSSES

- 1.) Boiler, produces dry saturated steam, is supplied with saturated liquid. 10 [kg/s] steam mass flow rate is produced by using 0.5 [kg/s] oil (lower calorific value of the fuel is 45 [MJ/kg]). The flue gas heat loss of the furnace is 10 [%], the combustion efficiency is 99 [%].

How much is the heat loss by radiation considering no other losses than the above mentioned? (heat needed to evaporate 1 [kg] of steam at this operational point is 1935 [kJ])

Solution:

The boiler efficiency can be calculated directly (the ratio of the useful and input heats) or indirectly (100% minus the sum of the loss factors in percent).

$$\dot{Q}_{in} = \dot{Q}_{useful} + \dot{Q}_{losses} \quad [\text{kW}]$$

where...

$$\dot{Q}_{in} = \dot{B}_{fuel} \cdot H_i \quad [\text{kW}]$$

$$\dot{Q}_{useful} = \dot{G}_{steam} \cdot (h_{steam}^{out} - h_{feed_water}^{in}) \quad [\text{kW}]$$

$$\dot{Q}_{losses} \cong \dot{Q}_{flue_gas} + \dot{Q}_{unburnt_fuel} + \dot{Q}_{radiation} + \dots \quad [\text{kW}]$$

Now, if we divide the above heat balance equation with the input heat we get the formulas to calculate the boiler efficiency by direct and indirect methods.

$$1 = \frac{\dot{Q}_{useful}}{\dot{Q}_{in}} + \frac{\dot{Q}_{flue_gas} + \dot{Q}_{unburnt_fuel} + \dot{Q}_{radiation} + \dots}{\dot{Q}_{in}}$$

and

¹ This collection was prepared by Group No.19 under TEMPUS JEP 1501, May 23rd, 1993.
(Ref.: BOILER_Q.DOC)

$$\eta_{boiler}^{direct} \equiv \frac{\dot{Q}_{useful}}{\dot{Q}_{in}}$$

$$\eta_{boiler}^{indirect} = 1 - \frac{\dot{Q}_{flue_gas} + \dot{Q}_{unburnt_fuel} + \dot{Q}_{radiation} + \dots}{\dot{Q}_{in}} = 1 - \sum_i \xi_i$$

where...

$$\text{- flue gas heat loss factor } \xi_{flue_gas} = \frac{\dot{Q}_{flue_gas}}{\dot{B}_{fuel} \cdot H_i} \quad [-]$$

$$\text{- unburnt heat loss factor } \xi_{unburnt_fuel} = \frac{\dot{Q}_{unburnt_fuel}}{\dot{B}_{fuel} \cdot H_i} \quad [-]$$

$$\dot{Q}_{unburnt_fuel} = \dot{B}_{unburnt_fuel} \cdot H_i$$

$$\text{- radiation heat loss factor } \xi_{radiation} = \frac{\dot{Q}_{radiation}}{\dot{B}_{fuel} \cdot H_i} \quad [-]$$

The combustion efficiency, which shows how complete and total the burning is, can be expressed by the unburnt heat loss factor...

$$\eta_{combustion} = \frac{\dot{Q}_{released}}{\dot{Q}_{in}} = \frac{\dot{B}_{burnt_fuel} \cdot H_i}{\dot{B}_{fuel} \cdot H_i} = \frac{\dot{B}_{fuel} - \dot{B}_{unburnt_fuel}}{\dot{B}_{fuel}} = 1 - \xi_{unburnt_fuel}$$

Now, we are able to answer the problem.

The useful heat is calculated considering only evaporation of feed water, because the drum is supplied with saturated liquid.

$$\eta_{boiler}^{direct} = \frac{\dot{G}_{steam} \cdot r}{\dot{B}_{fuel} \cdot H_i} = \frac{10 \cdot 1935}{0.5 \cdot 45 \cdot 10^3} = 0.86$$

$$\eta_{boiler}^{indirect} \equiv 1 - (\xi_{flue_gas} + \xi_{radiation} + \xi_{unburnt_fuel}) = 1 - (\xi_{flue_gas} + \xi_{radiation}) - (1 - \eta_{combustion})$$

$$\xi_{radiation} = 1 - \xi_{flue_gas} - \left(1 - \eta_{combustion}\right) - \eta_{boiler}^{indirect} = 1 - \frac{10}{100} - \left(1 - \frac{99}{100}\right) - 0.86 = 0.03$$

It means that the heat loss of the boiler towards the surroundings by radiation is:

$$\xi_{radiation} = \underline{\underline{3 \text{ [\%]}}}$$

- 2.) How much is the outlet steam enthalpy produced by a boiler if the flue gas heat loss is 10 [%] and the heat loss by radiation and by unburnt combustibles is 2-2 [%] respectively?

- feed water enthalpy : 461.1 [kJ/kg]
- mass flow rate of steam : 10.0 [kg/s]

Consider that the steam pressure is constant and a fuel is oil (mass flow rate of the fuel: 0.5 kg/s ; lower calorific value of the fuel is 45 [MJ/kg]).

Solution:

The outlet steam enthalpy is found with calculating the boiler efficiency and the useful heat.

$$\eta_{boiler} = 1 - (\xi_{flue_gas} + \xi_{radiation} + \xi_{unburnt_fuel}) = 1 - (0.1 + 0.02 + 0.02) = 0.86$$

$$\dot{Q}_{useful} = \eta_{boiler} \cdot \dot{B}_{fuel} \cdot H_i = 0.86 \cdot 0.5 \cdot 45 = 19.35 \text{ [MW]}$$

$$h_{steam} = h_{feed_water} + \frac{\dot{Q}_{useful}}{\dot{G}_{steam}} =$$

$$= 461.1 \text{ [kJ / kg]} + \frac{19.35 \cdot 10^3 \text{ [kJ / s]}}{10 \text{ [kg / s]}} = \underline{\underline{2396.1 \text{ [kJ / kg]}}}$$

FLUE GAS TEMPERATURE

3.) A fire tube boiler produces 12 [t/h] dry saturated steam at 12 [bar]. The feed water temperature is 105 [°C] (feed water enthalpy: 440.5 [kJ/kg] ; dry saturated steam enthalpy at the given pressure: 2784.5 [kJ/kg]).

- a.) Determine the needed power of the required burner if the boiler efficiency is 88 [%] and the lower calorific value of the fuel is 34.0 [MJ/kg].
- b.) How much is the outlet flue gas temperature if the ambient air temperature is 25 [°C] and the flue gas mass flow rate is 8.5 [kg/s]?

