Energy Conversion & Power Generation from Solid Fuel

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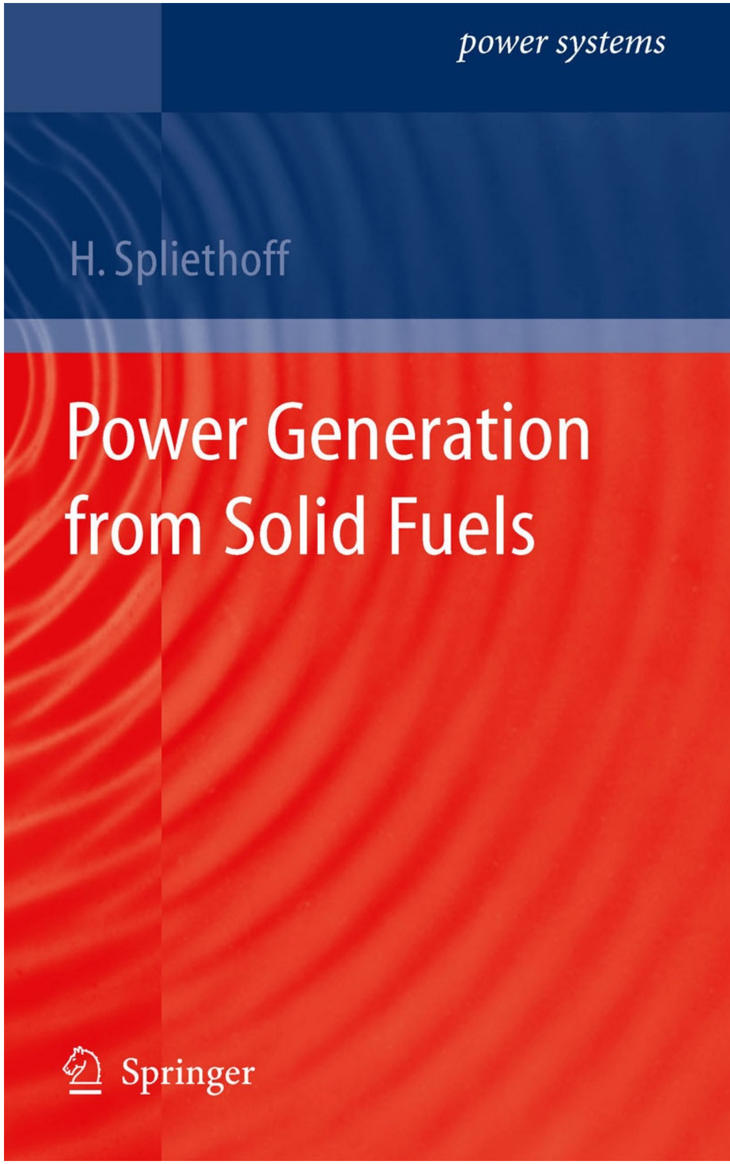
Main-text : Internal Combustion Engine

1. Motivation
2. Solid Fuels
3. Thermodynamic Fundamentals
4. Steam Power Station for Electricity and Heat Generation
5. Combustion System for Solid Fuels
6. Power Gen. from Biomass &Waste
7. Combined Cycle Power Plants
8. Carbon Capture and Storage(CCS)



Steam Power Station for Electricity and Heat Generation

1. Pulverised Hard Coal Fired Steam Power Plants
2. Steam Generators
3. Design of a Condensation Power Plant
4. Possibilities for Efficiency Increases in the Development of a Steam Power Plant
5. Effects on Steam Generator Construction
6. Developments – State of the Art and Future

Ref./ Homework

4.1 Pulverised Hard Coal Fired Steam Power Plants

4.1.1 Energy Conversion and System Components

Power plants for process heat generation or combined heat and power (CHP) stations generate electrical power, steam and district heat as their main products. Simultaneous heat and/or steam utilisation, along with power generation, is an effective method to diminish waste heat losses at the cold end of the turbine.

Main components of a modern coal-fired power plant can be divided into the following plant sections:

- Fuel supply and preparation
- Steam generator with furnace
- Turbine and generator
- Heat rejection unit, condenser, cooling tower
- Units for emissions reduction and disposal

The major loss, of 50% of the fuel heat input, occurs during the energy conversion in the turbine. Further significant losses occur in the steam generator, mainly as flue gas losses of about 6%. The auxiliary power requirements of about 9% of the fuel energy input add to these losses.

4.1.1 Energy Conversion and System Components

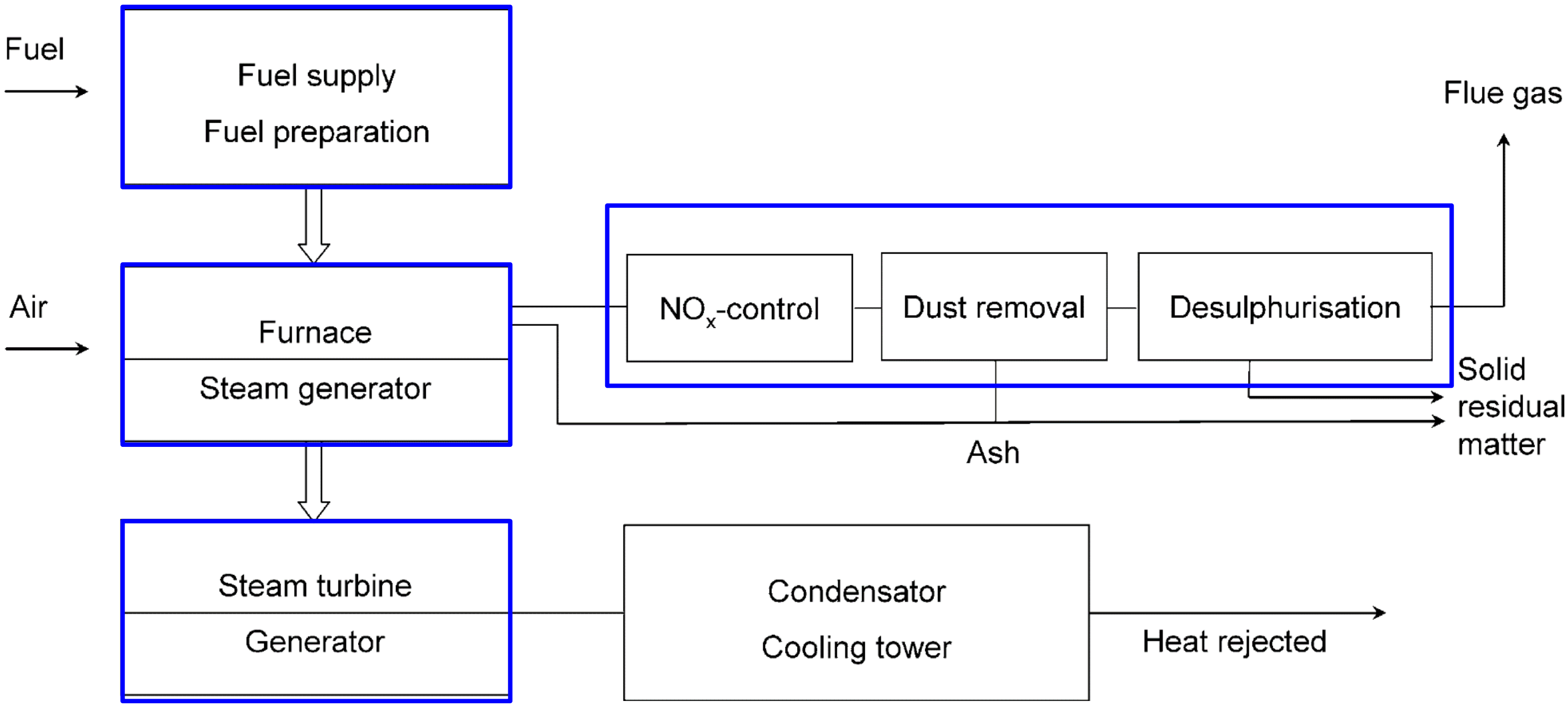
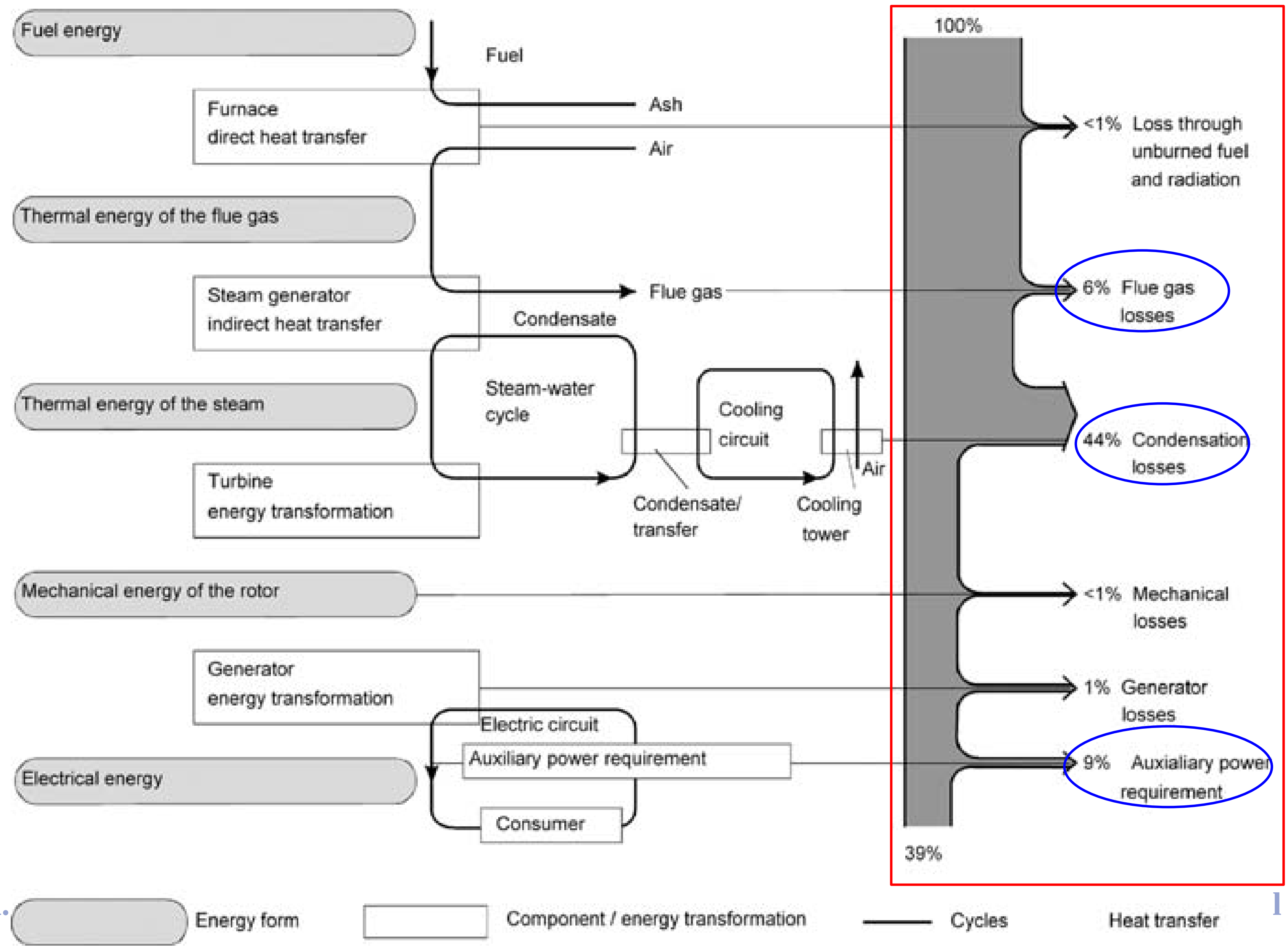


Fig. 4.1 Components of a steam power plant

4.1.1 Energy Conversion and System Components



4.1.2 Design of a Condensation Power Plant

- The fuel, coal, feeders transport coal from the bunkers to the mills for drying and pulverising. The combined drying and pulverising process of hard coal fired furnaces uses hot air that is heated up to 350–400◦C in an air preheater.
- The transport air is further used in the combustion process as primary air. Complete combustion of the fuel is achieved by injecting secondary air, heated in the preheater to 300–400◦C, into the furnace; Wall firing 610K, Standard case 540K.
- In the furnace, the pulverised coal burns almost completely, radiating heat to the furnace walls, producing flame temperatures between 1,400 and 1,600◦C.
- After the flue gases are cooled to about 1,200–1,300◦C at the end of the furnace, they are further cooled down by the convective heating surfaces of the superheater (SH), the reheater (RH) and the feed water preheater, also called the economiser. Then nitrogen is removed from the flue gas in a DeNO*x* unit at a temperature range of 300–400◦C.
- In the air heaters the flue gases transfer their residual heat to the combustion air, during which they are cooled to the exit flue gas temperature of the steam generator.
- The flue gas is conducted through an electrostatic precipitator (ESP) to remove dust and, through a flue gas desulphurisation unit, to meet the allowed sulphur dioxide emission standards. With flue gas desulphurisation and DeNO*x* units from the outset, one or more induced-draught fans are connected in parallel to overcome the pressure loss of all installations and components in the flue gas train.

4.1.2 Design of a Condensation Power Plant

- Condensate pumps transport the condensate to the feed water tanks via low pressure preheaters (LP preheaters), which are heated by steam from the lower pressure-staged turbine extraction.
- The high-pressure feed water pump sets the operating pressure in the water – steam section of the boiler and transports the feed water to the boiler inlet via the high-pressure preheaters, which are heated by steam from the upper pressure turbine extraction stages. The feed water is preheated to the entry temperature of the boiler in 6–9 stages.
- The last preheating stage before the boiler is fed with steam taken from the cold reheater in a conventional design or from the HP turbine extraction in an advanced design.
- In the boiler, the preheated feed water is further heated in the economiser, the last convective heating surface in the flue gas path, and then conducted to the evaporator heat exchanger surface.
- The level of the turbine entry temperature is slightly lower, by the amount of the temperature drop in the connecting high-pressure steam piping. After partial expansion in the HP turbine, most power plants heat the steam up to levels such as the live steam temperature or higher in a so-called reheater (exchanging heat with the flue gas). Higher temperatures in the reheater are possible due to the lower pressure.

4.1.2 Design of a Condensation Power Plant

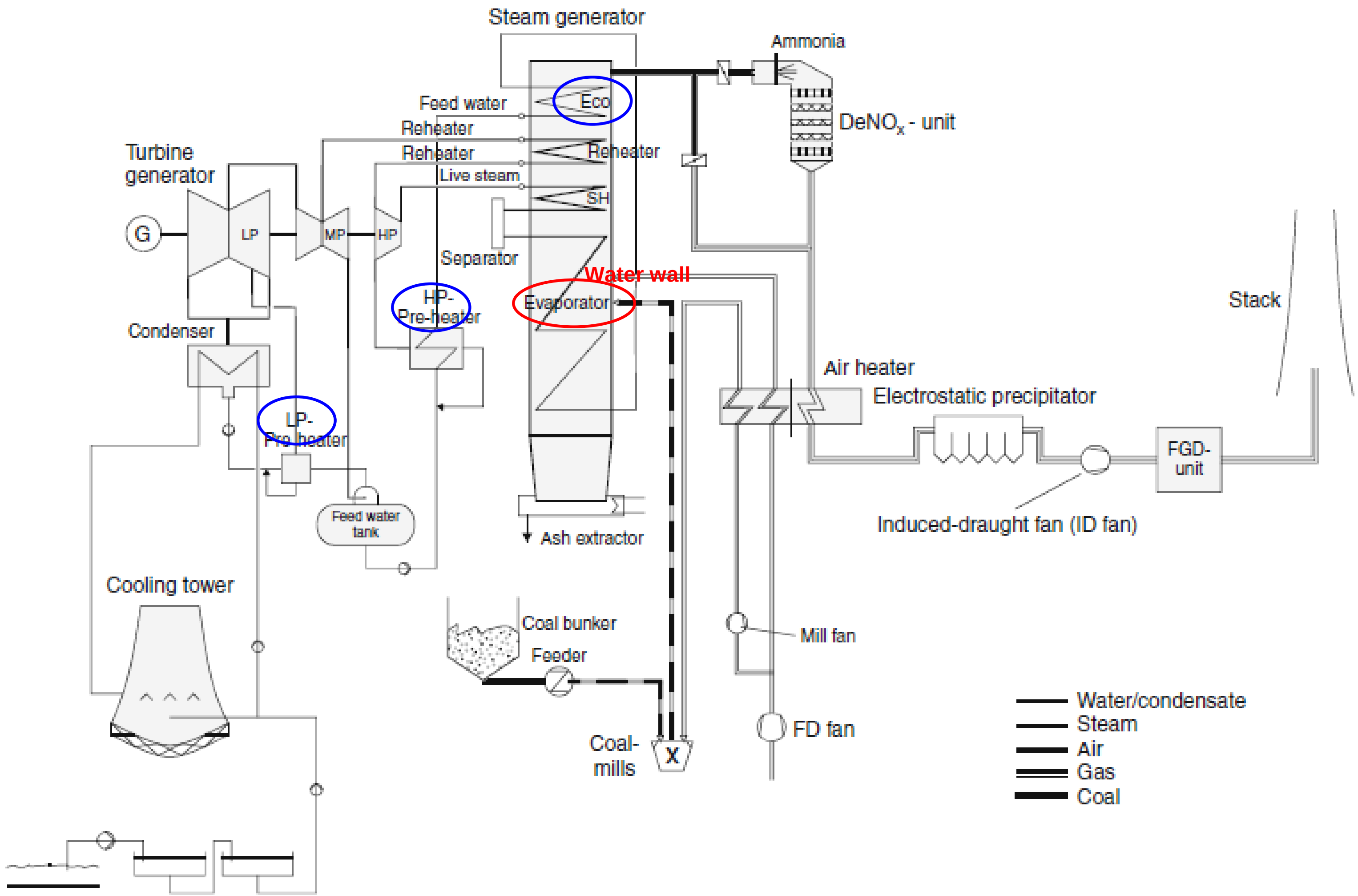


Fig. 4.3 Schematic diagram of a hard coal fired thermal power station

4.1.2 Design of a Condensation Power Plant

영흥화력 1,2호기 동영상

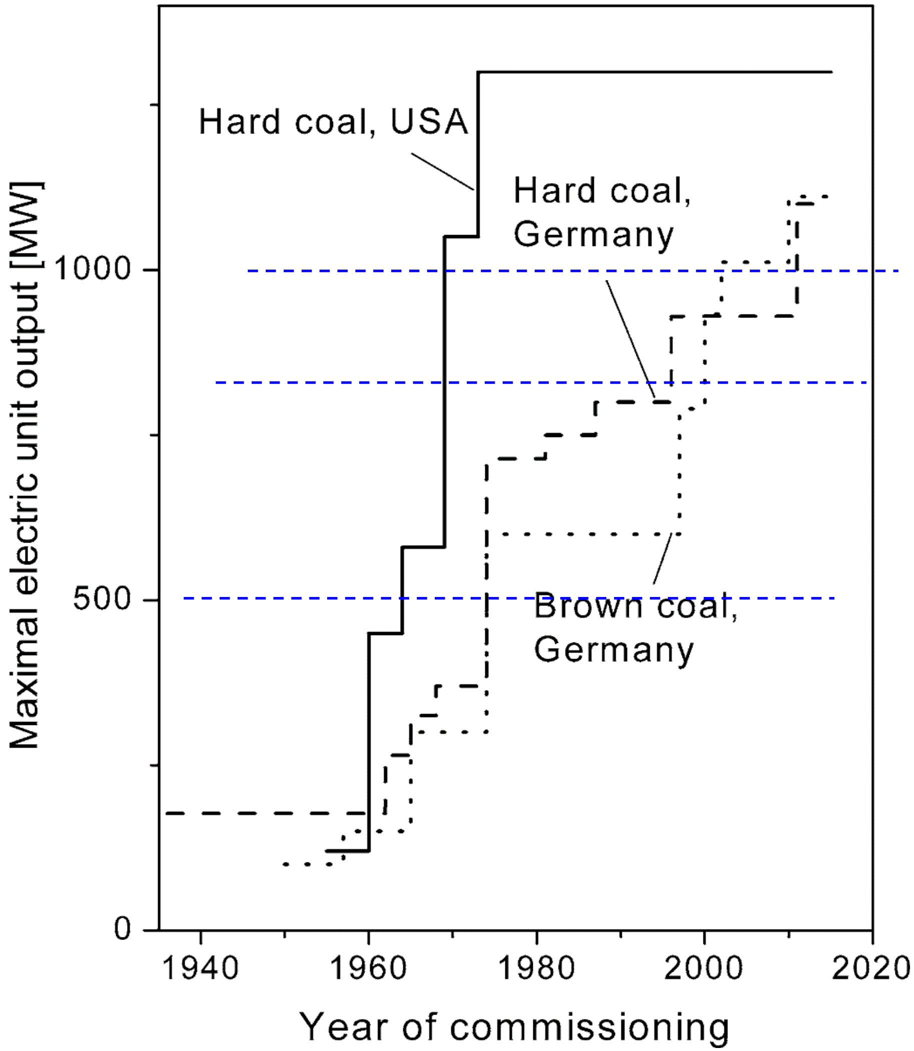
Fig. 4.3 Schematic diagram of a hard coal fired thermal power station

4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency

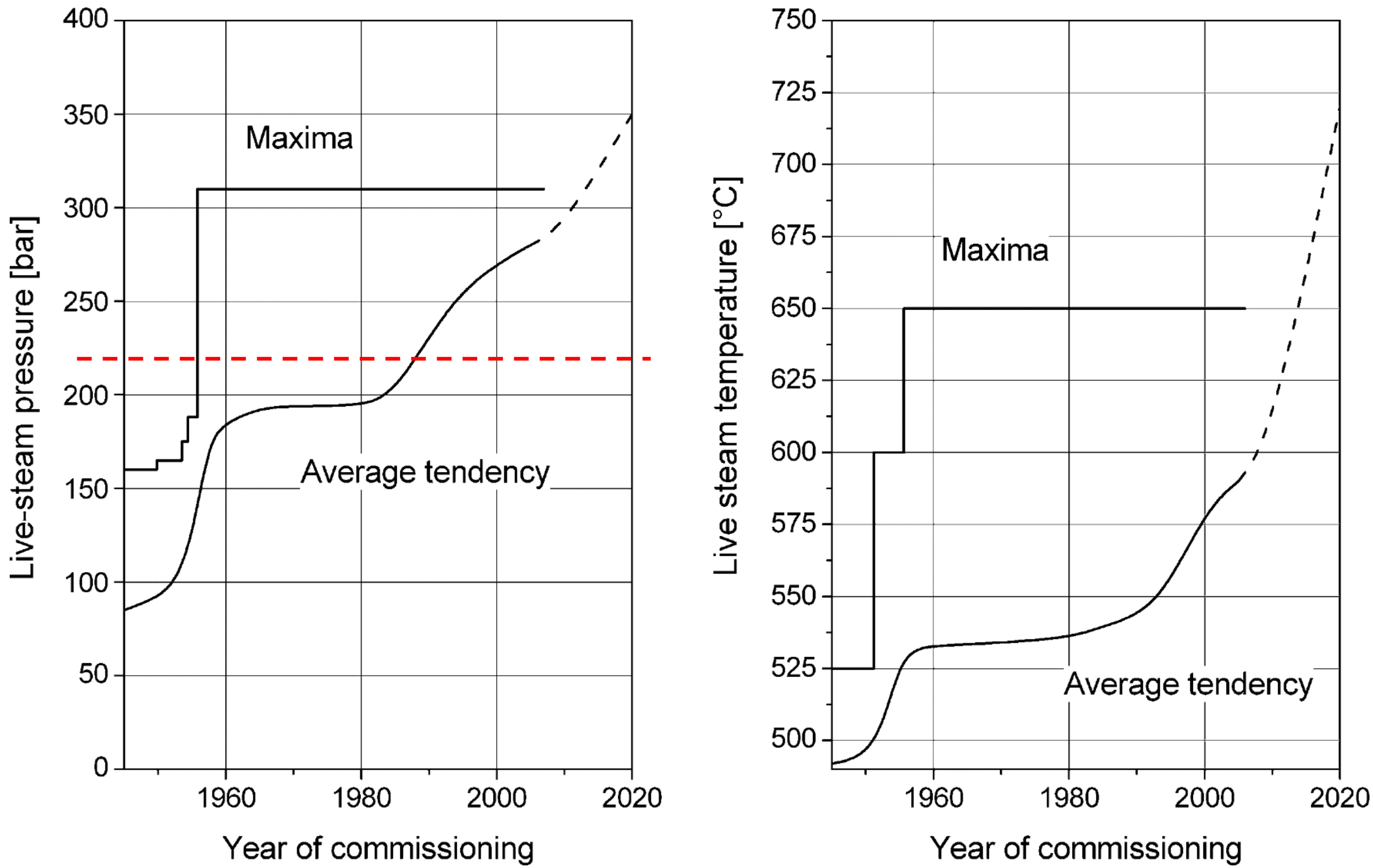
- The block power station was born out of the need for higher power plant capacities (due to increasing energy demands), changing expectations with respect to lower investment costs and the desire for a higher reliability in power supply. Besides other parameters, it is, in particular, the • unit output, • efficiency and • availability that describe the development of the block power station unit.

- From the early 1950s, condensation power plants were built as block units with simple reheating for base and for intermediate loads. At the beginning, the unit capacities were some 60MW or more; live steam and reheater temperatures were at 525◦C, while the live steam pressure was at about 125 bar.
- The maximum block capacity rose step by step, the power station unit has been supplemented by additional components and plants. Today, the largest unit capacities are 1,010MW in Europe, which will increase to 1,100MW by 2010, and 1,300MW in the USA (Eitz 1996; Smith 1996). Conventional live steam conditions proven in operation are 180–250 bar and 540◦C, with reheater temperatures at 540◦C as well. All over the world, one can see a trend towards higher live steam conditions.

4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency

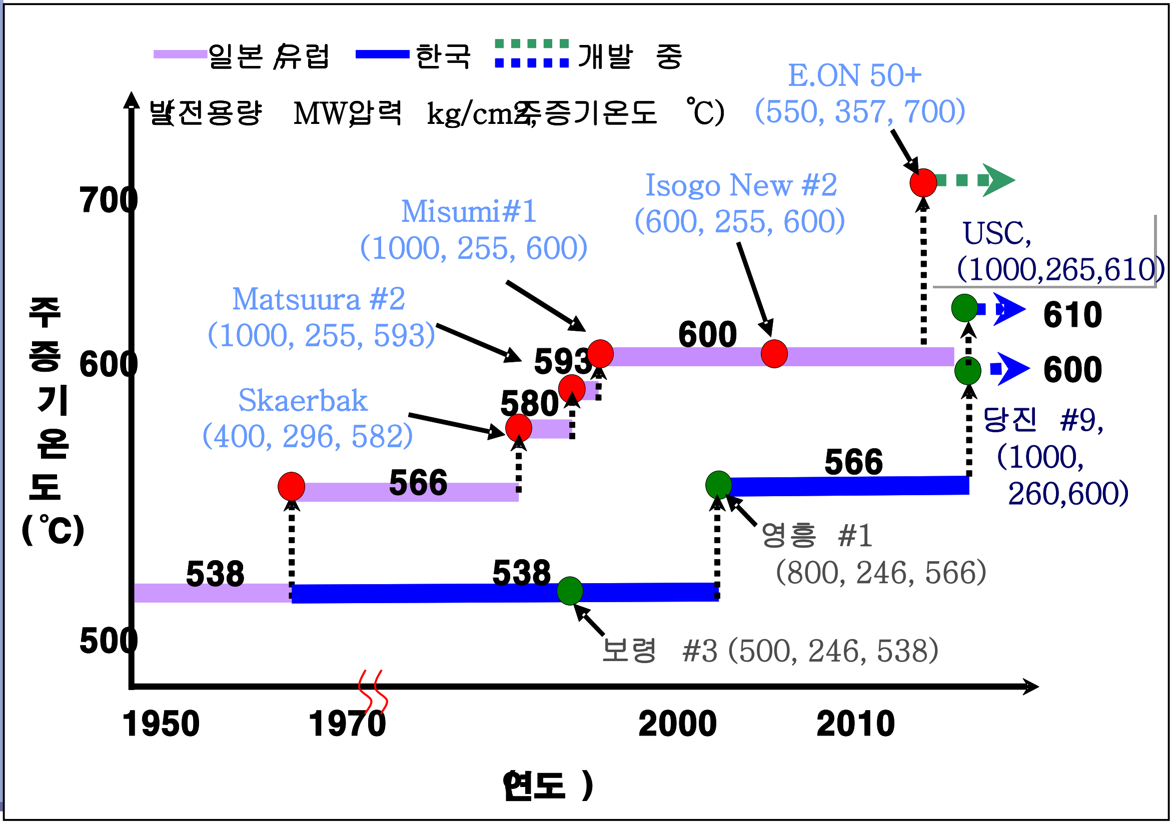


4.1.3 Development History of Power Plants –
Correlation Between Unit Size, Availability and Efficiency



Critical : 220 bar , 374C

4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency

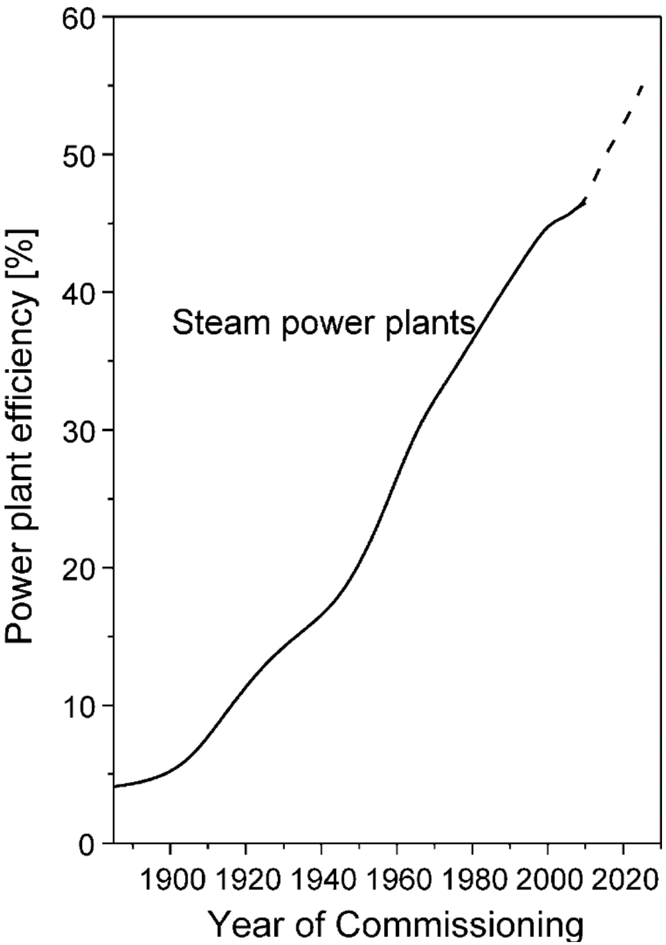


USC 기술 수준

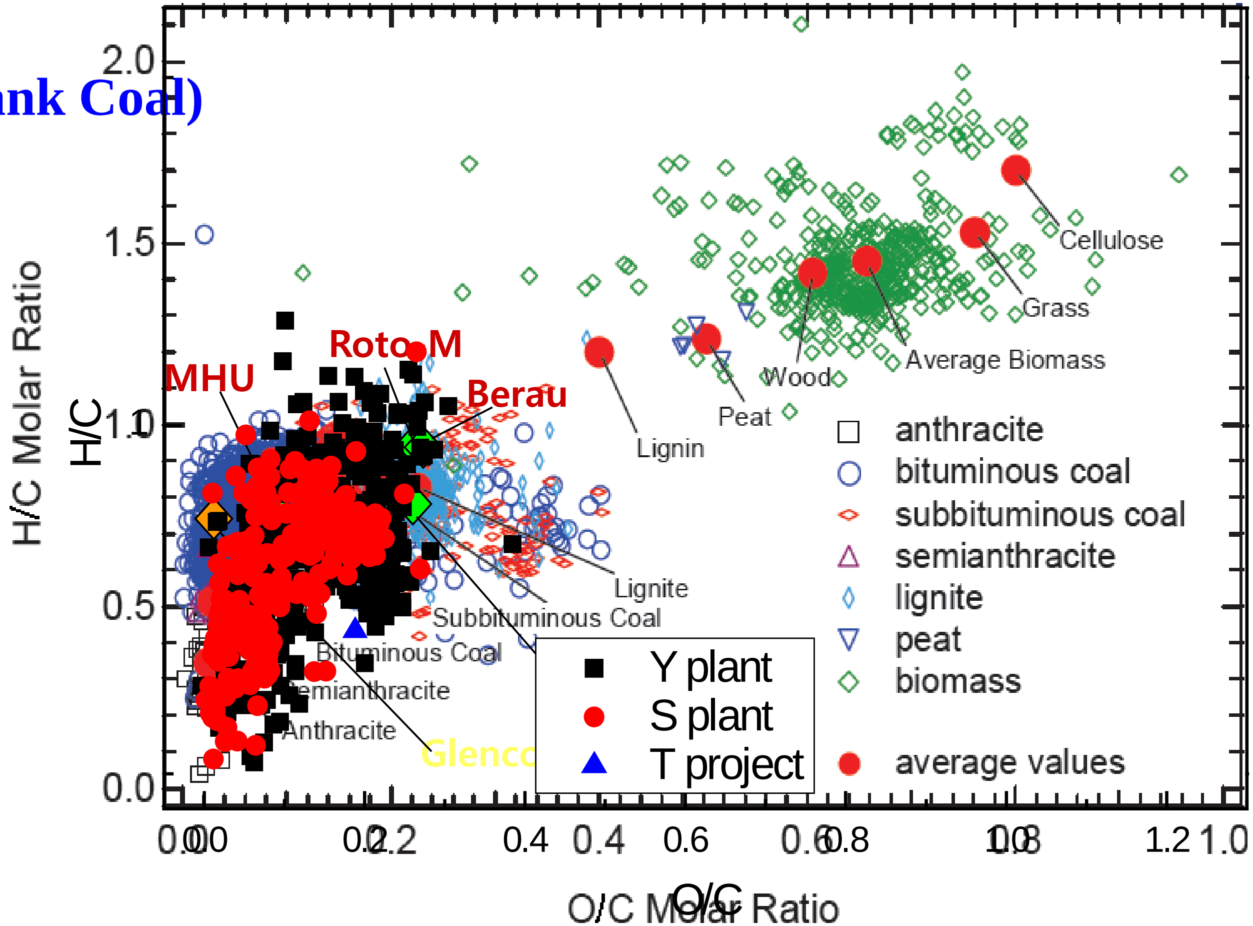
- 일본 : Isogo New No.2
  - 600MW, 255kg/cm2, 600/620℃, COD 2009년
- 유럽 : Niederaussem K
  - 1000MW, 278kg/cm2, 580/600℃, COD 2002년
- 한국 : 실증사업(중부발전)
  - 1000MW, 265kg/cm2, 610/621℃, COD 2015년

4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency

- Higher efficiencies of large units can also be explained physically: specific surface heat losses of boilers and losses of rotating machinery due to leakiness diminish with higher capacities.
- Availability of technology becomes important with increasing capacities, the need for more pollution control equipment and the desire for technical developments towards higher efficiency levels.
- Besides being economically significant, availability also has an environmental impact. The lowest CO2 emission level is achieved by a generation system when the power plants with the highest efficiency are of comparably high availability.
- In the 1970s, the efficiency was further enhanced along with increasing unit sizes from 150 via 300 to more than 600MW. At the same time, the availability rate was increased and thus the effect of the efficiency enhancement improved.

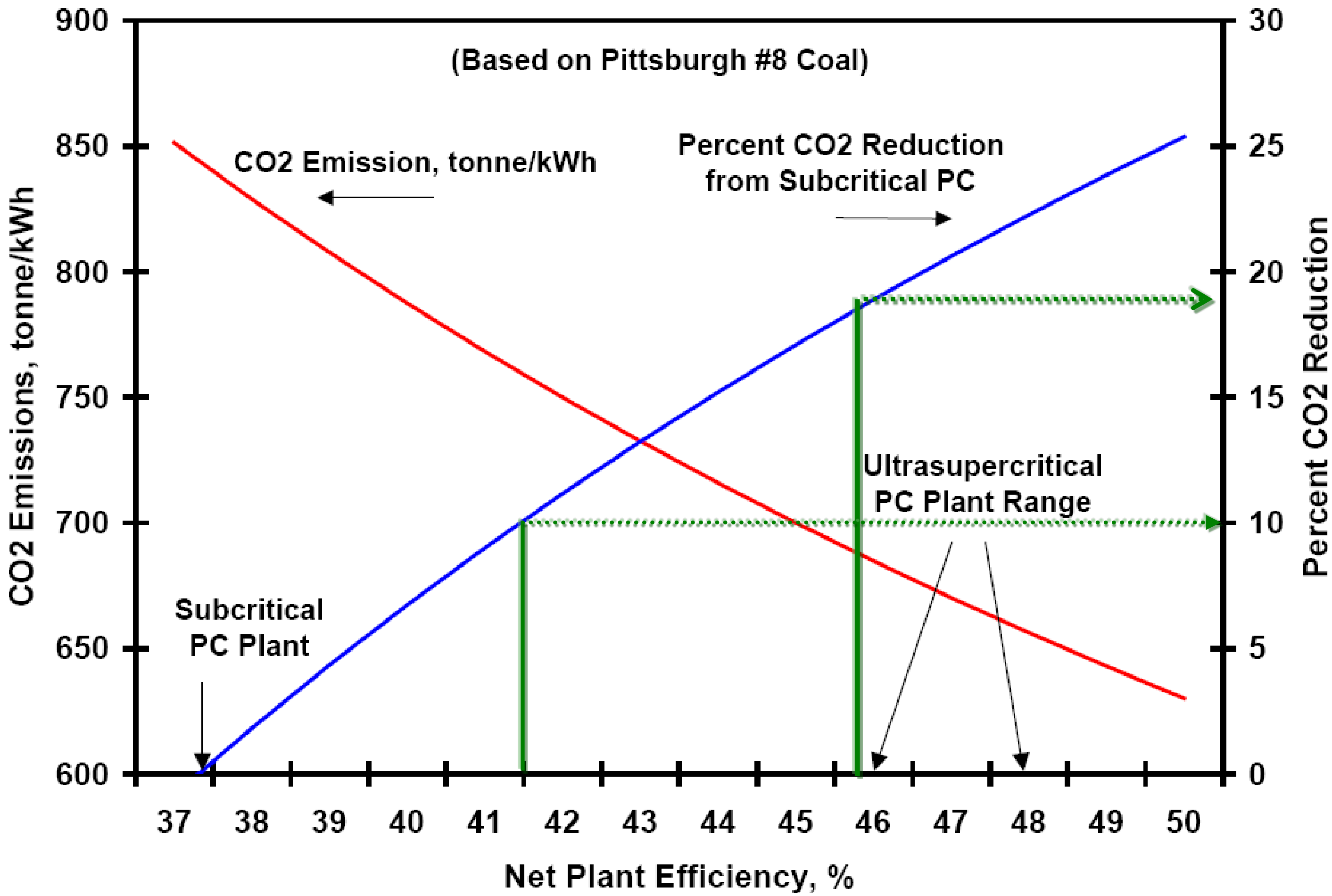


LRC
(Low Rank Coal)



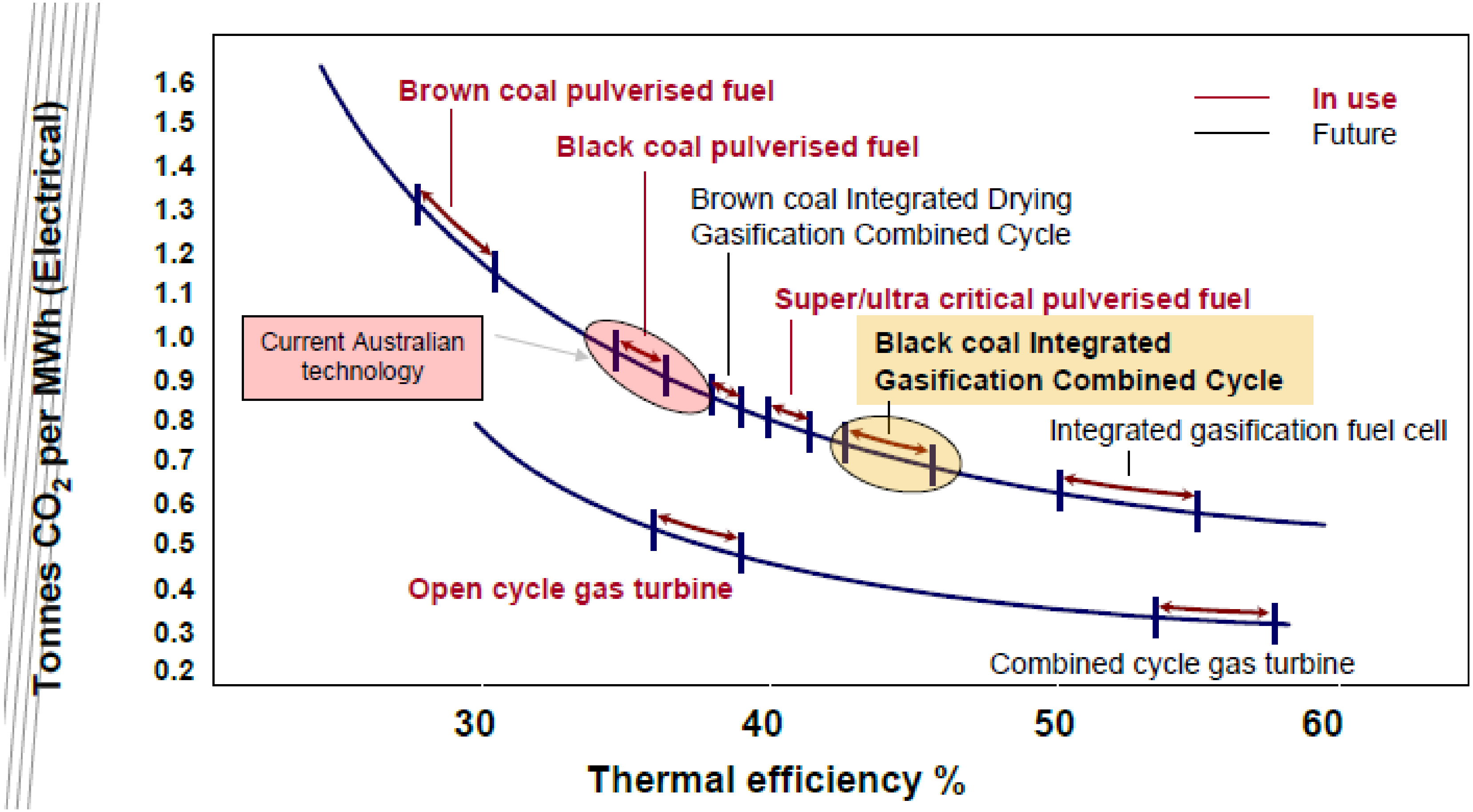
Coal	Proximate (wt. %)				Ultimate (daf, wt. %)					HHV (Kcal/kg)
	Moi.	VM	FC	Ash	C	H	S	N	O	
Collie	24.4	27.1	38.5	10	74.8	4.4	0.9	1.3	18.8	4538
GLENCORE(인니)	19.67	35.5	37.18	7.6	74.0	5.0	1.0	2.0	18.0	4603
ABN(인니)	15.77	38.47	38.86	6.91	74.0	5.0	1.0	1.0	19.0	4613

4.1.3 Development History of Power Plants –
Correlation Between Unit Size, Availability and Efficiency

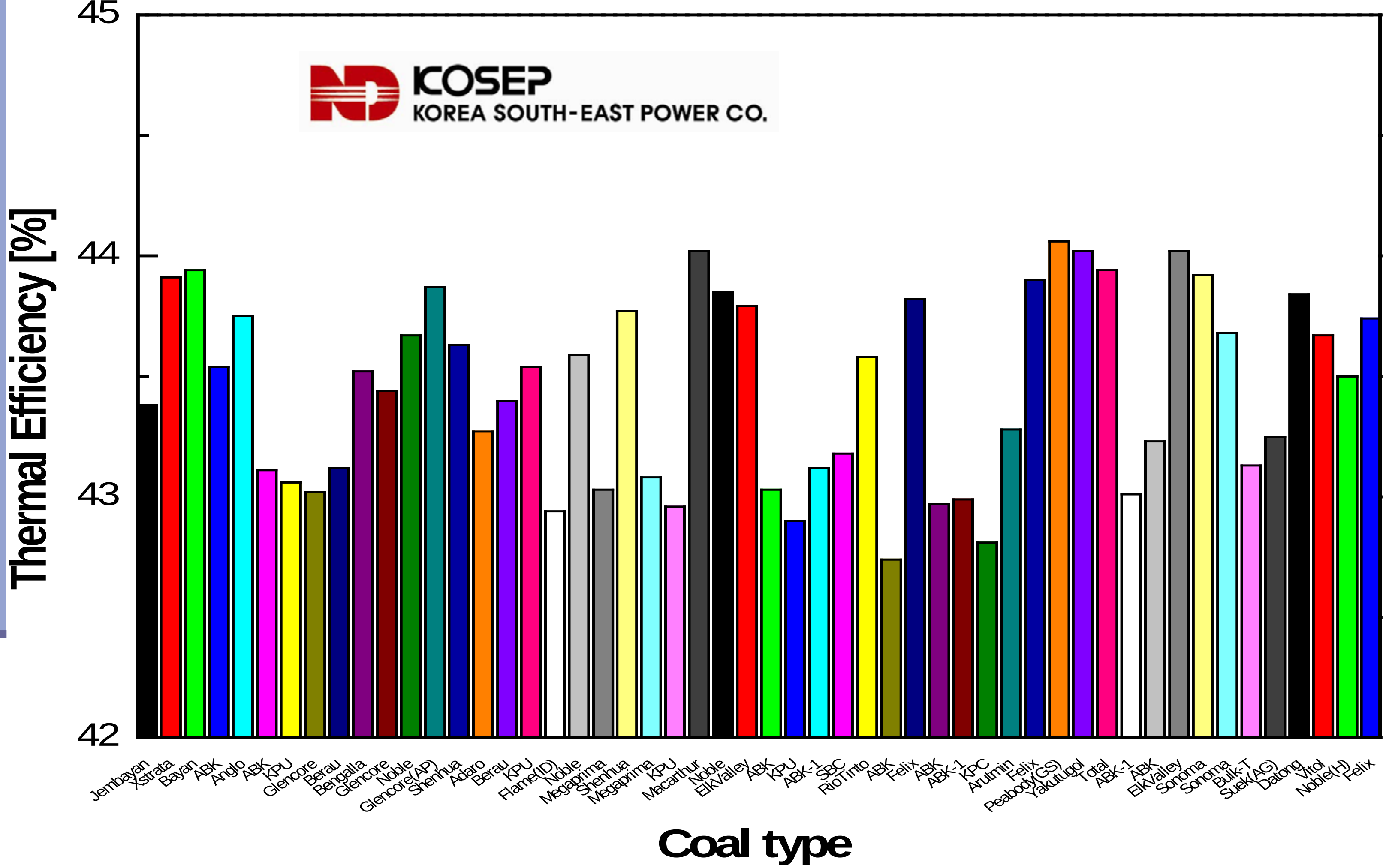


Combustion Technology University Alliance Workshop, August 4, 2003, Columbus, OH

EPRI


Tonnes CO_2 per MWh (Electrical)
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
Brown coal pulverised fuel
Black coal pulverised fuel
Brown coal Integrated Drying Gasification Combined Cycle
Super/ultra critical pulverised fuel
Black coal Integrated Gasification Combined Cycle
Integrated gasification fuel cell
Current Australian technology
Open cycle gas turbine
Combined cycle gas turbine
In use
Future
30
40
50
60
Thermal efficiency %

4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency



4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency

❖ Indirect method

$$\eta_B = 100 - \frac{(L_T \pm \Delta L)}{H_f + B_e} \times 100\,(\%)$$

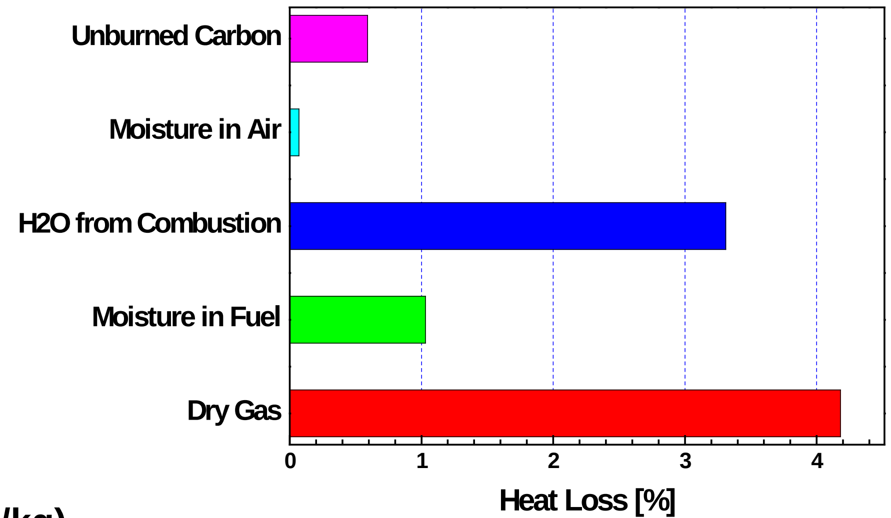
η_B : Boiler Efficiency (%)

L_T : Total Heat Loss from Boiler (kcal/kg)

ΔL : Heat Loss Correction (kcal/kg)

H_f : Heat in Fuel, Higher Heating Value (kcal/kg)

B_e : Total Heat Credit (kcal/kg)



Heat loss for Yonghung power plant

❖ Heat Losses

1) **Unburned Carbon Loss:** $L_{UC} = Cu \times LHV\ (kcal/kg\ \text{Fuel})$ **-> LOI reduction**

2) **Dry Flue Gas Loss:** $L_G = W_G^{'} \times cpg^{'} \times (TGO - TRA)$ **-> Combustion kinetics**

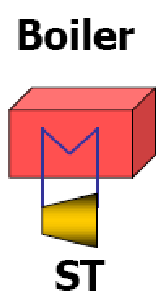
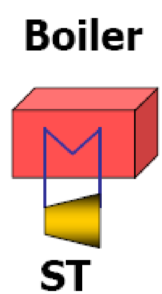
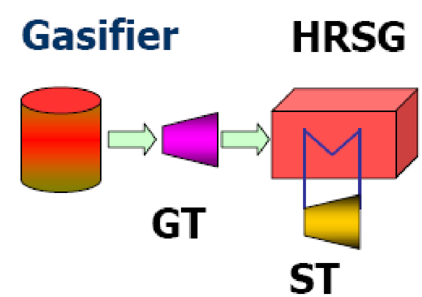
3) **Moisture Loss in Fuel:** $L_{mf} = m_f \times (He - HRW)$ **-> Drying technology**

4) **Hydrogen Combustion** $L_H = 8.936 \times H \times (He - HRW)$ **-> Combustion kinetics**

4.1.3 Development History of Power Plants – Correlation Between Unit Size, Availability and Efficiency

○ **Future development of the high-efficiency coal fired thermal power generation**

- **A-USC: Advanced ultra super critical pressure power generation**
- **IGCC: Integrated Coal Gasification Combined Cycle**

	USC	A-USC	IGCC(1,500degC)
Configuration			
Thermal Efficiency	42%	46%	46~48%
CO2 Emission Reduction	Base	▲11%	▲13%

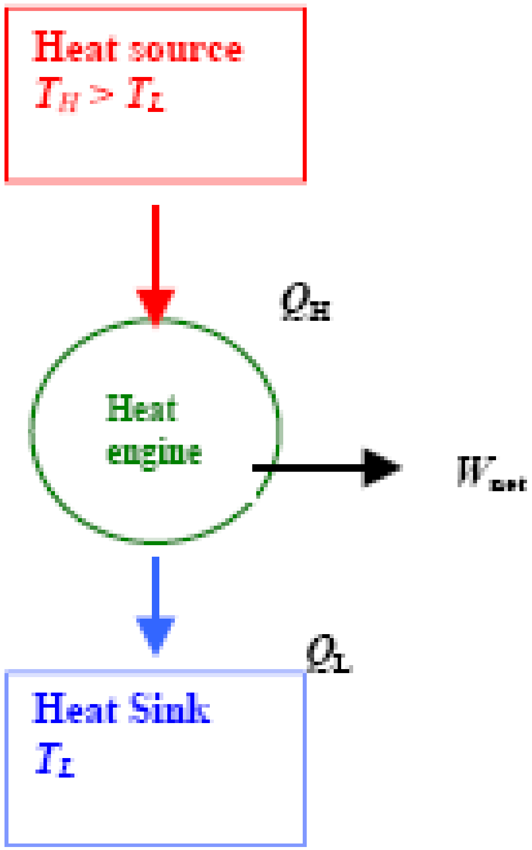
Thermodynamic Fundamentals

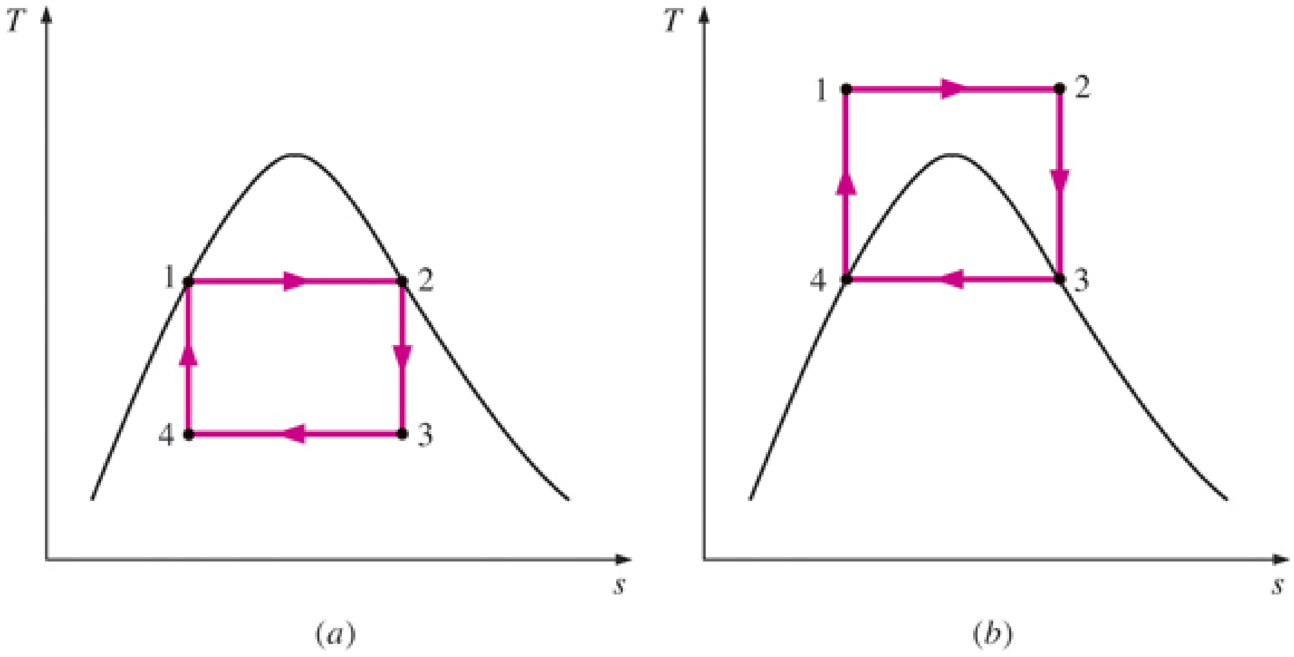
1. Cycles
   - Carnot Cycle / Joule-Thomson Process
   - Clausius-Rankine Cycle
2. Steam Power Cycle : Exergy Consideration
   - Steam Generation Exergy Efficiencies
   - Energy and Exergy Cycle Efficiencies
   - Total Cycle Efficiency

Ref./ Homework

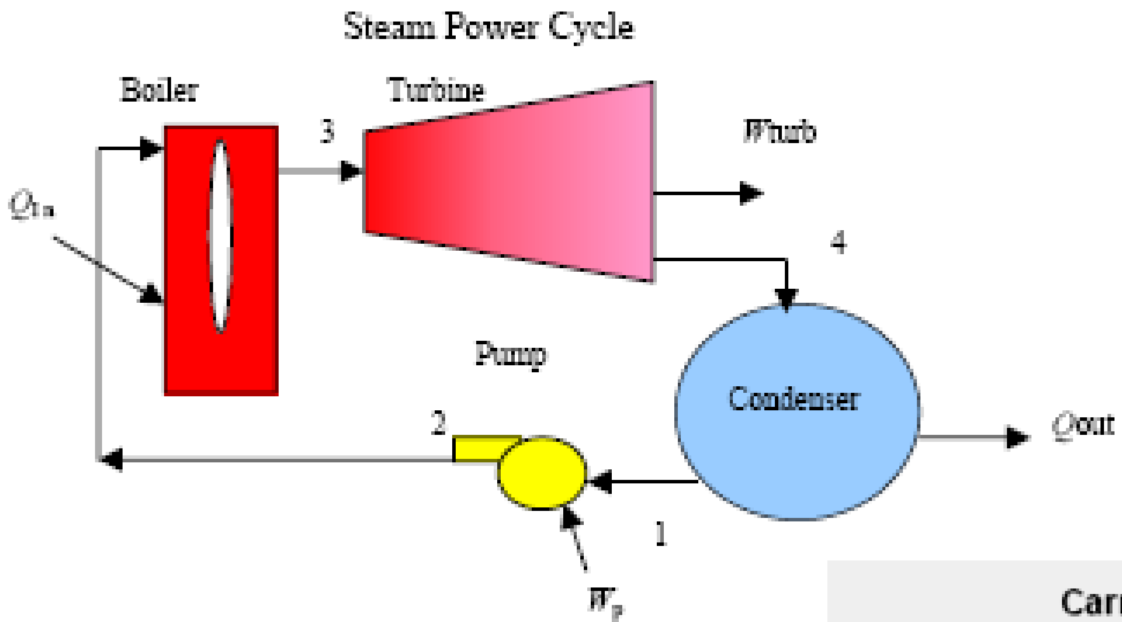
3.1 Cycles

Two Carnot vapor cycles and its Engineering value

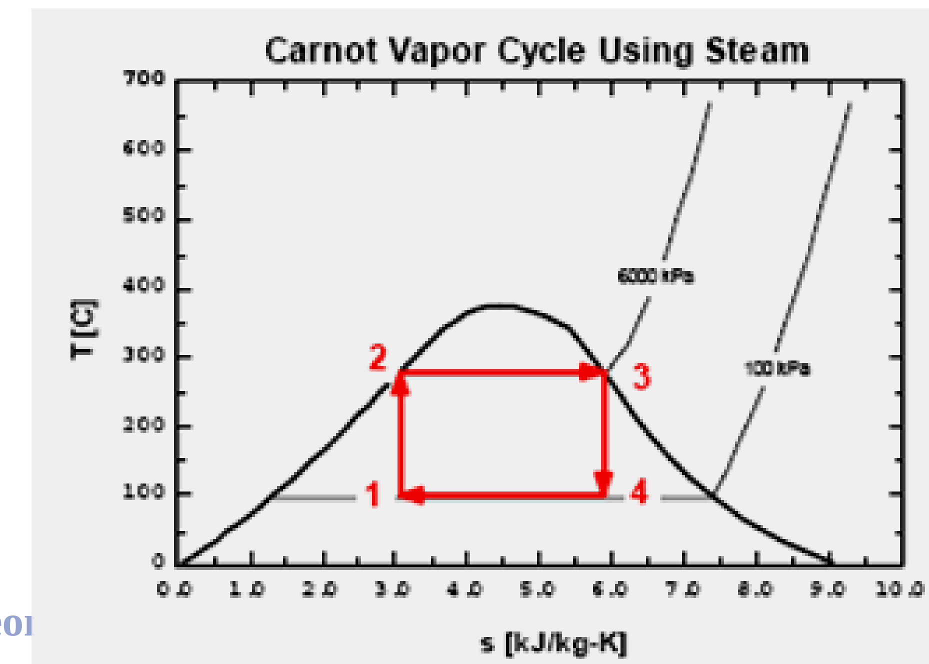




3.1.1 Carnot Cycle

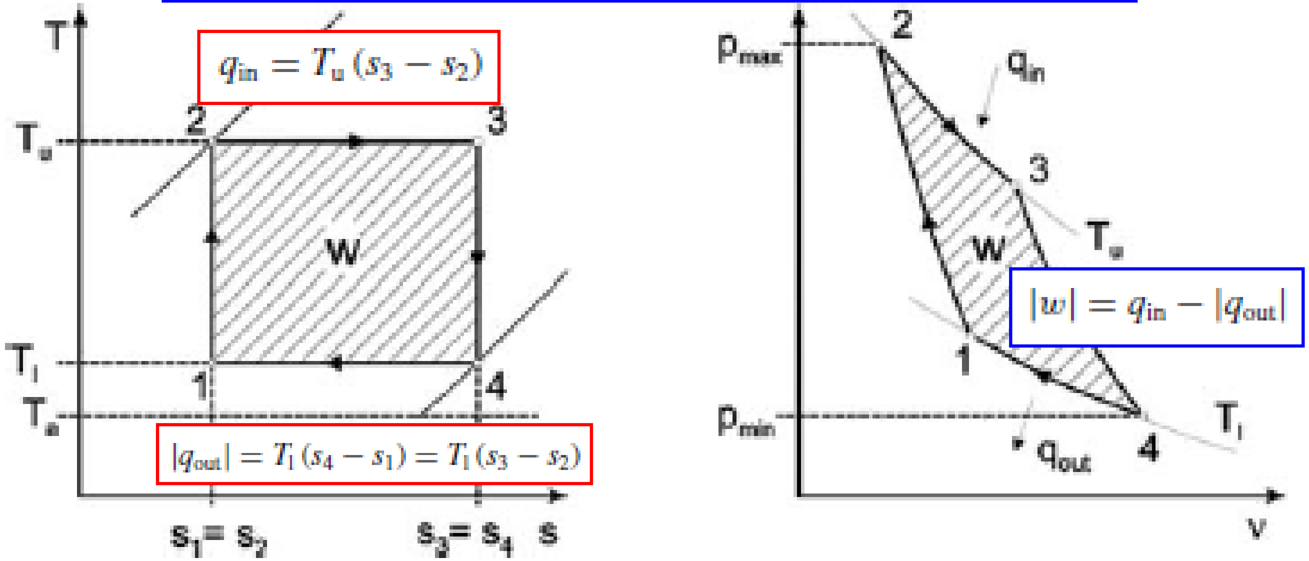


$$\eta_{th} = \frac{|w|}{q_{in}} = \frac{q_{in} - |q_{out}|}{q_{in}} = 1 - \frac{|q_{out}|}{q_{in}}$$



3.1.1 Carnot Cycle

$$\eta_{th} = \frac{T_u(s_3 - s_2) - T_l(s_3 - s_2)}{T_u(s_3 - s_2)} = \frac{T_u - T_l}{T_u} = 1 - \frac{T_l}{T_u}$$



$$q_{in} = T_u(s_3 - s_2)$$

$$|q_{out}| = T_l(s_4 - s_1) = T_l(s_3 - s_2)$$

$$|w| = q_{in} - |q_{out}|$$

Fig. 3.1 Carnot cycle $T - s$ and $p - V$ diagrams

1–2: isentropic compression with work input w_{12},
2–3: isothermal expansion at a constant upper process temperature T_u with heat input $q_{23} = q_{in}$,
3–4: isentropic expansion with work output w_{34},
4–1: isothermal compression at a constant lower process temperature T_l with heat output $q_{41} = q_{out}$.

3.1.2 Joule-Thomson Process

The Joule–Thomson process is the idealised reference process for gas turbines. A simple, open gas turbine process, shown in Fig. 3.2, consists of a compressor, a combustion chamber and a gas turbine.

For the ideal Joule–Thomson process, the assumption is that both the compression and the expansion processes are isentropic, i.e. reversible.

The Joule–Thomson process therefore consists of two isentropes and two isobars.

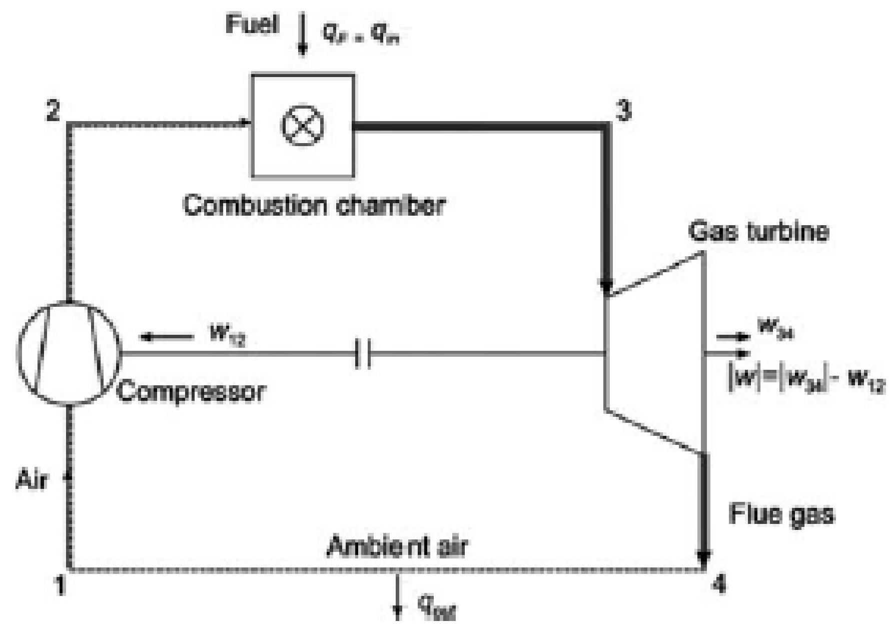
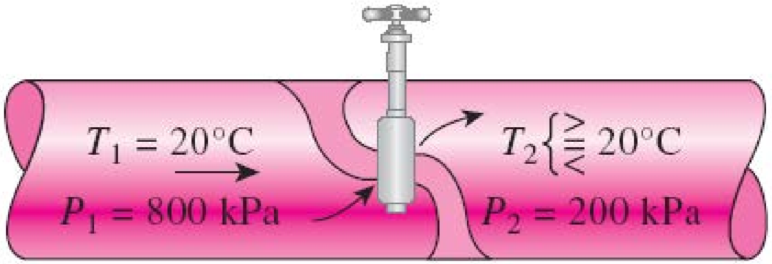


Fig. 3.2 Schematic diagram of an open gas turbine process

5. The Joule-Thompson Coefficient



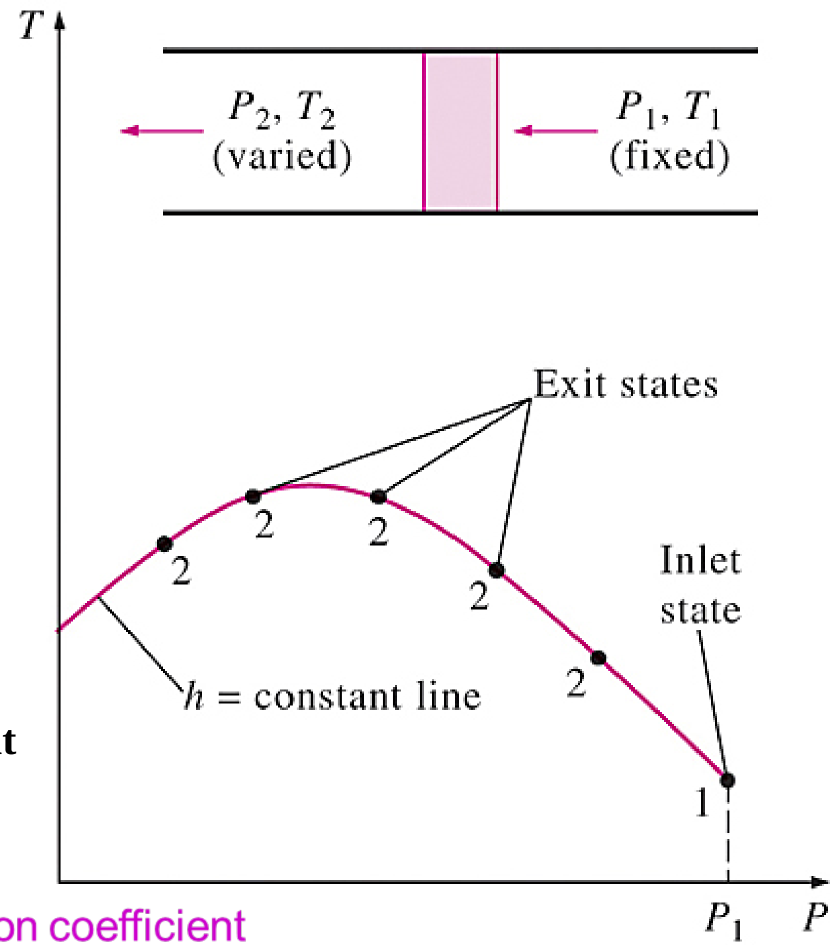
The temperature of a fluid may increase, decrease, or remain constant during a throttling process (h=constant)

Joule-Thomson coefficient

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h$$

$$\mu_{JT} \begin{cases} < 0 & \textbf{Temperature increases} \\ = 0 & \textbf{Temperature remains constant} \\ > 0 & \textbf{Temperature decreases} \end{cases}$$

The development of an *h* = constant line on a T-*P* diagram.



The Joule-Thomson coefficient represents the slope of *h* = constant lines on a *T-P* diagram.

Chung H. Jeon

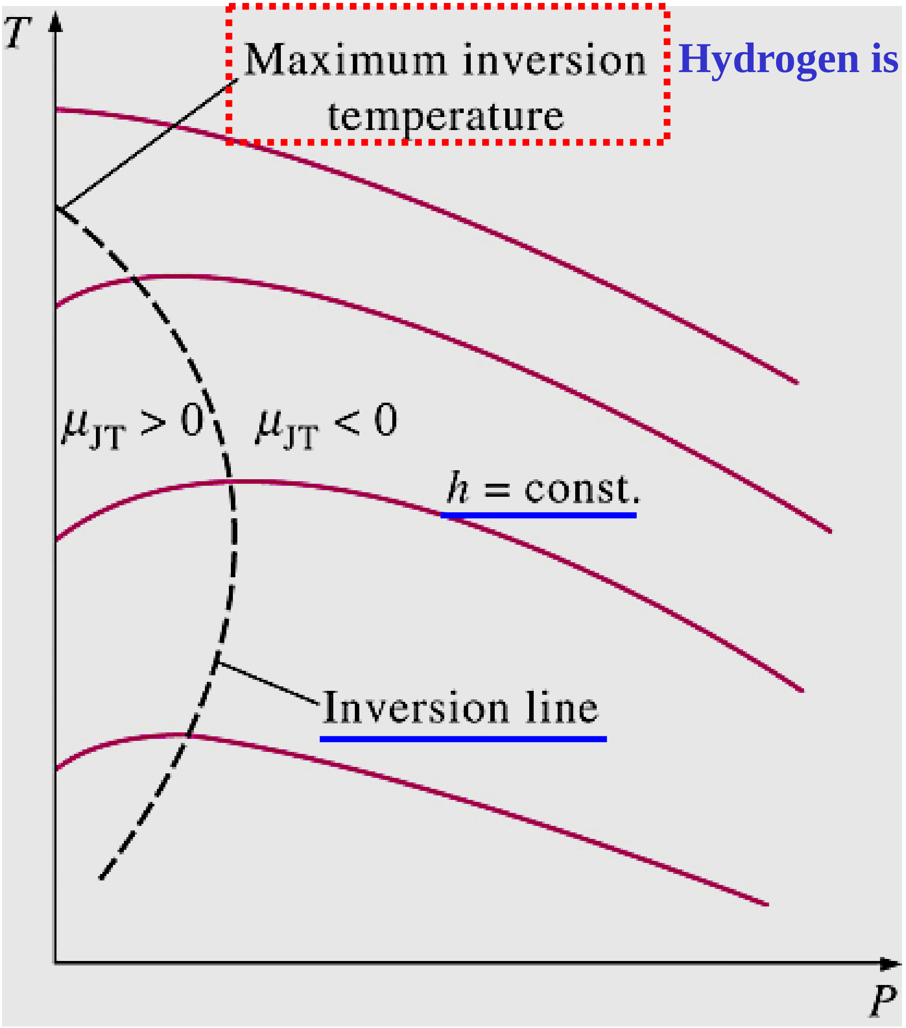
Constant-enthalpy lines of a substance on a *T-P* diagram

$$dh = c_P dT + \left[v - T\left(\frac{\partial v}{\partial T}\right)_P \right] dP$$

$$h = const. \rightarrow dh = 0$$

$$-\frac{1}{c_P}\left[v - T\left(\frac{\partial v}{\partial T}\right)_P \right] = \left(\frac{\partial T}{\partial P}\right)_h = \mu_{JT}$$

$$\mu_{JT_{IDEALGAS}} = -\frac{1}{c_P}\left[v - T\frac{R}{P} \right] = 0$$



Hydrogen is -68

EXAMPLE 12–10 Joule-Thomson Coefficient of an Ideal Gas

Show that the Joule-Thomson coefficient of an ideal gas is zero.

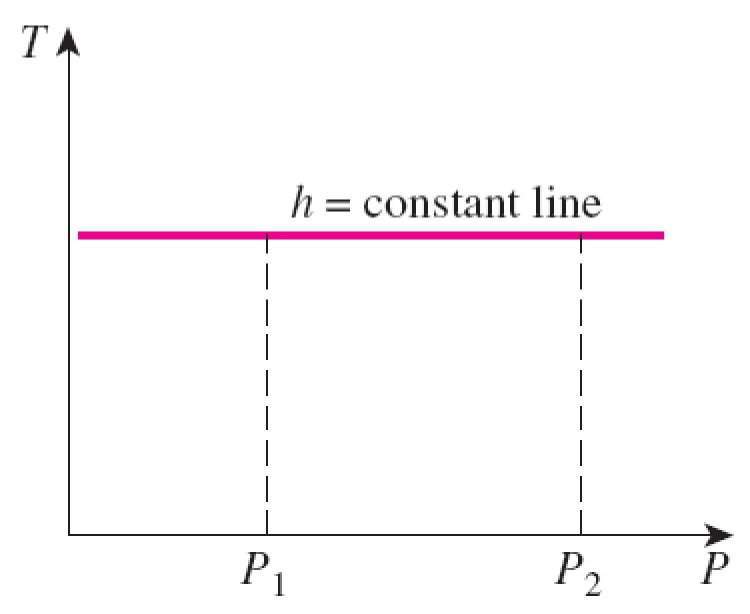
Solution It is to be shown that $\mu_{JT} = 0$ for an ideal gas.

Analysis For an ideal gas $v = RT/P$, and thus

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

Substituting this into Eq. 12–52 yields

$$\mu_{JT} = \frac{-1}{c_p}\left[v - T\left(\frac{\partial v}{\partial T}\right)_P\right] = \frac{-1}{c_p}\left[v - T\frac{R}{P}\right] = -\frac{1}{c_p}(v - v) = 0$$



The temperature of an ideal gas remains constant during a throttling process since h = constant and T = constant lines on a T-P diagram coincide.

3.1.2 Joule-Thomson Process

$$\eta_{\text{th}} = \frac{|w|}{q_{\text{in}}} = 1 - \frac{|q_{\text{out}}|}{q_{\text{in}}} = 1 - \frac{c_p\,(T_4 - T_1)}{c_p\,(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

$$|w| = |w_{34}| - w_{12}$$

$$q_{\text{in}} = h_3 - h_2 = c_p\,(T_3 - T_2)$$

$$w_{34} = h_4 - h_3$$

$$w_{12} = h_2 - h_1$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$|q_{\text{out}}| = h_4 - h_1 = c_p\,(T_4 - T_1)$$

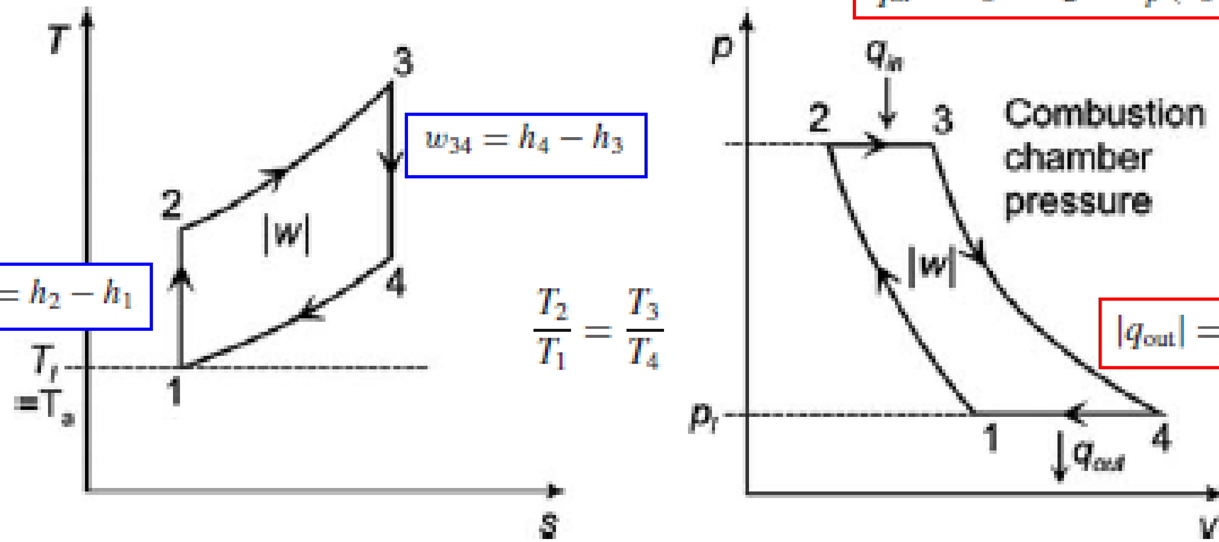


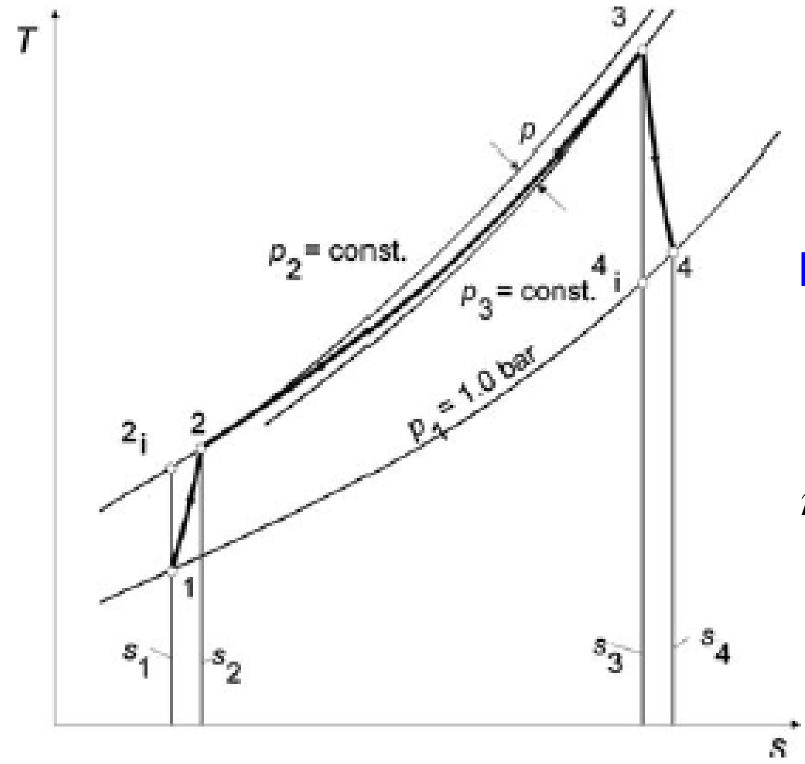
Fig. 3.3 $p - V$ and $T - s$ diagrams for the ideal Joule – Thomson process

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}$$

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\kappa-1}{\kappa}}$$

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{p_1}{p_2}\right)^{\frac{\kappa-1}{\kappa}}$$

3.1.2 Joule-Thomson Process



Isentropic efficiency of pump

$$\eta_{i,c} = \frac{w_s}{w_a} = \frac{h_{2,id} - h_1}{h_2 - h_1}$$

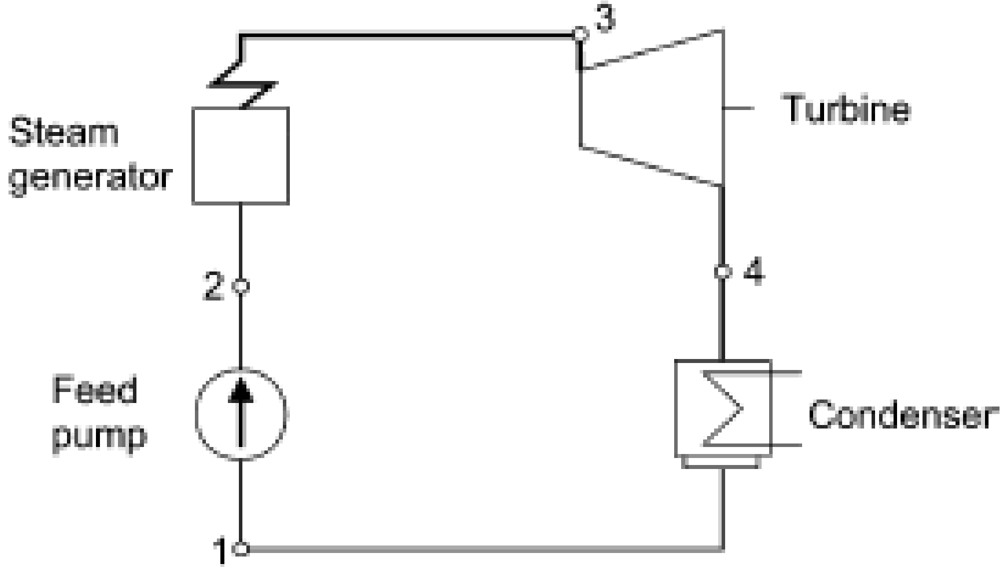
Isentropic efficiency of steam turbine

$$\eta_{i,T} = \frac{w_a}{w_s} = \frac{h_3 - h_4}{h_3 - h_{4,id}}$$

$$w_{pump,in} = \frac{w_{s,pump,in}}{\eta_p} = \frac{v_1(P_2 - P_1)}{\eta_p}$$

Fig. 3.4 $T - s$ diagram of the real Joule – Thomson process

3.1.3 Clausius-Rankine Cycle



1–2: isentropic compression in the feed pump by work input w_{12}
2–3: isobaric heat supply $q_{23} = q_{in}$ in the steam generator (preheating, evaporation, superheating)
3–4: isentropic expansion in the turbine with work output w_{34}
4–1: isobaric heat dissipation $q_{41} = q_{out}$ in the condenser (Hahne 2004).

The 2nd Law of Thermodynamics ; Clausius Statement

Kelvin-Planck Statement

- It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work
- This statement means that no heat engine can have a thermal efficiency of 100 %, or as for a power plant to operate, the working fluid must exchange heat with the environment as well as the furnace

Clausius Statement

- It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a lower-temperature body to a higher-temperature body.
- This statement means that energy from the surroundings in the form of work or heat has to be expended to force heat to flow from a low-temperature medium to a high-temperature medium. (COP$<\infty$)

3.1.3 Clausius-Rankine Cycle

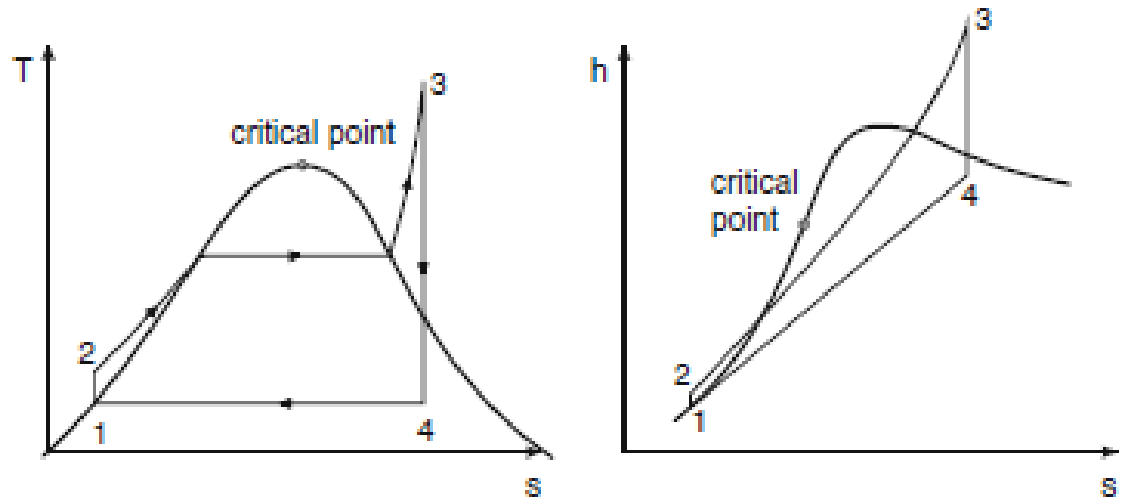


Fig. 3.6 Ideal Clausius–Rankine cycle $T-s$ and $h-s$ diagrams

$$|w| = |w_{34}| - w_{12} = (h_3 - h_4) - (h_2 - h_1)$$

$$\eta_{th} = \frac{|w|}{q_{in}} = \frac{h_3 - h_4 - (h_2 - h_1)}{h_3 - h_2}$$

Thermodynamic mean temperature

$$T_{m,in} = \frac{q_{in}}{s_3 - s_2} = \frac{h_3 - h_2}{s_3 - s_2}$$

Heat extraction

$$T_{m,out} = \frac{|q_{out}|}{s_4 - s_1} = \frac{h_4 - h_1}{s_4 - s_1}$$

The Carnot factor

$$\eta_{th} = \frac{T_{m,in} - T_{m,out}}{T_{m,in}} = 1 - \frac{T_{out}}{T_{in}}$$

3.1.3 Clausius-Rankine Cycle

The thermal efficiency of the Clausius–Rankine cycle thus becomes greater the higher the mean thermodynamic temperature of the heat supply and the lower the mean thermodynamic temperature of the heat extraction.

Another consequence is that, at a given maximal heat supply temperature and a minimal heat dissipation temperature, the result of the idealised isothermal heat exchange processes of the Carnot cycle is in each case the highest efficiency.

In the Clausius–Rankine cycle, feed water preheating, evaporation and superheating inevitably result in a lower average temperature of the heat input, so the efficiency of the Rankine cycle is lower than the Carnot factor.

Therefore, measures to raise the thermal efficiency of the steam power cycle can be assessed with reasonable adequacy by means of the average thermodynamic heat input temperature.

3.2 Steam Power Cycle : Energy and Exergy Consideration

The energy efficiency, η, is the ratio of the power delivered or produced by a process to the power which is supplied to it (Adrian et al. 1986).
The gross installed capacity P_{gr} is the capacity measured at the generator, whereas the net output capacity P_{ne} is the power output delivered to the network. The difference between the gross and net output capacities is given by the so-called electrical auxiliary power $P_{aux,el}$ which is needed to supply all electrical auxiliary devices, e.g. for coal milling, for driving the feed pump (only when there is an electrical feed pump), the combustion air and flue gas fans and to cover the loss of the station service transformer:

$$P_{\mathrm{ne}} = P_{\mathrm{gr}} - P_{\mathrm{aux,el}}$$

$$\boxed{\eta_{\mathrm{ne}} = \frac{P_{\mathrm{ne}}}{\dot{m}_{\mathrm{F}} \cdot \mathrm{LHV}}}$$

$$\boxed{\eta_{\mathrm{ne}} = \eta_{\mathrm{B}} \cdot \eta_{\mathrm{th}} \cdot \eta_{\mathrm{m}} \cdot \eta_{\mathrm{Gen}} \cdot \eta_{\mathrm{aux}} \cdot \eta_{P}}$$

η_B is the steam generator efficiency, η_{th} is the thermal efficiency demand (if not included in η_{aux} already)
η_m reflects the mechanical losses of the turbine
η_{Gen} the generator efficiency covers electrical and mechanical losses of the generator
η_{aux} auxiliary power efficiency takes into account the electrical and the mechanical power
η_R represents the heat losses of the live steam and reheater pipes which connect the steam generator and the turbine

3.2 Steam Power Cycle : Energy and Exergy Consideration

For the boiler or steam generator, the efficiency

$$\eta_{\mathrm{B}} = \frac{\sum \dot{m}_{\mathrm{S},j} \cdot \Delta h_j}{\dot{m}_{\mathrm{F}} \cdot \mathrm{LHV}} = \frac{\dot{m}_{\mathrm{S}}(h_3 - h_2)}{\dot{m}_{\mathrm{F}} \cdot \mathrm{LHV}}$$

$m_{s,j}$ are the individual mass flows of the working medium(water/steam) supplied with heat from combustion in the steam generator. Δh_j are the increases of enthalpy attained in each mass flow.

Steam generator efficiency determined indirectly–by the losses of the steam generator.