($\overline{c_{p,fg}} = 1.03$ [kJ / kgK]; $t_{ref} = 25$ [°C] consider that the combustion efficiency is 100 [%] and the specific heat loss factor due to radiation can be calculated as...

$$\xi_{radiation} = \sqrt{\frac{100}{\dot{G}_{steam} [t/h]}} \quad [\%]$$

Solution:

- a.) The required mass flow rate of fuel of the burner ("power") is calculated from the efficiency of the boiler.

$$\eta_{boiler}^{direct} = \frac{\dot{Q}_{useful}}{\dot{Q}_{in}} = \frac{\dot{G}_{steam} \cdot (h_{12bar}^{@@} - h_{feed_water})}{\dot{B}_{fuel} \cdot H_i}$$

$$\dot{Q}_{useful} = 12 \cdot \frac{1000}{3600} [\text{kg/s}] \cdot (2784.5 - 440.5) [\text{kJ/kg}] = 7813.3 [\text{kW}]$$

$$\dot{B}_{fuel} = \frac{\dot{Q}_{useful}}{\eta_{boiler}^{direct} \cdot H_i} = \frac{7813.3 [\text{kW}]}{\frac{88}{100} [-] \cdot 34 \cdot 10^3 [\text{kJ/kg}]} = 0.2611 [\text{kg/s}] \approx 940 [\text{kg/h}]$$

- b.) The outlet flue gas temperature is calculated with the help of a heat balance equation on the boiler furnace and the enthalpy change of the fuel-air mixture.

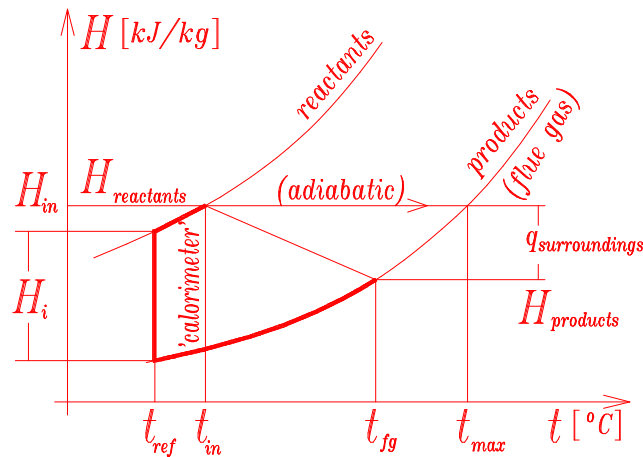


Fig. 1

The enthalpy, temperature diagram of reactants and flue gas.
The real enthalpy change of the fuel-air system is between the adiabatic and the 'calorimeter' processes.

Applying the first law of thermodynamic on the burning process...

$$dq = dh - vdp$$

$$p \approx \text{const. which means that } dp = 0$$

The heat towards the surroundings...

$$q_{\text{surroundings}} = (H_{\text{products}}(t_{fg}) - H_{\text{reactants}}(t_{in})) \text{ [kJ/kg fuel]}$$

Considering that the given lower calorific value of the fuel is released only if the flue gases are cooled back to the reference temperature (see Fig.1, this is the situation in a calorimeter when we determine the lower and upper heating value of an unknown fuel) we can divide the combustion for two parts. In the 1st part chemical reactions are completed at the constant reference temperature and in the 2nd part the heat of combustion is released and increases the flue gas temperature and gives useful heat. (If the useful heat is zero, which means that the whole released heat increases only the flue gas temperature, the process is called adiabatic.) The heat flow towards the surroundings can only be negative from the view point of the fuel-air mixture ($q_{\text{surroundings}} \leq 0$ (!), which means that the energy content of the mixture ('system') is decreased.

$$q_{\text{surroundings}} = (H_{\text{products}}(t_{fg}) - H_{\text{products}}(t_{ref})) + H_i + (H_{\text{reactants}}(t_{ref}) - H_{\text{reactants}}(t_{in}))$$

The last part of the above equation is zero because the reference and inlet temperatures are equal. The heat towards the surroundings includes the useful heat and the losses as well, except the flue gas heat loss, because it is included in the

product enthalpy. The enthalpy of combustion (H_i) is the lower calorific value of the fuel multiplied by minus one. This is because when this heat is released the energy of the 'system' is decreased.

The first part on the right hand side of the above equation is

$$(H_{products}(t_{fg}) - H_{products}(t_{ref})) = \mu_{fg} \cdot \overline{c_{p,fg}} \cdot (t_{fg} - t_{ref})$$

where

μ_{fg} : the specific produced flue gas amount; [kg flue gas / kg fuel]

$$\mu_{fg} = \frac{\dot{m}_{fg}}{\dot{B}_{fuel}}$$

$\overline{c_{p,fg}}$: the average specific heat of flue gas; [kJ/kgK]

t_{fg} : the outlet flue gas temperature; [K]

t_{ref} : the reference temperature at which the lower calorific value of the fuel is determined; [K]

The heat towards the surroundings is calculated:

$$q_{surroundings} = -(q_{useful} + q_{radiation}) \text{ [kJ/kg fuel]}$$

where

$$q_{useful} = \frac{\dot{Q}_{useful}}{\dot{B}_{fuel}} = \eta_{boiler}^{direct} \cdot H_i = \frac{88}{100} \cdot 34 \cdot 10^3 = \underline{29.92 \cdot 10^3} \text{ [kJ / kg fuel]}$$

$$q_{radiation} = \frac{\dot{Q}_{radiation}}{\dot{B}_{fuel}} = \xi_{radiation} \cdot H_i$$

and

$$\xi_{radiation} = \sqrt{\frac{100}{\dot{G}_{steam} [t/h]}} = \sqrt{\frac{100}{12}} = 2.89 \text{ [%]}$$

$$q_{radiation} = \frac{2.89}{100} \cdot 34 \cdot 10^3 = \underline{982.6} \text{ [kJ / kg fuel]}$$

The final equation to be solved:

$$-(q_{useful} + q_{radiation}) = \mu_{fg} \cdot \overline{c_{p,fg}} \cdot (t_{fg} - t_{ref}) + H_i$$

$$t_{fg} = \frac{-(q_{useful} + q_{radiation}) - H_i}{\mu_{fg} \cdot c_{p,fg}} + t_{ref} =$$

$$= \frac{-(29.92 \cdot 10^3 + 982.6) - (-34 \cdot 10^3)}{\frac{8.5}{0.2611} \cdot 1.03} + 25 = \underline{\underline{117.4}} \text{ [}^\circ\text{C]}$$

Similar equation can be derived to calculate the outlet flue gas temperature writing a heat balance equation on the whole boiler.