- loss through unburned combustibles (κ_{U}),
- loss through sensible heat of the slag (κ_{S}),
- flue gas loss (κ_{FG}) and
- loss through radiation and convection of the steam generator (κ_{RC}).

$$\boxed{\eta_B = 1 - \kappa_{\mathrm{U}} - \kappa_{\mathrm{S}} - \kappa_{\mathrm{FG}} - \kappa_{\mathrm{RC}}}$$

3.2 Steam Power Cycle : Energy and Exergy Consideration

For the thermal efficiency of the real

$$\eta_{th} = \frac{P_i}{\sum \dot{m}_{S,j} \bullet \Delta h_j} = \frac{P_i}{\dot{m}_S(h_3 - h_2)}$$

the inner power output of the turbine *P*i (the power of the turbine without mechanical

The isentropic turbine efficiency η

$$\eta_{i,T} = \frac{\eta_{th}}{\eta_{th,0}} = \frac{h_3 - h_4}{h_3 - h_{4,id}}$$

Mechanical efficiency η_m

$$\eta_m = \frac{P_m}{P_i}$$

the generator efficiency

$$\eta_{Gen} = \frac{P_{Gen}}{P_m}$$

for the auxiliary power efficiency

$$\eta_{aux} = \frac{P_{ne}}{P_{Gen}}$$

3.2 Steam Power Cycle : Energy and Exergy Consideration

Often, the turbine or turbine generator efficiency η_T is used, which represents the ratio of the gross electrical output and, if necessary, the mechanical power output (in the case of feed pumps with a steam turbine drive) to the steam energy input :

$$\eta_{\mathrm{T}} = \frac{P*_{\mathrm{Gen}}}{\sum \dot{m}_{\mathrm{s},j} \cdot \Delta h_j} = \eta_{\mathrm{th}} \cdot \eta_{\mathrm{m}} \cdot \eta_{\mathrm{Gen}} \qquad P^*_{\mathrm{Gen}} = P_{\mathrm{Gen}} + P_{\mathrm{aux,m}}$$

If the feed pump is driven by a steam turbine, the power output of the turbine generator $P*_{\mathrm{Gen}}$ increases, surpassing the gross output P_{Gen} by the amount of the mechanical output of the turbine drive $P_{\mathrm{aux,m}}$.

Where the feed pump is driven electrically, the power output $P*_{\mathrm{Gen}}$ equals the generator output P_{Gen}. The turbine generator efficiency, in contrast to the thermal efficiency of the cycle, also takes into account the losses occurring in the turbine and the generator.

the auxiliary power efficiency
$$\eta_{\mathrm{aux}} = \frac{P_{\mathrm{ne}}}{P*_{\mathrm{Gen}}} = \frac{P*_{\mathrm{Gen}} - P_{\mathrm{aux,el}} - P_{\mathrm{aux,m}}}{P*_{\mathrm{Gen}}}$$

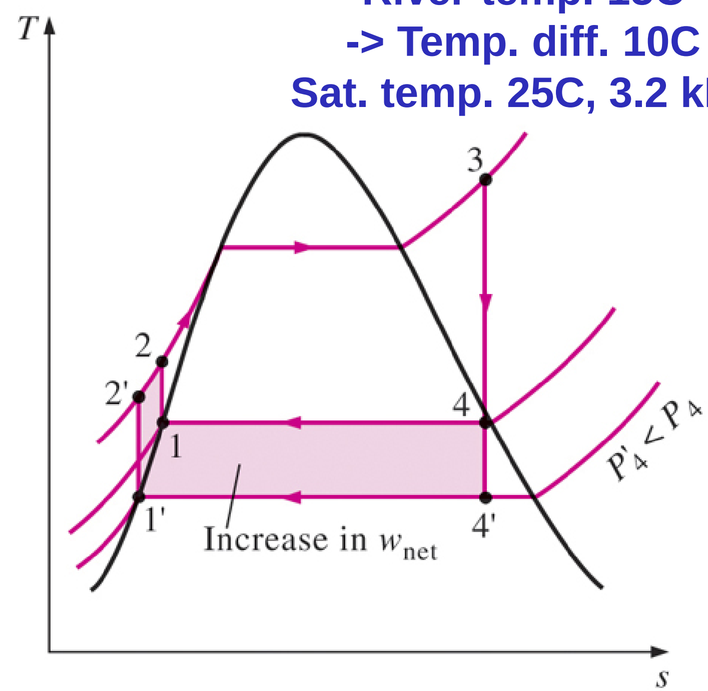
for the total and single exergy efficiencies:
$$\zeta_{\mathrm{ne}} = \zeta_{\mathrm{B}} \cdot \zeta_{\mathrm{th}} \cdot \zeta_{\mathrm{Gen}} \cdot \zeta_{\mathrm{aux}} \cdot \zeta_{\mathrm{m}} \cdot \zeta_{P}$$

4. How Can We Increase the Efficiency of the Rankine Cycle?

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: *Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser.*

Lowering the Condenser Pressure (*Lowers* $T_{low,avg}$)

River temp. 15C
-> Temp. diff. 10C
Sat. temp. 25C, 3.2 kPa

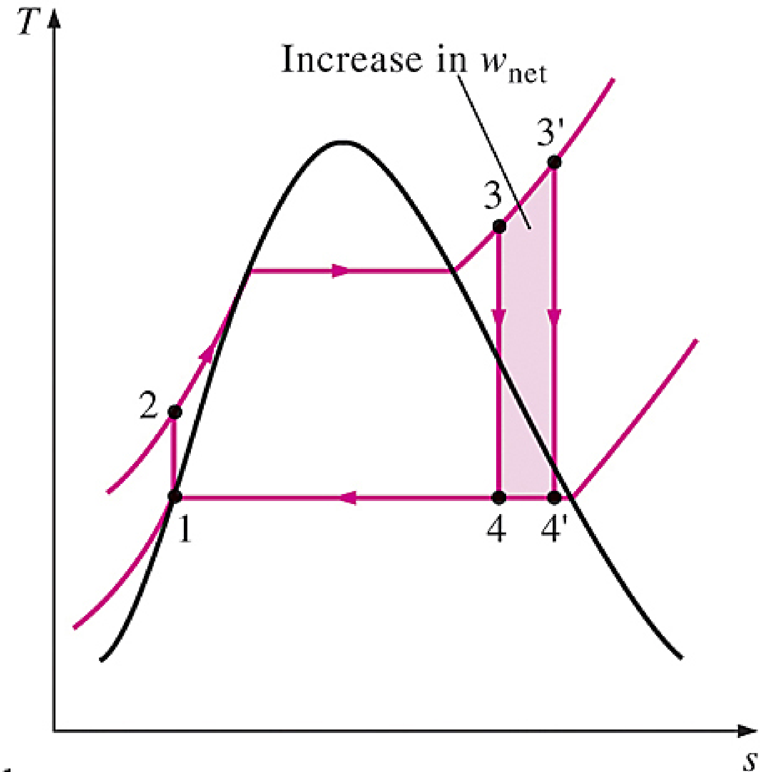


To take advantage of the increased efficiencies at low pressures, the condensers of steam power plants usually operate well below the atmospheric pressure. There is a lower limit to this pressure depending on the temperature of the cooling medium

Side effect: Lowering the condenser pressure increases the moisture content of the steam at the final stages of the turbine.

Superheating the Steam to High Temperatures (*Increases* $T_{high,avg}$)

Both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency since the average temperature at which heat is added increases.



Superheating to higher temperatures decreases the moisture content of the steam at the turbine exit, which is desirable.

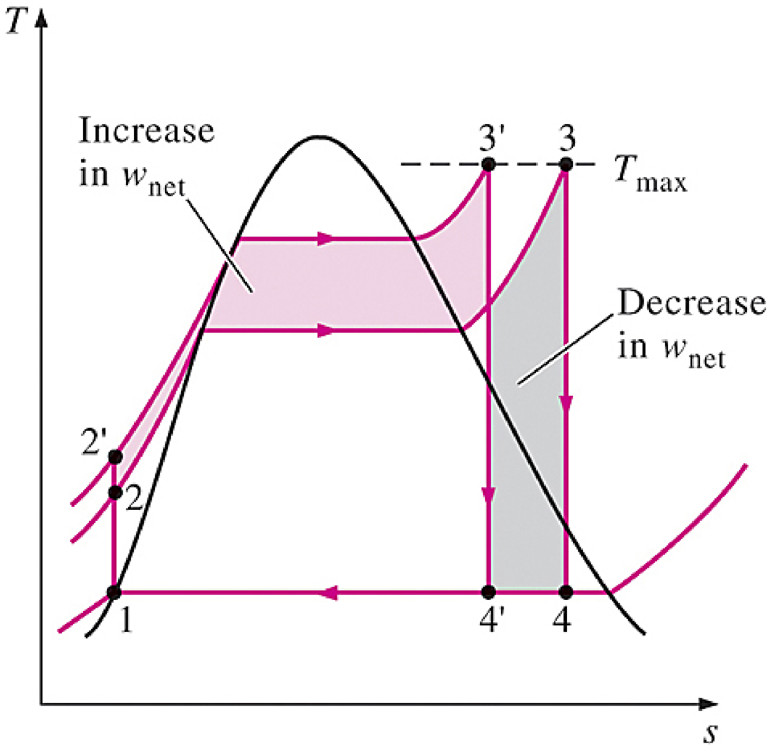
- Superheat the vapor
  - Average temperature is higher during heat addition.
  - Moisture is reduced at turbine exit (we want x_4 in the above example > 85 percent).

The temperature is limited by metallurgical considerations. Presently the highest steam temperature allowed at the turbine inlet is about 620°C in USC Bolier

Increasing the Boiler Pressure (*Increases* $T_{high,avg}$)

For a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This side effect can be corrected by reheating the steam.

-> Reheating has both benefits



- Increase boiler pressure (for fixed maximum temperature)
  Availability of steam is higher at higher pressures.
  Moisture is increased at turbine exit.

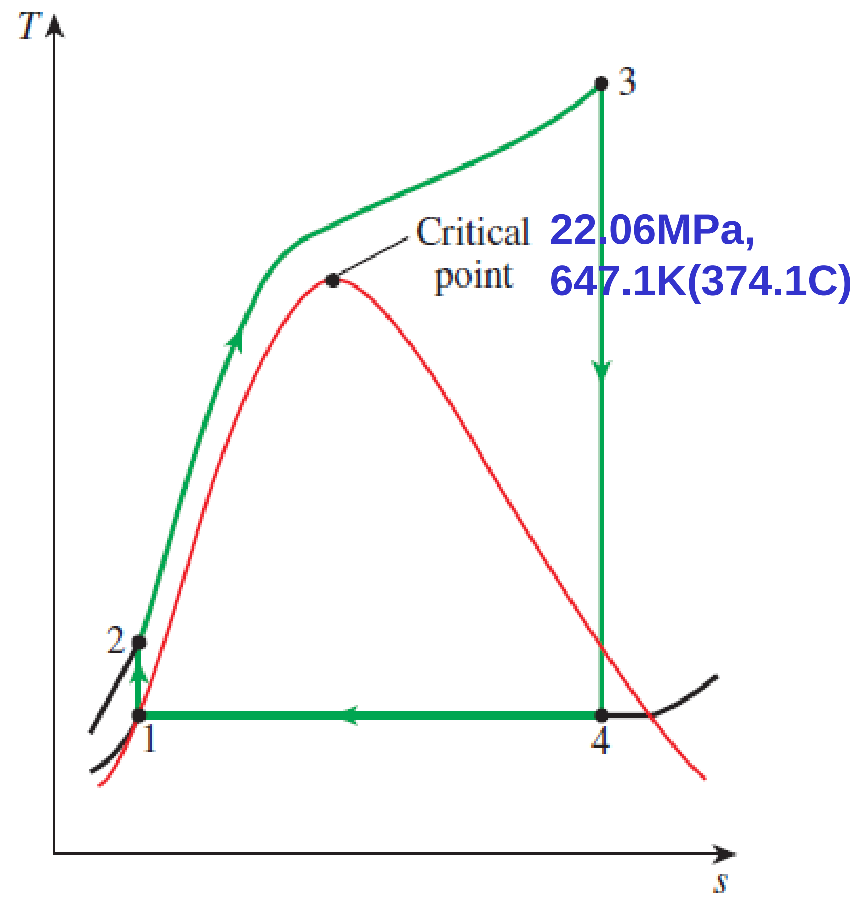
A supercritical Rankine cycle

Today many modern steam power plants operate at supercritical pressures (*P* > 22.06 MPa) and have thermal efficiencies of about 40% for fossil-fuel plants and 34% for nuclear plants.

- 2.7MPa in 1992 to over 30MPa today

- output 1000MW

- 150 Super-critical Plants



Assignment #4 : Boiler

Coals Power Plant Boiler in Korea

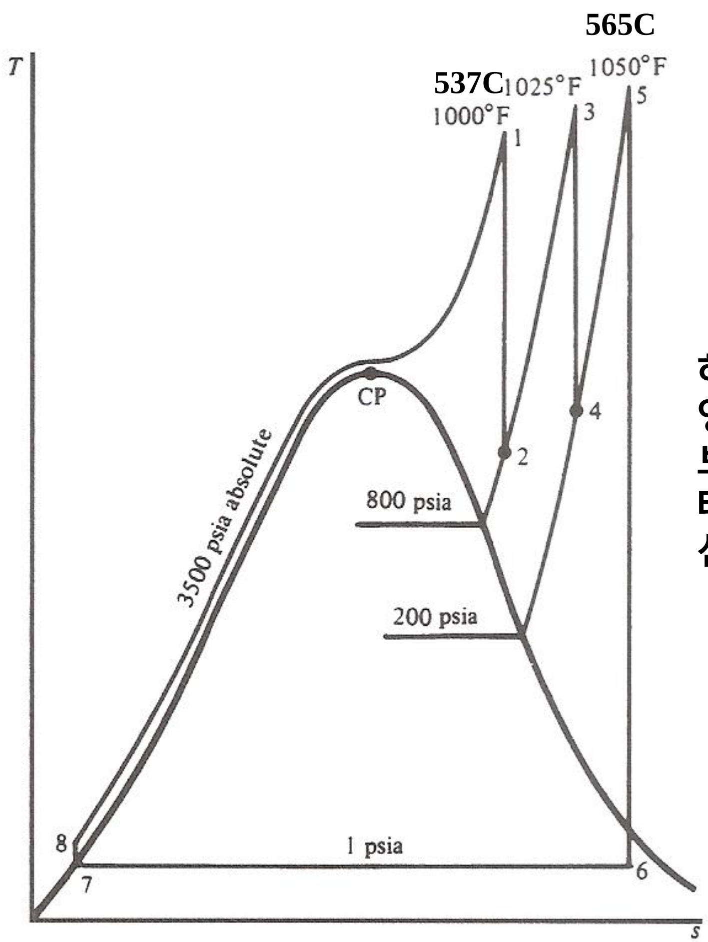


FIGURE 2.13 *T-s* diagram of an ideal supercritical, double-reheat 3500/1000/1025/1050 steam cycle.

하동(KOSPO)
영흥(KOSEP)
보령(KOMPO)
태안(KOWPO)
신보령(KOMPO)

T-s curve
Efficiency?

4.1.4 Reference Power Plant

Table 4.1 Data for the reference power plant (Spliethoff and Abröll 1985)

Power plant unit	
Gross rated power	740 MW
Net rated power	690 MW
Efficiency	39%
Mechanical capacity of the feed pump	21 MW
Auxiliary power requirement	50 MW
Mode of service	Intermediate load range (170 starts p.a.)
Steam generator	
Capacity	2250 t/h (625 kg/s)
Construction	Once-through boiler
Live steam condition	209 bar, 535°C
Steam condition after reheater	39.6 bar, 535°C
Entry temperature of feed water	250°C
Firing	
Air ratio	1.3
Flue gas temperature	130°C
Coal mills	4 × 74 t/h
Forced-draught fan (FD fan)	1 × 100%
Induced-draught fan (ID fan)	1 × 100%
Range of control	40–100%
Steam generator efficiency	94%
Boiler feed pump	1 × 100% duty turbine-driven pump 1 × 50% duty motor-driven pump
Steam turbine generator	
Construction	Condensation turbine with single reheating
Operational mode	modified sliding-pressure operation with throttling of the intake valves (5%)
Turbine pressure sections/number of extractions	4 (1 × HP, 1 × MP, 2 × LP)/6
Live steam condition	190 bar/530°C
Exhaust steam pressure	0.0549 bar
Back-cooling system	
Cooling tower construction	Natural-draught wet-type cooling tower
Heat rejection	894 MW
Air temperature	10°C, max. 35°C
Cold water temperature	16.6°C, max. 29°C
Flue gas cleaning unit	
Particulate collector	Electrostatic precipitator (ESP)
Nitrogen oxide control device	High-dust catalyst before air preheater
Desulphurisation unit	Wet desulphurisation with limestone
Flue gas off-take	Stack, reheat after FGD unit

Developments build upon this state of the art. For this reason, a reference power plant, For further discussion in this chapter, this power plant corresponds to the state of the art from the 1980s in Germany.
The reference power plant will be used in the following sections to explain fundamentals and design by way of comparison with the further development of steam power plants.

Homework#4 Draw T-S diagram comparing with your power plant.

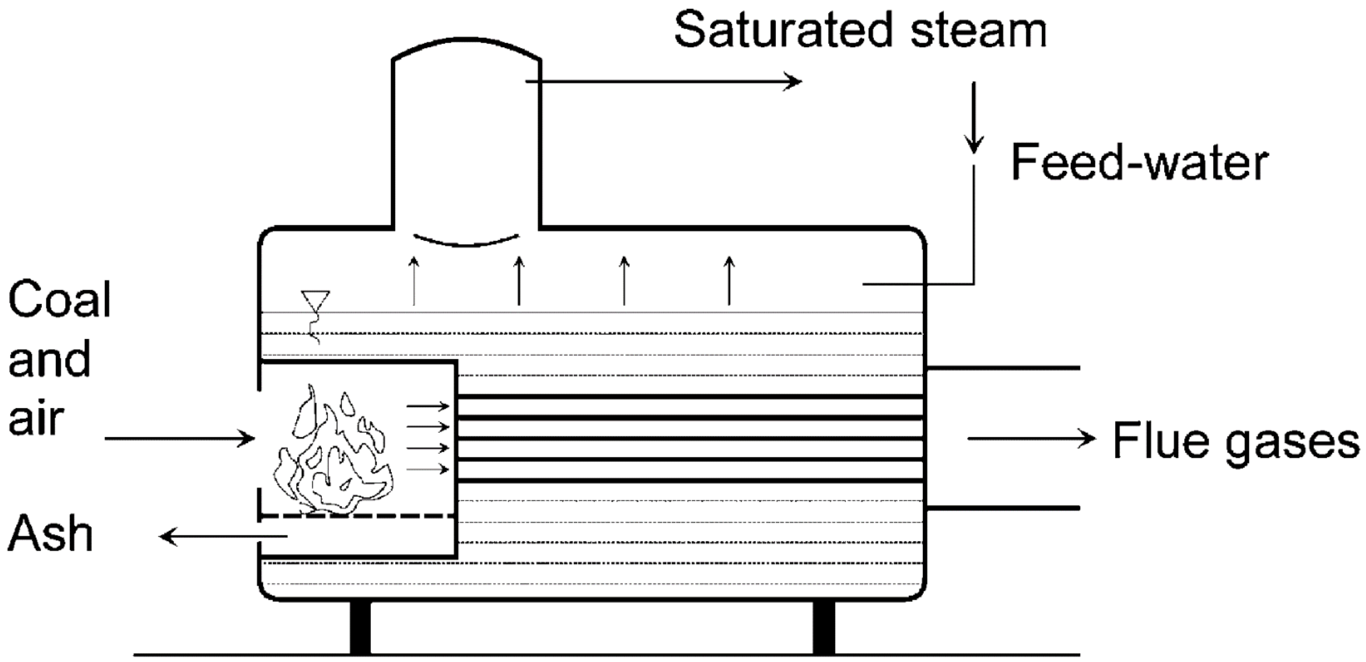
4.2 STEAM GENERATORS

The high-pressure water is evaporated and superheated.

The capacity range of steam generators lies between 0.4 t/h for process steam generators, up to 4,500 t/h for large power plant boilers for electricity production.

New plants are designed for live steam pressures up to 300 bar and live steam and reheat temperatures up to 600◦C/*620◦C.*

Shell boilers are suited to low steam pressures only and so are utilised for low capacities up to 54 t/h of steam output and steam pressures up to 35 bar (Sobbe 2004).



4.2 STEAM GENERATORS

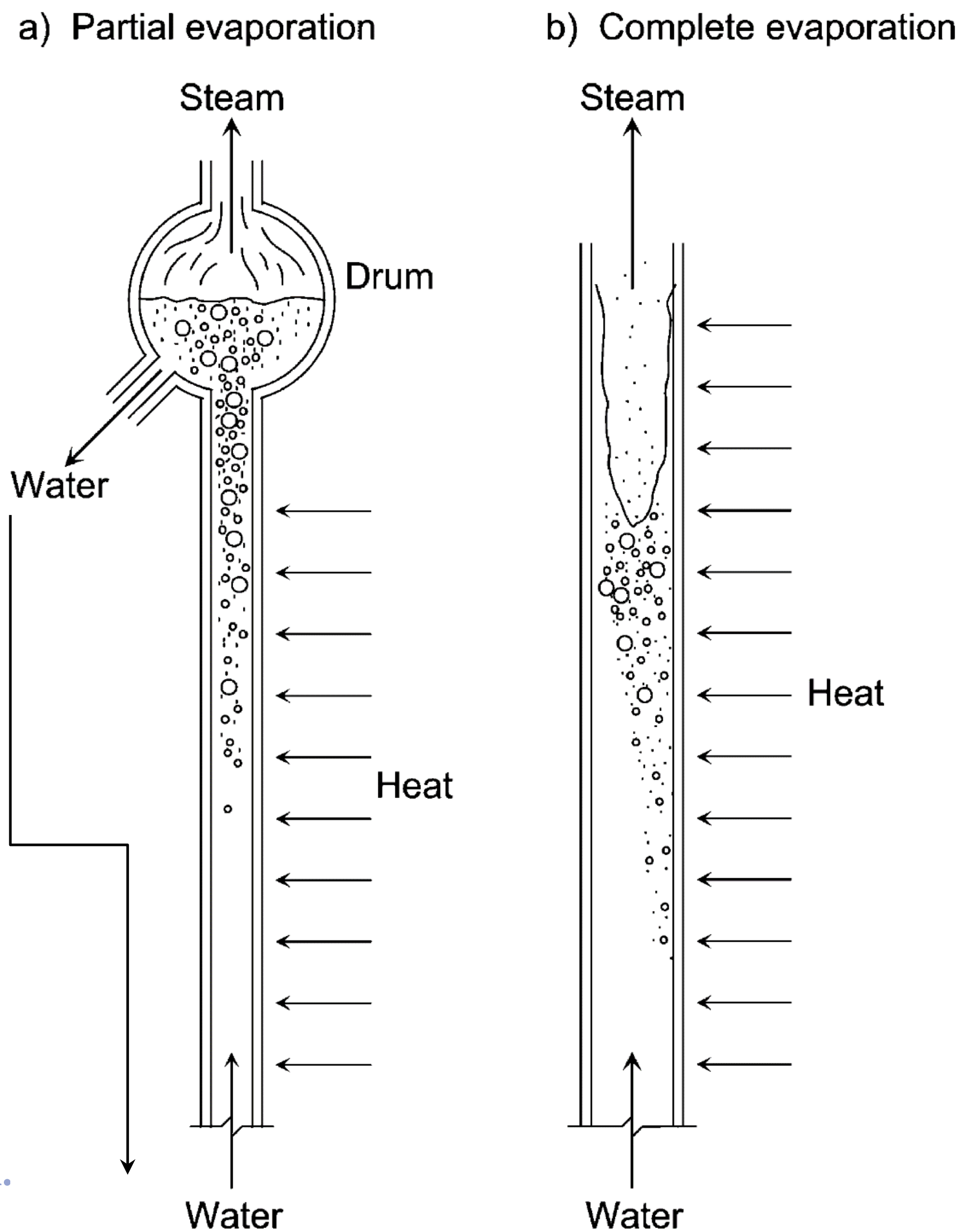
In steam generators of higher capacity, the heat exchange surfaces consist of complex parallel tube systems. A great number of small water and steam flows, conducted through tubes with a small inner diameter, take up heat along the heated stretches of the tubes.

Accordingly, a steam generator consists of various heat exchange surfaces, such as the feed water heater or economiser, evaporator, superheater and reheater, which operate with different heat flux densities depending on the firing and the hot flue gases.

The increases in the volumetric flows are provided for by branching of the heated single tubes, introducing more flow capacity.

The relative heat absorptions of the economiser, evaporator and superheater are dependent on the pressure, as the evaporation enthalpies decrease with higher pressures. The heat absorptions of the economiser and the superheater increase with higher pressures.

4.2 STEAM GENERATORS



The various steam generator systems differ in the configuration of the evaporator, while there is no difference in the superheater and economiser units.

A distinction is made between circulation(partial) and once-through(complete) systems.

4.2 STEAM GENERATORS

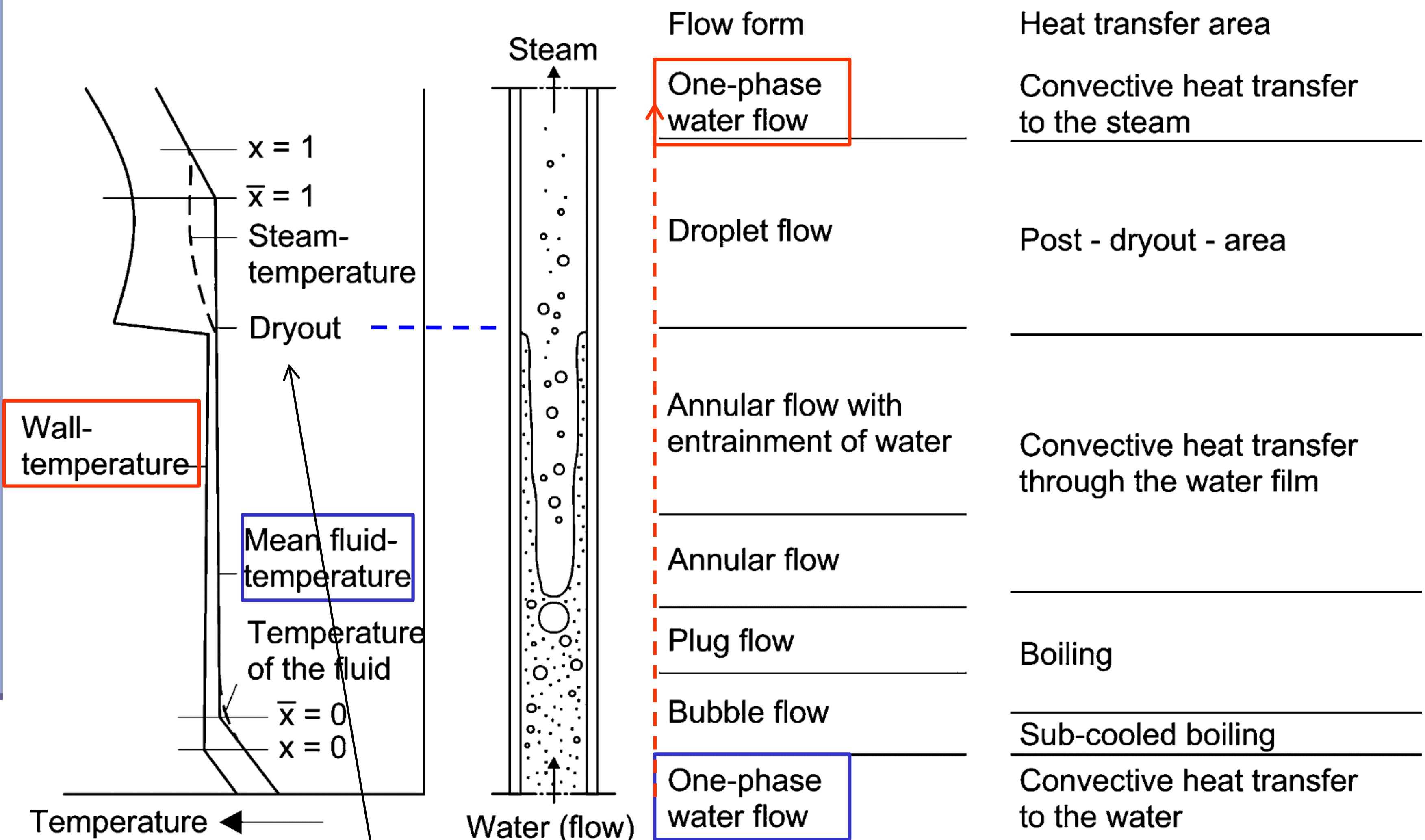
In circulation steam generators, water is heated to boiling temperature in the heated vertical evaporator tubes. In the drum mounted above the heated tubes, the rising water – steam mixture is divided, and the water flowing back through downcomer pipes to re-enter the heated evaporator tubes. Complete evaporation is achieved only after several recirculations. Circulation systems have a fixed liquid – vapour phase transition point in the drum.

In contrast, in once-through steam generators, the water in the evaporator tube is in one stage preheated, evaporated and partially superheated. This system does not need a water–steam division drum. In once-through systems, the liquid–vapour phase transition point is not fixed.

The required heat for steam generation is transferred to the heat exchange surfaces by radiation and convection.

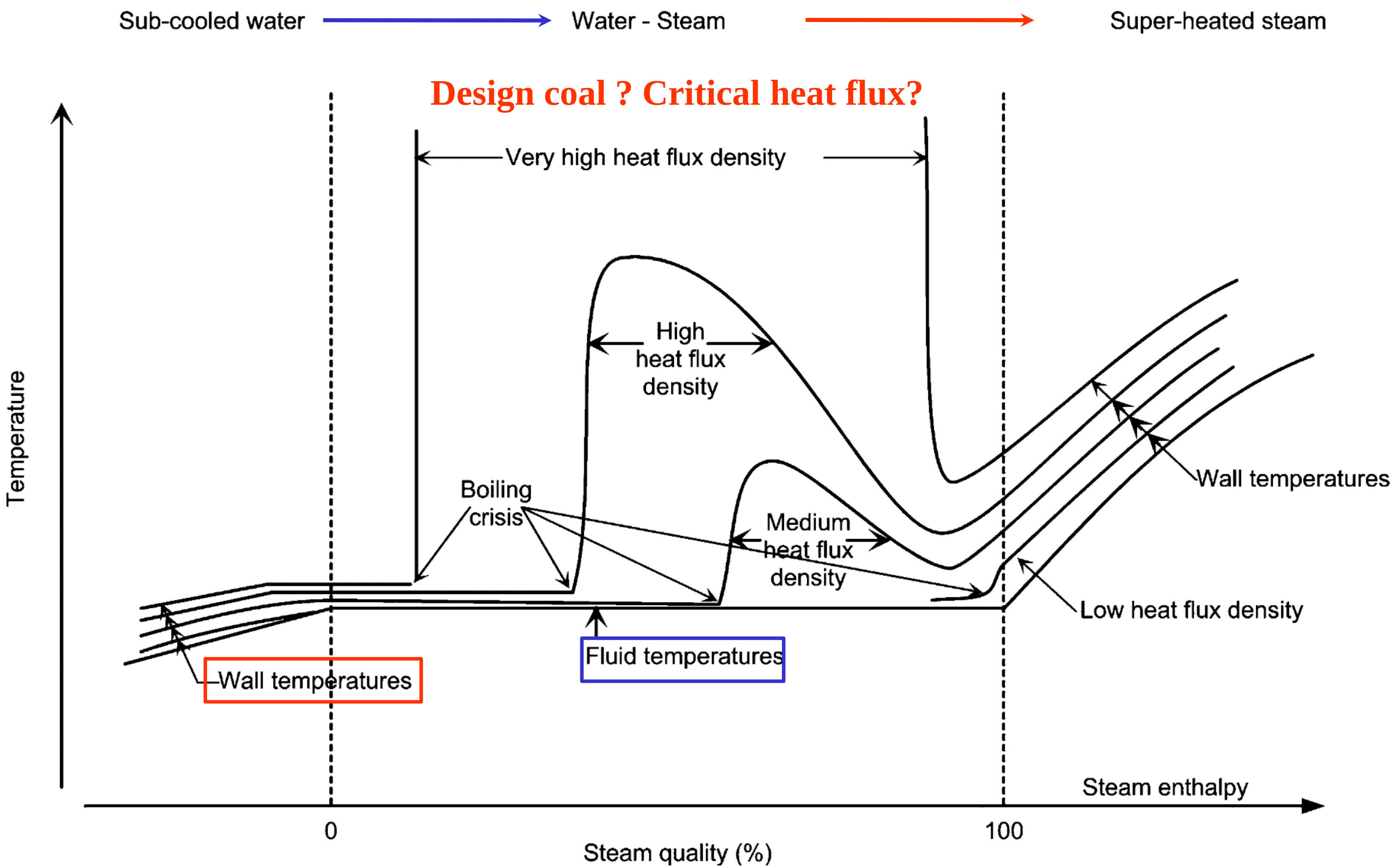
The heat exchange conditions in the evaporator –water-wetted tube walls and high mass flow densities – make it possible to achieve high heat transfer rates in the evaporator. The furnace walls, which have the highest heat flux density in a steam generator, due to the flame radiation, are therefore mostly designed as evaporative heating surfaces.

4.2.1 Flow and Heat Transfer Inside a Tube



The location of the boiling crisis and the level the wall temperature rises to depend on numerous factors, <u>such as the heat flux density, the mass flow density, the tube design and the steam quality.</u>

4.2.1 Flow and Heat Transfer Inside a Tube



For the design of steam generators, boiling crises are of great importance, because they can lead to excess temperatures in the tube walls, which have to be taken into account in the design stage.

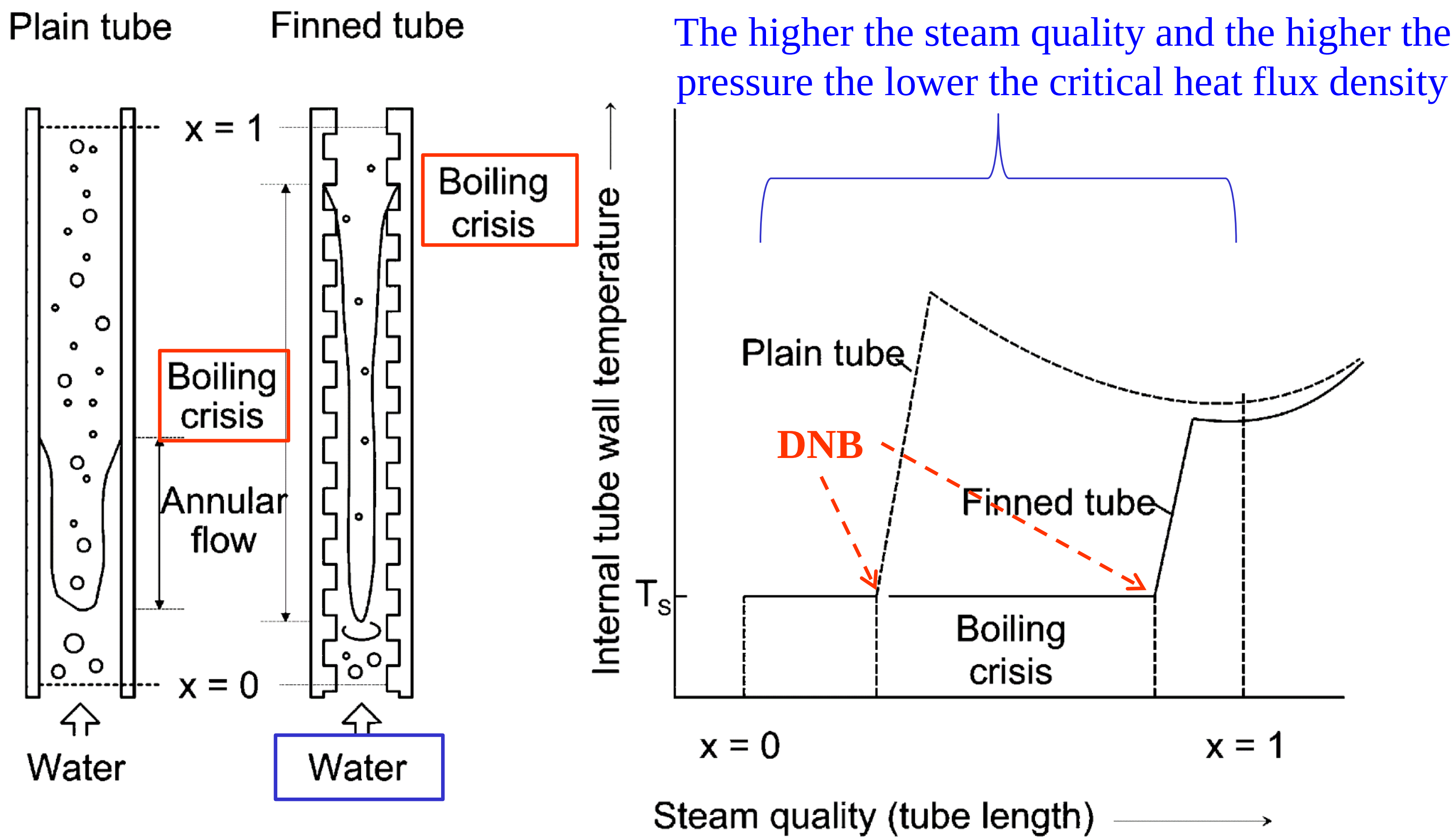
4.2 STEAM GENERATORS

The “first kind” of boiling crisis, called DNB, from “departure from nucleate boiling”, is caused by excessively stressed heat exchange surfaces. This crisis can occur anywhere in the evaporation area, from the sub-cooled boiling region to the annular flow region, when a so-called critical heat flux density is reached and then exceeded. The higher the steam quality and the higher the pressure the lower the critical heat flux density. A steam film forms on the wall, which impedes the heat transfer.

During design of a steam generator, the DNB crisis has to be designed out. By improved cooling of the tubes, e.g. by using smaller tube diameters or internally finned tubes, the critical heat flux can be raised.

The “second kind” of boiling crisis occurs during the transition from annular to droplet flow, through a drying out of the water film. The effects of this boiling crisis, though, are of minor consequence compared to the DNB crisis. They are a systematic phenomenon with once-through steam generators. In circulation steam generators, due to the partial evaporation, the liquid – vapour phase transition point is not reached (Strauß 2006).