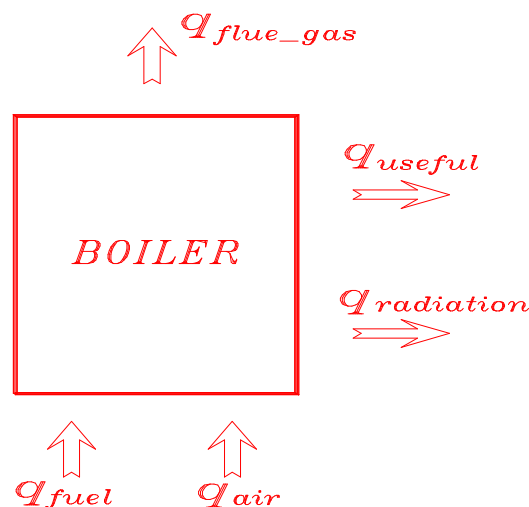


Fig. 2
Heat flows into and out of the boiler.

$$q_{fuel} + q_{air} = q_{flue_gas} + q_{useful} + q_{radiation} \quad [\text{kJ / kg fuel}]$$

where

$$q_{fuel} = H_i + \overline{c_{fuel}} \cdot (t_{fuel} - t_{ref})$$

: input heat by the fuel flow

$\overline{c_{fuel}}$: average sp. heat of the fuel; [kJ/kgK]

t_{fuel} : fuel inlet temperature; [K]

$$q_{air} = \lambda \cdot \mu_{o,air} \cdot \overline{c_{p,air}} \cdot (t_{air} - t_{ref})$$

: input heat by the air flow

λ : excess air factor of combustion; [-]

$\mu_{o,air}$: stoichiometrically needed air to burn 1 kg of the fuel; [kg air/kg fuel]

$\overline{c_{p,air}}$: average sp. heat of air; [kJ/kgK]

t_{air} : air input temperature; [K]

$$q_{flue_gas} = \mu_{fg} \cdot \overline{c_{p,fg}} \cdot (t_{fg} - t_{ref})$$

: outlet heat flow from the boiler by the flue gas

$$\mu_{fg} = \mu_{o,fg} + (\lambda - 1) \cdot \mu_{o,air}$$

μ_{fg} : sp. produced flue gas amount; [kg flue gas / kg fuel]
 $\mu_{o,fg}$: stoichiometrically produced flue gas amount; [kg flue gas / kg fuel]
 $\overline{c_{p,fg}}$: average sp. heat of flue gas; [kJ/kgK]
 t_{fg} : outlet flue gas temperature; [K]

$q_{useful}, q_{radiation}$: they have already been explained above (in case of adiabatic combustion these are zero)

Presently, the input temperature of fuel-air mixture is equal with the reference temperature, so...

$$H_i = \mu_{fg} \cdot \overline{c_{p,fg}} \cdot (t_{fg} - t_{ref}) + q_{useful} + q_{radiation}$$

$$t_{fg} = \frac{H_i - (q_{useful} + q_{radiation})}{\mu_{fg} \cdot \overline{c_{p,fg}}} + t_{ref} =$$

$$= \frac{34 \cdot 10^3 - (29.92 \cdot 10^3 + 982.6)}{\frac{8.5}{0.2611} \cdot 1.03} + 25 = \underline{\underline{117.4 [^\circ C]}}$$

FEED WATER INJECTION

- 4.) In a water tube boiler feed water injection between the radiant and convection superheaters is applied to regulate the outlet steam temperature. How much is the needed injection if we want the temperature to be decreased by 10 [°C] after the injection (see below)?

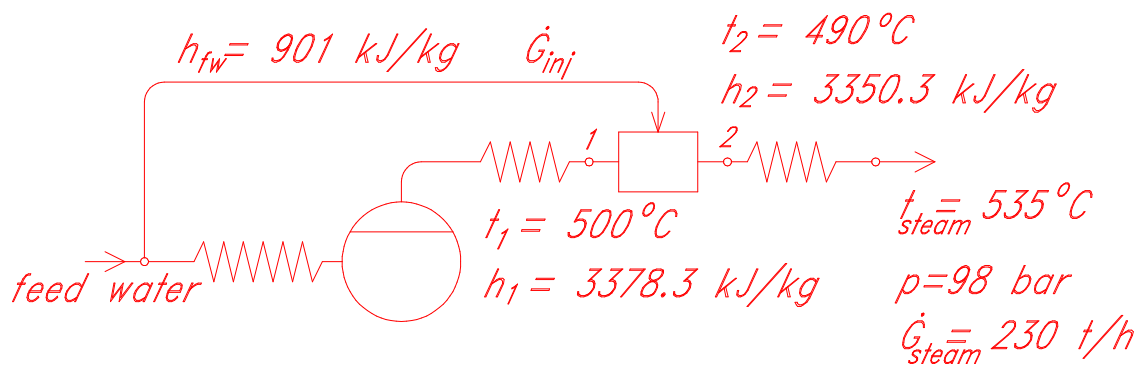


Fig. 3

Schematic layout of the water-steam path of a boiler

Solution:

The mass flow rate of the injected feed water is determined with the help of heat and mass balance equations written on the injection process.

a.) The heat balance equation...

$$\dot{G}_{inj} \cdot h_{fw} + \dot{G}_1 \cdot h_1 = \dot{G}_2 \cdot h_2$$

b.) The mass balance equations...

$$\dot{G}_2 = \dot{G}_{steam}$$

$$\dot{G}_1 = \dot{G}_{steam} - \dot{G}_{inj}$$

hence ...

$$\dot{G}_{inj} \cdot h_{fw} + \left(\dot{G}_{steam} - \dot{G}_{inj} \right) \cdot h_1 = \dot{G}_{steam} \cdot h_2$$

$$\dot{G}_{inj} = \dot{G}_{steam} \cdot \frac{h_1 - h_2}{h_1 - h_{fw}} = 280 \cdot \frac{3378.3 - 3350.3}{3378.3 - 901.0} = \underline{\underline{3.165 \text{ [t/h]}}}$$

STEAM STORAGE CAPACITY (FLEXIBILITY) OF FIRE TUBE BOILER

- 5.) Nominal steam mass flow rate of a fire tube boiler is 12 [t/h] at 12 [bar]. The volume of the drum is 30 [m³] and the mass of the steam water mixture in the drum at maximum load is 23 [t].

How long can the boiler produce the same amount of steam if the feed water pump and the firing system stop. The min. steam pressure can only be 8 bar required by the consumer.

Data at 12 [bar]:

saturated liquid enthalpy	:	798.40	[kJ/kg]
saturated liquid specific volume	:	1.14	[dm ³ /kg]
saturated dry steam enthalpy	:	2782.70	[kJ/kg]
saturated dry steam specific volume	:	163.20	[dm ³ /kg]

Data at 8 [bar]:

saturated liquid enthalpy	:	720.90	[kJ/kg]
saturated liquid specific volume	:	1.12	[dm ³ /kg]
saturated dry steam enthalpy	:	2767.50	[kJ/kg]
saturated dry steam specific volume	:	240.30	[dm ³ /kg]

Solution:

The dryness factor of the water-steam mixture in the drum at the beginning of the process:

The specific volume of the mixture:

$$v_{12[\text{bar}]} = \frac{V}{G} = \frac{30}{23} = 1.3043 \text{ [dm}^3 \text{ / kg]}$$

The dryness factor of the mixture (steam in the mixture):

$$x_{12[\text{bar}]} [\%] = 100 \cdot \frac{v_{12[\text{bar}]} - v'_{12[\text{bar}]}}{v_{12[\text{bar}]} - v'_{12[\text{bar}]}} = 100 \cdot \frac{1.3043 - 1.14}{163.20 - 1.14} = 0.101 [\%]$$

It means that the pure steam content of the drum is so low that we can neglect it without a significant error and the calculation can be based on pure saturated water.