4.2.1 Flow and Heat Transfer Inside a Tube



In tubes with internal helicoid fins, the flow is set into a twisting movement by the helical guidance of the fins. Centrifugal force makes the dispersed water droplets settle on the wall, which keep the wall covered with a wet coating up to high steam qualities of $x > 0.9$

4.2.1 Flow and Heat Transfer Inside a Tube

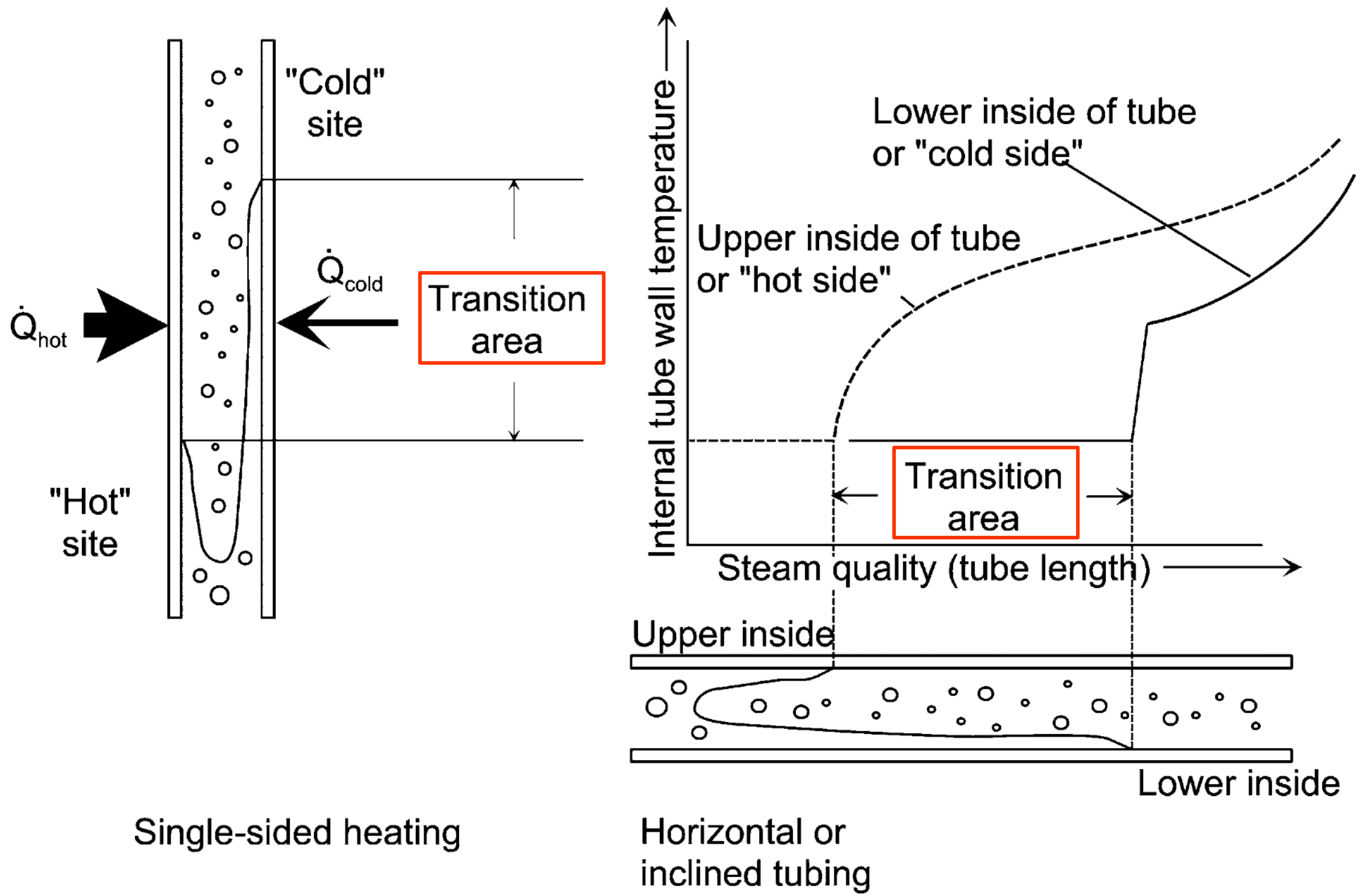
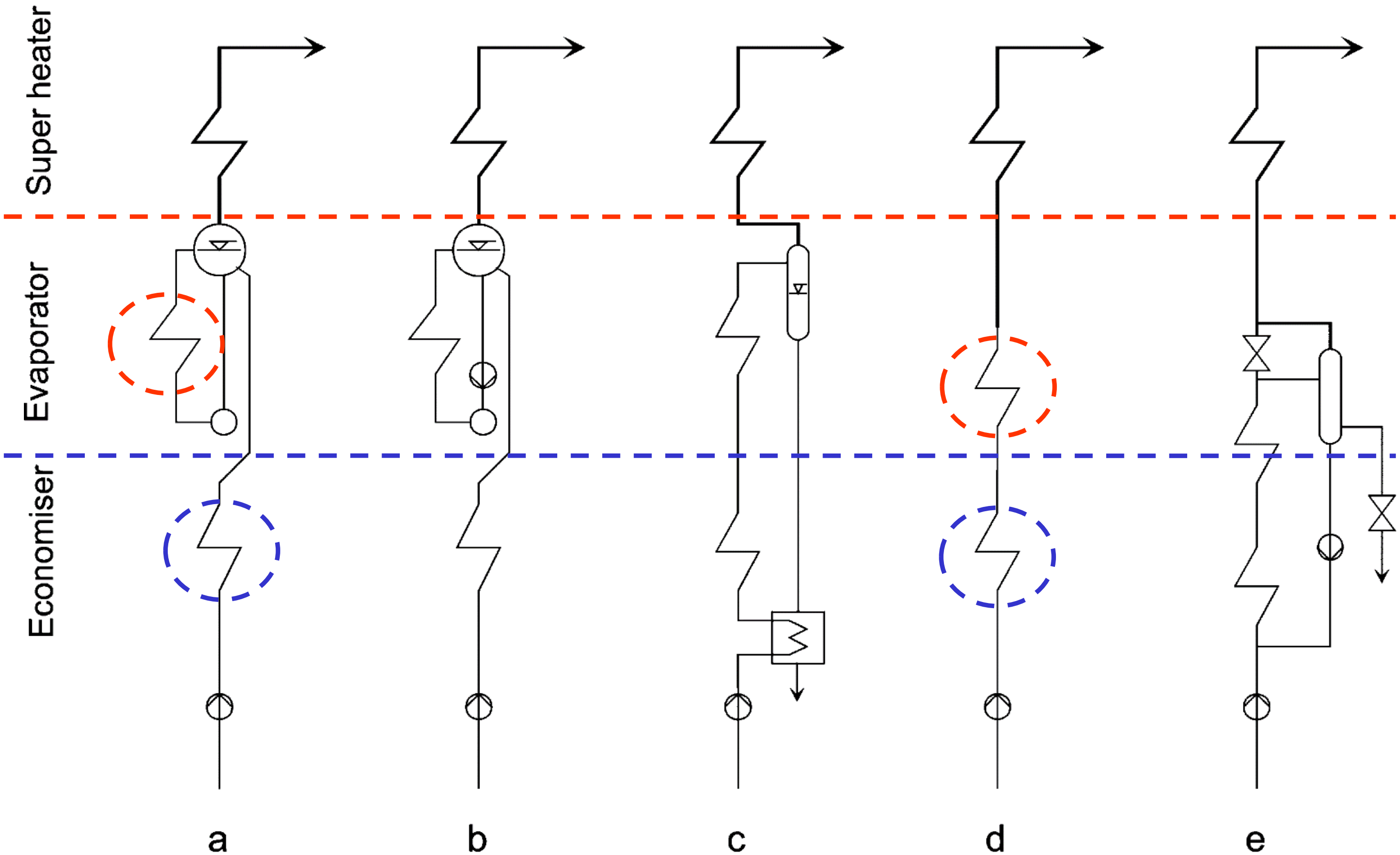


Fig. 4.12 Flow patterns and wall temperatures in a single-sided heated, horizontal or inclined evaporator tube (Kefer et al. 1990)

4.2.2 Evaporator Configurations

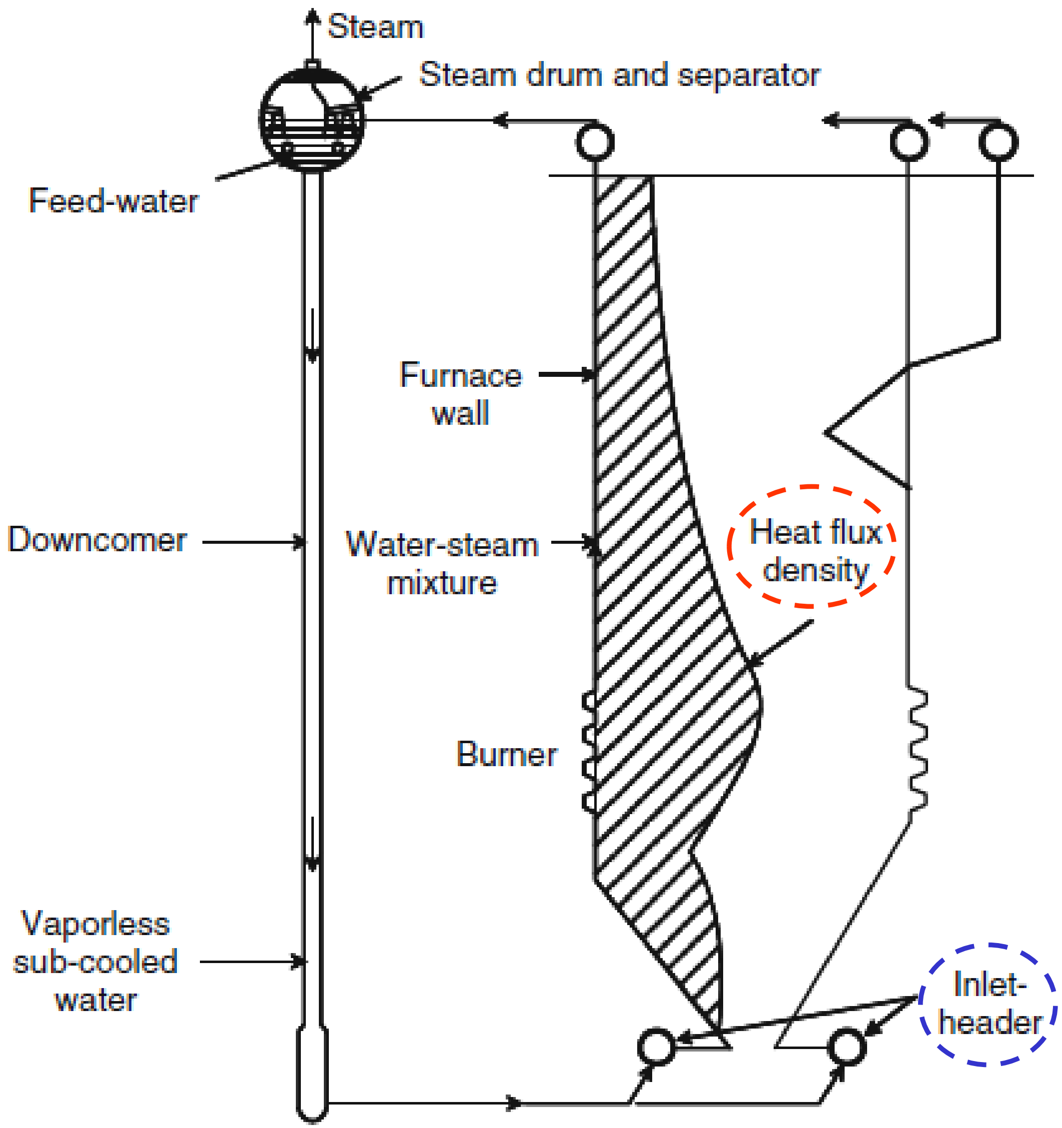
4.2.2.1 Natural Circulation



a Natural circulation
b Forced circulation
c Once-through and residual water separation (Sulzer)
d Once-through (Benson)
e Once-through with low-load circulating system

The various state-of-the-art designs of steam generator systems

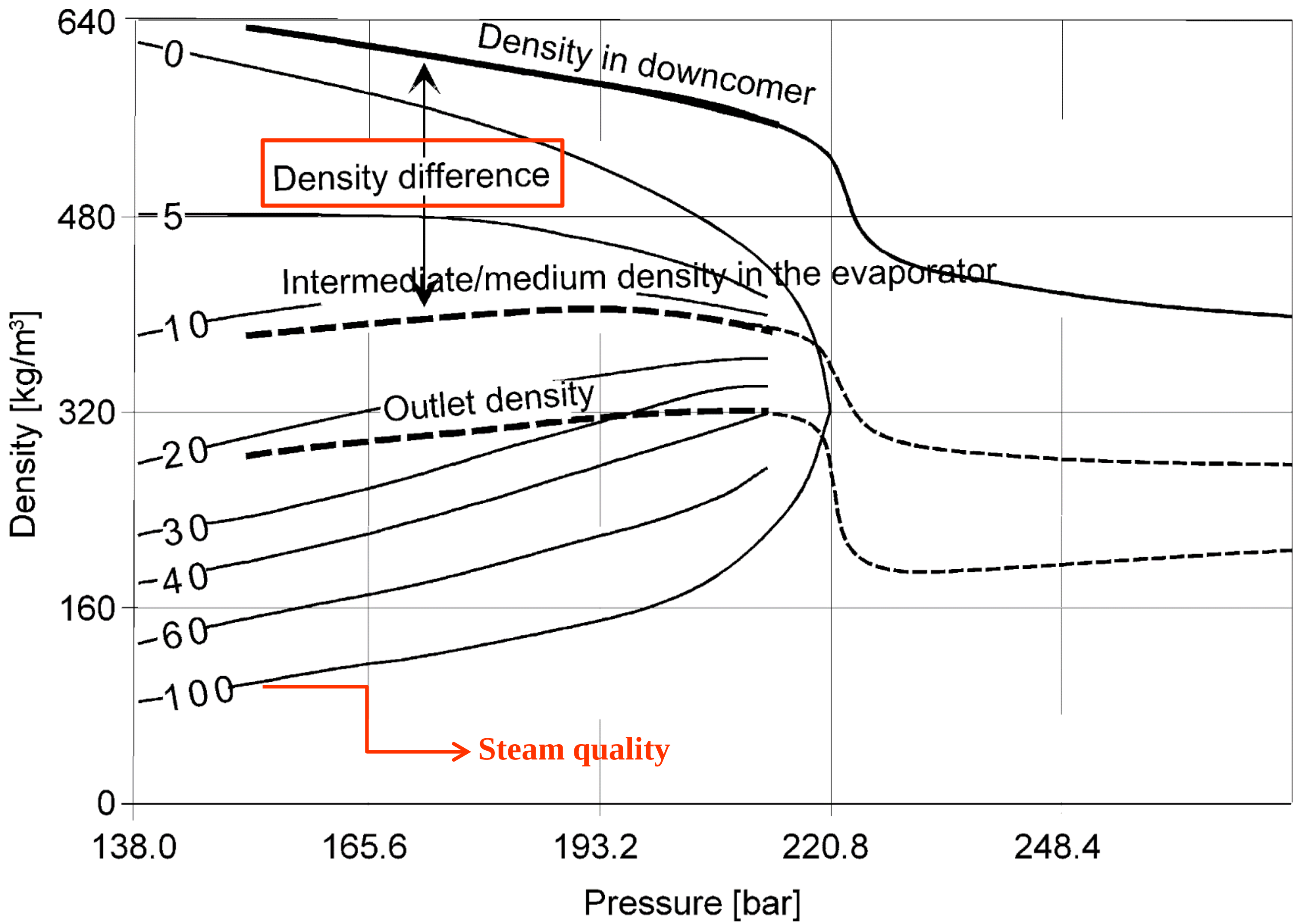
4.2.2.1 Natural Circulation



Schematic diagram of a natural-circulation steam generator . (Stultz and Kitto 1992)

Natural-circulation steam generators typically consist of economisers and an evaporator with risers that form the heated furnace wall, a drum for the separation of water from steam and unheated down pipes and superheaters.

4.2.2.1 Natural Circulation



Circulating flow forms because of the density difference between the falling water in the unheated downcomer and the water – steam mixture in the heated riser

4.2.2.2 Forced Circulation

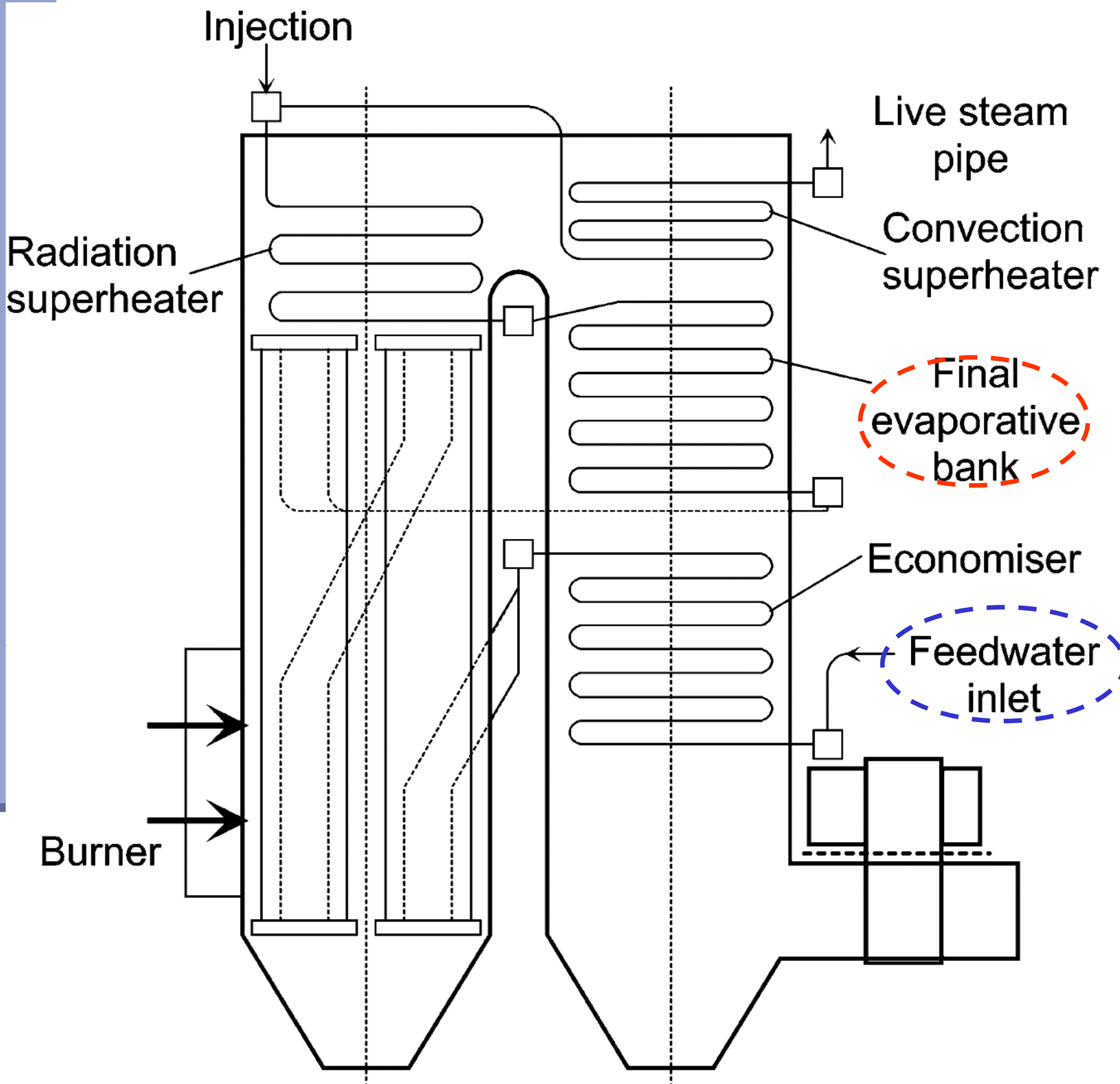
Forced circulation is limited to a range of about 200 bar to ensure sufficient water – steam separation in the drum (Adrian et al. 1986).

Since the circulating pump can balance out the pressure losses in the riser and downcomer parts of the evaporator, it is possible for the design to include components with higher pressure losses. It allows the choice of narrower tubes for better cooling; forced distribution of the water at the inlet of the evaporator tube; and drum inserts that are more effective for water separation but have higher pressure losses.

The applications of forced-circulation steam generators, like natural-circulation systems, are low-pressure and intermediate-pressure plants with capacities up to 500 t/h and also heat recovery steam generators, whereas for high-pressure steam generators, in Germany, it is preferred to use once-through forced circulation.

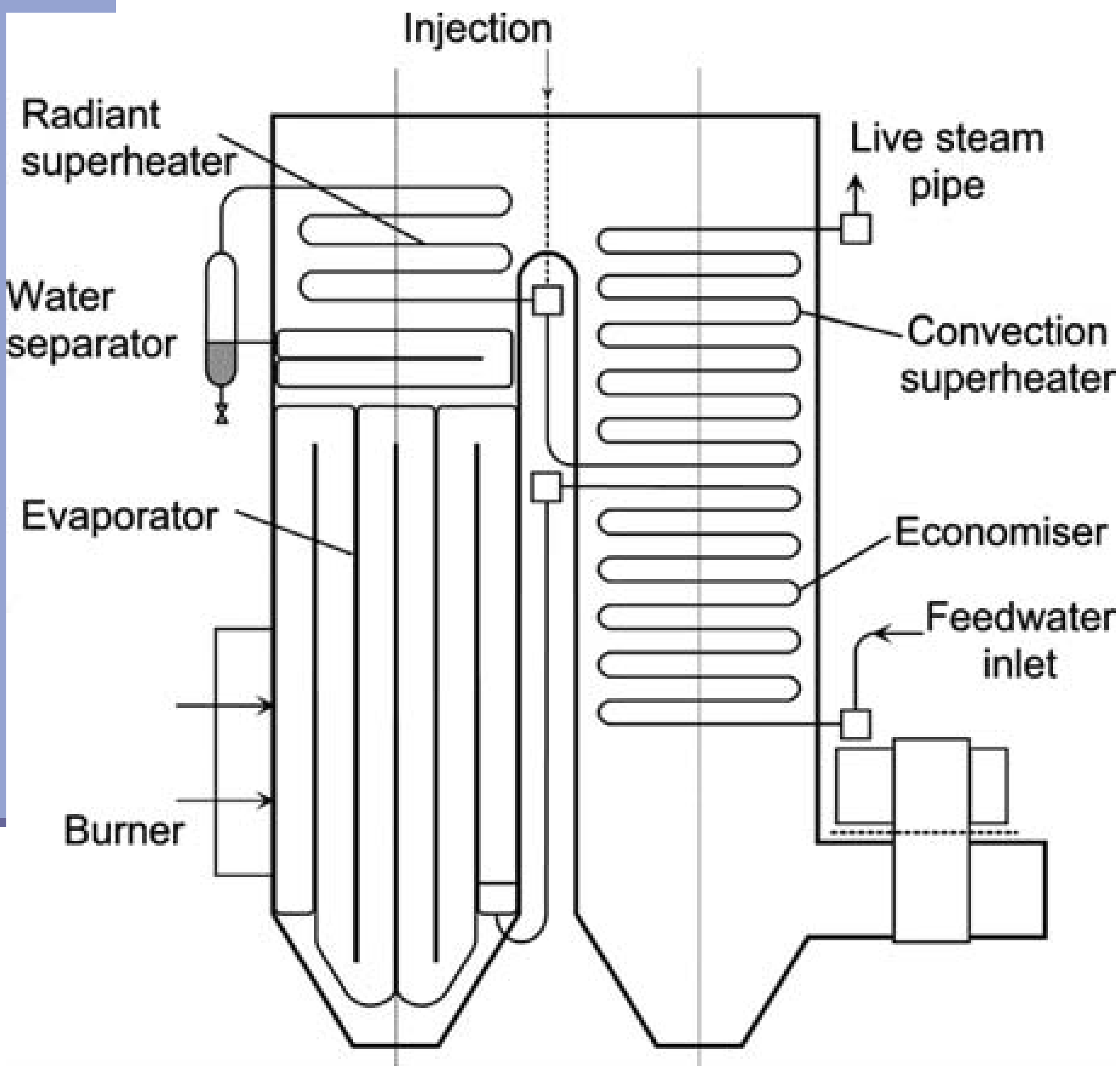
In several countries, though, and in the USA in particular, the forced-circulation system is the preferred system even for large plants, with capacities up to 2,000 t/h and pressures up to 170 bar.

4.2.2.3 Once-Through Systems



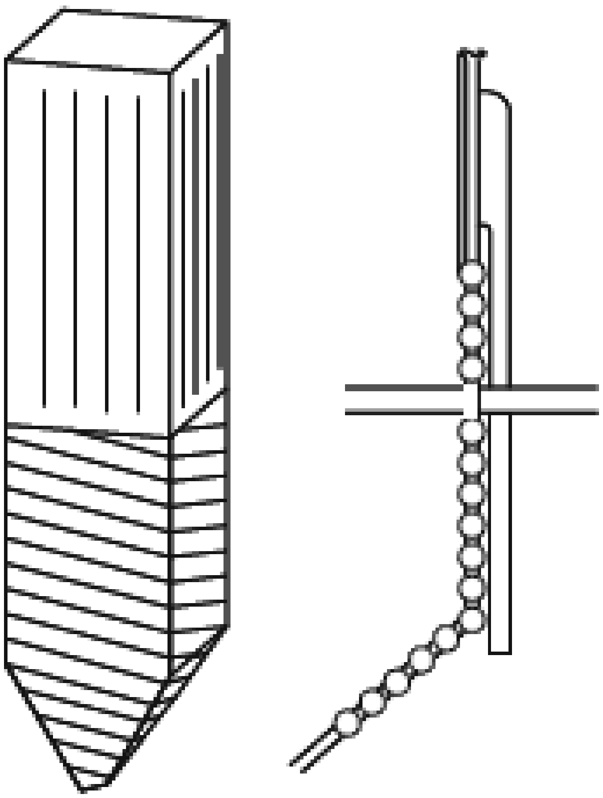
The Benson boiler had an evaporator consisting of several vertical tubes with upward flow, mounted in series-connected banks, which at the same time defined the furnace perimeter. The liquid–vapour phase transition point was in the so-called final evaporative bank which, for salt deposit considerations, had been installed after the furnace in the convective heat exchanger range, with low heat transfer rates.

4.2.2.3 Once-Through Systems

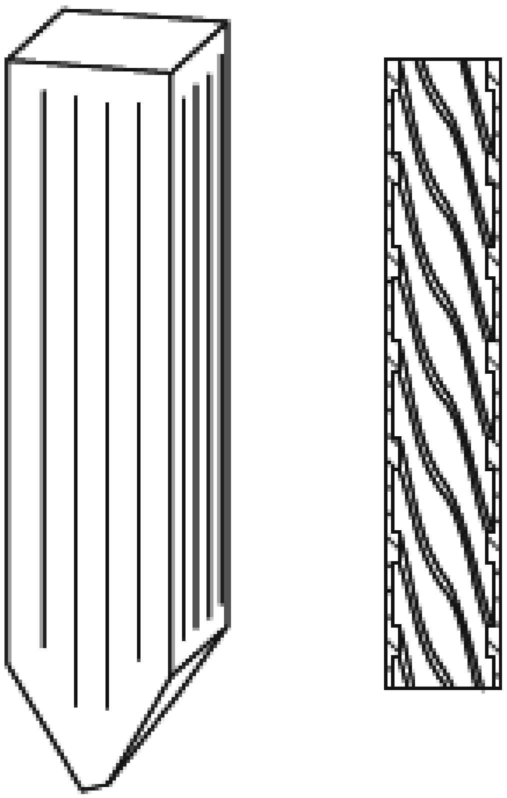


In the Sulzer boiler, several parallel evaporator tubes meandering through the furnace formed the evaporator (Doleˇzal 1990; Wauschkuhn 2001). The difference to the Benson boiler was that this way each tube ran the entire length of the evaporator. Typical features of the Sulzer boiler were the wet operating regime of the evaporator and the following downstream water separator, which was designed to separate a residual water content of 5%.

4.2.2.3 Once-Through Systems



Furnace with wound-pattern walls and girders

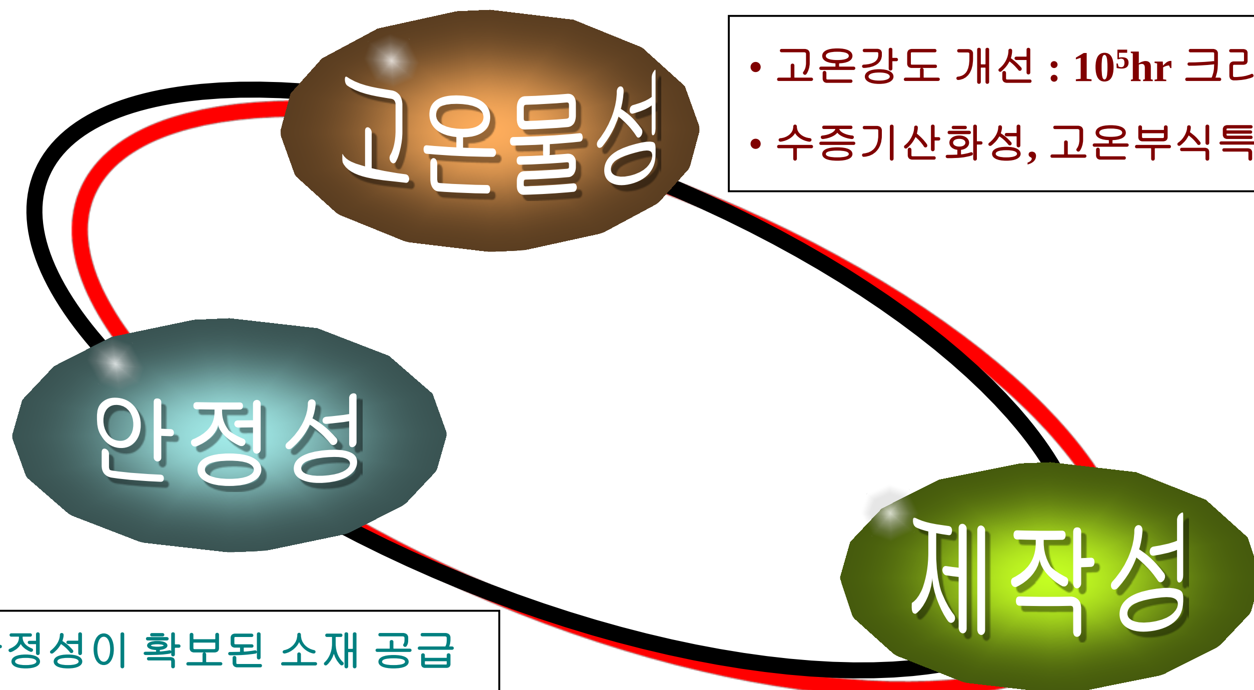


Furnace with vertical internally finned tubes

Membrane tube walls in a helically wound pattern, however, are not able to carry, w/o additional support, the weight of the furnace, the structural bracings, the insulation and the water contained within it, as well as the possible fouling and slagging deposits.

❖ 1000MW USC 기술 – 소재 선정 기준

Improvement of Efficiency ➡ High Pressure & Temperature



- 고온강도 개선 : 10^5hr 크리프강도
- 수증기산화성, 고온부식특성 개선

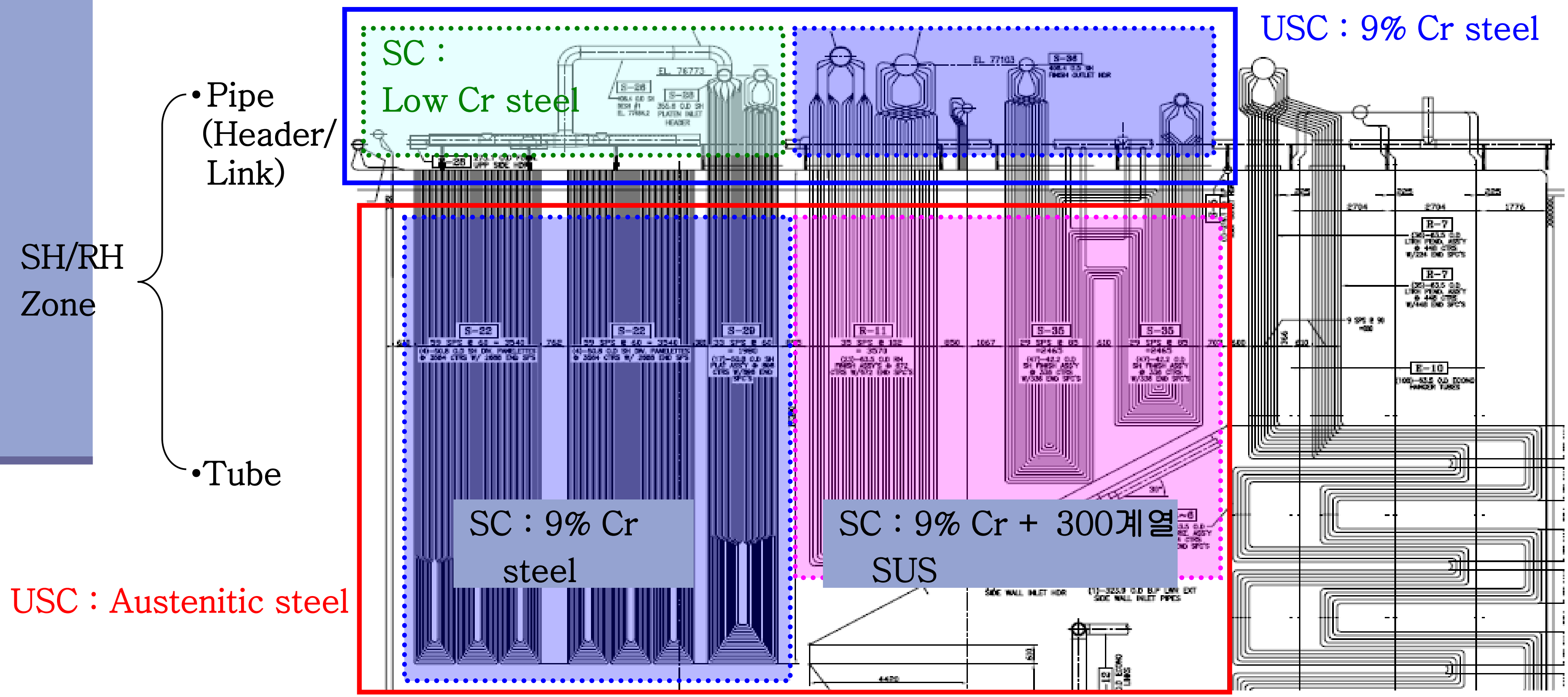
- 고온 안정성이 확보된 소재 공급
- **Proven material** 사용

- 용접성 : **Matching filler material** 개발
- 고온 성형/제관성

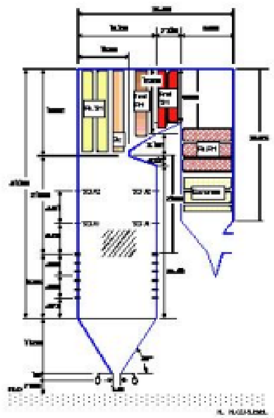
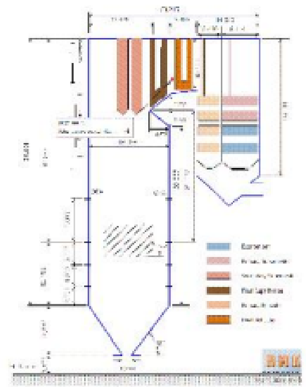
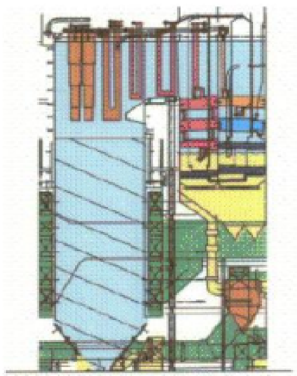
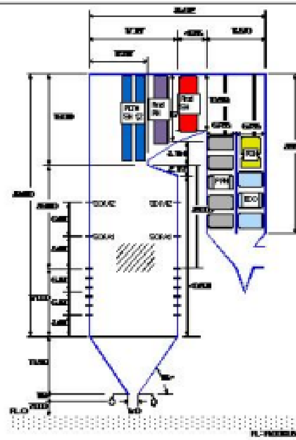
❖ 1000MW USC 기술 – 보일러 소재

•고온 안정성 확보를 위해 메탈온도 저감설계 수행 및 FSH/FRH의 가열부(화로내부) 재질을 Stainless Steel 적용

- 566°C Class : 9% Cr강이 주로 적용, SH/RH에 300계열 SUS강 일부 사용
- 600°C Class : SH/RH부위에 300계열 및 HR3C급(25Cr-20Ni class)의 SUS 튜브 사용



❖ 1000MW USC 기술 – 발전소별 열부하

구 분	보령 #7,8	당진 #9,10	Tachinanawan #2	신보령 #1,2
제 작 사	두산중공업, CE	BHK	BHK	두산중공업, 밥콕
설비용량	500 MW × 2	1,000 MW × 2	1,050 MW × 1	1,000 MW × 2
투영면적 (EPRS)				
	6,079	12,430	-	12,507
	177,733	184,800	289,200	136,440(≤190,000)
	좌 동 (튜브뱅크 간격 672㎜≥)	좌 동 (튜브뱅크 간격 450㎜≥)	-	Final RH 전단까지 (튜브뱅크 간격 450㎜≥)
노체적 (Vol)	14,021	29,700	-	31,744
	74,019	73,300	-	68,456 (≤75,000)
	Final RH 전단까지	Final RH 전단까지	-	Final RH 전단까지
노단면적 (PA)	271 ㎡	516 ㎡	444 ㎡	522
	3,830,232	4,217,200	4,900,000	4,164,451 (≤4,300,000)
버너지역면적 (BZS)	725.5 ㎡	1,408 ㎡	-	1,904
	1,430,523	1,545,500	1,835,000	1,141,453 (≤1,550,000)
고위발열량 (㎉/㎏)	6,080	5,626	6,518	5,600
	Min 5,610	Min 5,250	-	Min 5,200
회융점 (℃)	1,250	1,223	1,250	1,200
	Min 1,160	Min 1,150	-	Min 1,050

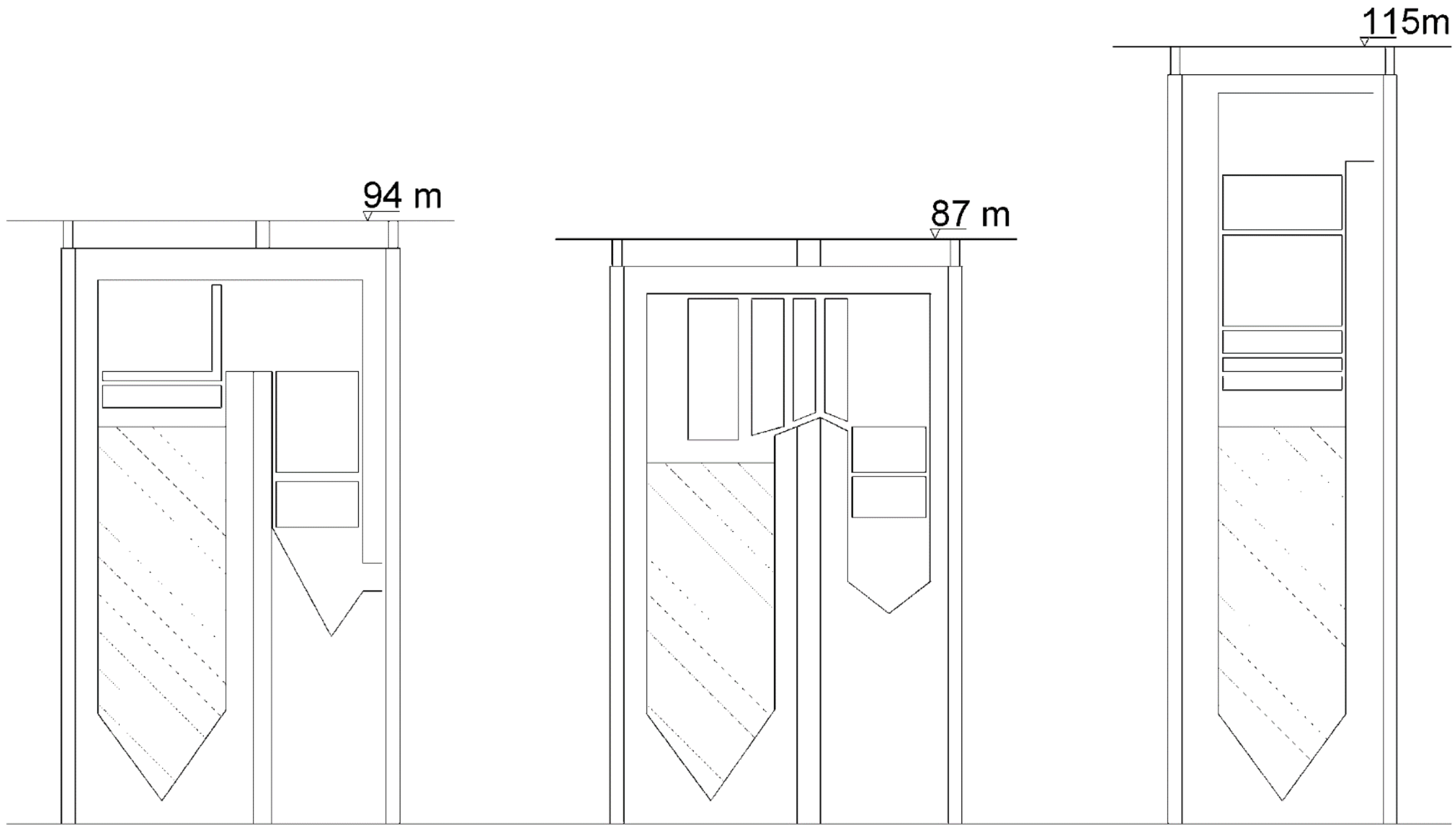
※ HRR : Heat Release Rate (열부하율)

4.2.3 Steam Generator Construction Types

In single-pass or tower boilers, the convective heating surfaces (the superheater, reheater and economiser) are mounted above the furnace, so that the flue gases only have to be redirected after the last water vapour/steam heating surfaces. This helps to minimise erosion, in particular with high-ash coal types. Only after being cooled down to 400◦C are the flue gases conducted to the air preheater through an uncooled blank pass.

Two-pass boilers offer more favourable conditions for heat transfer by introducing a second pass and adapting its cross-section to the volumetric flow through it, which decreases with falling flue gas temperatures. Two-pass boilers can be built with hanging superheater surfaces – the so-called plate heating surfaces – with wide spacings of about 1m, hanging from the ceiling of the first pass. These heating surfaces are suited to high temperatures of more than 1,230◦C. Incorporated into the design, they create a rather compact boiler construction, meaning 5–10% lower investment costs in comparison to single-pass boilers. Hanging heating surfaces, though, are not suitable for a frequent start-up/shutdown operation mode, because they cannot be drained.

4.2.3.1 Single-Pass Boilers and Two-Pass Boilers

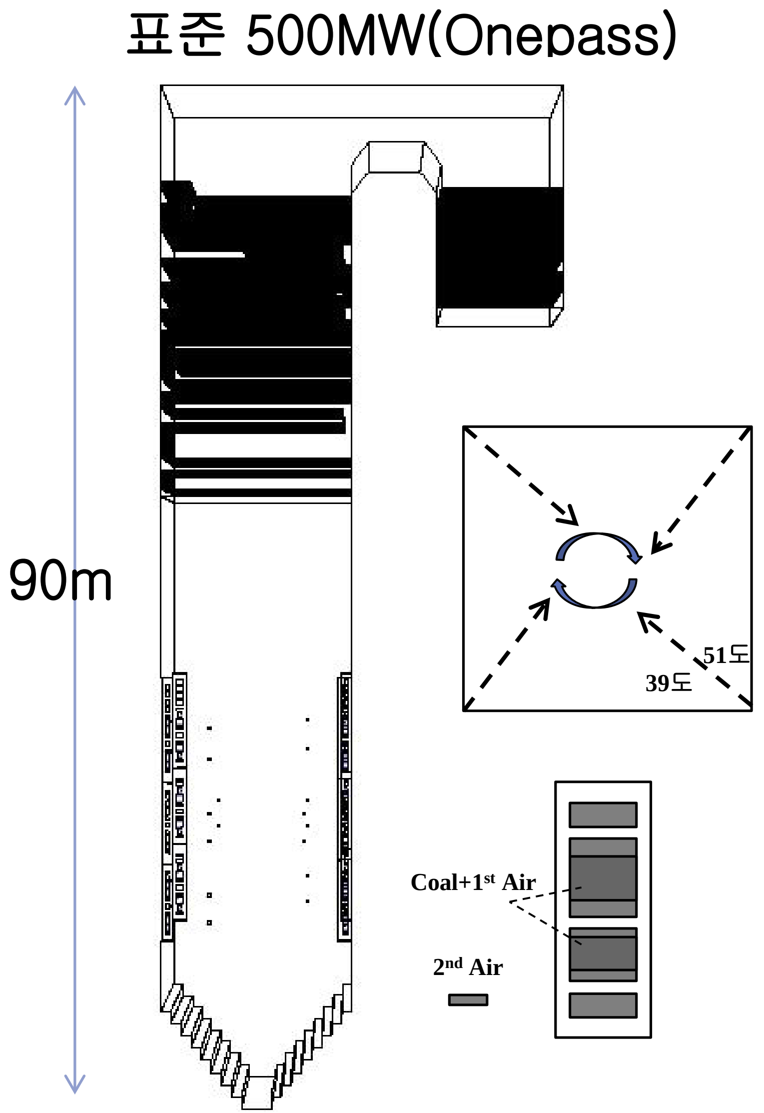


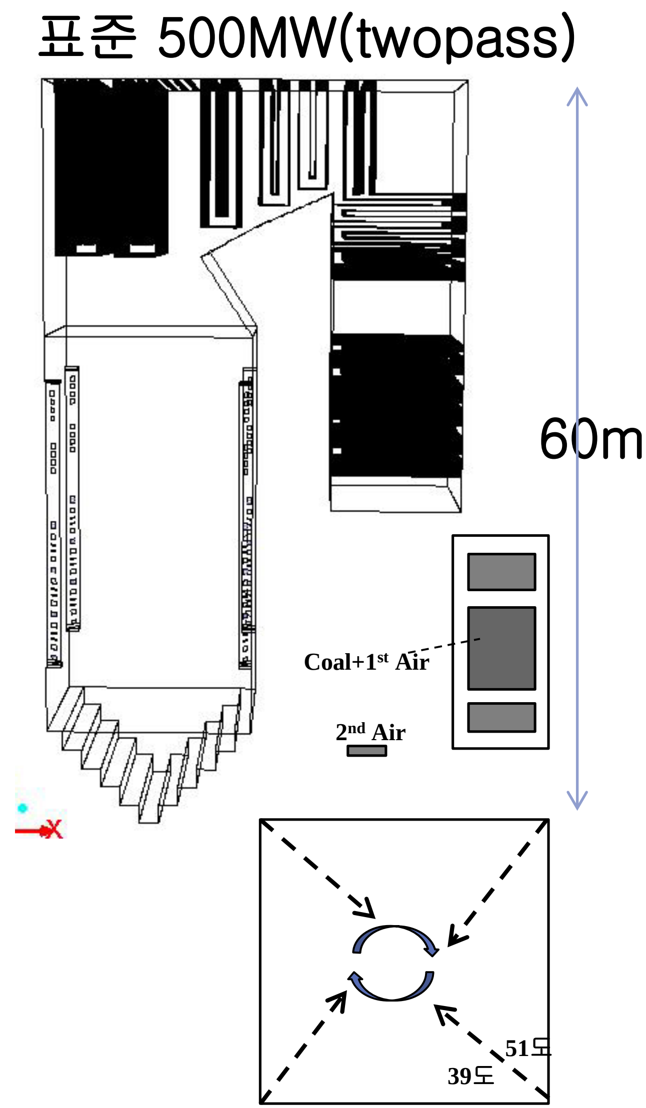
Two-pass construction with horizontal heating surfaces

Two-pass construction with hanging heating surfaces

Single-pass construction (Tower boiler)

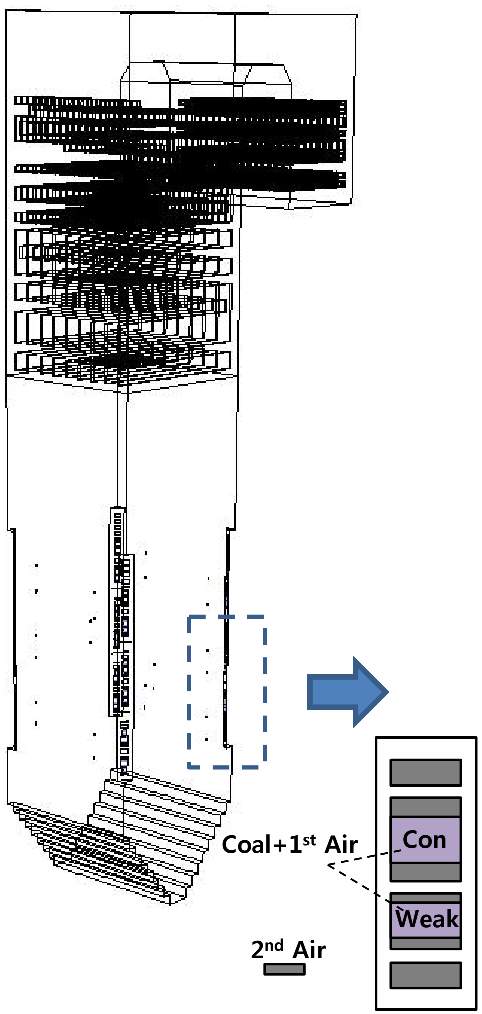
4.2.3.1 Single-Pass Boilers and Two-Pass Boilers





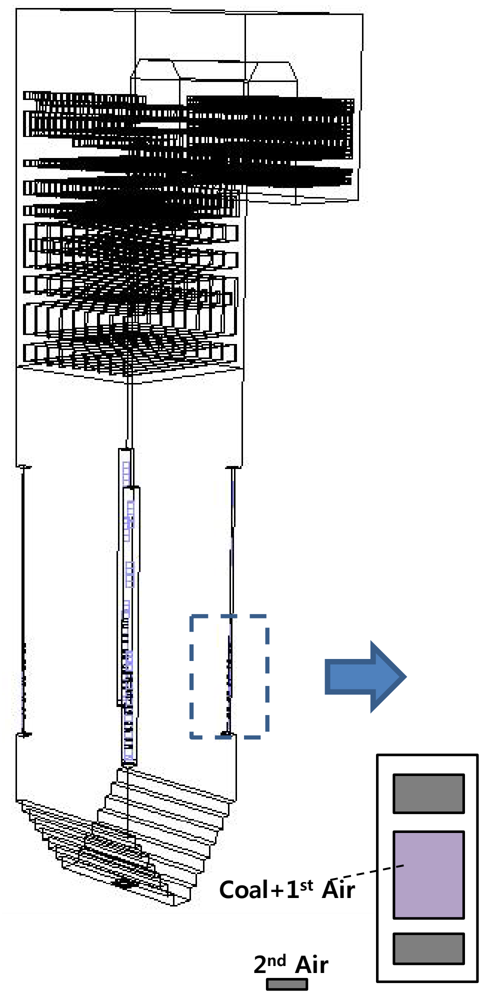
4.2.3.1 Single-Pass Boilers and Two-Pass Boilers

기존의 내부 형상 Geometry



Aux air
Aux air
F Weak coal
F Conc.coal
Aux air
Oil
Aux air
E Conc. Coal
E Weak coal
Bottom air
Aux air
Aux air
D Weak coal
D Conc.coal
Aux air
Oil
Aux air
C Conc. Coal
C Weak coal
Bottom air
Aux air
Aux air
B Weak coal
B Conc.coal
Aux air
Oil
Aux air
A Conc. Coal
A Weak coal
Bottom air

버너부가 개조된 Geometry



CROTCH
CC OFA
CC OFA
End Air
F COAL
Aux Oil
CFS
E COAL
CFS
Aux Air
CFS
D COAL
CFS
Aux Oil
CFS
C COAL
CFS
Aux air
CFS
B COAL
CFS
AUX Oil
CFS
A COAL
END Air
U.F.A
CROTCH

열 부하율 :

연소로 크기를 좌우하는 열 부하율은 증기조건과는 무관하며 연소 되는 석탄 및 회분 의 특성, 즉 회분양이나 용융온도 , Slagging / Fouling 정도에 따라 결정된다.

1. 노 단면적 열 부하율 (Plan Area Heat Release Rate : 연료 입열/ 노 단면적)

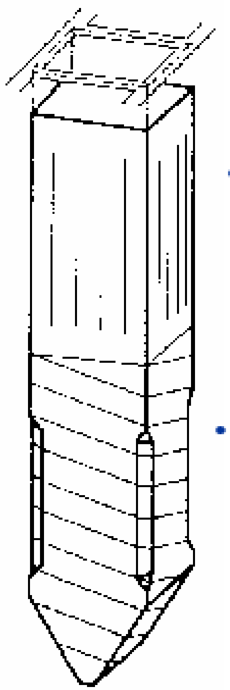
노 단면적 : 버너 지역을 통과하는 노의 평면적

열 부하율	단 위	표준 500MW	후속기 500MW
노 단면적	kcal/m^{2}h	4,300,000	4,690,000

2. 투영 열 면적 열 부하율 (Projected Area Heat Release Rate : 순수 입열/투영 열 면적)

투영 열 면적 : 연소로 벽 튜브 중심을 통과하는 모든 튜브의 면적과 튜브간의 간격이 450 mm이하 인 첫번째 대류 열 전열 면적이전에 설치되는 튜브군의 투영 열 전열 면적

열 부하율	단 위	표준 500MW	후속기 500MW
투영 열면적	kcal/m^{2}h	300,000	200,000



- 상 부

  Vertical Wall

- 하 부

  Spiral Wall

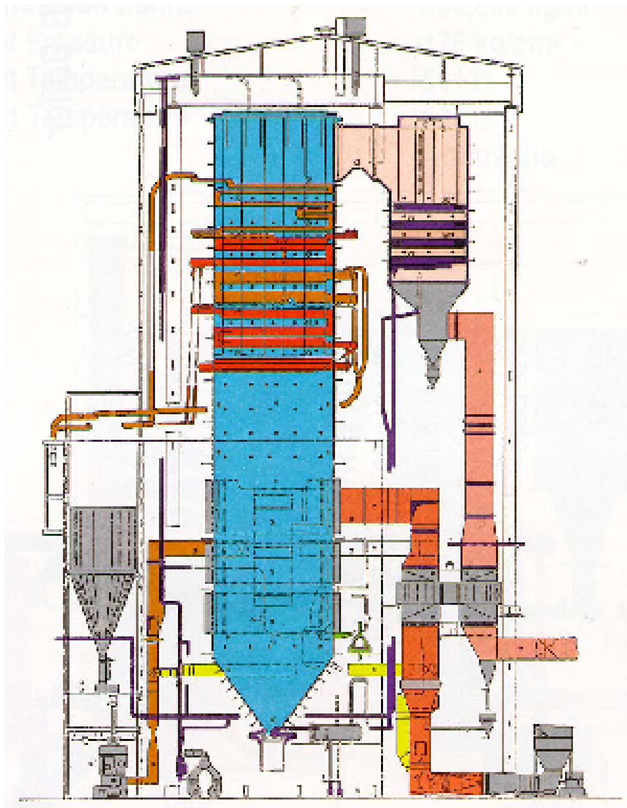
Evaporator 설계 :

항 목	단위	표준 500MW	후속기 500MW
Spiral wall tube O.D. / Spacing	mm	38 / 54	38 / 54
Spiral wall tube angle	Degree	14.2086	13.95
Vertical wall tube O.D. / Spacing	mm	31.8 / 55	34,38 / 56
No of Vertical tube / Spiral tube		4	4

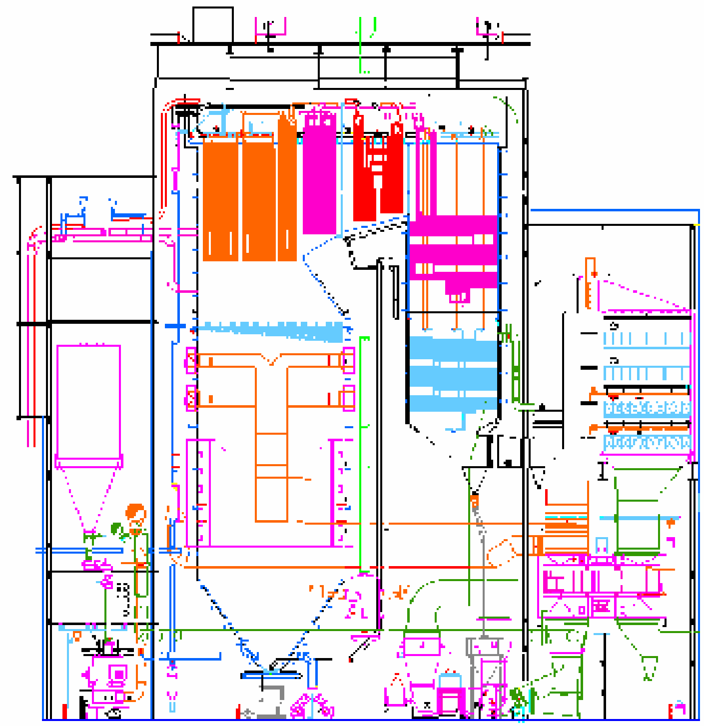
라. 화로 설계 및 화로 출구 온도계산

항목	나선형 관벽 (Spiral wall)	수직형 관벽 (Vertical wall)	비고
튜브형식	Smooth tube	Rifled tube	
장점.	- 모든 튜브가 노벽을 동일한 조건으로 통과하므로 튜브간 열흡수가 차이가 없어 부하변동이나 다양한 석탄 연소시에도 튜브 출구온도가 균일하고 열응력이 최소화되는 우수한 성능 유지. - 기저부하나 변동 부하 모두에 적합.	- 구조가 수직형으로 간단하여 노벽 제작 및 설치비가 상대적으로 낮음. - 노내 slagging 이 심한 탄인 경우 ash slag 처리에 유리. - 나선형에 비해 압손이 다소 낮음.	국내수입탄은 대부분 저 slagging 탄임.
단점	- 노벽 구조가 나선형으로 노벽지지대 (buckstay)가 복잡하고 field 용접수가 많아 노벽 제작 및 설치비가 상대적으로 높음.	- 튜브간 불균일한 열분포특성을 보완하기 위해 특정 부하에서 유효한 유량제어용 orifice 를 사용하여야 하며 부하변동시나 다양한 석탄연소시, 노벽 slagging 특성이 설계와 다를시 튜브출구 온도 편차 및 튜브간 열응력이심화. - 튜브보호를 위해 튜브내경이 적은 튜브가 사용되어 수질이 불량시 급격한 압손증가등 문제가능성 있음. - 과거 초기운전시 Orifice 교체 경우 많음. - 기저부하에 보다 적합. - 사용 용량에제한 (실적은 700MW 이상)	

마. SH, RH, Eco., GAH 등의 전열면 배치 및 가스온도 선정



표준 500MW
One Pass Arrangement



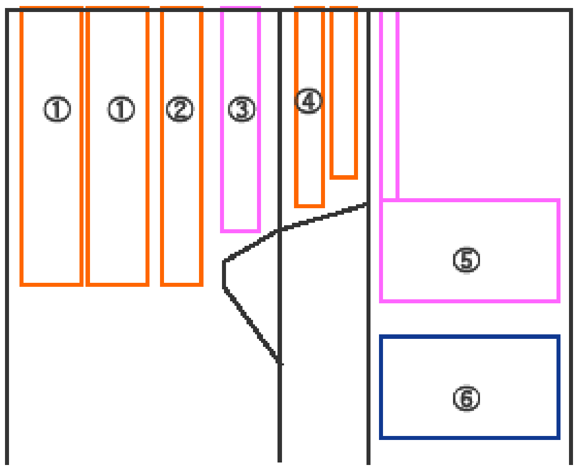
후속기 500MW
Two Pass Arrangement

배치 방식 비교 :

형식	One pass type	Two pass type
주 제작사	Alstom power (구 Sulzer), 독일 EVT 사, 두산	Alstom power (구 C-E), 일본업체, 두산
일반	유럽지역에 주로 설치	일본지역에 주로 설치
장점	- 튜브가 수평식 배열로 배수가 용이해 기동 시간이 짧음. - 연소 개스 속도가 낮아 튜브 마모율이 낮음.	- 튜브가 수직식 배열로 보일러 높이가 낮아 경제적인 설계가 가능. - 보일러 지지 구조가 간단함
단점	- 보일러 높이가 높아 지진이 심하거나 바람이 심한 지역은 사용 제한 - 보일러 지지 구조가 복잡함.	- 튜브가 수직식 배열로 배수가 어려워 기동 시간이 김. - 연소 개스 흐름변화로 보일러 후부에서 마모 발생 가능성 있음.

마. SH, RH, Eco., GAH 등의 전열면 배치 및 가스온도 선정

TWO PASS TyPE 전열면 배치

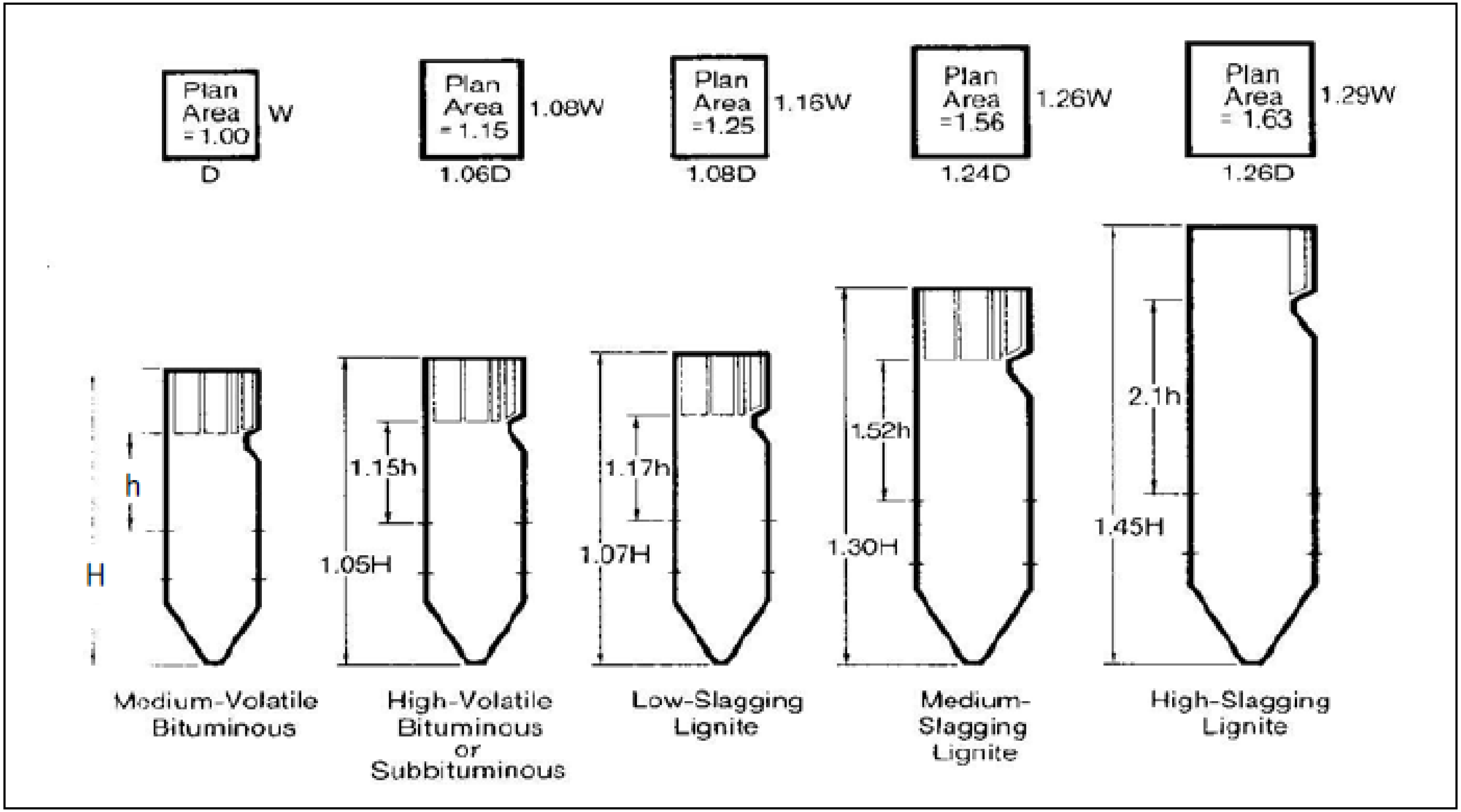


튜브 군에서의 연소 개스 최대 속도 : **15 m/sec**

튜브 군에서의 튜브 간격 :

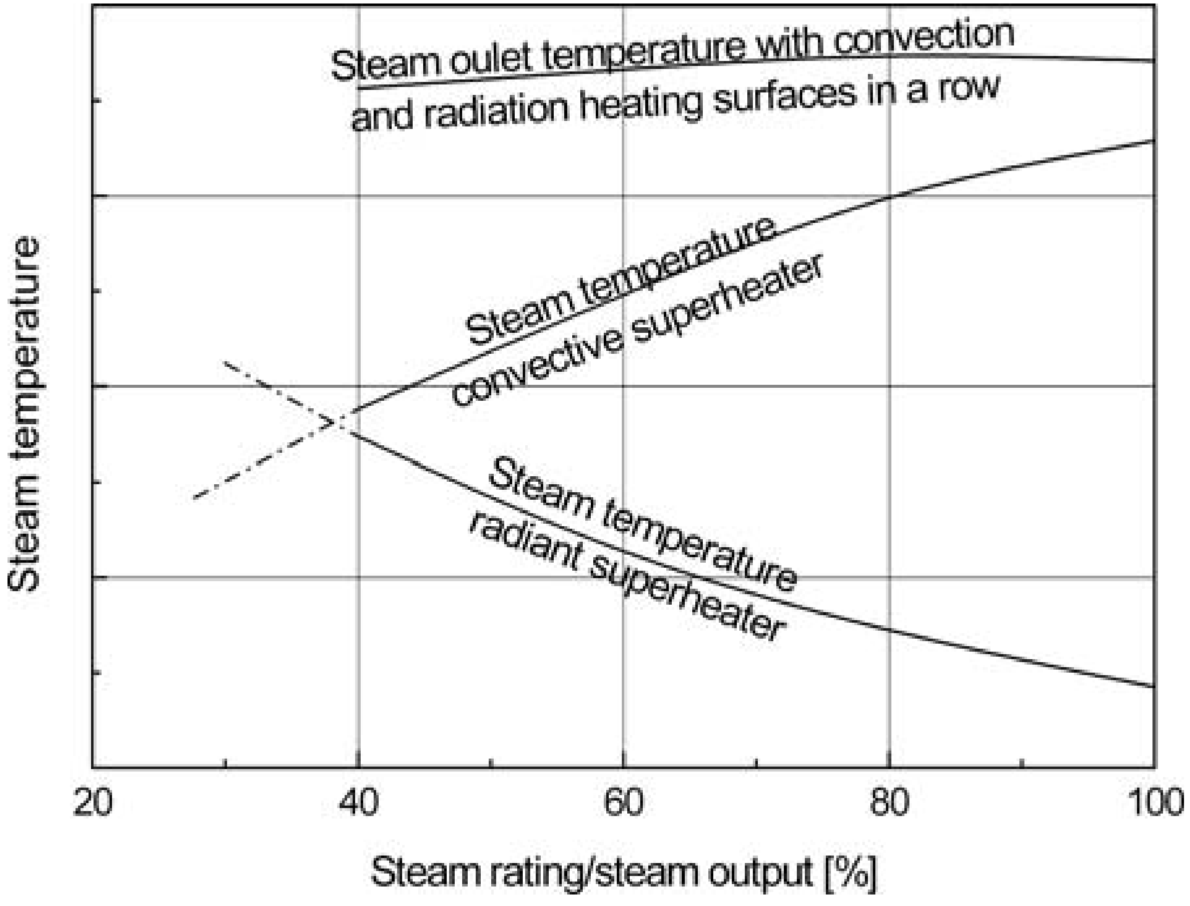
No	Section	Transverse Spacing (mm)	Longitudinal Spacing (mm)
1	SH Panel	3,584	OD + 10
2	SH Platen	896	OD + 10
3	RH Final	672	2 x OD
4	SH Final	336	2 x OD
5	Horizontal RH	168	2 x OD
6	Economizer	112	2 x OD

4.2.3.1 Single-Pass Boilers and Two-Pass Boilers



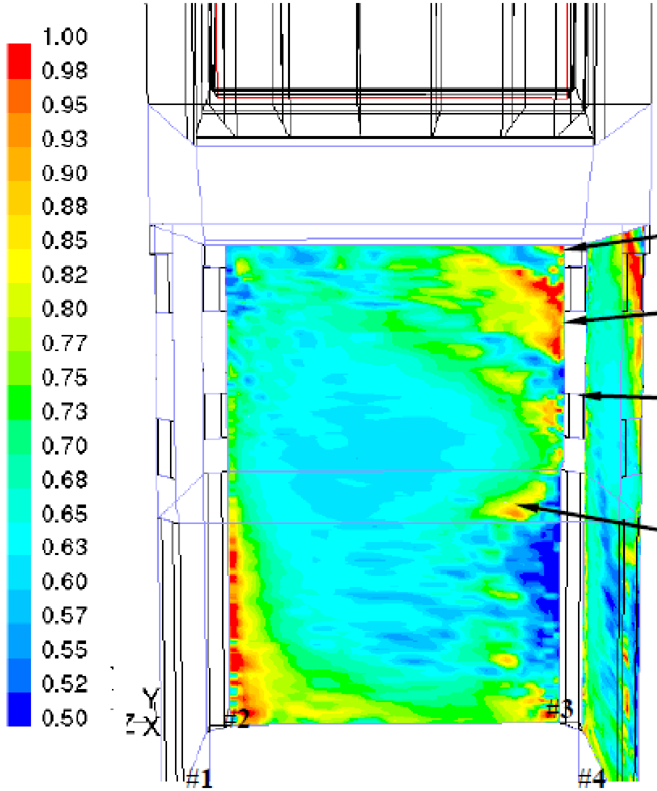
[그림 3-1] 미분탄 노의 석탄등급에 따른 크기

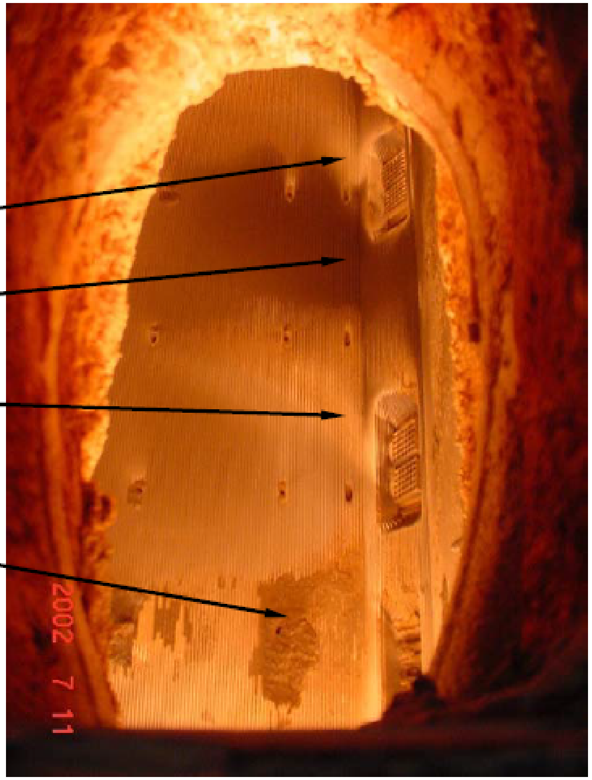
4.2.3.1 Single-Pass Boilers and Two-Pass Boilers



4.2.3.1 Single-Pass Boilers and Two-Pass Boilers

Furnace Wall Slagging Deposit

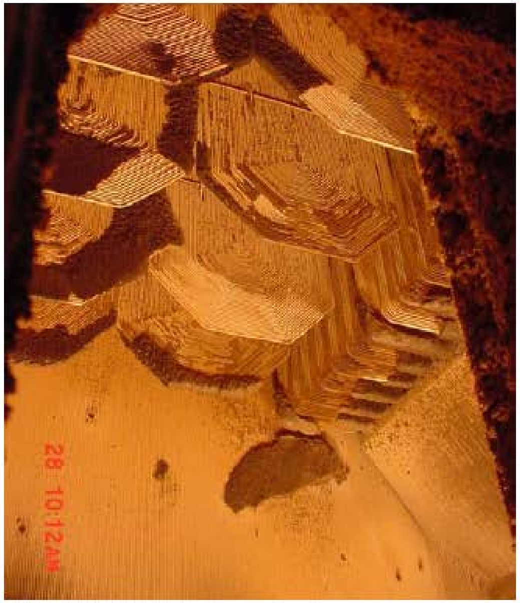




Deposit thickness (inch) 6 hour operation w/o Sootblowing

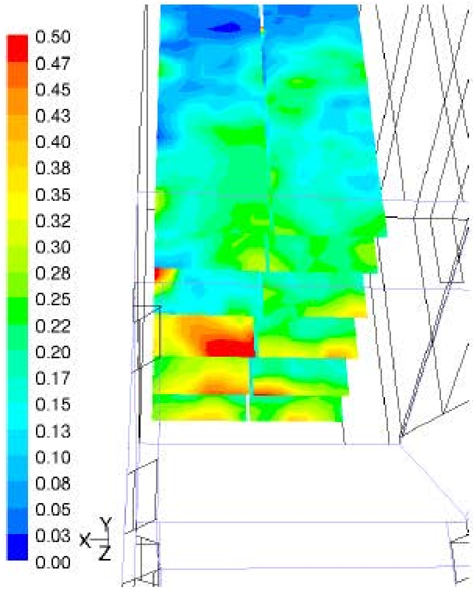
4.2.3.1 Single-Pass Boilers and Two-Pass Boilers

SH Division Panel Slagging Deposit



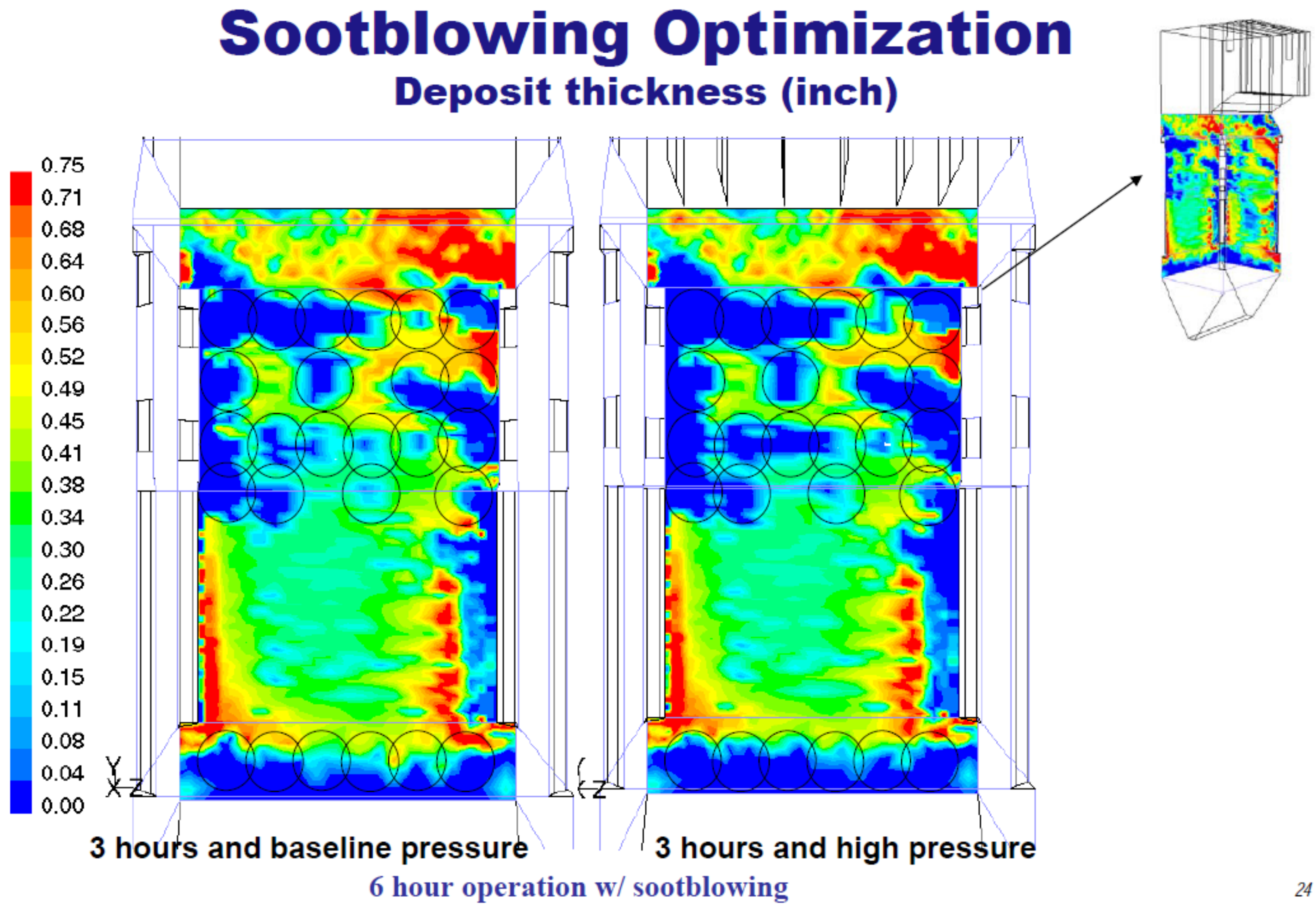
Picture taken on Feb 28th , 2007

Picture taken on March 3rd, 2007

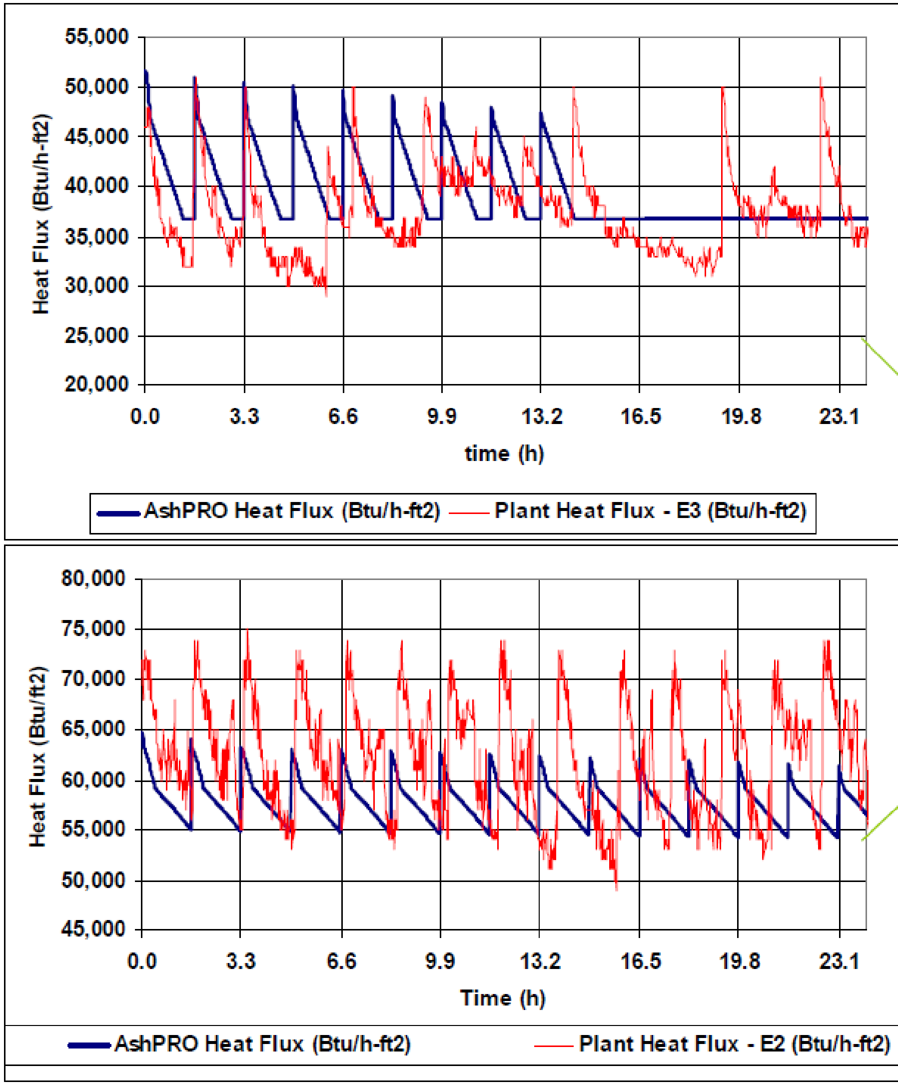


Deposit thickness (inch) one-hour operation w/o sootblowing

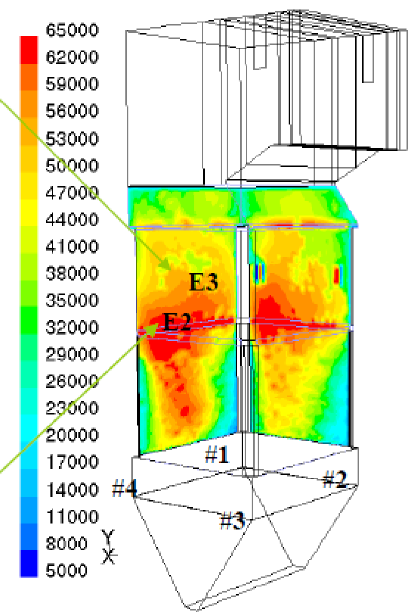
4.2.3.1 Single-Pass Boilers and Two-Pass Boilers



4.2.3.1 Single-Pass Boilers and Two-Pass Boilers



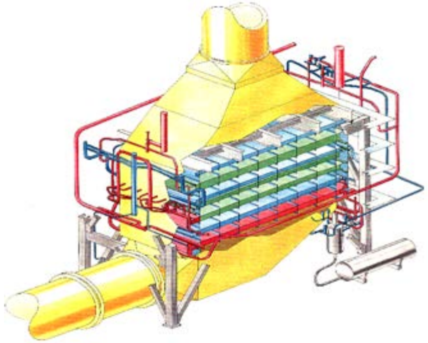
Heat Flux Data Comparison



Heat Flux (Btu/hr-ft^2)

HFO-HRSG의

Ash fouling의 전산 해석



2012. 11. 26

부산대학교

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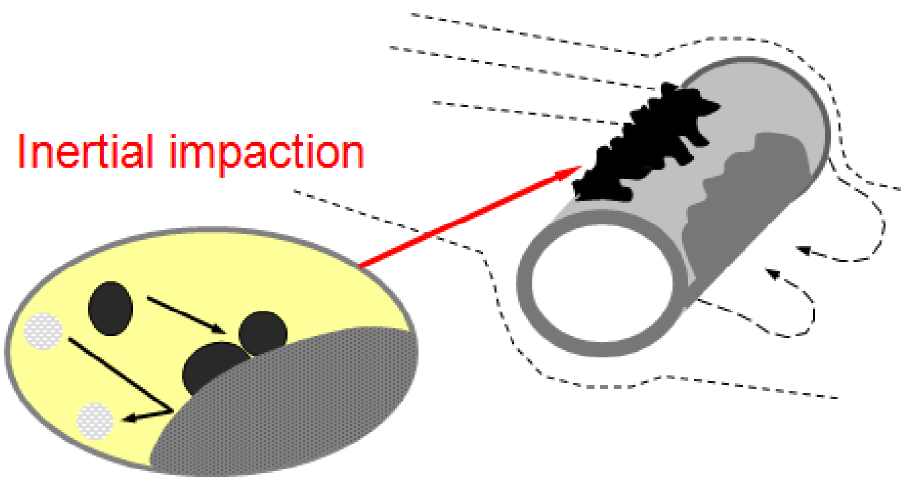
PUSAN NATIONAL UNIVERSITY

DOOSAN 두산중공업

Contents

Deposition mechanisms



[ash deposition rate]

$$\frac{dm}{dt} = I \cdot G + E + T + C + R$$

I = Inertial impaction rate
G = Capture efficiency
E =Eddy impaction
T =Thermophorestic deposition rate
C =condensation rate
R =Chemical reaction rate

Deposition mechanism	Driving forces
Inertial impaction	Momentum
Eddy impaction	Turbulence intensity
Thermophoresis	Temperature gradient
Condensation	Vapor pressure of alkali salt
Chemical reaction rate	Species concentration gradient

Ref. Bryers, Prog. Energ. Sci, 22(1):29-120, 1996

HRSG model 1

- "Modeling the slag layer in solid fuel gasification and combustion – Formulation and sensitivity analysis" Fuel 92 (2012) 162–170

$$We \equiv \frac{Particle\_kinetic\_Energy}{Surface\_tension\_Energy} = \frac{\rho_p v_p^2 d_p}{\sigma_{sp}}$$

We >= 1 , Rebound

We < 1 , Deposit to the surface

[Ghoniem et al, Fuel, MIT]

$$\rho_p = \frac{\sum M_i x_i}{(1+0.0001(T_{particle} - 1773))\sum \overline{V}_i x_i}$$

$$\sigma_{sp} = \sigma_p - \sigma_s \cos\theta$$

$$\sigma_p = (\sum \overline{\sigma}_i x_i - 0.15(T_{particle} - 1733))10^{-3}$$ (Particle surface tension)

$$\sigma_s = (\sum \overline{\sigma}_i x_i - 0.15(T_{slag} - 1733))10^{-3}$$ (Slag surface tension)

[Mills & Rhine]

$$T_{slag} = \frac{5T_s}{8} + \frac{3T_{cv}}{8}$$

Critical Viscosity Temperature
(chungen, 1998)

HRSG model 1

- "Modeling the slag layer in solid fuel gasification and combustion – Formulation and sensitivity analysis" Fuel 92 (2012) 162–170

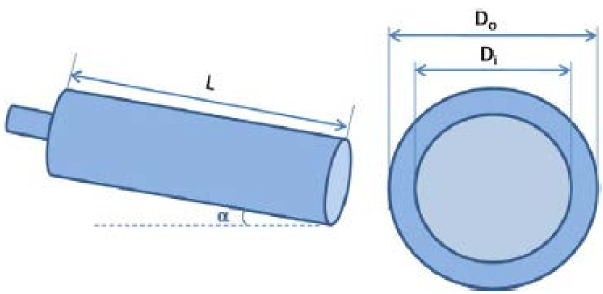
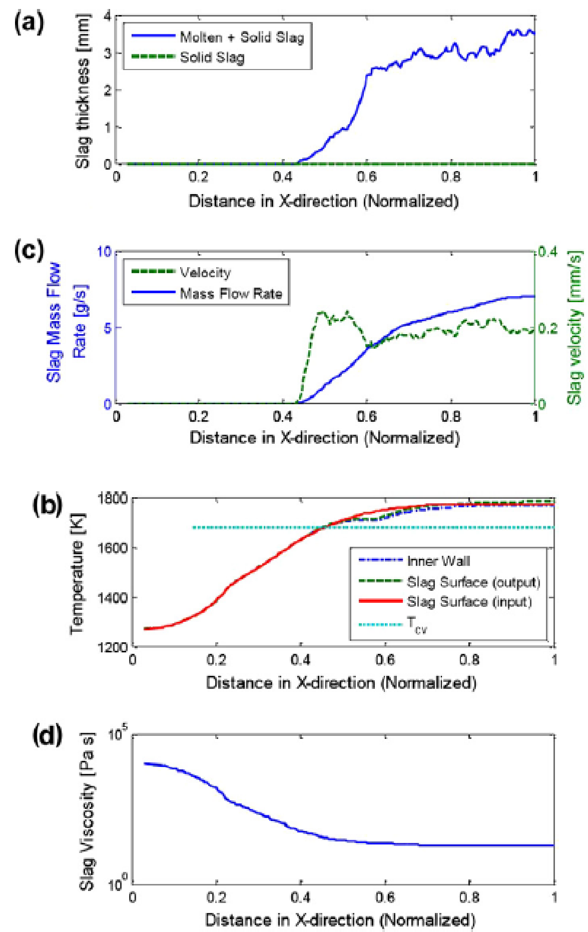


Table 2
Properties of raw coal and ash.

Coal Proximate Analysis		Oxide wt% of ash	
Moisture (%)	6.4	SiO_2	44.35
Ash (%)	7	TiO_2	1.56
Volatile matter (%)	33.1	Al_2O_3	30.88
Fixed carbon (%)	53.5	CaO	3.82
		MgO	3.14
Coal Ultimate Analysis		Na_2O	0.76
Carbon (%)	71.1	K_2O	0.67
Hydrogen (%)	4.7	P_2O_5	1.027
Moisture (%)	6.4	Mn_3O_4	0.1
Ash (%)	7	SO_3	0.85
Sulphur (%)	0.5	Fe_2O_3	4.51
Nitrogen (%)	1.2		
Oxygen (%)	9.086		
Chlorine (%)	0.014		
Fluorine (ppm)	34.6		
Properties		Range	
T_{cv} (K)		1680	
Viscosity (Pa s)		6.21 – 334.47	
Density (kg/m³)		2779.9-2887.9	
Specific heat (kJ/kg K)		1.3825	
Thermal conductivity (W/m K)		1.7294 – 1.7966	



HRSG model 4

- Krzysztof Wacławiak, Sylwester Kalisz, "A practical numerical approach for prediction of particulate fouling in PC boilers," Fuel 97 (2012) 38–48

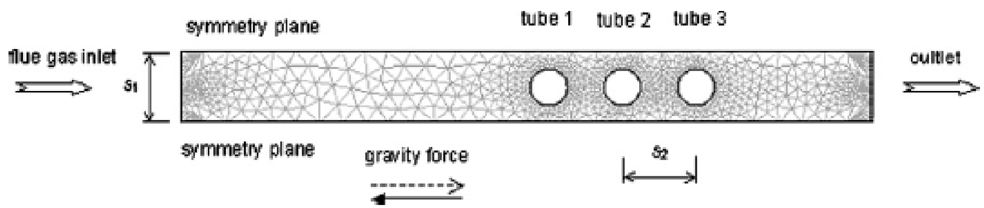


Fig. 6. Mesh for modeled in-line superheater tube bank.

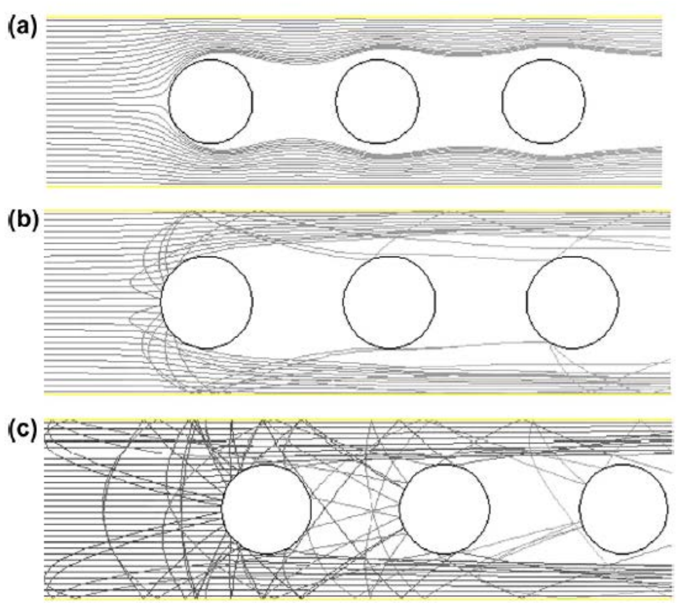
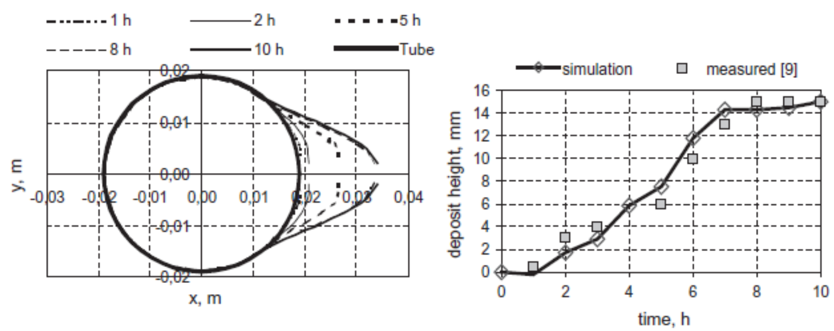


Fig. 8. Trajectories of particles of 10 μm (a), 50 μm (b), 150 μm (c) diameter and velocity of 7 m/s.



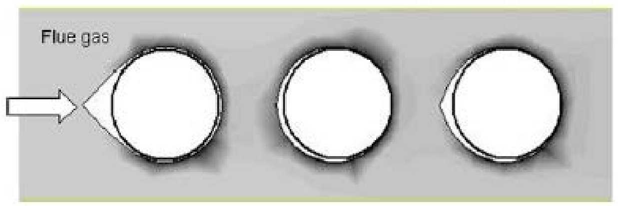
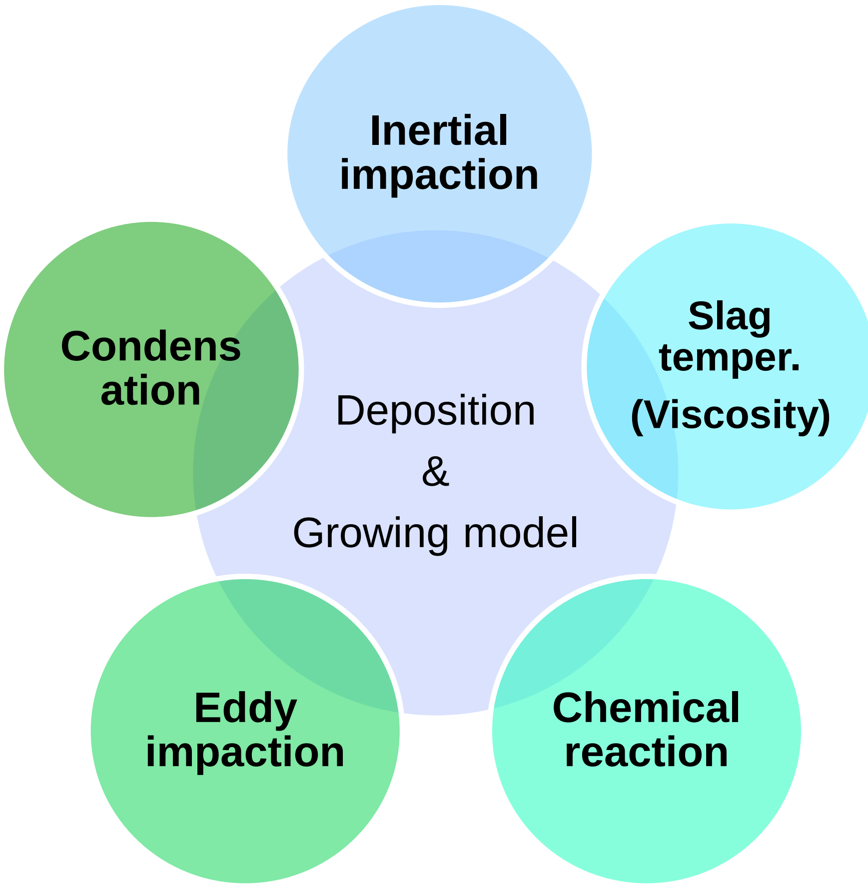


Fig. 15. Simulations of deposit shape after 32 h of fouling on the surface of SRH.