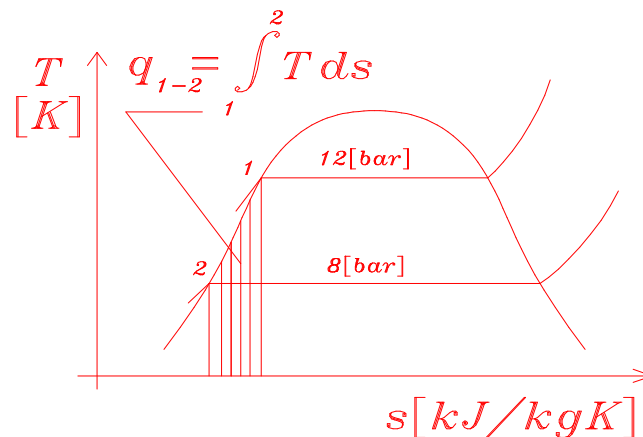


Fig. 4

The hatched area in T-s chart shows the heat covering steam consumption after stopping the feed water pump.

The heat represented by the steam consumption during the pressure drop in the drum from 12 bar to 8 bar is:

$$Q \cong \dot{G}_{steam} \cdot \bar{r} \cdot \tau \quad [\text{kJ}]$$

where

\dot{G}_{steam} : is the 12 [t / h] steam consumption; 3.33 [kg/s]

\bar{r} : is the average latent heat of evaporation; [kJ/kg]

$$\bar{r} = \frac{(h''_{12[\text{bar}]} - h'_{12[\text{bar}]}) + (h''_{8[\text{bar}]} - h'_{8[\text{bar}]})}{2} =$$

$$\frac{(2782.7 - 798.4) + (2767.5 - 720.9)}{2} = 2015.45 \quad [\text{kJ/kg}]$$

τ : is the time during the pressure drop; [s]

This heat is produced by the saturated feed water due to the pressure drop.

$$Q \cong G_{feed_water} \cdot (h'_{12[\text{bar}]} - h'_{8[\text{bar}]}) = 23 \cdot 10^3 \cdot (798.4 - 720.9) = 1782.5 \quad [\text{MJ}]$$

Hence, the time available for the 12 [t/h] dry steam consumption until the pressure in the drum reaches 8 [bar]:

$$\tau = \frac{Q}{\dot{G}_{steam} \cdot \bar{r}} = \frac{1782.5 \cdot 10^3 [\text{kJ}]}{12 \cdot \frac{1000}{3600} [\text{kg/s}] \cdot 2015.45 [\text{kJ/kg}]} = \frac{265.3 \cdot [\text{sec}]}{4 \cdot [\text{min}], 25 \cdot [\text{sec}]}$$

COMPLEX PROBLEMS (WATER TUBE BOILER)

6.) In a coal fired industrial water tube boiler the flue gas temperature is $t_{fg} = 173$ [°C]. The volume flow rate of the gas at this temperature is $V_{fg} \Big|_{t_{fg}} = 66.71$ [m³ / s].

a.) Determine the boiler efficiency knowing that the loss factors of the furnace are...

- heat loss through the walls : $\xi_{walls} = 0.9$ [%]
- heat loss by the flue gas : $\xi_{flue_gas} = 9.4$ [%]
- heat loss due to unburnt combustibles : $\xi_{unburnt_fuel} = 3.0$ [%]
- heat loss by slag and flying ashes : $\xi_{slag+ashes} = 1.3$ [%]

b.) How high is the adiabatic flame temperature of combustion if the useful heat of the boiler is $\dot{Q}_{useful} = 85417$ [kW] and the entering air temperature after the preheater is $t_{air} = 230$ [°C] ?

$$\left(\overline{c_{p,air}} = 1.295 \text{ [kJ/Nm}^3 \text{ K]} ; \overline{c_{p,fg}} = 1.41 \text{ [kJ/Nm}^3 \text{ K]} \right)$$

b.) Calculate the excess air factor of the present operation.

- lower calorific value of the fuel : $H_i = 11279$ [kJ / kg]
- stoichiometrically needed air to burn 1 kg of the fuel : $\mu_{o,air} = 3.198$ [Nm³ / kg]
- theoretically produced flue gas amount from 1 kg fuel : $\mu_{o,fg} = 3.787$ [Nm³ / kg]

Solution:

a.) The boiler efficiency can be calculated indirectly from the heat loss factors.

$$\eta_{boiler}^{indirect} = 1 - \frac{(\xi_{walls} + \xi_{flue_gas} + \xi_{unburnt_fuel} + \xi_{slag+ashes})}{100} =$$

$$= 1 - \frac{(0.9 + 9.4 + 3.0 + 1.3)}{100} = \underline{\underline{0.854}}$$

- b.) If the burning process is adiabatic the total released heat increases the flue gas temperature only. The heat balance equation on the fuel-air mixture and flue gas is:

$$\dot{Q}_{fuel} + \dot{Q}_{air} = \dot{Q}_{flue_gas}$$

Heat input by the fuel:

$$\dot{Q}_{fuel} = \dot{B}_{fuel} \left[H_i + \overline{c}_{fuel} \cdot (t_{fuel} - t_{ref}) \right] \text{ [kW]}$$

where

$$\begin{aligned} \dot{B}_{fuel} &= \frac{\dot{Q}_{useful}}{\eta_{boiler} \cdot H_i} = \\ &= \frac{85417 \text{ [kW]}}{0.854 [-] \cdot 11279 \text{ [kJ/kg]}} = 8.868 \text{ [kg/s]} \approx 32.0 \text{ [t/h]} \end{aligned}$$

The second part of the fuel heat can be neglected comparing it to the heat represented by the lower calorific value of the fuel...

$$\dot{Q}_{fuel} \cong \dot{B}_{fuel} \cdot H_i = 8.868 \text{ [kg/s]} \cdot 11279 \text{ [kJ/kg]} \cong 100.0 \text{ [MW]}$$

Heat input by the air:

$$\dot{Q}_{air} = \dot{B}_{fuel} \cdot \lambda \cdot \mu_{o,air} \cdot \overline{c}_{p,air} \cdot (t_{air} - t_{ref}) \text{ [kW]}$$

where

λ : excess air factor of combustion

$$\lambda \equiv \frac{\mu_{air}}{\mu_{o,air}}$$

and

$$\dot{V}_{fg}^N = \dot{B}_{fuel} \cdot \mu_{fg} = \dot{B}_{fuel} \cdot \left(\mu_{o,fg} + (\lambda - 1) \cdot \mu_{o,air} \right) \text{ [Nm}^3\text{/s]}$$

applying the ideal gas state equation on the flue gas

$$\dot{V}_{fg}^N = \dot{V}_{fg} \cdot \frac{T_N}{T_{fg}} = 66.71 \text{ [m}^3\text{/s]} \cdot \frac{273.15 \text{ [K]}}{(173 + 273.15) \text{ [K]}} = 40.84 \text{ [Nm}^3\text{/s]}$$

$$\lambda = \frac{\dot{V}_{fg}^N / \dot{B}_{fuel} - \mu_{o,fg}}{\mu_{o,air}} + 1 =$$

$$= \frac{40.84 [\text{Nm}^3 / \text{s}] / 8.868 [\text{kg} / \text{s}] - 3.878 [\text{Nm}^3 / \text{kg}]}{3.198 [\text{Nm}^3 / \text{kg}]} + 1 = \underline{\underline{1.23 [-]}}$$

hence

$$\dot{Q}_{air} = 8.868 \cdot 1.23 \cdot 3.198 \cdot 1.295 \cdot (230 - 25) = 9260.5 [\text{kW}]$$

Heat flow by the outlet flue gas:

$$\dot{Q}_{flue_gas} = \dot{V}_{fg}^N \cdot \overline{c_{p,fg}} \cdot (t_{fg} - t_{ref}) [\text{kW}]$$

Expressing the outlet flue gas temperature from the heat balance equation...

$$t_{fg} = \frac{\dot{Q}_{fuel} + \dot{Q}_{air}}{\dot{V}_{fg}^N \cdot \overline{c_{p,fg}}} + t_{ref} = \frac{100 \cdot 10^3 + 9260}{40.84 \cdot 1.41} + 25 = \underline{\underline{1922.4 [^\circ\text{C}]}}$$

- c.) The excess air factor was needed to calculate the adiabatic flame temperature under point b.).

- 7.) A water tube pulverised coal fuelled boiler produces 120 [t/h] steam at 100 [bar] and 500 [°C]. The feed water temperature is 190 [°C], the preheated air temperature is 233 [°C] and the outlet flue gas temperature is 173 [°C] (ambient air temperature is 20 [°C]).

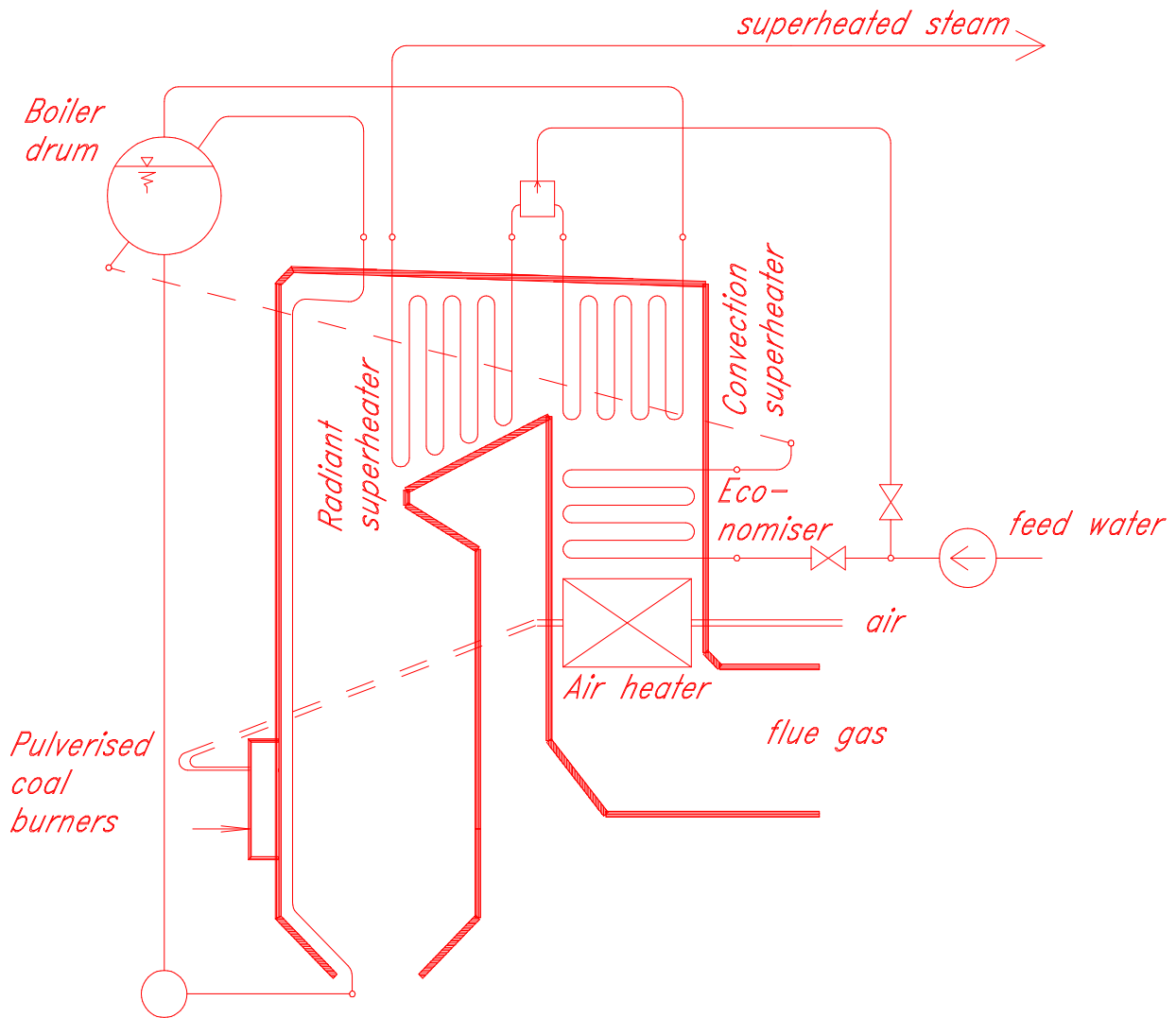


Fig. 5
Schematic layout of a water tube boiler with radiant and convection superheaters

- a.) Check the required steam mass flow rate with the help heat balance equations on the boiler drum and on the whole boiler. How much is the injected feed water between the convection and radiant superheater? Calculate the enthalpy of the superheated steam before and after the injection.

Given data:

- feed water enthalpy	:	h_{fw}	=	811.6	[kJ/kg]
- saturated liquid enthalpy	:	h'	=	1408.6	[kJ/kg]
- saturated dry steam enthalpy	:	h''	=	2724.4	[kJ/kg]
- superheated steam enthalpy at the exit of the boiler	:	h_{steam}	=	3374.1	[kJ/kg]

Useful heat flow through ...

- the economiser	:	\dot{Q}_{ECO}	=	15817	[kW]
- the convection superheater	:	\dot{Q}_{SH1}	=	13506	[kW]
- the radiant superheater	:	\dot{Q}_{SH2}	=	12030	[kW]
- the evaporator water walls	:	\dot{Q}_{EVAP}	=	44064	[kW]

b.) Calculate the needed mass flow rate of pulverised coal if the different heat loss factors of the boiler are ...