What is important factor in our system ?

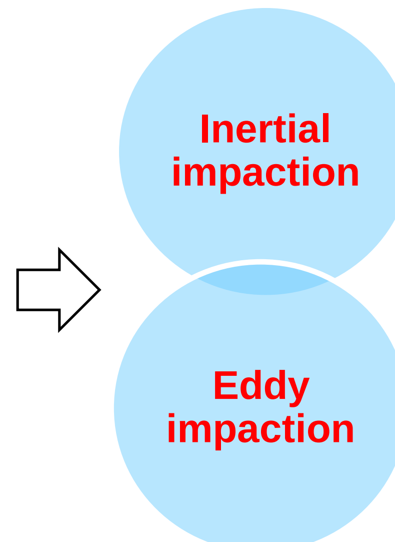


▪ 배기가스 온도 : **800K**

배기가스 조성	N2	O2	CO2	H2O	Ar	SO2
WT%	71.74	15.09	7.01	3.94	1.22	0.99
Vol%	74.09	13.64	4.61	6.33	0.89	0.45

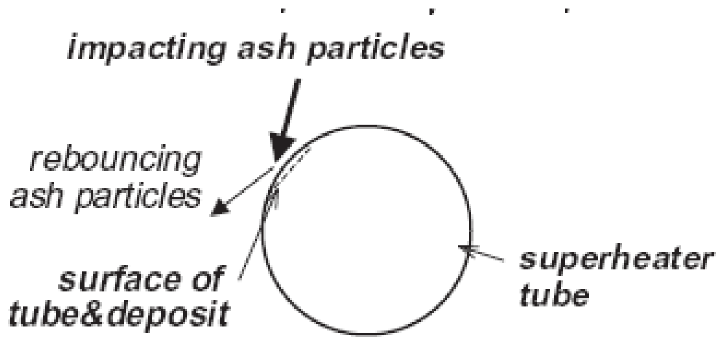
▪ 미립자 조성

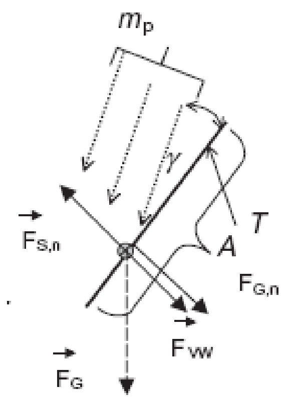
Element	Total
MgO	17.4015
SO_3	51.0805
V_2O_5	5.2025
NiO	1.3105
Al_2O_3	0.903
SiO_2	0.8395
Fe_2O_3	13.0295
Na_2O	0.0305
CaO	0.023
C	10.1795
Sum	100



Melting temperature 2470K (Fact-sage results)

Fouling criteria





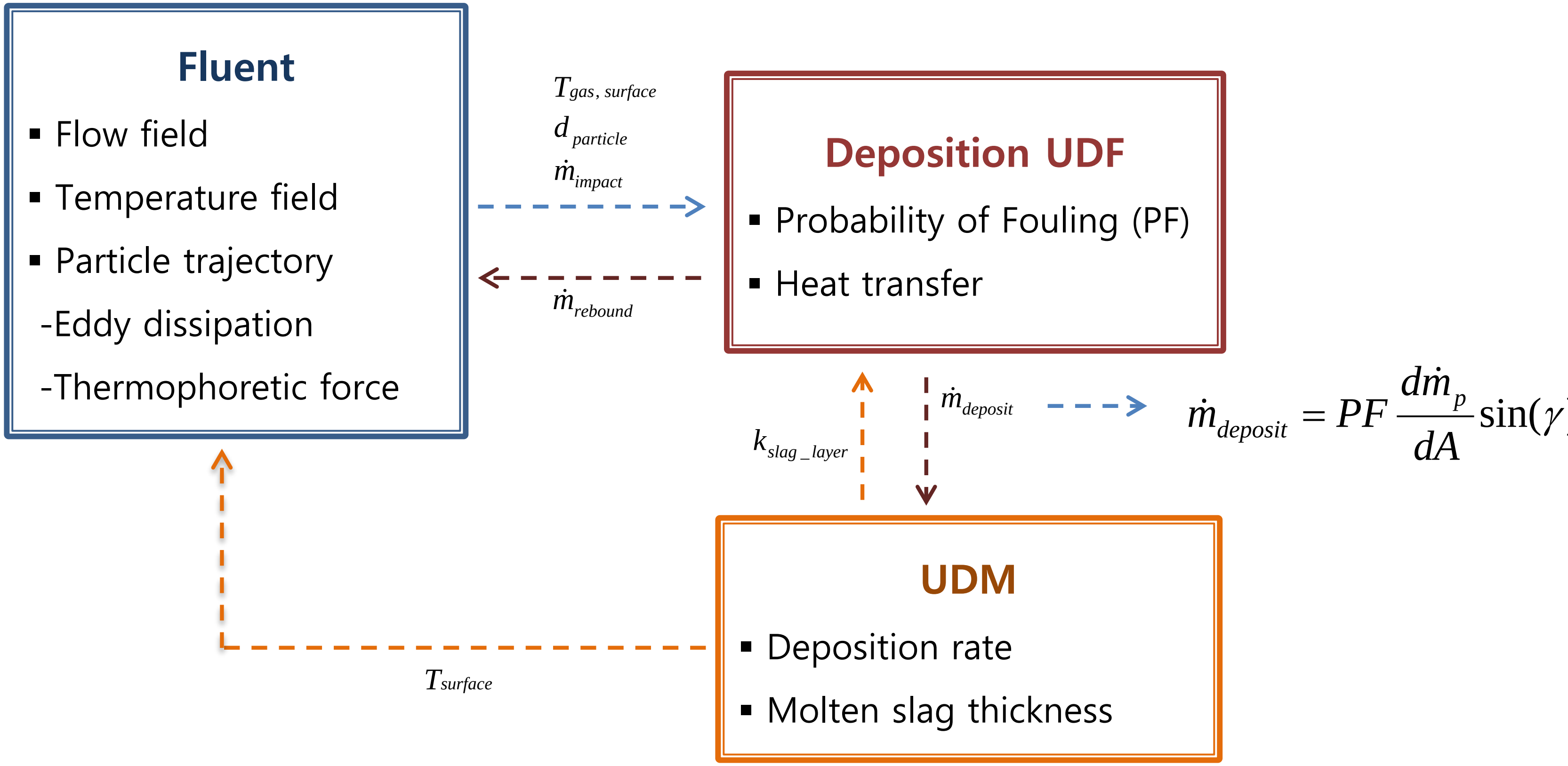
$$PF \equiv \frac{Elastic\_rebound\_force}{Van\_del\_waals\_force} = \frac{Kd_p^2 w_p^{1,2}}{\dfrac{Bd_p}{6\delta^2}}$$

$$PF = \frac{0, \quad if\,(F_{ER} > F_{VW})}{1, \quad if\,(F_{ER} \le F_{VW})}$$

$$\dot{m}_{deposit} = PF \frac{d\dot{m}_p}{dA} \sin(\gamma)$$

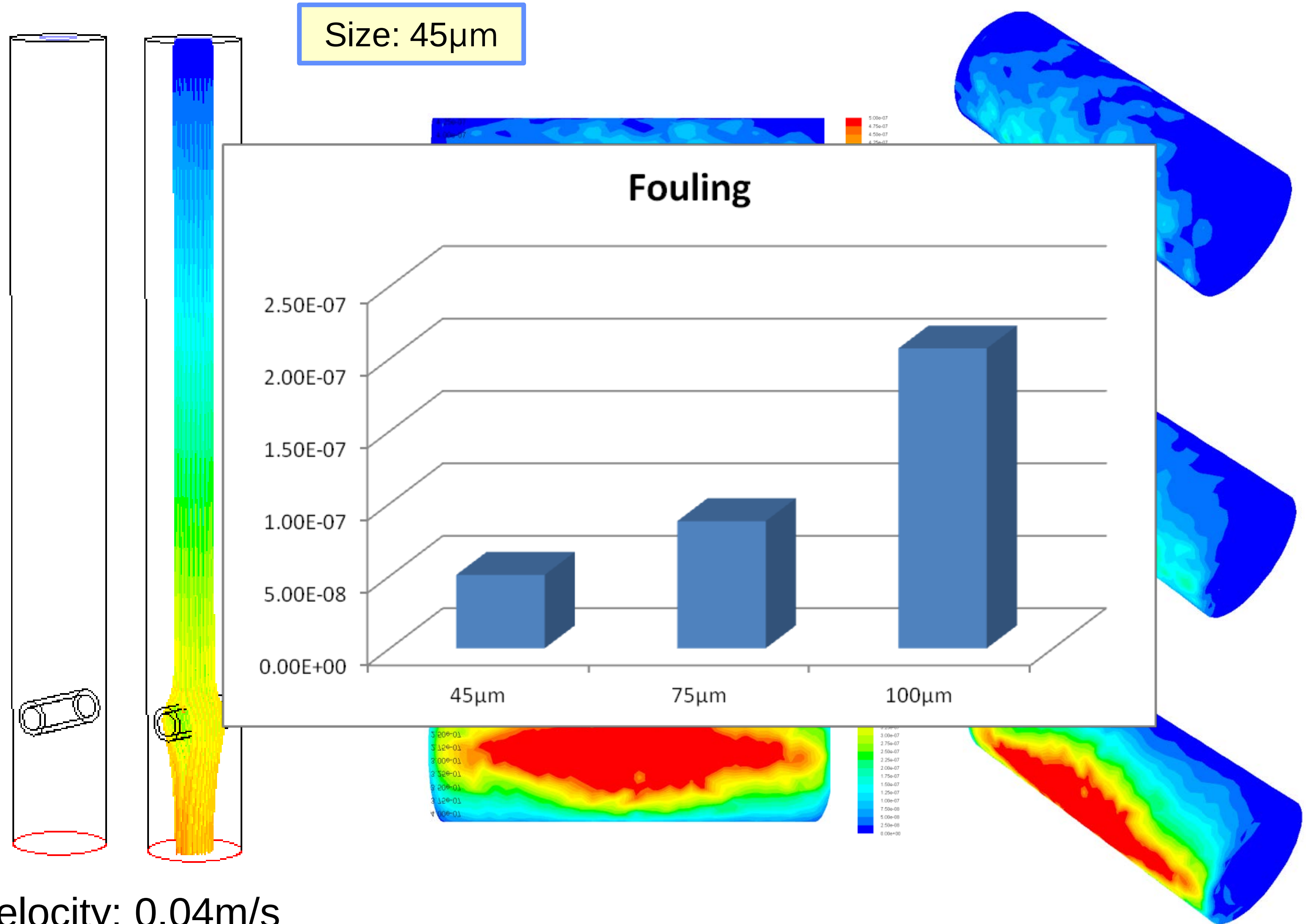
- K : proportionality constant depending on elasticity of particle and deposition surface [K=1.0 * (kgs$^{-0.8}$m$^{-2.2}$)]
- d_p : diameter of particle
- w_p : velocity of particle
- B : molecular interaction constant [B = 1e-17* (kg m^2 s^{-2})]
- δ : distance between particle and deposition surface [0.5 d_p]

Implementation of model in Fluent



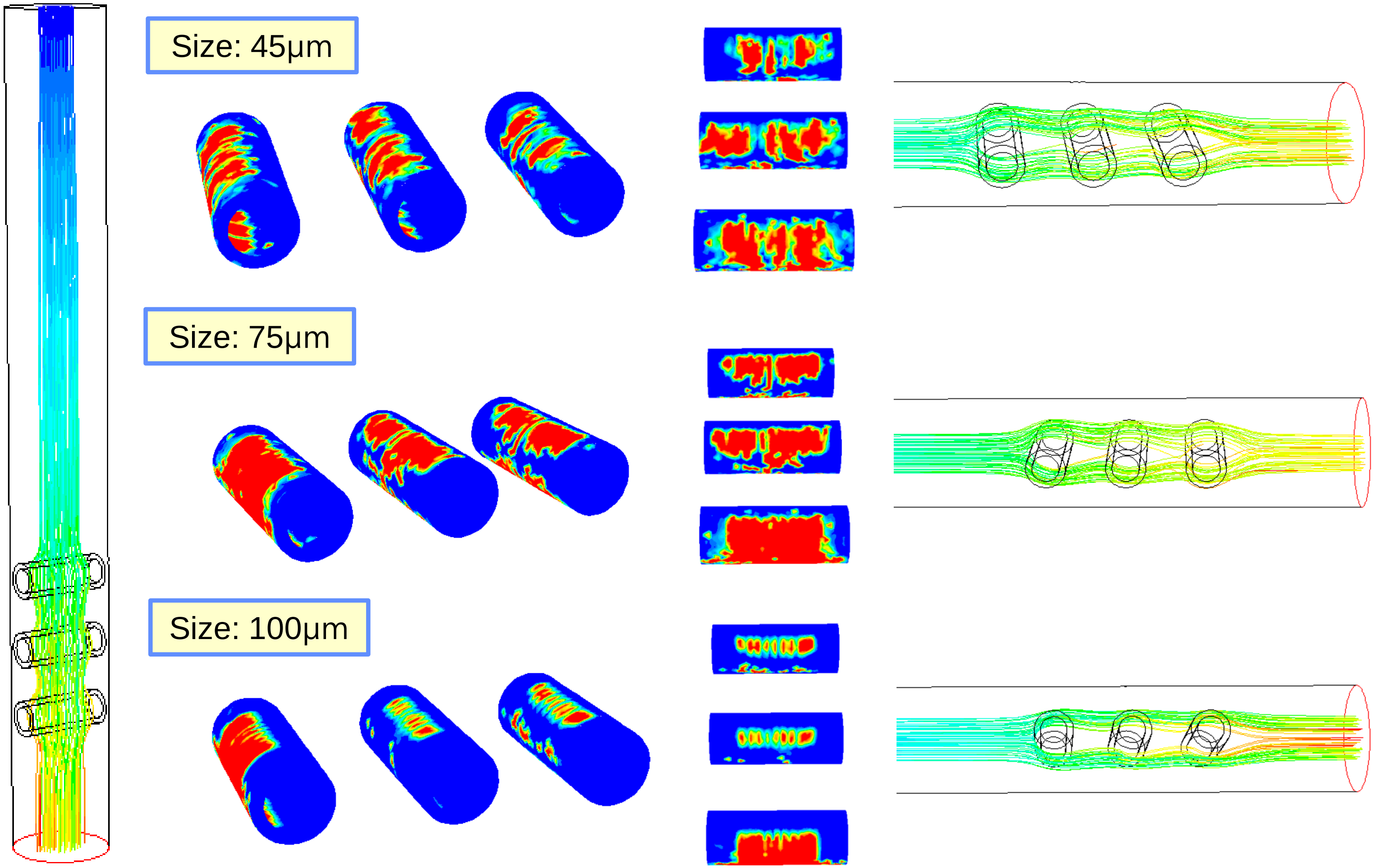
DTF _ Effect of size in single coupon

Size: 45μm

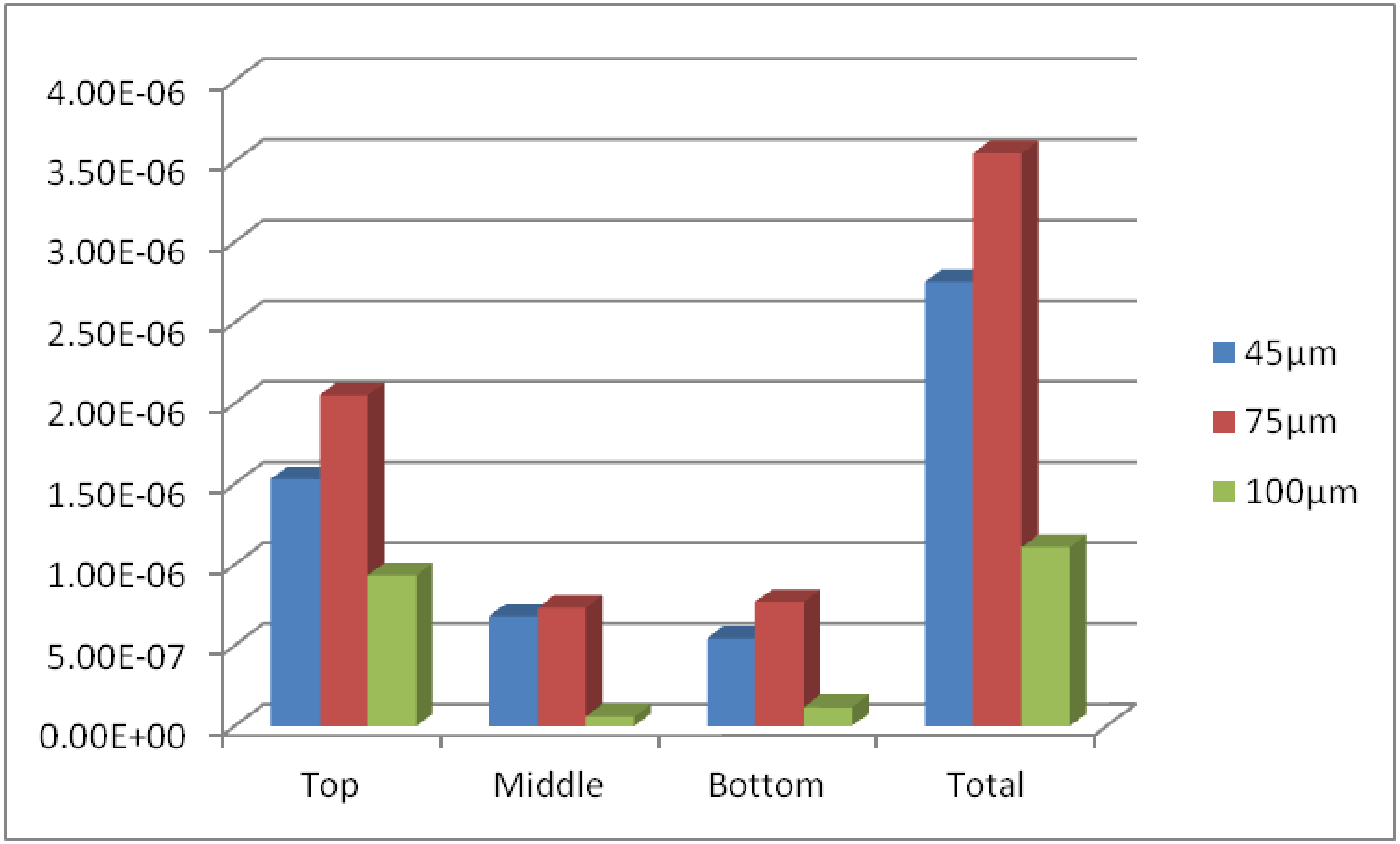


Velocity: 0.04m/s

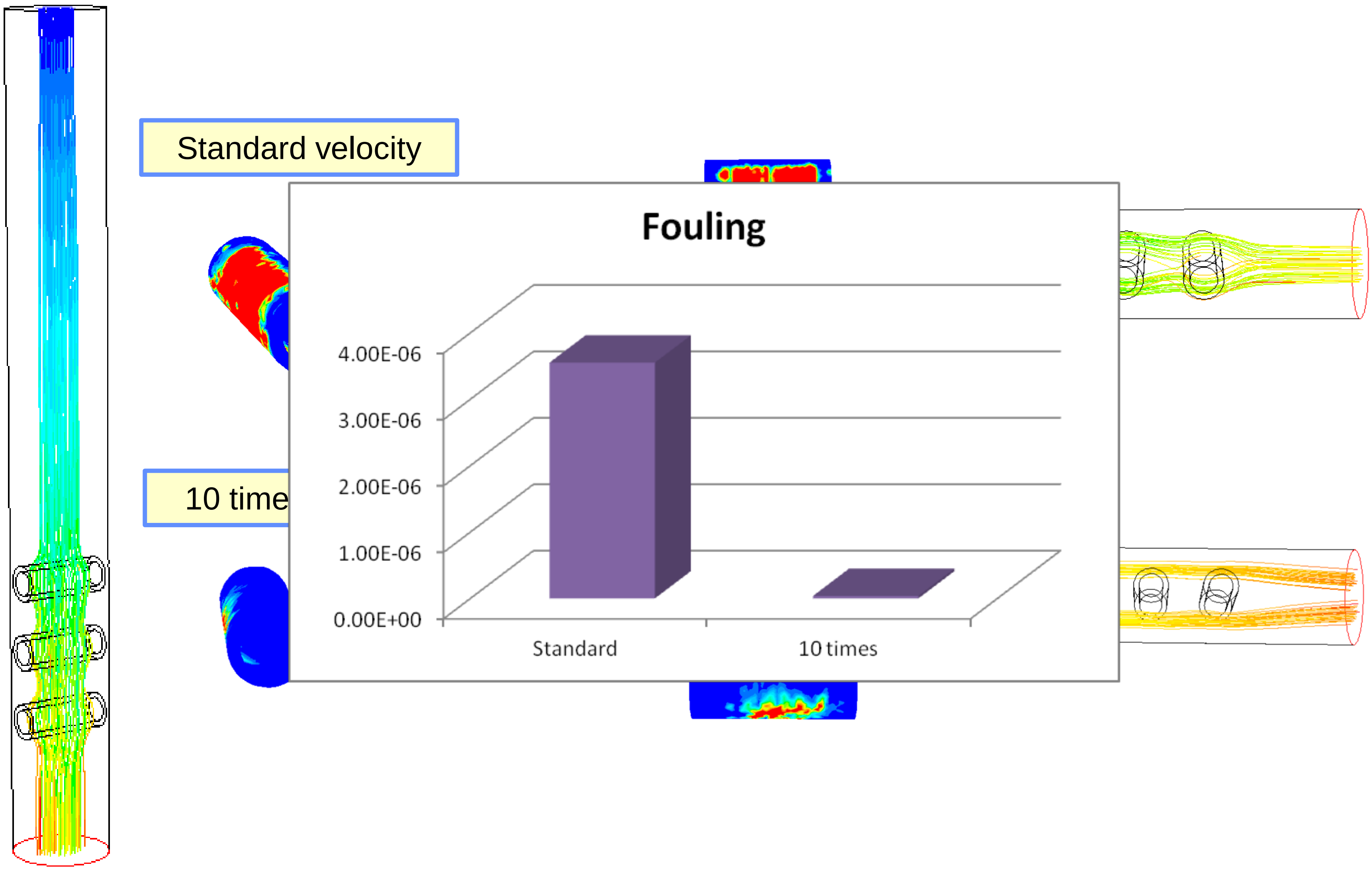
DTF _ Effect of size in three coupons



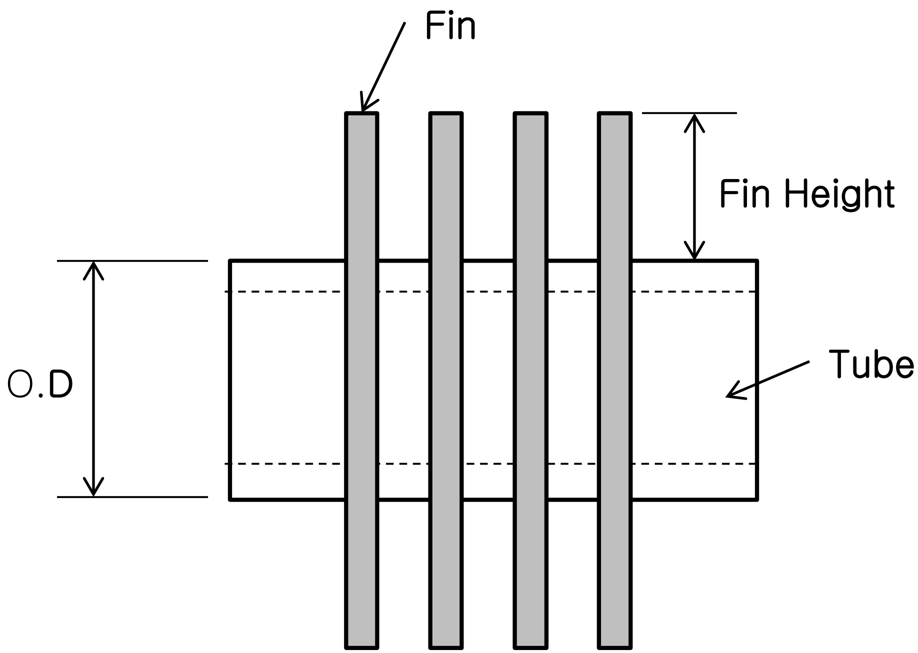
DTF _ Fouling characteristics

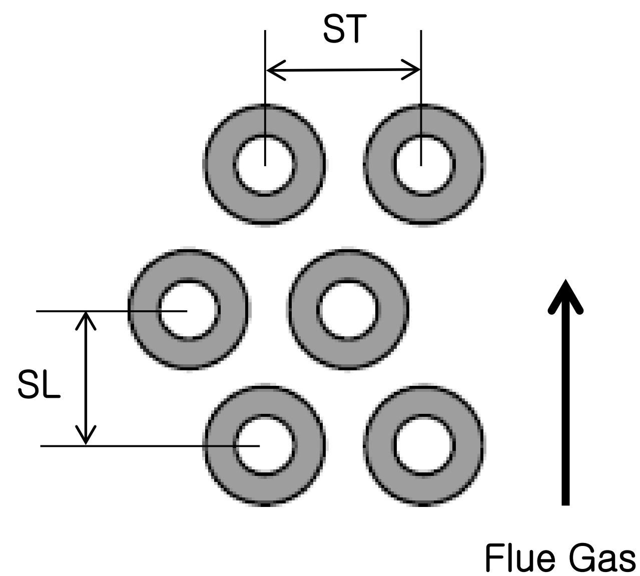


DTF _ Effect of velocity



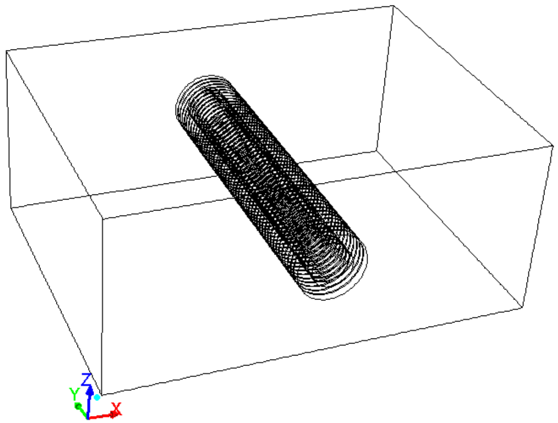
Fin tube _ Geometry information

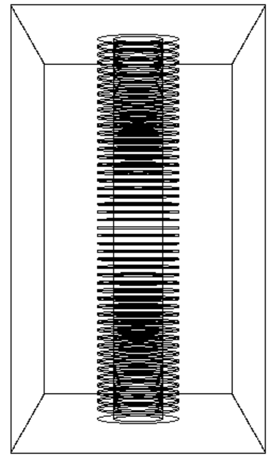


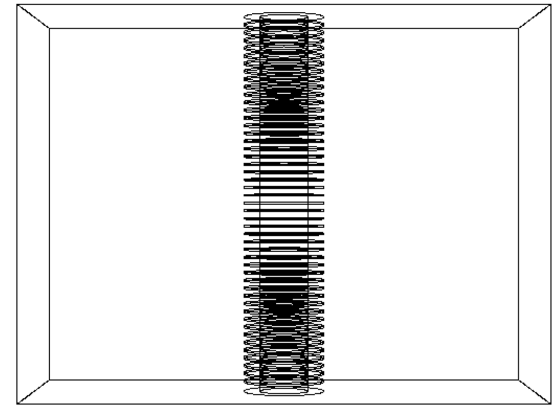


Tube Data			Fin Data		
ST	92	mm	Type	Solid Fin	
SL	79	mm	Thickness	1.3	mm
O.D	38.1	mm	Height	13	mm
Thickness	2.8	mm	Density	160	ea/m
Material	A213-T22		Material	A240-TP409	

Fin tube _ Creation of geometry







-Fin 과 Fin사이 간격 6.21mm

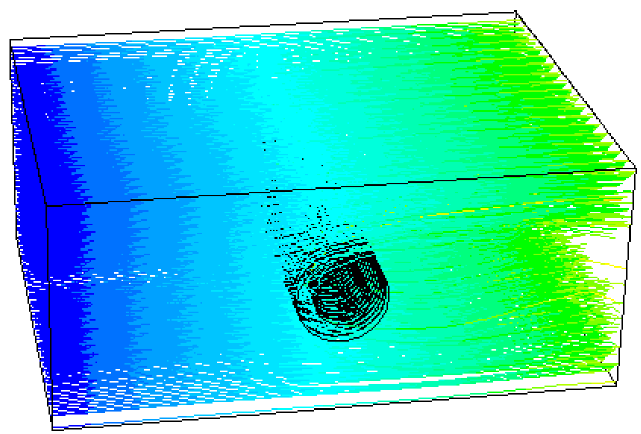
-Fin 두께 1.3mm

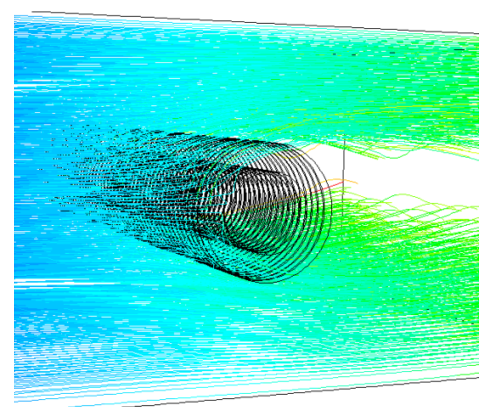
-튜브거리 : 92mm

- Fin 밀도 = 160 ea/m

- 1차적으로 1/3 fin tube 해석 수행

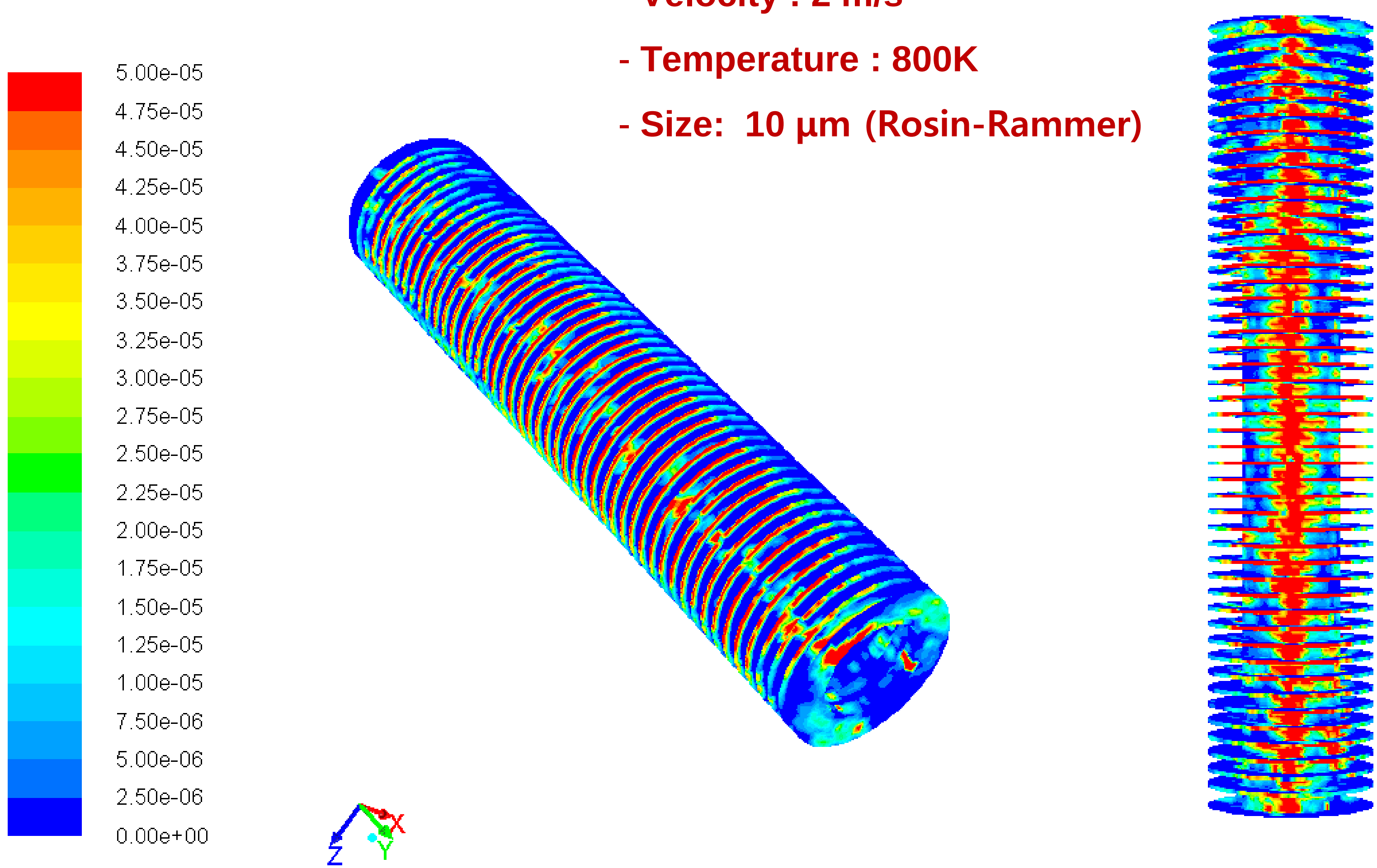
(The number of cell: 1 million)





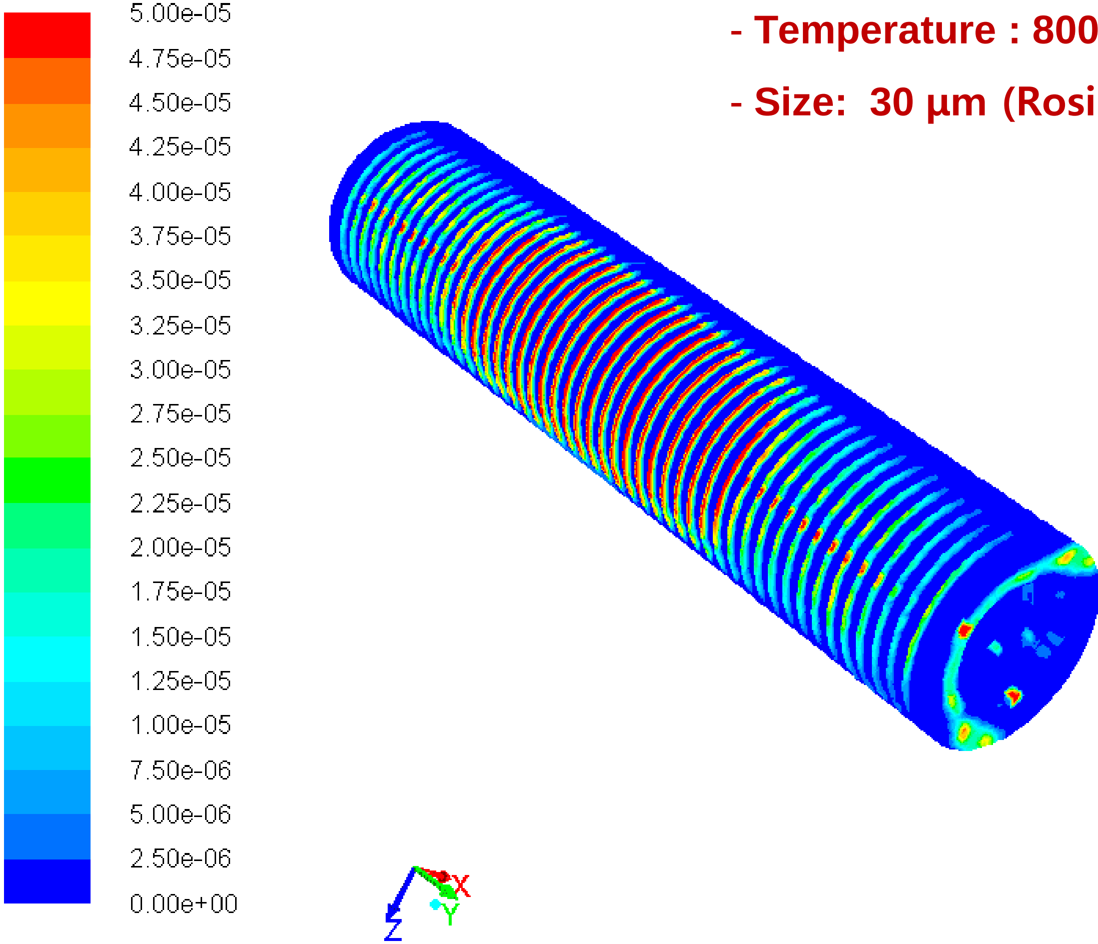
Fin tube _ Fouling characteristics

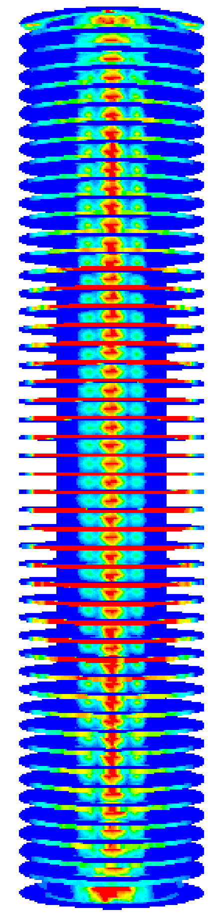
- Velocity : 2 m/s
- Temperature : 800K
- Size: 10 μm (Rosin-Rammer)



Fin tube _ Fouling characteristics

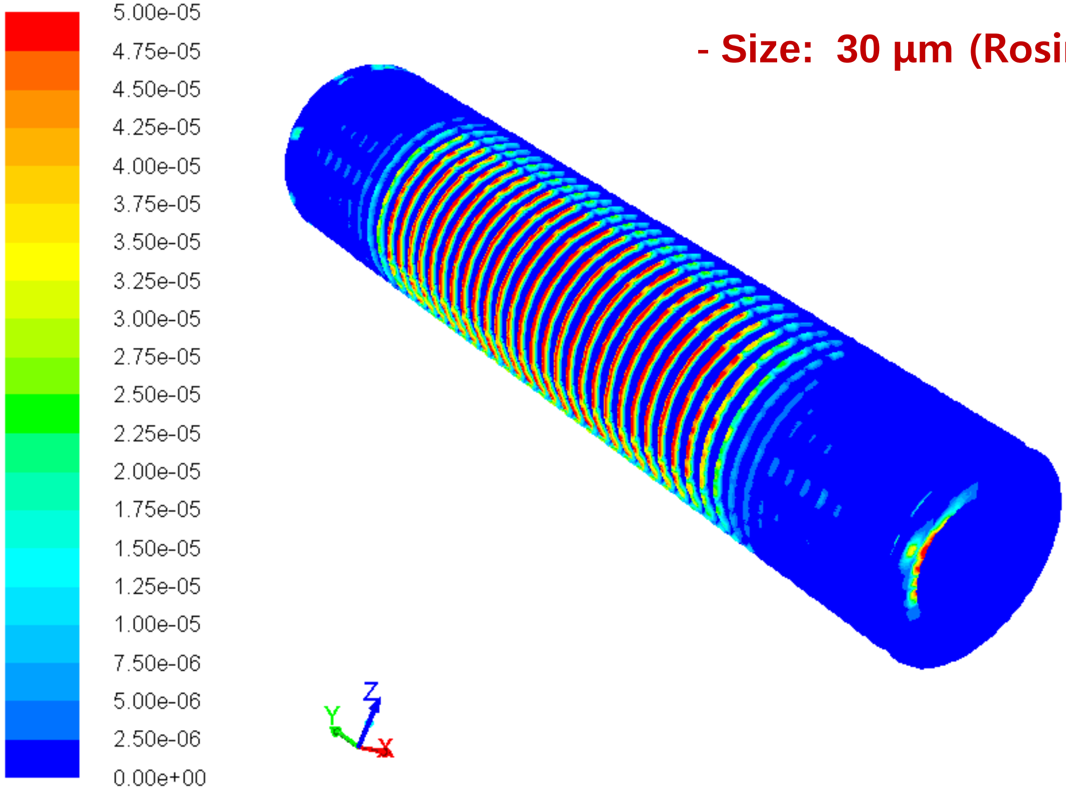
- **Velocity : 2 m/s**
- **Temperature : 800K**
- **Size: 30 μm (Rosin-Rammer)**

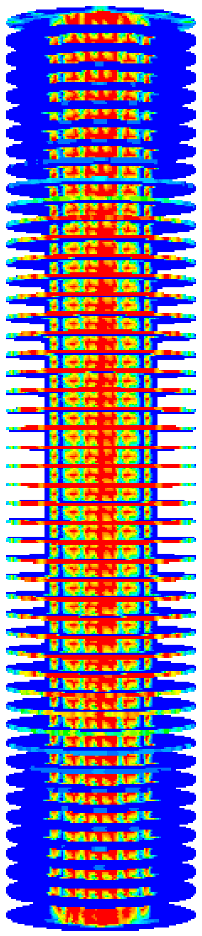




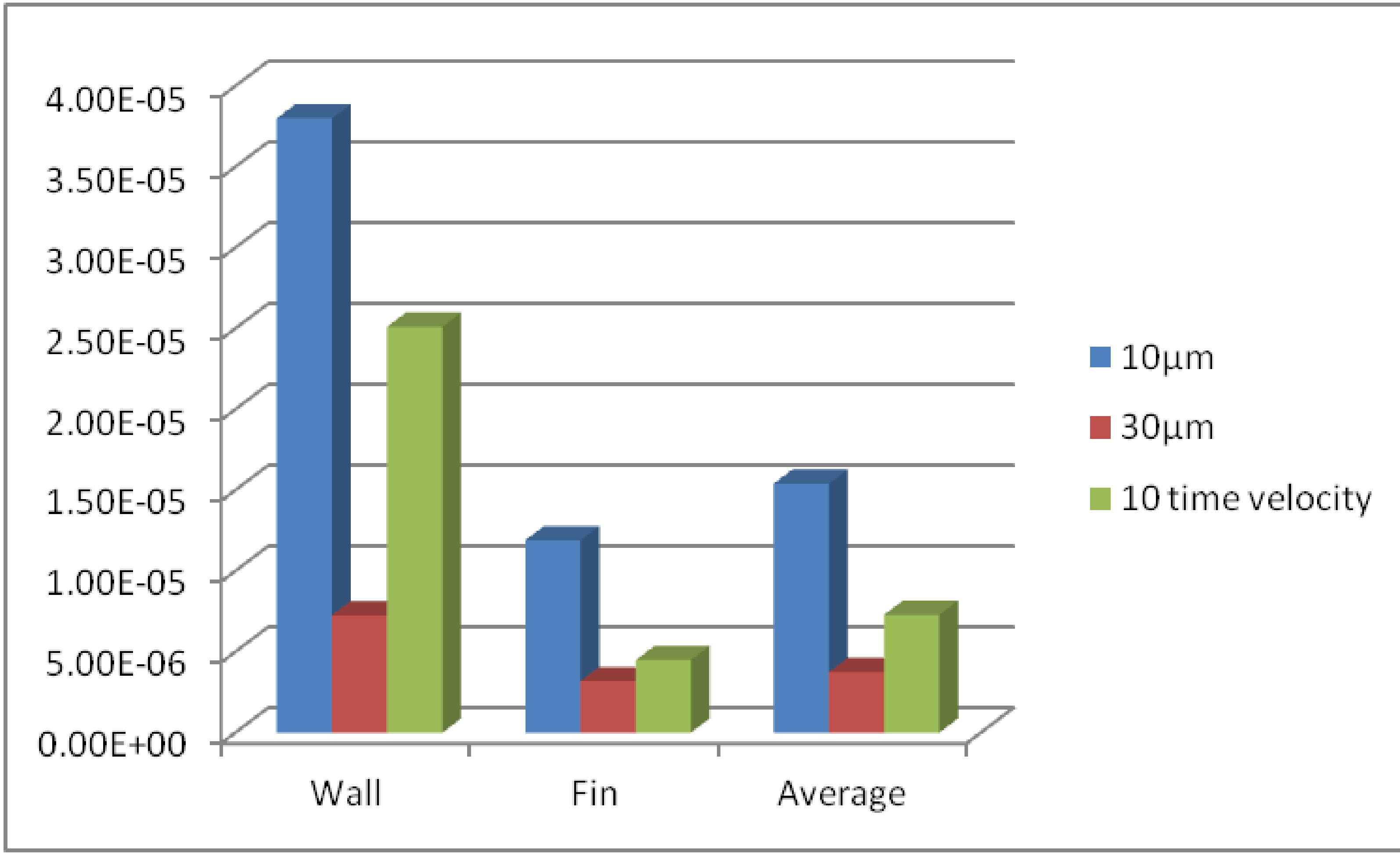
Fin tube _ Fouling characteristics

- Velocity : 20 m/s
- Temperature : 800K
- Size: 30 μm (Rosin-Rammer)





Fin tube _ Comparison Fouling



Research Plans

- 개발된 Fouling 모델을 적용하여 튜브 열수에 따른 Fouling 특성 분석
- 시간에 따른 Fouling height 모델링 개발 및 update
- DTF 및 Fin-tube에 적용하여 특성 분석
- 전열 효율 저감 Curve 제시 ➔ 무차원화된 curve 도출

4.2.4 Operating Regimes and Control Modes

4.2.4.1 Operating Regimes

Power stations can be categorised according to the duty they operate under: peak, intermediate and base loads.
Based on the operating regime of the power plant, Start-ups are classified into cold, warm and hot start-ups:

– Hot start-up: after an outage of maximum 8h. Such an outage typically occurs overnight. For a hard coal fired power station operated in the mid-range such as the reference power plant, about 3,000–4,500 hot start-ups are scheduled for the lifetime of 40 years.
– Warm start-up: after an outage of 8–72 h. The outage is typically over the weekend. For a medium-range power plant the number of warm start-ups is about 1,000 over the station lifetime.
– Cold start-up: after an outage of more than 72 h. This start-up is quite rare; the total number for the medium-range power plant is about 200 (Zehtner 2009).