- heat loss through the walls	:	ξ_{walls}	=	0.9	[%]
- heat loss due to the high heat content of the flue gas	:	ξ_{flue_gas}	=	9.4	[%]
- unburnt combustibles losses	:	$\xi_{unburnt_fuel}$	=	3.0	[%]
- slag and flying-ashes losses	:	$\xi_{slag+ashes}$	=	1.3	[%]

(lower calorific value of the coal is 11279 [kJ/kg])

c.) How much is the required air and flue gas volume flow rate (at the exit of the boiler) if the excess air factor of combustion is 1.3 ?

- stoichiometrically needed air to burn 1 kg of the fuel	:	$\mu_{o,air}$	=	3.198	[Nm ³ /kg]
- stoichiometrically produced flue gas amount	:	$\mu_{o,fg}$	=	3.787	[Nm ³ /kg]

d.) Calculate the adiabatic flame temperature in the furnace.

$$\left(\overline{c_{p,air}} = 1.295 \text{ [kJ / Nm}^3\text{K]} ; \overline{c_{p,fg}} = 1.41 \text{ [kJ / Nm}^3\text{K]} \right)$$

- e.) How much is the flue gas temperature between the two superheaters?
(Consider that the ξ_{walls} and $\xi_{slag+ashes}$ represent the heat loss factors until this point.)
- f.) Calculate the overall heat transfer coefficient of the convection superheater.

($t_s|_{p=100[bar]} = 310.96 [^{\circ}C]$; exit steam temperature from the conv. sup.heater is $419 [^{\circ}C]$; surface area of the heat exchanger is $649 [m^2]$)

Solution:

- a.) The heat balance equation on the drum, considering that the steam leaving the drum is 100% dry.

$$\dot{G}_{steam}^{drum} \cdot h_{fw} + \dot{Q}_{ECO} + \dot{Q}_{EVAP} = \dot{G}_{steam}^{drum} \cdot h''$$

where \dot{G}_{steam}^{drum} is the mass flow rate of dry steam leaving the drum.

$$\dot{G}_{steam}^{drum} = \frac{\dot{Q}_{ECO} + \dot{Q}_{EVAP}}{h'' - h_{fw}} = \frac{15817 + 44064}{2724.4 - 811.6} = 31.3 \text{ [kg/s]} \approx 112.7 \text{ [t/h]}$$

$$\dot{G}_{steam}^{drum} < \dot{G}_{steam}^{required} = 120 \text{ [t/h]} \quad (!)$$

The heat balance on the whole boiler:

$$\dot{G}_{steam} \cdot h_{fw} + \left(\dot{Q}_{ECO} + \dot{Q}_{EVAP} + \dot{Q}_{SH1} + \dot{Q}_{SH2} \right) = \dot{G}_{steam} \cdot h_{steam}$$

$$\begin{aligned} \dot{G}_{steam} &= \frac{\dot{Q}_{ECO} + \dot{Q}_{EVAP} + \dot{Q}_{SH1} + \dot{Q}_{SH2}}{h_{steam} - h_{fw}} = \\ &= \frac{15817 + 44064 + 13506 + 12030}{3374.1 - 811.6} = 33.3 \text{ [kg/s]} \approx 120 \text{ [t/h]} \end{aligned}$$

The heat balance on the injection process:

$$\dot{G}_{steam}^{drum} \cdot h'' + \dot{Q}_{SH1} + \dot{G}_{fw_inj} \cdot h_{fw} + \dot{Q}_{SH2} = \dot{G}_{steam} \cdot h_{steam}$$

$$\begin{aligned}\dot{G}_{fw_inj} &= \frac{\dot{G}_{steam} \cdot h_{steam} - \left(\dot{G}_{steam}^{drum} \cdot h^{@@} + \dot{Q}_{SH1} + \dot{Q}_{SH2} \right)}{h_{fw}} = \\ &= \frac{33.3 \cdot 3374.1 - (31.3 \cdot 2724.4 + 13506 + 12030)}{811.6} = 2.027 \text{ [kg/s]} \\ &\approx \underline{\underline{7.3 \text{ [t/h]}}}\end{aligned}$$

This result can be found with the help of a mass balance equation as well.

$$\dot{G}_{fw_inj} = \dot{G}_{steam} - \dot{G}_{steam}^{drum} = 120 \text{ [t/h]} - 112.7 \text{ [t/h]} = \underline{\underline{7.3 \text{ [t/h]}}}$$

The enthalpy of the steam before feed water injection:

$$h_1 = \frac{\dot{G}_{steam}^{drum} \cdot h'' + \dot{Q}_{SH1}}{\dot{G}_{steam}^{drum}} = \frac{31.3 \cdot 2724.4 + 13506}{31.3} = \underline{\underline{3155.8 \text{ [kJ/kg]}}}$$

The enthalpy of the steam after feed water injection:

$$h_2 = \frac{\dot{G}_{steam} \cdot h_{steam} - \dot{Q}_{SH2}}{\dot{G}_{steam}} = \frac{33.3 \cdot 3374.1 - 12030}{33.3} = \underline{\underline{3013.2 \text{ [kJ/kg]}}}$$

So, the feed water injection decreased the enthalpy by $\approx 143 \text{ [kJ/kg]}$.

- b.) The mass flow rate of the fuel is calculated with the help of indirect boiler efficiency definition knowing the useful heat amount.

$$\begin{aligned}\eta_{boiler}^{indirect} &= 1 - \sum_i \xi_i = 1 - \left(\xi_{walls} + \xi_{flue_gas} + \xi_{unburnt_fuel} + \xi_{slag+ashes} \right) = \\ &= 1 - \frac{0.9 + 9.4 + 3.0 + 1.3}{100} = 0.854\end{aligned}$$

The input heat to the furnace by the fuel chemical energy.

$$\begin{aligned}\dot{Q}_{in} &= \frac{\dot{Q}_{\text{useful}}}{\eta_{\text{boiler}}^{\text{indirect}}} = \frac{\dot{Q}_{ECO} + \dot{Q}_{EVAP} + \dot{Q}_{SH1} + \dot{Q}_{SH2}}{\eta_{\text{boiler}}^{\text{indirect}}} = \\ &= \frac{15817 + 44064 + 13506 + 12030}{0.854} \cong 100020 \text{ [kW]}\end{aligned}$$

hence the mass flow rate of fuel...

$$\dot{B}_{\text{fuel}} = \frac{\dot{Q}_{in}}{H_i} = \frac{100020}{11279} = 8.868 \text{ [kg/s]} \cong \underline{\underline{31.92 \text{ [t/h]}}}$$

The above mass flow rate of the fuel is the total, but only $(1 - \xi_{\text{unburnt_fuel}})$ [%] will burn during combustion (it gives the combustion efficiency as well).

$$\begin{aligned}\dot{B}_{\text{burnt_fuel}} &= (1 - \xi_{\text{unburnt_fuel}}) \cdot \dot{B}_{\text{fuel}} = \\ &= (1 - 0.03) \cdot 8.868 = \underline{\underline{8.602 \text{ [kg/s]} \cong 31.90 \text{ [t/h]}}}\end{aligned}$$

- c.) The volume flow rate of air and flue gas are calculated from the definition of excess air factor.

$$\lambda \equiv \frac{\mu_{\text{air}}}{\mu_{\text{o,air}}} = \frac{\dot{V}_{\text{air}}^N}{\dot{B}_{\text{burnt_fuel}} \cdot \mu_{\text{o,air}}} = \frac{\dot{V}_{\text{air}}^N}{\dot{B}_{\text{burnt_fuel}} \cdot \mu_{\text{o,air}}}$$

$$\dot{V}_{\text{air}}^N = \lambda \cdot \dot{B}_{\text{burnt_fuel}} \cdot \mu_{\text{o,air}} = 1.3 \cdot 8.602 \cdot 3.198 = \underline{\underline{35.76 \text{ [Nm}^3\text{/s]}}}$$

This volume flow rate of air is valid at normal state (1 [atm] = 1.01325 [bar] and 0 [°C]).

The volume flow rate of flue gas from combustion:

$$\begin{aligned}\dot{V}_{\text{fg}}^N &= \dot{B}_{\text{burnt_fuel}} \cdot (\mu_{\text{o,fg}} + (\lambda - 1) \cdot \mu_{\text{o,air}}) = \\ &= 8.602 \cdot (3.787 + (1.3 - 1) \cdot 3.198) = \underline{\underline{40.83 \text{ [Nm}^3\text{/s]}}}\end{aligned}$$

Certainly, unburnt fuel is also included in the total flue gas volume, but its volume is so small that can be neglected.

The flue gas volume at the outlet flue gas temperature from the ideal gas state equation:

$$\dot{V}_{fg} \Big|_{t_{fg}} = \dot{V}_{fg}^N \cdot \frac{T_{fg}}{T_N} = 40.83 \cdot \frac{173 + 273.15}{273.15} \cong \underline{\underline{66.70 \text{ [m}^3 \text{ / s]}}}$$

This normal and real state consideration is very important in selection of flue gas fan.

- d.) In case of adiabatic condition the released heat increases the flue gas temperature only, which means that the input heat represented by the fuel and air flow is equal with the output heat of the flue gas.

$$\dot{Q}_{fuel} + \dot{Q}_{air} = \dot{Q}_{flue_gas}$$

Heat input by the fuel, neglecting the fuel heat capacity:

$$\dot{Q}_{fuel} = \dot{B}_{burnt_fuel} \cdot \left[H_i + \frac{\overline{c_{fuel}} \cdot (t_{fuel} - t_{ref})}{\eta_{combustion}} \right] \approx 8.602 \cdot 11279 = \underline{\underline{97022 \text{ [kW]}}}$$

Heat input by the air:

$$\dot{Q}_{air} = \dot{V}_{air}^N \cdot \overline{c_{p,air}} \cdot (t_{air} - t_{ref}) = 35.76 \cdot 1.295 \cdot (233 - 25) = \underline{\underline{9632.3 \text{ [kW]}}}$$

Heat flow by the outlet flue gas neglecting unburnt combustion:

$$\dot{Q}_{flue_gas} \cong \dot{V}_{fg}^N \cdot \overline{c_{p,fg}} \cdot (t_{fg} - t_{ref}) \text{ [kW]}$$

Expressing the outlet adiabatic flue gas temperature from the heat balance equation...

$$t_{fg} = \frac{\dot{Q}_{fuel} + \dot{Q}_{air}}{\dot{V}_{fg}^N \cdot \overline{c_{p,fg}}} + t_{ref} = \frac{97022 + 9632.3}{40.83 \cdot 1.41} + 25 = \underline{\underline{1877.6 \text{ [}^\circ\text{C]}}}$$

e.) The heat balance equation until the point between the two superheaters:

$$\dot{Q}_{fuel} + \dot{Q}_{air} = \dot{Q}_{EVAP} + \dot{Q}_{SH2} + \dot{Q}_{losses} + \dot{Q}_{flue_gas}^{SH}$$

Above we have already calculated \dot{Q}_{fuel} and \dot{Q}_{air} . The useful heats until this point (\dot{Q}_{EVAP} , \dot{Q}_{SH2}) are known.

The heat loss is calculated from its definition:

$$\begin{aligned} \dot{Q}_{losses} &= \dot{B}_{fuel} \cdot H_i \cdot (\xi_{walls} + \xi_{slag+ashes}) = \\ &= 8.868 \cdot 11279 \cdot \frac{0.9 + 1.3}{100} = 2200.5 \text{ [kW]} \end{aligned}$$

The heat content of the flue gas between the two superheaters is calculated with the same specific heat.

$$\dot{Q}_{flue_gas}^{SH} \cong \dot{V}_{fg}^N \cdot \overline{c_{p,fg}} \cdot (t_{fg}^{SH} - t_{ref}) \text{ [kW]}$$

Expressing the flue gas temperature between the superheaters...

$$\begin{aligned} t_{fg}^{SH} &= \frac{\dot{Q}_{flue_gas}^{SH}}{\dot{V}_{fg}^N \cdot \overline{c_{p,fg}}} + t_{ref} = \frac{(\dot{Q}_{fuel} + \dot{Q}_{air}) - (\dot{Q}_{EVAP} + \dot{Q}_{SH1} + \dot{Q}_{losses})}{\dot{V}_{fg}^N \cdot \overline{c_{p,fg}}} + t_{ref} \\ t_{fg}^{SH} &= \frac{(97022 + 9632.3) - (44064 + 12030 + 2200.5)}{40.83 \cdot 1.41} + 25 = \underline{\underline{865.0 \text{ [}^\circ\text{C]}}} \end{aligned}$$

- f.) The heat transfer coefficient of the convection superheater is calculated with the help of the inlet and outlet temperatures in steam and flue gas side.

The flue gas temperature after the convection superheater has to be calculated first.