4.2.4.2 Primary, Secondary and Tertiary Control

The generation of power within a network needs to be controlled and monitored for a secure and high-quality supply of electricity. The goal of the control is to maintain a balance between generation and consumption (demand) of electricity. The key control variable is the frequency of the net, which should be kept stable at 50Hz, or 60 Hz in the USA or parts of Japan. In case of a drop in the frequency, caused by a higher consumption in comparison to the generation, power plants have to increase their load in order to stabilise the frequency.

Primary Control

The objective of primary control is to rapidly re-establish the balance between generation and consumption within the synchronous area by using turbine speed or turbine governors. Primary control is activated if the frequency deviation exceeds ±20 mHz.

All power stations have to be capable of delivering a maximum primary control reserve of 2% of the rated power within 30 s. The maximum reserve has to be activated at a frequency deviation of 200 mHz and has to be maintained over a period of 15 min.

4.2.4.2 Primary, Secondary and Tertiary Control

Secondary Control

Secondary control restores primary control reserves and maintains a balance between generation and consumption of electricity within each control area in a timeframe of seconds to, typically, 15min. Accordingly, load variations of differing magnitudes must be corrected in the control area within this timeframe. Secondary control is based on secondary control reserves which are under automatic control by the operator of the network area. Secondary control is accomplished by increasing the fuel input of a power plant and thus puts requirements on the dynamic behaviour of power plants.

Tertiary Control

Tertiary control reserve is required to restore the secondary control reserves. Tertiary control reserve is usually activated manually after activation of secondary control and frees secondary reserve. Tertiary control is achieved by re-scheduling power generation of operating plants or start-up of additional plants. Tertiary control thus corresponds to the operation planning of all power plants within a network area.

4.2.4.3 Constant-Pressure and Sliding-Pressure Operation

The output of a condensation power station is set by means of the live steam mass flow m_{LS}. The mechanical power, P_m, of the turbine shaft depends on the live steam pressure p_{LS}, the cross-section of the opening A, or the lifting of the turbine intake valves, and the live steam temperature, T_{LS}, according to the following relation:

$$P_m \approx \dot{m}_{LS} \approx A \frac{p_{LS}}{\sqrt{T_{LS}}}$$

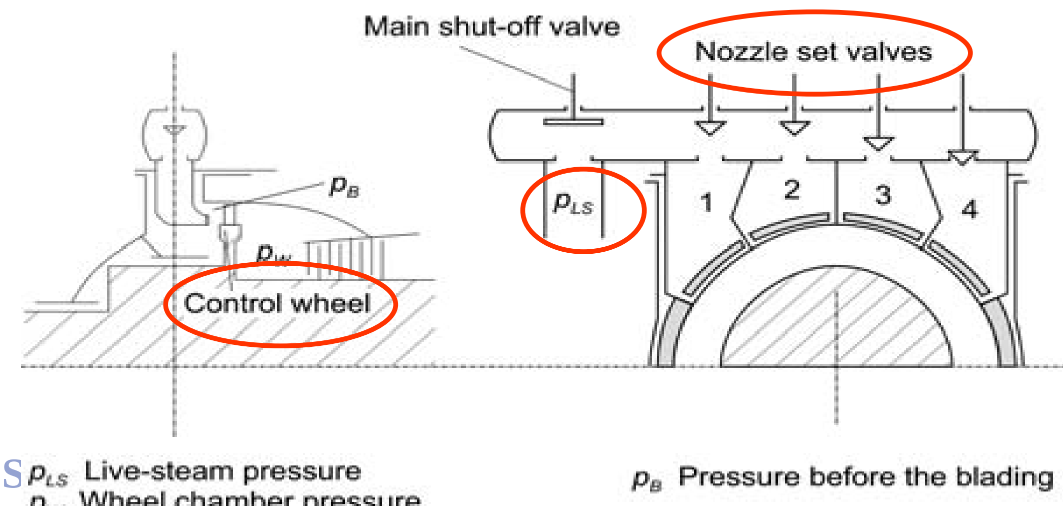
The live steam temperature should remain constant throughout the whole load control range, so that a high efficiency rate is also achieved during part load and to avoid stress on the turbine caused by temperature changes. The turbine output and the live steam mass flow to the turbine are set during steady-state conditions, either when the live steam pressure is at a constant cross-section of the turbine intake valves (sliding or variable pressure) or when the intake cross-section is at a constant steam pressure (constant or fixed pressure).

4.2.4.3 Constant-Pressure and Sliding-Pressure Operation

Constant-Pressure Operation

The control in constant-pressure operation is subdivided into throttle control and governing control. In constant-pressure governing control, the first turbine stage is designed as a control wheel and is preceded by sets of nozzle valves (see Fig. 4.20). As the load increases, the nozzle valves are sequentially opened.

The first stage of the turbine, the control stage, is charged by a high pressure p_B only for part of the circumference, where this pressure is slightly lower than the live steam pressure (i.e. constant pressure). The control stage cuts the pressure back to the wheel chamber pressure p_W and homogenises the steam distribution over the blading of the circumference of the following turbine stage.



4.2.4.3 Constant-Pressure and Sliding-Pressure Operation

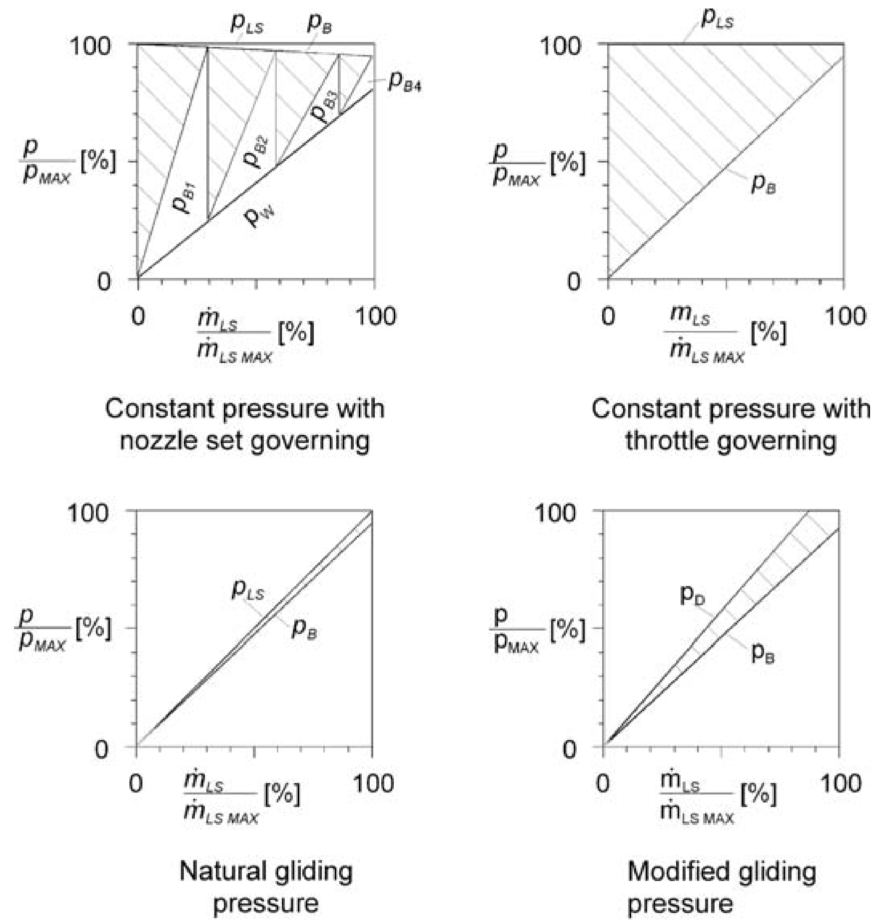
Sliding-Pressure Control

In sliding-pressure operation, the turbine output and the steam flow are adjusted by the pressure at the outlet of the boiler. In natural sliding-pressure operation, the live steam valves of the turbine are completely opened, and the cross-section of the turbine intake is constant throughout the whole load range. An output change using this control type can only be carried out by changing the fuel flow, a consequence of which is a long delay control characteristic of a change in the steam generator. Given that, in sliding-pressure operation, the pressure rises with increasing output, it is necessary that an increased steam flow is produced by the boiler before the output of the turbine increases. Advantages of sliding-pressure control are a load-independent temperature distribution in the turbine, a lower pressure stress on the steam generator and a lower power demand of the boiler feed water pump in part-load operation. Disadvantages are the changes of the boiling temperature in the evaporator, due to the pressure changes. The advantage of the decreasing power requirement for boiler feed pumping is stronger when the live steam pressure becomes higher.

4.2.4.4 Impacts on the Turbine by Sliding-Pressure

The comparison of the different control modes shows that, in constant pressure operation with the nozzle set controlling, the pressure p_B after the turbine inlet valves and before the blading remains almost constant over the load range. In sliding-pressure control, in contrast, and also in constant-pressure operation with throttle control, the pressure shows a linear rise with the output.

Both in sliding-pressure and in constant-pressure operation with throttle control, the stage pressures change to the same degree depending on the output, so that the stage temperatures are constant.



Throttling losses

p [bar]: Live steam pressure

Index LS: Live steam,
B: before blading
W: wheel chamber
1–4: referring to valve or nozzle set

4.2.4.5 Impacts on Circulation or Once-Through Steam Generators by Sliding-Pressure or Constant-Pressure Operation

Circulation steam generators, however, are not operated with sliding pressure because it would involve considerable restrictions on load changes. Circulation systems are almost exclusively operated with constant-pressure

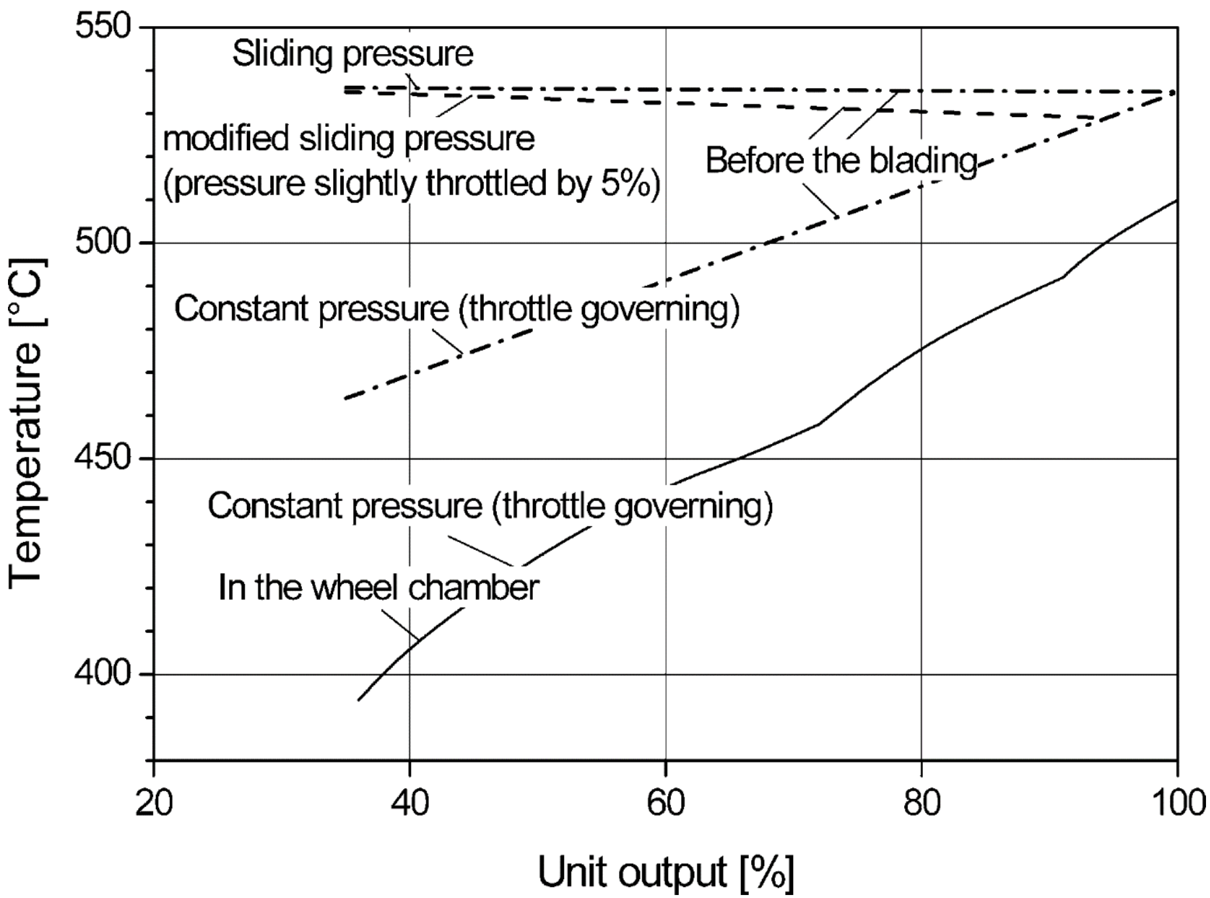


Fig. 4.22 Temperatures in the high-pressure section of the turbine with different control modes (Wittchow 1982)

4.2.4.5 Impacts on Circulation or Once-Through Steam Generators by Sliding-Pressure or Constant-Pressure Operation

Different system characteristics determine different degrees of suitability of drum boilers and once-through boilers for rapid load changes. While the thick-walled drums of circulation steam generators limit the allowable rate of load change, the stress of a once-through boiler is lower at the same pressure rating due to the thinner walls of the separators. However, with higher pressures and temperatures involved, thick-walled construction parts of once-through steam generators, such as separators, do limit the allowable load change rates.

In the case of a short-term increased power output demand of about 5%, the output can be increased by opening the turbine valves, which is possible both using modified sliding pressure and at constant-pressure control. Steam released in the first 20 s comes essentially from the live steam pipe and the superheater. Only afterwards does the evaporator add to the extra steam supply. The greater storage capacity of the drum boiler is an advantage in this case compared to once-through boilers. Delays in steam production if a step load change occurs can be bridged for a longer period until the compensation by the firing rate takes effect (Wittchow 1982).

4.2.4.6 Start-Up

The reliable cooling of all superheater surfaces is a prerequisite for a rapid increase in the firing rate. It is ensured by an adequate turbine bypass system (see Fig. 4.23, 1986). Separated bypass systems for the high-pressure section (HPS), and the intermediate- and the low-pressure sections (IPS, LPS) of the turbine allow independent charging of the turbine parts while maintaining the cooling of the reheater.

The pressure systems of the boiler and the turbine parts are decoupled. This makes it possible to operate them in independent regimes, such as during the start-up and the shutdown processes, and in accidents.

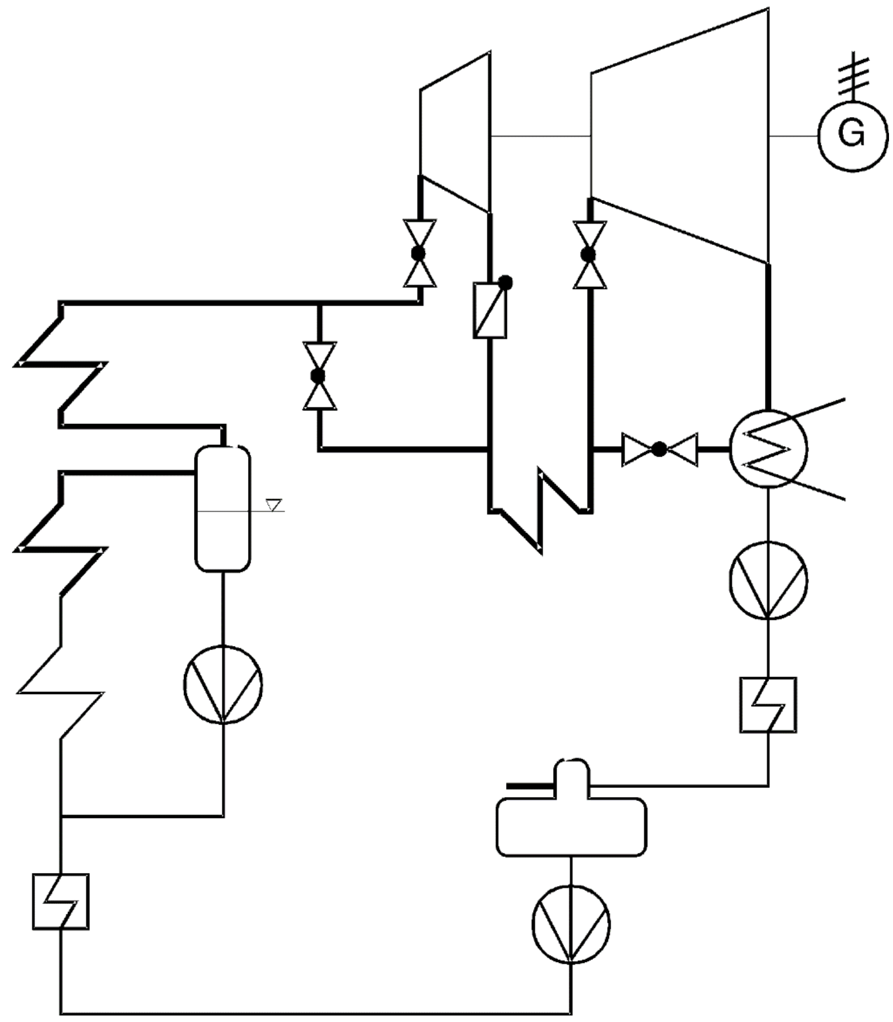


Fig. 4.23 Startup system of a power plant unit (Wittchow 1982)

4.2.4.6 Start-Up

During start-up, the bypass heats those plant components which are unheated to wall temperature, so they can be charged with steam in duty operation and interconnected rapidly.

This system has advantages for other boiler systems as well. The usual long startup times for units with drum boilers in other countries can be put down to the start-up systems used, which often lack turbine bypasses with sufficiently large dimensions.

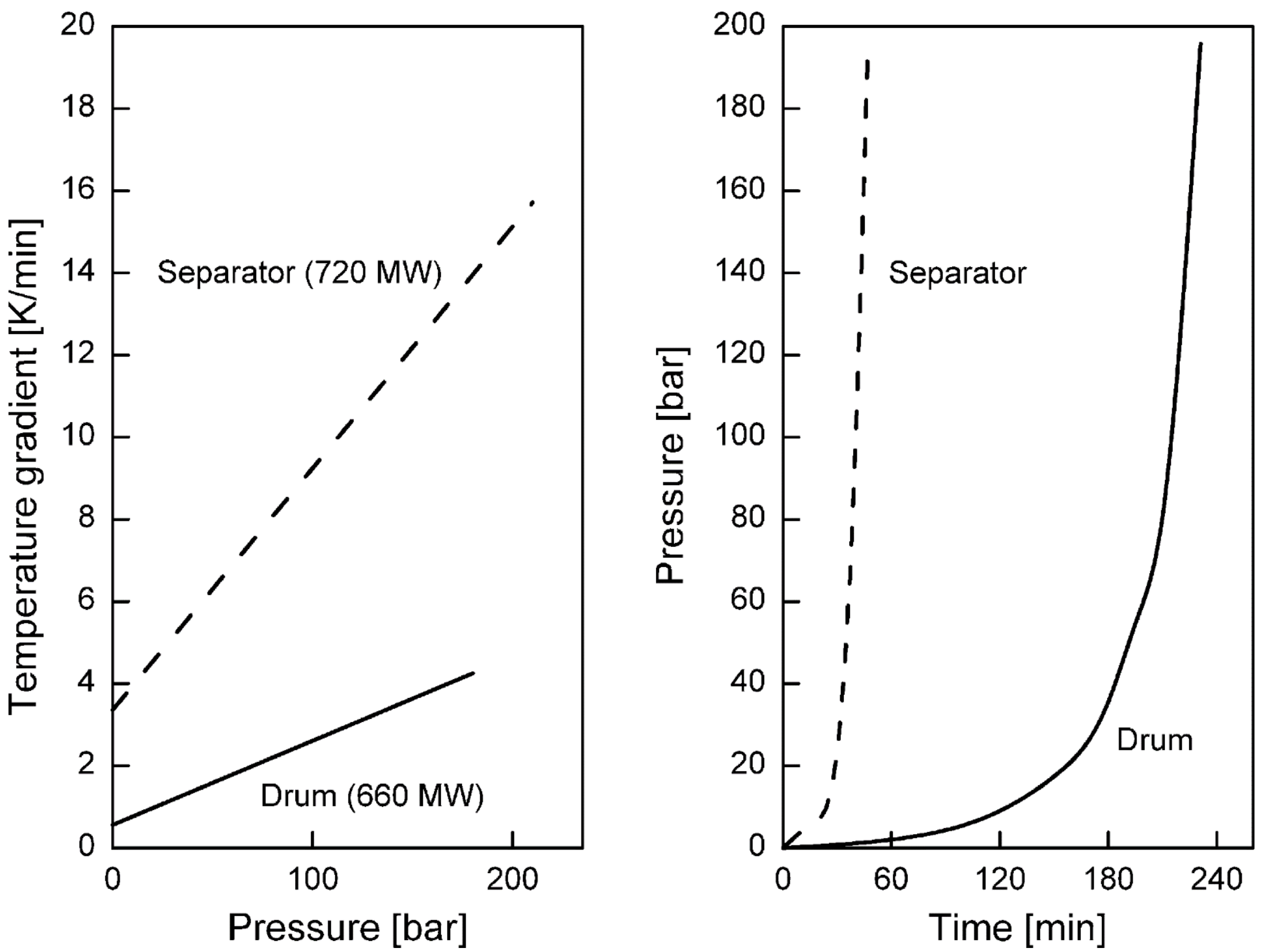
A further criterion for assessment of the start-up process for each plant is the allowable temperature gradient across the thick-walled construction parts.

Figure 4.24 shows the corresponding values for the drum of a 660MW boiler and for the separators of a 720MW once-through boiler.

Correlated with the pressure dependent boiling temperature, it is possible to calculate the warm-up times for these parts. Owing to the thermal flexibility of its construction, the once-through boiler, compared to drum boilers, has advantages when starting up from a cold state and after weekend shutdowns.

In contrast, there are no remarkable differences for warm or hot start-ups, provided that the pressure in the drum boiler has not dropped too low before start-up (Wittchow 1982).

4.2.4.6 Start-Up



4.3 Design of a Condensation Power Plant

4.3.1 Requirements and Boundary Conditions

The design of a condensation power plant and, in particular, the steam generator, is subject to a range of requirements with respect to the

- *Plant capacity*
- *Fuel*
- *Operating regime*
- *Boundary conditions and official directives*
- *Efficiency*
- *Availability*
- *Investment and operating costs*
- *Serviceability*
- *Service life, maintenance and repair (STEAG 1988; Baehr 1985)*

Because the requirements are partly contradictory, the design in each case is a compromise between the different requirements. The task of a plant design is the optimisation for the given case.

4.3.1.1 Fuel

The planned fuel is a key factor for the design of the plant. Compared to a gas-fired power plant, a coal-fired power plant is much more complex and requires additional, sophisticated components such as installations for the unloading, transport, storage and mixing of solid fuels, as well as machinery for fuel preparation, equipment for the cleaning of heating surfaces, devices for ash transportation and disposal and additional flue gas cleaning units. The design of the furnace, the steam generator and other components is dependent on the fuel. For this reason, designing a power plant includes the specification of a design fuel and the range of fuels fired.

4.3.1.2 Operating Regime

The plant design has to take into account the planned operating regime – base load, mid-range load or peak load (see Sect. 4.2.4). The number of expected start-ups per year, classified into cold, warm or hot starts, and the necessary load control ranges and daily load changes between the minimum and the rated power have to be determined prior to the design.

Both the fuel costs and the utilisation factor of the plant determine the economic optimum of the investment costs. For a base load power plant, the higher investment costs of the desired higher efficiency rates are more economic than for a mid-range load plant. If a plant is almost only full-load operated, thick-walled components and the resulting limits to the load change rate can be tolerated. It is sufficient to design such a plant for operating regimes with small load changes and a small number of start-ups.

The design of mid-range load plants, however, involves more compromise and therefore requires a more considered design with regard to the behaviour during load changes, start-ups and shutdowns, the minimum power and the efficiency over the load range. Modern hard coal fired plants can usually be operated in a load range from about 35 to 100% of the rated power. Loads below 35% are in general only possible with oil or gas as backup firing.

4.3.1.3 General Conditions and Official Directives

The conditions specific to the location have to be exactly determined prior to designing a power plant. An important part of these conditions, which have to be incorporated into the power plant design, is the legislative directives. The legislator stipulates allowable emission levels which have to be complied with by installing flue gas cleaning and noise insulation. Water withdrawal for process cooling and the discharge of wastewater have to be planned and carried out in compliance with the ordinances referring to water rights. In Germany, to give an example, the thermal stress that it would impart upon rivers may no longer allow the operation of once-through cooling in the summer. This restriction can be avoided by back-cooling processes, which are mostly used for inland locations. The height of natural-draught towers can also be limited by directives. Locations near the seaside allow once-through cooling with seawater. Aspects of the design that impact upon waterways, railways and highways have, as a rule, to comply with directives of local authorities as well.

4.3.1.4 Efficiency

High overall efficiencies of conventional steam power plants can be achieved by the following features:

- *High temperatures and pressures of the generated live steam before it enters the* turbine
- *High temperatures of the single or multiple reheat cycle in intermediate pressure* stages
- *Regenerative air heating and fuel drying*
- *Regenerative feed water heating*
- *Low exhaust steam pressures of the turbine before condensation*
- *Low losses of all plant components*
- *A low electric auxiliary power demand*

Plant components with higher efficiency rates require parts with thicker walls to withstand higher temperatures and pressures. When fast temperature changes occur, stronger thermal stresses evolve in these parts, leading to levels that can exceed the allowable design strength and consequently to a shorter service life of the components. Therefore, advanced power plants necessarily involve longer start-up times and thus greater start-up losses and lower load change rates.

4.3.1.5 Availability

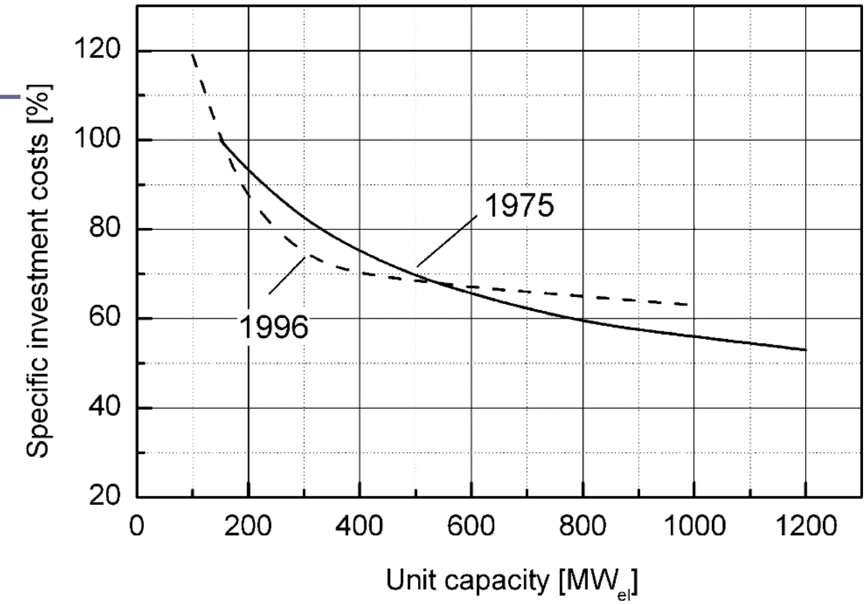
A high availability of technology implies a high-quality standard of plant components, standby components and care in operating, control and maintenance. For financial reasons and because of an achievable high level of availability, large single components such as boilers, forced-draught fans (FD fans), induced-draught fans(ID fans), turbines, cooling towers, generators and transformers are designed as mono-devices (i.e. one unit operating at full load instead of two or more at partial load).

As regards plant equipment designs, for example of FGD units and catalytic No*x* control units, there is a tendency towards single-line design. In the case of other plant components, standby options have to be discussed on the basis of their individual availability and the extra costs. In the case of an interconnected network system, the considerations about unit availability can include the existing reserve capacity of the network.

4.3.1.6 Costs

Costs are classified as variable costs, which depend on the operating period of the plant, and fixed costs. Variable costs are basically the fuel costs and the operating and maintenance costs. Fixed costs are the capital and personnel costs. The costs for the personnel depend on the serviceability of the plant.

Figure 4.25 shows how the unit investment costs of the entire plant and of its main components decrease as the capacity increases. The cost decrease lessens with high unit capacities, so a rise of the capacity will yield less financial advantages (STEAG 1988; Kotschenreuther and Klebes 1996).



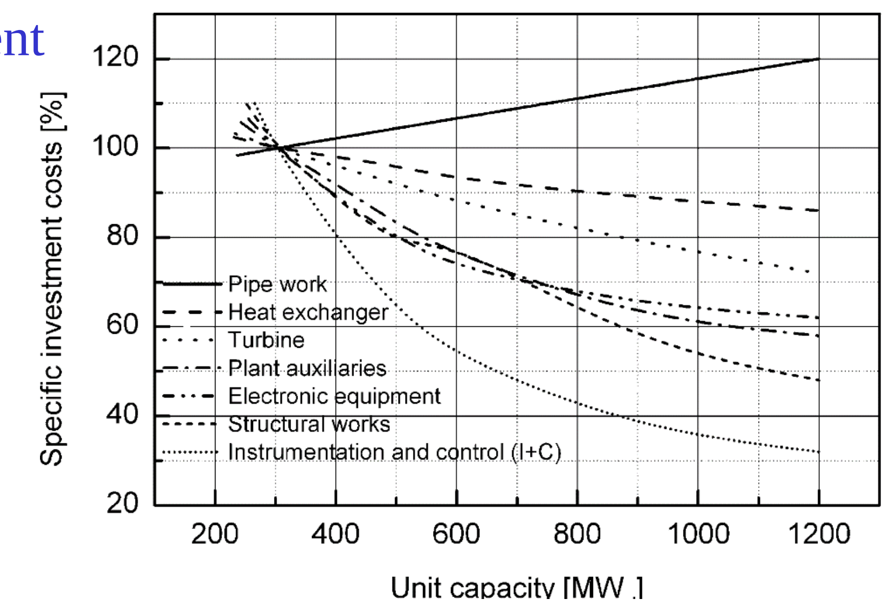


Fig. 4.25 Decrease of specific costs for the plant entity and for the plant components with increasing unit capacity (STEAG 1988; Kotschenreuther and Klebes 1996)

4.3.1.6 Costs

Figure 4.26 shows the breakdown of investment costs for a large hard coal fired power plant. In Germany, the specific investment costs of large power station units amounted to about 1,000 Euro/kWel around 2005. For power plants planned and built in Asia, the costs are about 30–40% lower due to lower manufacturing costs and less demanding directives/regulations. Competition in the past induced a decrease of the specific investment costs; however, recently the huge worldwide metal demand has caused an increase in investment costs by 50% (2008).

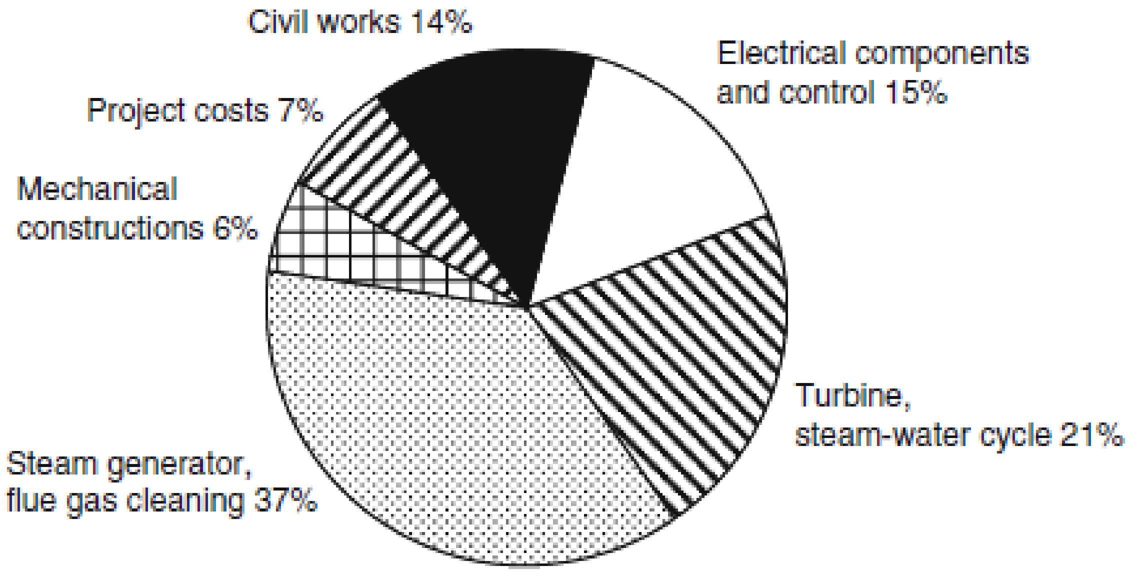


Fig. 4.26 Breakdown of investment costs of a large ulverised coal fired power plant

4.3.1.6 Costs

The economic optimum for a specific power plant configuration is determined by balancing the cost reductions achieved through higher efficiencies against the additional costs of the efficiency increase.
The correlation between the economically feasible investment *ΔI* and an advantage of consumption $\Delta HR/HR_0$ results from the following formula:

$$\frac{\Delta I}{\Delta HR/HR_0} = \frac{HR_0 \cdot P_{el} \cdot U \cdot C_F \cdot 10^{-5}}{CoC}$$

$\frac{\Delta I}{\Delta HR/HR_0}$ = economically feasible additional investments referring to the heat rate improvement [Euro/%]
HR_0 = basic heat rate (net) [kJ/kWh]
U = utilisation factor (full-load operating hours per year) [flh/a]
P_{el} = electric net power output at full load [MW]
C_F = fuel costs [€/GJ]
CoC = cost of capital/debt service factor as a function of financing and operation period (STEAG 1988) [1/a]

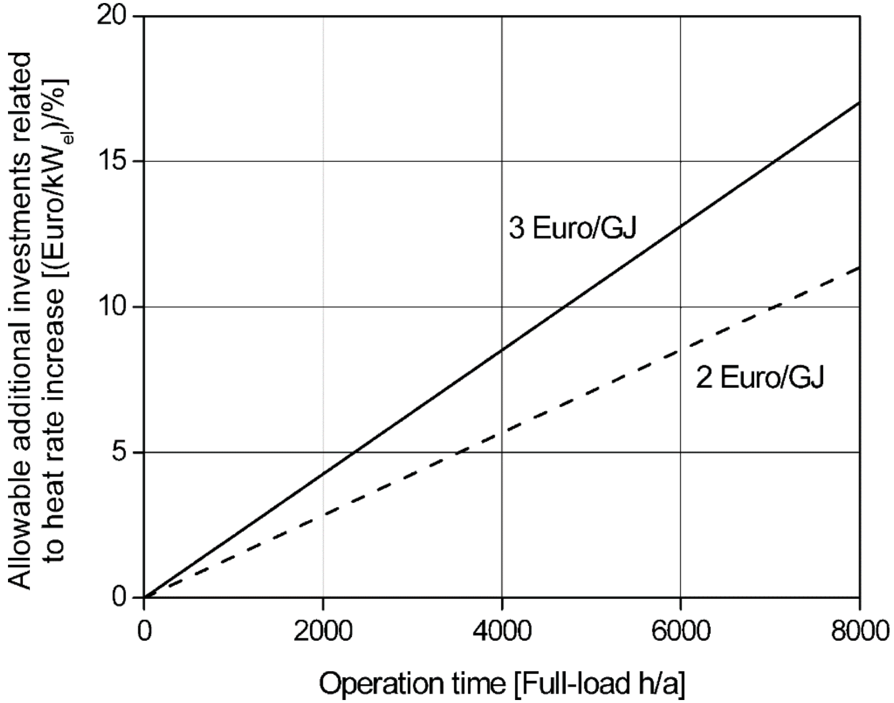


Fig. 4.27 Economically feasible additional investments per percentage of heat rate increase as a function of fuel price and operation time

4.3.1.7 Serviceability

The serviceability of the individual plant parts and the power plant installation as a whole are based on the applicability of the instrumentation and control (I + C) equipment. A more sophisticated power plant control system has to be balanced against the reduction of personnel costs.

4.3.1.8 Design Life

An important parameter in the power plant design is the planned lifetime. Conventional and advanced designs are planned for a lifetime of 200,000 h of operation. Together with the planned operating regime, the design life is mainly determined by the design of the main components, i.e. the steam turbine and, in particular, the high pressure and superheated steam pipe work and the respective steam generator components and vessels which are subject to regular inspection according to law. Base load power stations are mainly subject to creep rupture stresses, while mid-range load power stations are usually subject to alternating stress. Both types of stress result in consumption of the design life or fatigue of construction parts. The inspection of the components and the determination by calculation of the expended lifetime are laid down in technical rules such as the European Standard (or Norm) EN 12952 or formerly the “German Technical Rules on Steam Generators” (*Technische Regeln Dampferzeuger (TRD)). Apart from that, the design will take into account* regular scheduled outages for replacing worn parts or for improvement or retrofitting purposes, without factoring in the availability of such improved technology.

4.3.2 Thermodynamic Design of the Power Plant Cycle

The thermodynamic design of the cycle comprises the determination of the

- Process flow configuration
- Steam parameters
- Preheater configuration
- Heat dissipation (Baehr 1985)

The thermodynamic design determines the conditions of the closed steam – water cycle, yielding a power plant cycle diagram such as plotted in Fig. 4.28 for the reference power station (Spliethoff and Abr¨oll 1985).

The choice of the preheater configuration defines the number of stages and the design of the individual heater stages, thus determining the final feed water temperature. In the steam generator, the feed water is heated to boiling temperature, evaporated and superheated and reheated after a partial expansion step.

4.3.2 Thermodynamic Design of the Power Plant Cycle

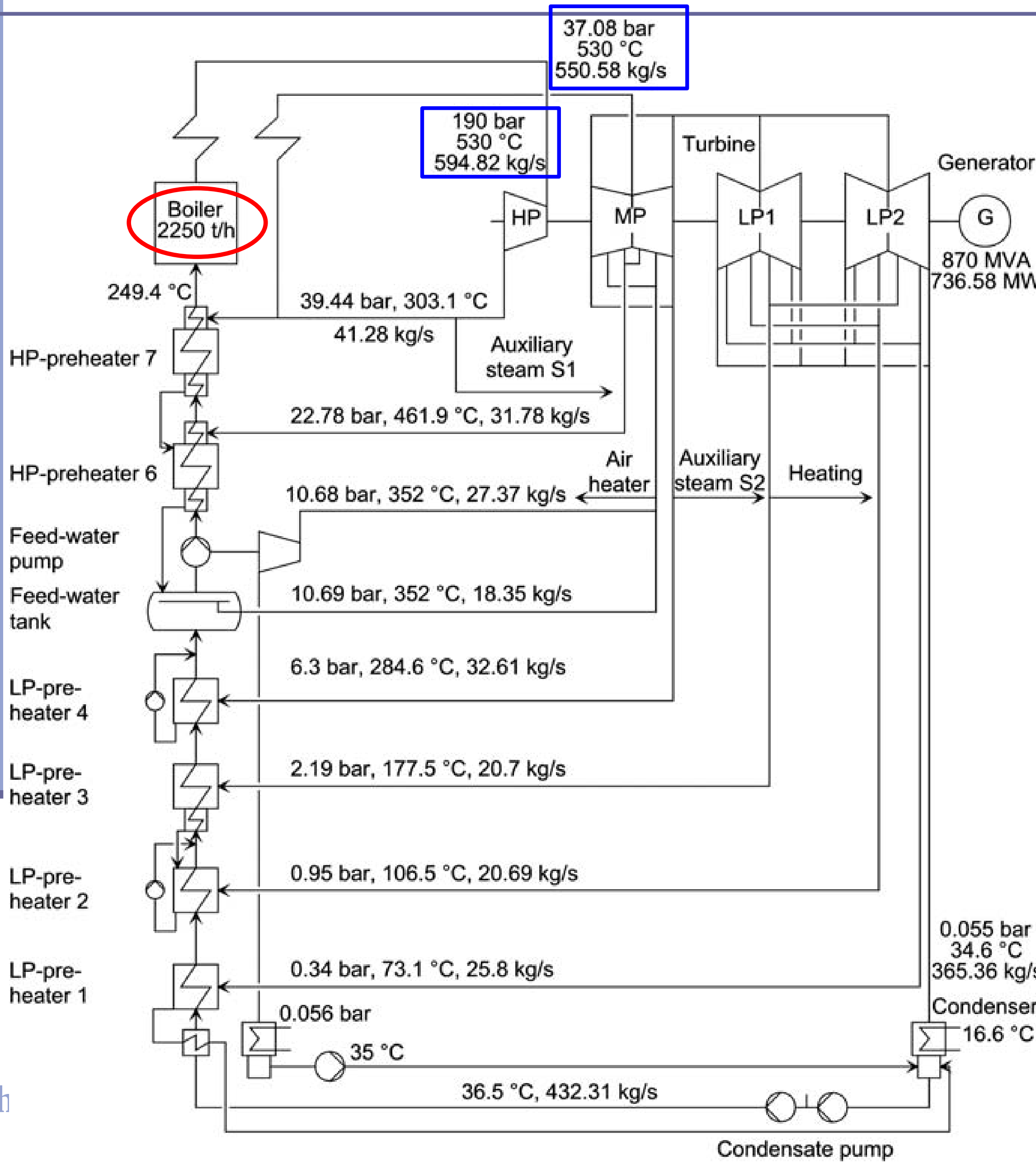


Fig. 4.28 Cycle of a conventional steam power plant with hard coal firing

In designing the process flow, only the live steam conditions and the conditions of the reheated steam are defined at first, without specifying the heating surfaces for the heat transfer from the flue gas to the water – steam system. The process flow design comprises the conversion of the thermal energy of the steam into the mechanical energy in the turbine. This includes the definition of the exhaust steam condition in the condenser, i.e. the exhaust steam temperature, as well as the type of drive of the feed water pump.