Lets denote the flue gas temperature before the convection superheater by:

$$t_{fg}^{(1)} = t_{fg}^{SH} = 865 \text{ [}^\circ\text{C]}$$

and the temperature of the flue gas after the convection superheater by:

$$t_{fg}^{(2)} \text{ [}^\circ\text{C]}$$

The heat balance equation on the convection superheater is:

$$\dot{Q}_{SH1} = \dot{V}_{fg} \cdot c_{p,fg} \cdot (t_{fg}^{(1)} - t_{fg}^{(2)}) \text{ [kW]}$$

hence the outlet flue gas temperature from the convection superheater:

$$t_{fg}^{(2)} = t_{fg}^{(1)} - \frac{\dot{Q}_{SH1}}{\dot{V}_{fg} \cdot c_{p,fg}} = 865 - \frac{13506}{40.83 \cdot 1.41} = 630.4 \text{ [}^\circ\text{C]}$$

The steam inlet temperature to the convection superheater is the saturated steam temperature, because the convection superheater follows the drum in the steam path.

$$t_{steam}^{(1)} = t_s \Big|_{p=100[\text{bar}]} = 310.96 \text{ [}^\circ\text{C]}$$

The steam outlet temperature can be found using water-steam T-s diagrams or tables, knowing the pressure (100 [bar]) and the enthalpy before feed water injection (which is to regulate the outlet steam temperature), $h_1 = 3155.8 \text{ [kJ/kg]}$.

$$t_{steam}^{(2)} \cong 419 \text{ [}^\circ\text{C]}$$

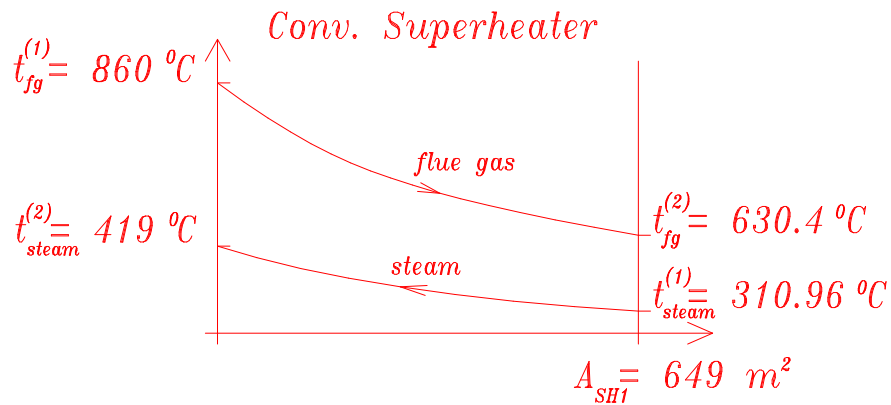


Fig. 6

Temperature variation along the surface of convective superheater

The logarithmic temperature difference of the heat exchange is defined by...

$$\overline{\Delta t_{Ln}^{SH1}} \equiv \frac{(t_{fg}^{(1)} - t_{steam}^{(2)}) - (t_{fg}^{(2)} - t_{steam}^{(1)})}{Ln \left(\frac{t_{fg}^{(1)} - t_{steam}^{(2)}}{t_{fg}^{(2)} - t_{steam}^{(1)}} \right)}$$

$$\overline{\Delta t_{Ln}^{SH1}} = \frac{(860 - 419) - (630.4 - 310.96)}{Ln \left(\frac{860 - 419}{630.4 - 310.96} \right)} \cong 377 \text{ [}^\circ\text{C]}$$

Knowing that the surface area of the heat exchanger is 649 [m²], the heat transfer coefficient is ...

$$k_{SH1} = \frac{\dot{Q}_{SH1}}{A_{SH1} \cdot \overline{\Delta t_{Ln}^{SH1}}} = \frac{13506 \cdot 10^3 \text{ [W]}}{649 \text{ [m}^2\text{]} \cdot 377 \text{ [}^\circ\text{C]}} = \underline{\underline{55.2 \text{ [W / m}^2\text{K]}}}$$

UNSOLVED PROBLEMS

8.) A boiler given with the following data...

- mass flow rate of fuel	:	720 [kg/h]	
- lower heating value of the fuel	:	13100 [kJ/kg]	
- stoichiometrically needed air	:	3.556	[m ³ /kg]
- stoichiometrically produced flue gas amount	:	4.243	[m ³ /kg]
- volume flow rate of the flue gas	:	8000 [m ³ /h]	
- boiler efficiency	:	75 [%]	

- Calculate the produced useful heat.
- How much is the excess air factor of the present operation?
- Calculate the adiabatic flame temperature of combustion.

(Results: 1965 [kW], 2.93 [-], 968 [°C])

9.) A fire tube boiler operates at 20 [bar]. After stopping the operation (all the valves are closed and firing stopped) the pressure decreased to 12 [bar] due to the cooling of the boiler. How much heat is needed to reach the operational pressure (12 [bar]) when we want to start the steam production again?

The volume of the boiler drum is 5 m³, and the total stored mass (steam + water) is 3000 kg.

State parameters at the two pressures...

Data at 20 [bar]:

saturated temperature	:	212.37	[°C]
saturated liquid enthalpy	:	908.57	[kJ/kg]
saturated liquid specific volume	:	1.1766	[dm ³ /kg]
saturated dry steam enthalpy	:	2797.4	[kJ/kg]
saturated dry steam specific volume	:	99.53	[dm ³ /kg]

Data at 12 [bar]:

saturated temperature	:	187.96	[°C]
saturated liquid enthalpy	:	798.42	[kJ/kg]
saturated liquid specific volume	:	1.1386	[dm ³ /kg]
saturated dry steam enthalpy	:	2783.4	[kJ/kg]
saturated dry steam specific volume	:	163.20	[dm ³ /kg]

(Result: 335.23 MJ)

10.) The nominal max. pressure of a fire tube boiler, which produces dry saturated steam, is 12 [bar]. It operates at max. load, but its pressure is only 10 [bar]. We want to reach the max. allowed pressure switching off a consumer with 2.16 [t/h] steam consumption.

- a.) How long does it take to reach the max. pressure?
- b.) How much additional fuel is needed to reach the max. pressure without limiting the steam consumption of the consumers?

Additional data

- volume of the drum	:	10	[m ³]
- stored total mass in the drum	:	7	[t]
- boiler efficiency	:	80	[%]
- lower heating value of the fuel	:	40	[MJ/kg]

(Results: 152 [s], 0.052 [kg/s])

11.) We know the following data about a water tube boiler.

Evaporator

- outer wall temperature of the surfaces	:	287	[°C]
- temperature drop in the wall	:	37	[°C]
- surface area	:	100	[m ²]
- radiation number $(c_s = \varepsilon \cdot \sigma \cdot \varphi_{12})$:	4	[W/m ² K ⁴]
- feed water inlet temperature	:	150	[°C]
- outlet superheated steam temperature	:	350	[°C]
- combustion temperature	:	1400	[°C]
- mass flow rate of fuel	:	1.4	[kg/s]
- lower heating value of the fuel	:	39608	[kJ/kg]
- latent heat of evaporation	:	r	[kJ/kg]

$$r(t_s [^{\circ}C]) = 179 \cdot \sqrt{374 - t_s} - 2.25 \cdot (374 - t_s) \text{ [kJ/kg]}$$

t_s : evaporation temperature

- specific heat of feed water	:	4.5343	[kJ/kgK]
- specific heat of steam	:	2.90	[kJ/kgK]

- a.) Calculate the useful heats in the boiler (ECO, evaporator, superheater).
- b.) How much is the boiler efficiency?

(Results: 8184.41 [kW], 30942.63 [kW], 5234.50 [kW], 80 [%])