4.3.2 Thermodynamic Design of the Power Plant Cycle

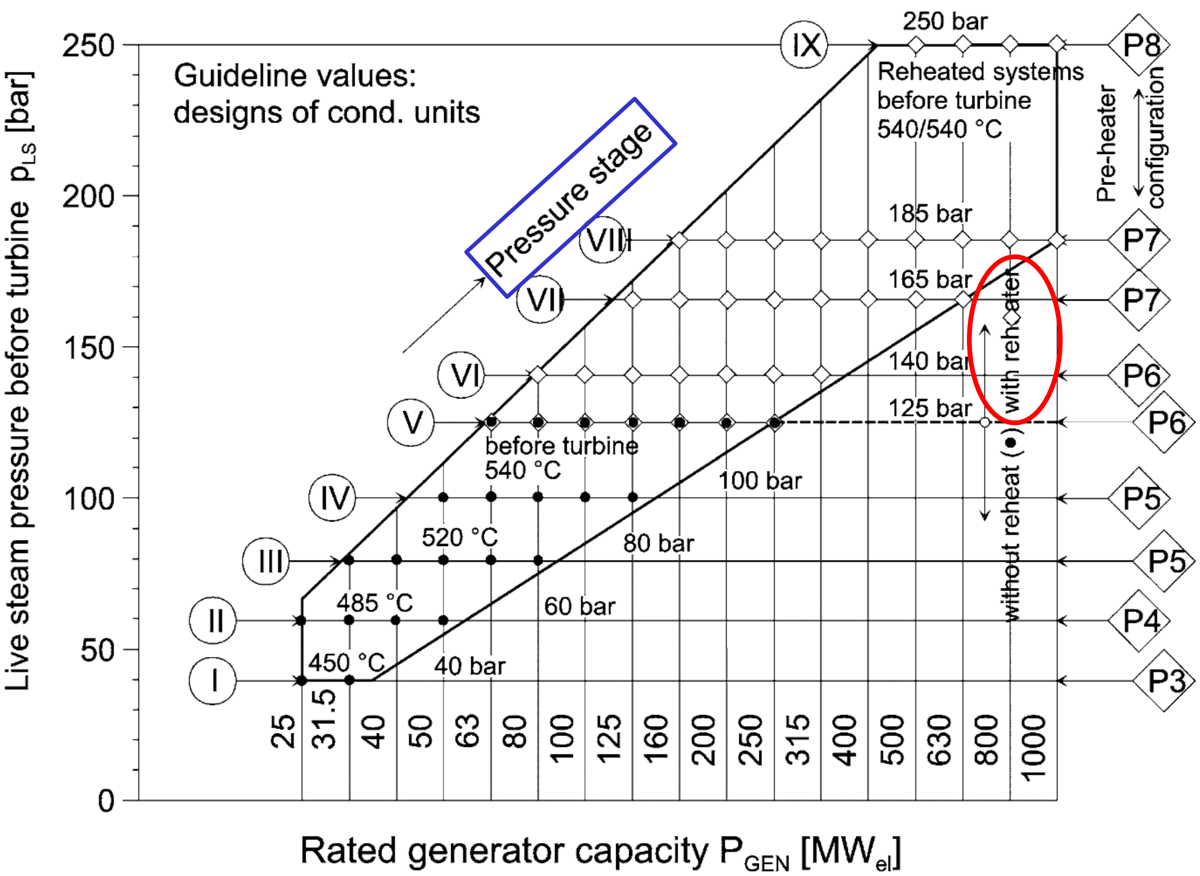


Fig. 4.29 Guideline values for the design of steam power plants (Baehr 1985)

Figure 4.29 presents several recommended pressure stages for a given generator capacity. The choice of higher pressure stages is reasonable for high fuel costs and full-load plants, the low-pressure stages for mid-range or peak load and low fuel costs. "Thermal cornerstones" – the live steam conditions, the reheater steam conditions, the regenerative feed water preheating by turbine extraction and the cold end of the turbine – determine the thermal efficiency and the heat rate of the condensation turbine.

4.3.2 Thermodynamic Design of the Power Plant Cycle

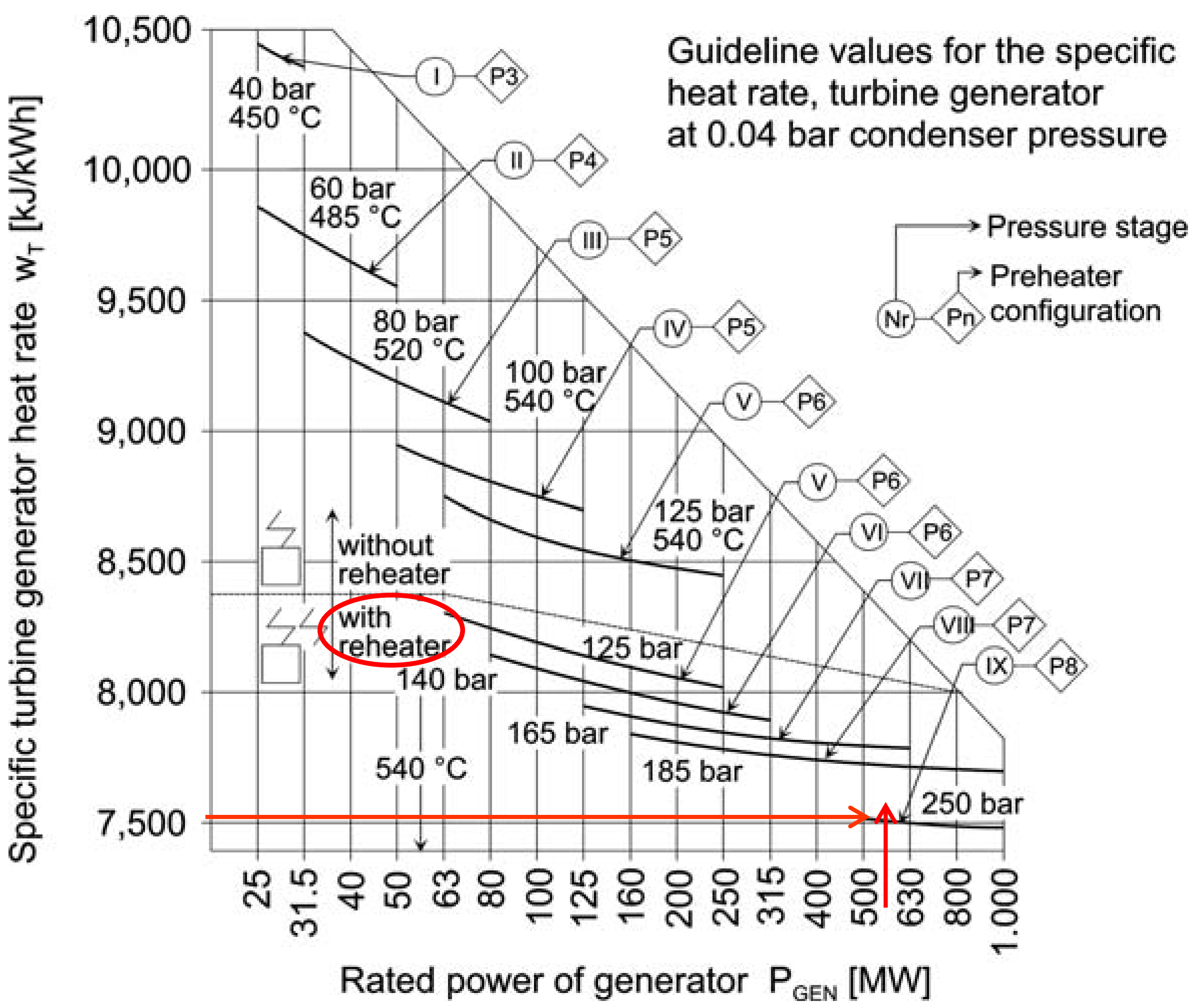


Fig. 4.30 Specific heat rate of the turbine generator (Baehr 1985)

Figure 4.30 shows the turbine heat rate for the turbine generator configurations shown in Fig. 4.29.
The net efficiency of a power plant is calculated by the various individual efficiencies:

$$\eta = \eta_B \cdot \eta_T \cdot \eta_{aux} \cdot \eta_P$$

$$\eta_T = \eta_{th,0} \cdot \eta_{i,T} \cdot \eta_m \cdot \eta_{Gen}$$

η_B is the boiler or steam generator efficiency / η_T is the efficiency of the steam turbine unit / η_{aux} is the auxiliary power efficiency takes the electrical and mechanical power equirements into account; the efficiency η_P comprises the heat losses of the live steam and the reheater pipework that connects the steam generator and the turbine.

4.3.3 Heat Balance of the Boiler and Boiler Efficiency

In the boiler or steam generator with a single reheating heat exchange stage, the heat balance can be calculated according to Fig. 4.31

$$\dot{Q}_F + \dot{Q}_A = \dot{m}_{LS}(h_{LS} - h_{FW}) + \dot{m}_{RS}(h_{RS2} - h_{RS1}) + \dot{Q}_{LOSS}$$

The boiler efficiency can be calculated directly when the steam conditions and flows and the heat addition into the furnace are known:

$$\eta_B = \frac{\dot{m}_{LS}(h_{LS} - h_{FW}) + \dot{m}_{RS}(h_{LS2} - h_{LS1})}{\dot{m}_F \cdot LHV + \dot{m}_A \bar{c}_{PA}(t_A - t_o)}$$

With the loss through unburned matter (KU), the loss through sensible heat of the slag (KS), the flue gas loss (KFG) and the loss through radiation and convection of the external surfaces of the boiler (KRC), the boiler efficiency can be calculated:

$$\eta_B = 1 - K_U - K_S - K_{FG} - K_{RC}$$

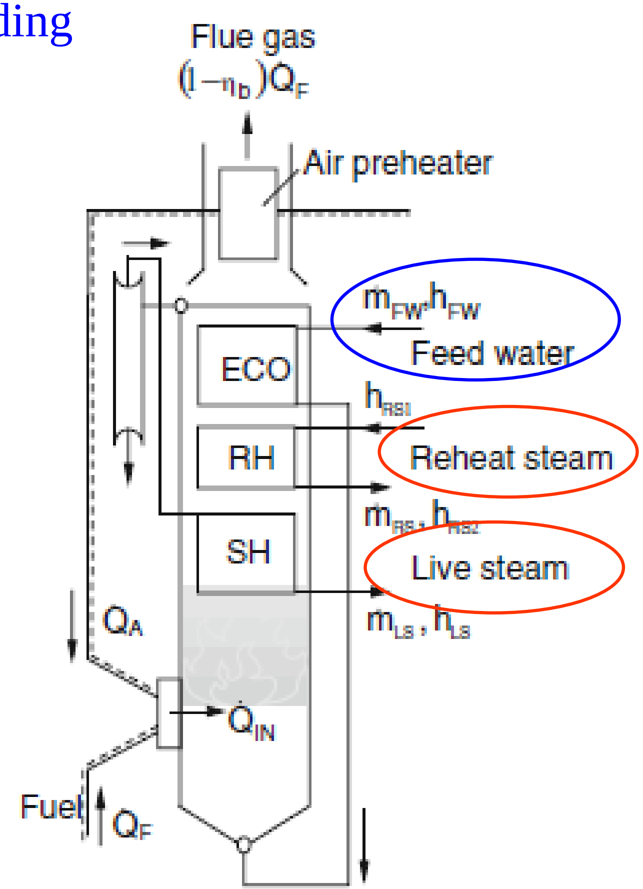


Fig. 4.31 Heat balance of a steam generator

4.3.4 Design of the Furnace

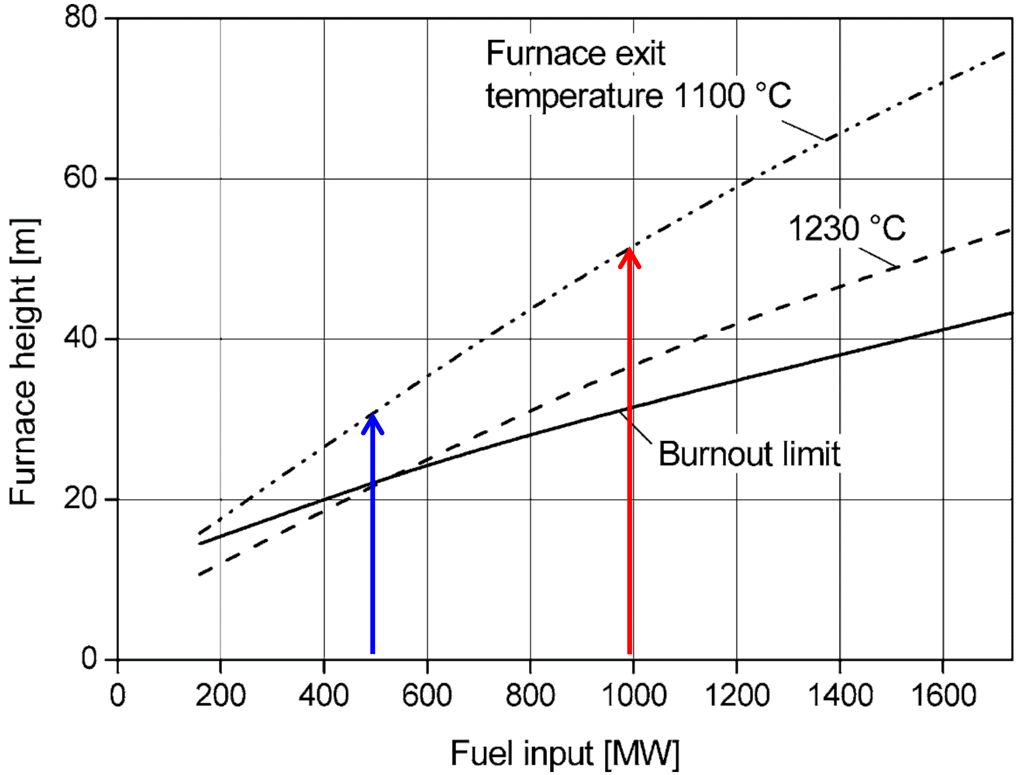
The essential parameters for the furnace design are the fuel mass flow, the primary fuel and the planned range of mixtures with secondary fuels. For solid fuels, other important design parameters besides the calorific value and the moisture and ash contents are the grindability of the coal, the fraction of volatile matter, the elemental and the ash composition and the ash melting behaviour. The choice of the firing system configuration (frontal firing, opposed firing, tangential firing, down-firing, bottom firing) is followed by the determination of the number and arrangement of the burners, including the mills. An important consideration in this process is the requirements for part-load performance. In setting the dimensions of the furnace, the following performance aims have to be taken into consideration:

- Stable ignition
- Complete burnout
- Prevention of slagging and corrosion inside the furnace
- Prevention of fouling and corrosion on the convective heating surfaces

The cross-section and height of the furnace have to be chosen according to the fuel type such that slagging and fouling inside the furnace, as well as on the subsequent heating surfaces, are within acceptable limits.

4.3.4 Design of the Furnace

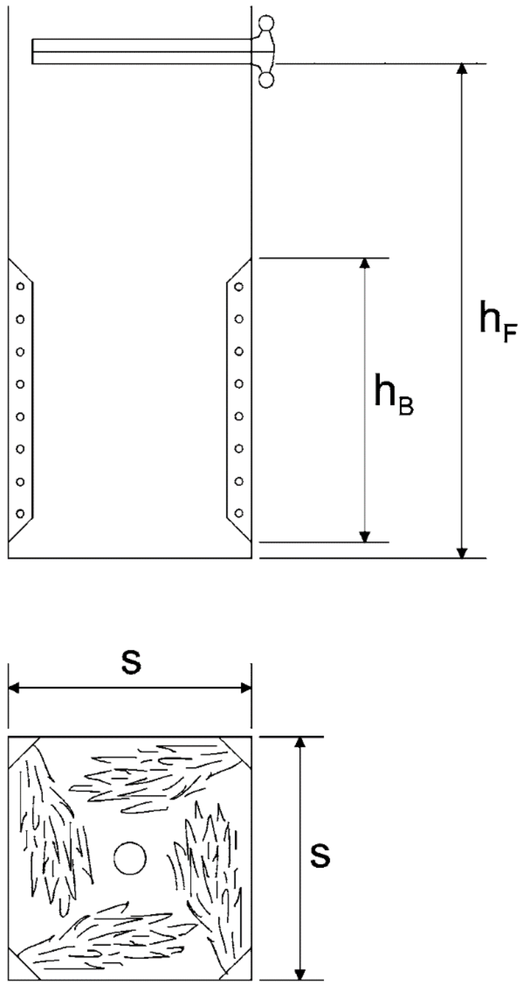
For coal types with a slagging tendency, a much larger cross-section will be chosen. The ash deformation temperature of the fuel defines the necessary furnace outlet temperature at the furnace end before the convective heating surfaces, in order to avoid sticky deposits on the convective heating surfaces.



Hard coal combustion systems have a furnace outlet temperature of about 1,250◦C and brown coal combustion systems about 1,050◦C.

Fig. 4.32 Burnout limits and furnace exit temperatures in hard coal fired tangential combustion systems (Strauß 2006)

4.3.4 Design of the Furnace



Volumetric heat release $q_V = \frac{\dot{Q}_{in}}{V_F} = \frac{\dot{Q}_{in}}{s^2 h_F}$ (MW/m³)

Cross-sectional area heat release $q_C = \frac{\dot{Q}_{in}}{A_C} = \frac{\dot{Q}_{in}}{s^2}$ (MW/m²)

Surface heat release $q_W = \frac{\dot{Q}_{in}}{A_F} = \frac{\dot{Q}_{in}}{4\,s\,h_F}$ (MW/m²)

Burner-belt heat release $q_B = \frac{\dot{Q}_{in}}{A_B} = \frac{\dot{Q}_{in}}{4\,s\,h_B}$ (MW/m²)

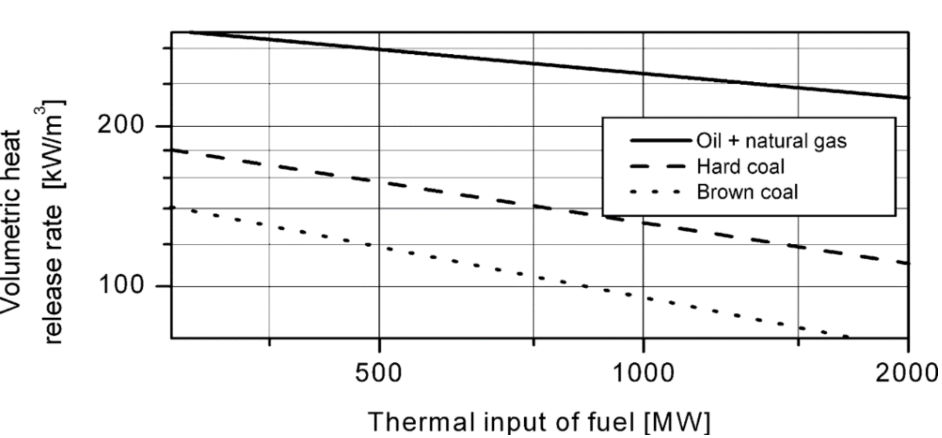
$\dot{Q}_{in}$ = Heat input
V_F = Furnace volume
A_C = Furnace cross-section
A_F = Furnace surface
A_B = Burner belt surface

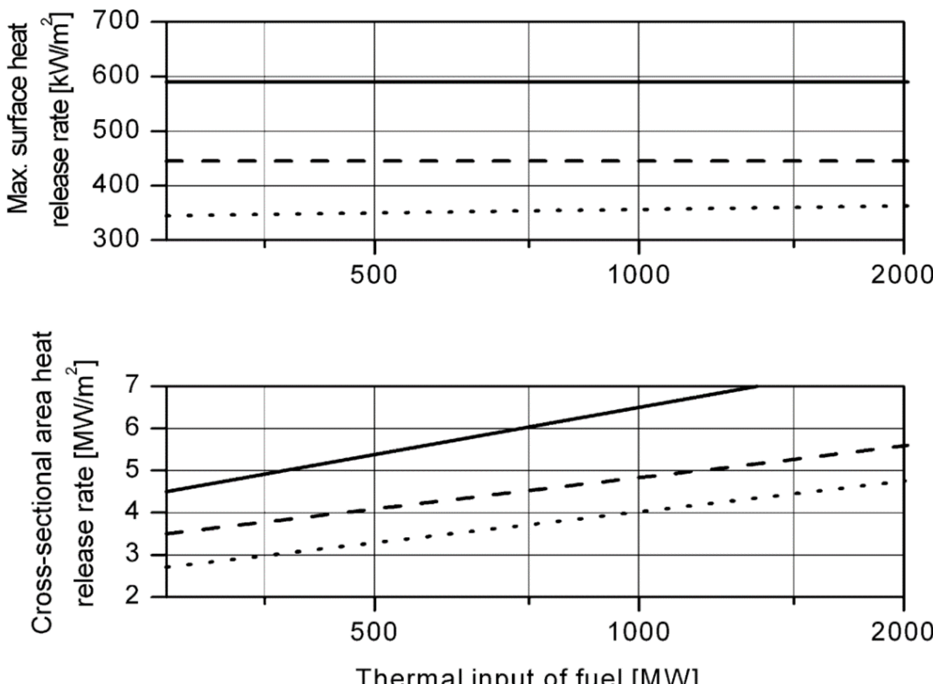
4.3.4 Design of the Furnace

4.3.4.1 Volumetric Heat Release Rate

The furnace volumetric heat release is a measure of the residence time in the furnace and thus makes it possible to evaluate the burnout. It is defined by the given crosssection and the furnace outlet temperature.

4.3.4.2 Cross-Sectional Area Heat Release Rate





4.3.4.3 Surface Heat Release Rate

The mean surface heat release rate is a measure of the average temperature decrease. It is determined by the furnace exit temperature at a given cross-section. The local allowable surface heat release maximum depends on the fuel.

4.3.4.4 Burner-Belt Heat Release Rate

The burner-belt heat release rate is an indication of the flame temperature in the burner area, as it represents the ratio of the thermal input to the cooling surface in the burner area. Its order of magnitude depends on the fouling rate of the fuel, among other parameters. For normal hard coal not prone to slagging, the value is about 1MW/*m2*.

Figure 4.34 gives guideline values for the volumetric, cross-sectional and maximum surface heat release rates. The comparative values mentioned above give reference values for the design of a furnace, but are not a substitute for the calculation of the heat transfer processes (Baehr 1985).

4.3.4.5 Calculation of the Flue Gas Cooling

Whereas the cross-section of the furnace is defined by the chosen firing system and the allowable cross-sectional heat release, the furnace height or (wall) heating surface area of large steam generators is determined by the necessary flue gas cooling to the furnace exit temperature. The height defines the threshold between radiative and convective heating surfaces.

For assessing the heat exchange between the flue gases in the furnace and the enclosing walls, one starts from a mean flue gas temperature in the furnace T_{FG} and a mean wall temperature T_W (Doležal 1990; Strauß 2006).

The flue gases in the furnace transfer the heat flux $\dot{Q}_F$ to the furnace walls (evaporator) by radiation:

$$\boxed{\dot{Q}_F = \underline{\varepsilon_{FW}} \cdot C_0 \cdot \underline{A_{FL}} \underline{\left(T_{FG}^4 - T_W^4\right)}}$$

ε_{FW} = emissivity between flame and wall
C_0 = coefficient of radiation of the black body (5.67×10^{-8} W/m^2 K^4)
T_W = the mean wall temperature
T_{FG} = the mean flue gas temperature in the furnace
A_{FL} = the flame surface
A_W = the wall surface

If a flame fills the furnace completely, the surface of the flame A_{FL} equals the surface of the furnace A_W. In other cases, ratios are given between the two surfaces (Ledinegg 1966).

과제 : Flue Gas 단계별로 계산해서 제출하기

4.3.4.5 Calculation of the Flue Gas Cooling

The emissivity between the flame and the wall depends on the emissivities of the surface wall and the flame and can be calculated:

$$\varepsilon_{FW} = \left(\frac{1}{\varepsilon_F} + \frac{1}{\varepsilon_W} - 1\right)^{-1}$$

The surface emissivity of an oxidised steel surface is between 0.6 and 0.8. Furnace ash deposits affect the heat transfer. The emissivity of deposits depends on the chemical composition, structure and porosity of the layer. The apparent emissivity, which describes the combined deposit and substrate emissivity, lies between 0.5 and 0.8 for most deposits (Stultz and Kitto 1992).

The flame emissivity can be calculated by

$$\varepsilon_F = \varepsilon_\infty(1 - \exp(-ks))$$

where ε_∞ is the emissivity for a very thick flame. The parameter s is the thickness of the flame or beam length and k depends on the character of the flame; the parameter k varies between 0.75 for luminous flames and 0.5 for blue flames.

Typical values for the emissivity ε_∞ are as follows:

Hard coal, brown coal	0.55–0.8
Oil	0.6–0.85
Natural gas	0.4–0.6

The resulting emissivity is, for a hard coal fired furnace, in the range of 0.4–0.7, mainly depending on fouling and slagging.

4.3.4.5 Calculation of the Flue Gas Cooling

The mean furnace temperature of the dry bottom furnaces is calculated as the geometric mean of the adiabatic combustion temperature T_{ad} and the furnace outlet temperature T_{FE}:

$$\boxed{T_{FG} = \sqrt{T_{ad} \cdot T_{FE}}}$$

The heat flux in the furnace Q_F is transferred from the flue gas mass flow m_{FG}, having a specific heat c_{pF6}, which cools from the adiabatic flame temperature T_{ad} down to furnace exit temperature T_{FE}:

$$\dot{Q}_F = \dot{m}_{FG} \cdot \bar{c}_{p_{F6}} (T_{ad} - T_{FE})$$

$$\boxed{\varepsilon_{FW} \cdot C_0 \cdot A_W \left(T_{ad}^2 \cdot T_{FE}^2 - T_W^4\right) = \dot{m}_{FG} \cdot \bar{c}_{p_{F6}} (T_{ad} - T_{FE})}$$

$$\left(\frac{T_{FE}}{T_{ad}}\right)^2 + Ko \cdot \frac{T_{FE}}{T_{ad}} = \left(\frac{T_W^4}{T_{ad}}\right)^2 + Ko$$

$$Ko = \frac{\dot{m}_{FG} \cdot \bar{c}_{p_{F6}}}{\varepsilon_{FW} \cdot C_0 \cdot A_W \cdot T_{ad}^3}$$

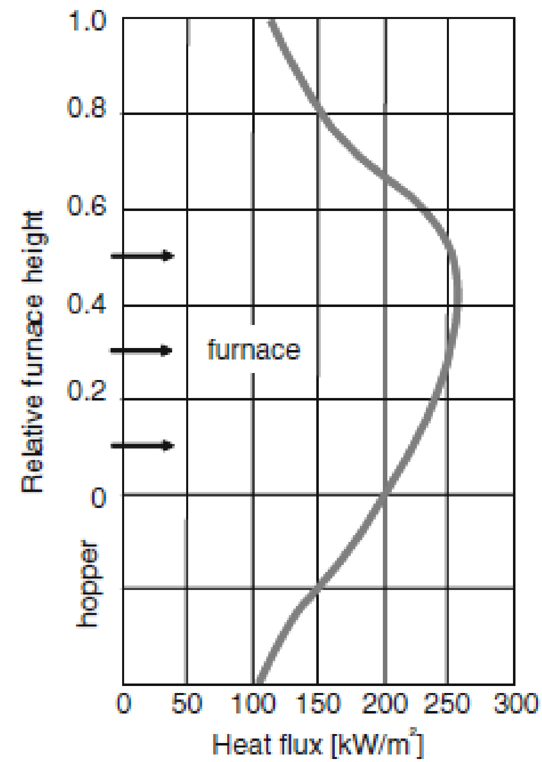
Ko is an undimensional similarity coefficient, called the Konakow number.

4.3.4.5 Calculation of the Flue Gas Cooling

In the calculation of modern steam generators with water-cooled tubes and vaporisation temperatures below 400◦C, T_W^4 can be neglected. Fouling and slagging of furnace walls make the temperatures rise considerably. The calculation of furnace wall heating surfaces and the preselected form (design) and dimensions of the cross-section together define the furnace height.

The objective of such a calculation is to determine the local heat fluxes towards the furnace walls and to determine the distribution of the temperature and heat flux densities inside the furnace and at the furnace end (Baehr 1985).

In most cases, simpler, partially empirical models are employed. The results of a one-dimensional plug flow model based upon a mean cross-sectional temperature are shown in Fig. 4.35. The maximum heat flow density in the upper burner area ranges around 0.27MW/m2 during standard operation. Firing conditions deviating from standard operation, such as those during fuel changes, when changing burner combinations, while there are unbalanced fuel and air distributions, during load change, or furnace wall fouling, can lead to locally higher heat flow densities. In the design and calculations of firing and heat transfer conditions, these cases are usually taken into account using empirical values (Stultz and Kitto 1992).



4.3.5 Design of the Steam Generator and of the Heating Surfaces

In designing the steam generator, it is necessary to dimension the heating surfaces such that the temperatures and mass flows defined in the cycle design can be met while taking the allowable material temperatures into consideration. Designing the thermal configuration and the steam generator is an iterative procedure. The design and construction regulations for steam generators are specified in the "ASME Boiler and Pressure Vessel Code" by the American Society of Mechanical Engineers (ASME) can be used (even in Europe).

The steam generator heating surfaces are the membrane furnace walls and the flue gas pass, as well as the tube banks across the flue gas cross-section. These walls are only exposed to thermal radiation on one side; the other side is insulated against the outside in order to avoid heat loss.

The components in the flue gas path that follow after the furnace are the convective heating surfaces of the superheater, the reheater and the economiser. Heat is mostly transferred by convection.

Evaporator and superheater surfaces are exposed to much higher temperatures on the side facing the fire and the flue gas than on the water/steam-cooled side.

The allowable tube wall temperatures can be above the temperature of the working medium by a maximum of 50K for radiant heating surfaces and of 20K for convective heating surfaces. The temperatures must not exceed the tube wall temperature limits, which are dependent on the materials and the design pressure.

4.3.5 Design of the Steam Generator and of the Heating Surfaces

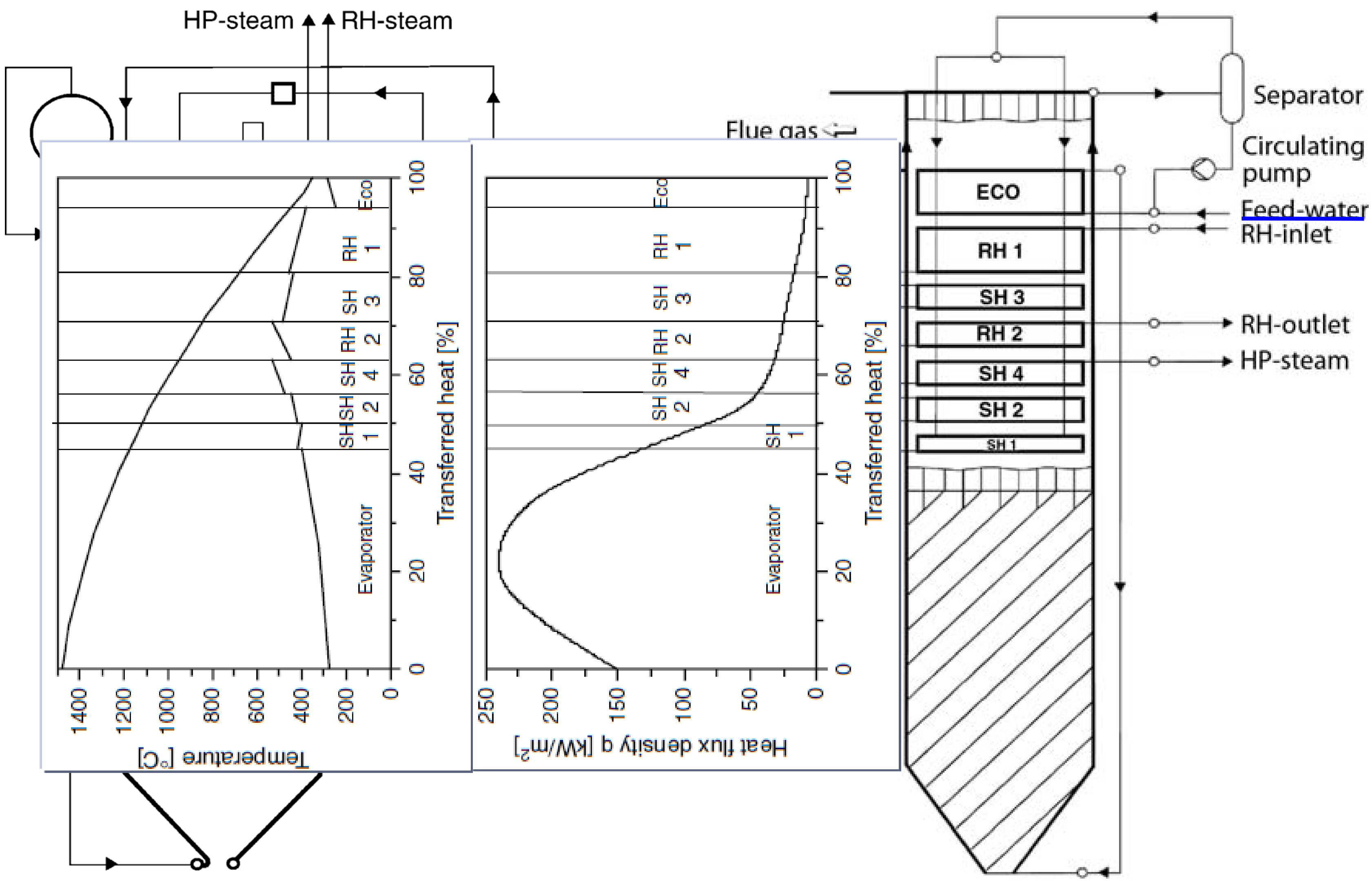


Fig. 4.37 Heating surface configuration of a two-pass boiler

Figure 4.36 Heating surface configuration of a single-pass or tower boiler (tower boiler)

4.3.5 Design of the Steam Generator and of the Heating Surfaces

Figure 4.38 charts the flue gas temperatures and the material temperatures along the flue gas path of the reference power plant. It can be observed that the heat flux density declines along the flue gas path. The heat transfer coefficient, too, shows a decrease towards the end of the steam generator, with the exception of the economiser stage.

The furnace walls are the heating surfaces with the highest temperatures (occurring on the flue gas side) and highest heat flux densities. The water/steam has to guarantee sufficient cooling in order keep the tube-furnace wall temperatures below the allowable material temperatures.

과제 : 보일러의 메탈온도와 스팀온도(실제데이터와 비교)를 단계별로 계산해서 제출하기

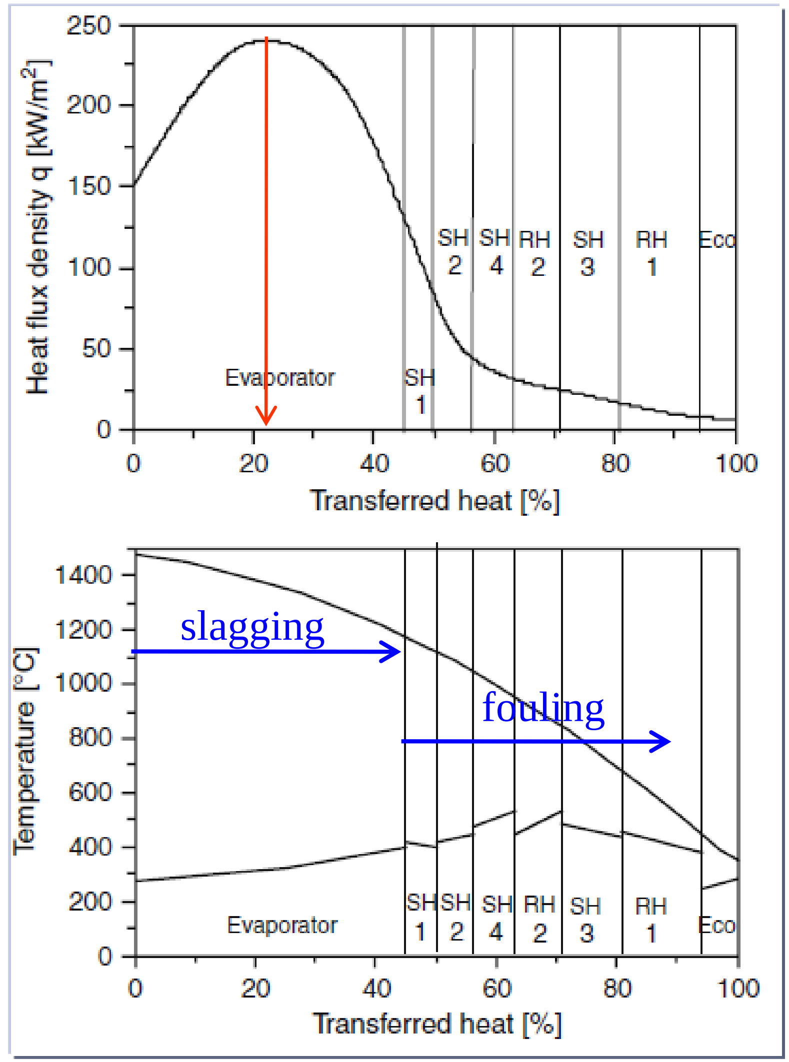


Fig. 4.38 Flue gas, temperature of the working medium and heat flux density of the reference power plant

4.3.5.1 Impact of the Live Steam Pressure

The next step is to integrate the live steam pressure defined by the steam conditions into the design of the heating surfaces.

Figure 4.39 shows h–p diagrams for a low-pressure and for a high-pressure boiler. At low pressures, the heat of evaporation predominates, whereas at higher pressures, the vaporisation enthalpy decreases and the heat demand for superheating increases (Doležal 1990).

In designing steam generation systems with a fixed liquid-vapour phase transition point, it is possible for over-determination to occur. This is because the vaporisation heat decreases with higher design pressures, while the flue gas cooling requirements and the evaporator capacity are fixed. As Fig. 4.40 shows, the entire furnace is required to act as an evaporative heating surface at low pressures, as the feed water is preheated and steam superheated only on convective heating surfaces.

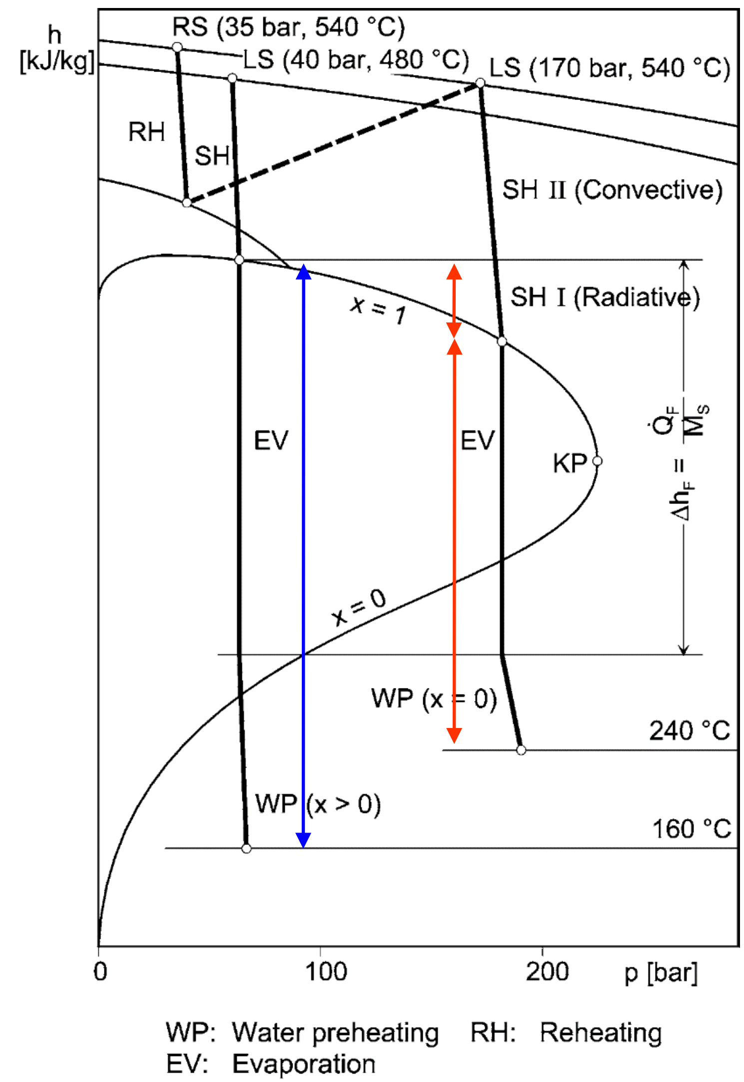


Fig. 4.39 h – p diagram for LP and HP boilers (Doležal 1990)

4.3.5.1 Impact of the Live Steam Pressure

Given the lower vaporisation enthalpy at high pressures, the flue gases are not sufficiently cooled as they flow towards the furnace end if no additional measures are taken. The resulting furnace outlet temperature at a live steam pressure of 170 bar thus amounts to 1,300◦C. But because a great number of coal types have lower ash deformation temperatures than this, the flue gas has to be further cooled by additional measures (Wittchow 1982).

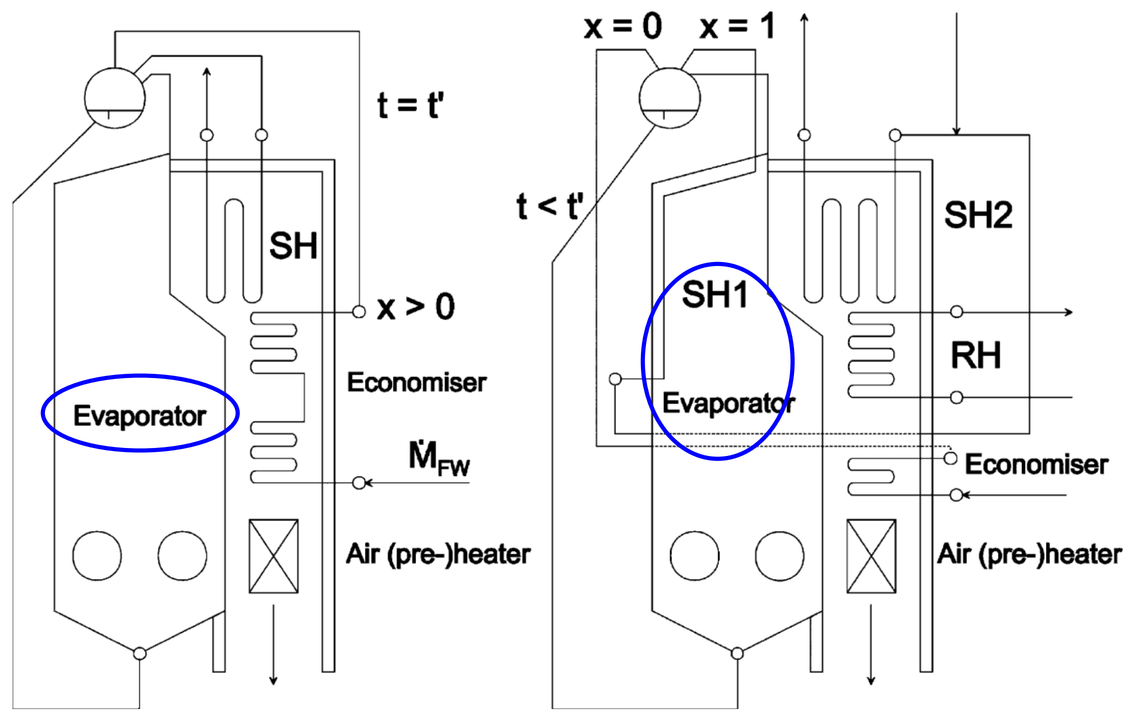


Fig. 4.40 Construction of a low-pressure and of a high-pressure drum boiler (Doleˇzal 1990)

4.3.5.2 Design of the Evaporator

For the design of a steam generator, it is necessary to know beforehand the maximum tube wall temperature, which is a function of the gas-side heat flux density (about 300–350kW for hard coal firing systems), and the mass flow density of the Steam-water mix. This is in order to avoid the allowable material temperatures being exceeded where the boiling crisis occurs. In general, the mass flow density of an evaporator with plain tubes lies between 700 and 800 kg/m2 s at a minimum output of 30–40% (Franke et al. 1993). The mass flow density at the rated power lies between 2,000 and 2,500 kg/m^2 s.

Figure 4.41 shows the inside wall temperatures as a function of the steam quality for plain tubes at different heat and mass flux densities. At a strong heating density of 450kW/m^2, a too-low mass flow density of 900 kg/m^{2}s causes a strong rising of the tube wall temperatures. With reduced heat flux densities, such as occur in partial-load conditions, the temperature rise is less dramatic (Franke et al. 1993).

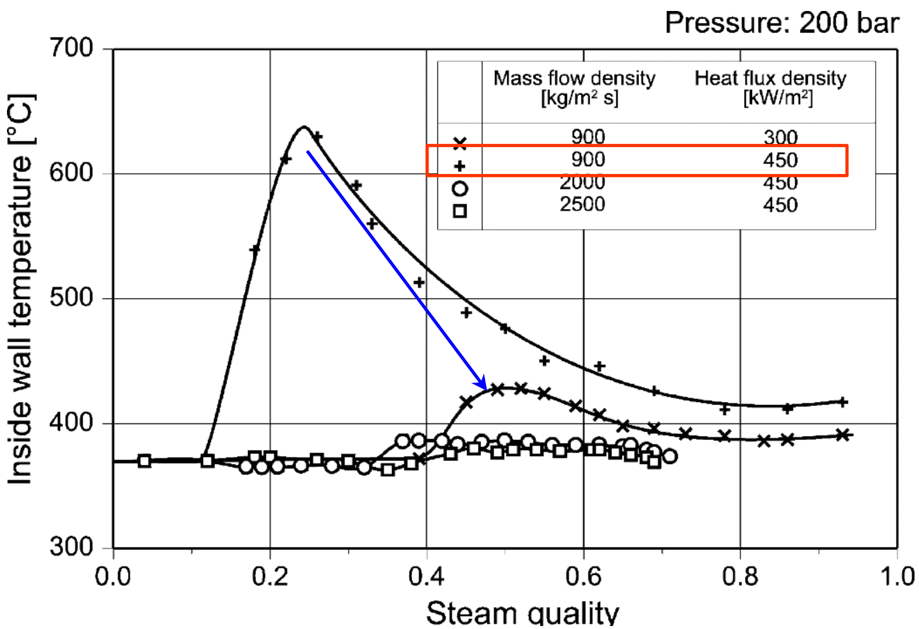
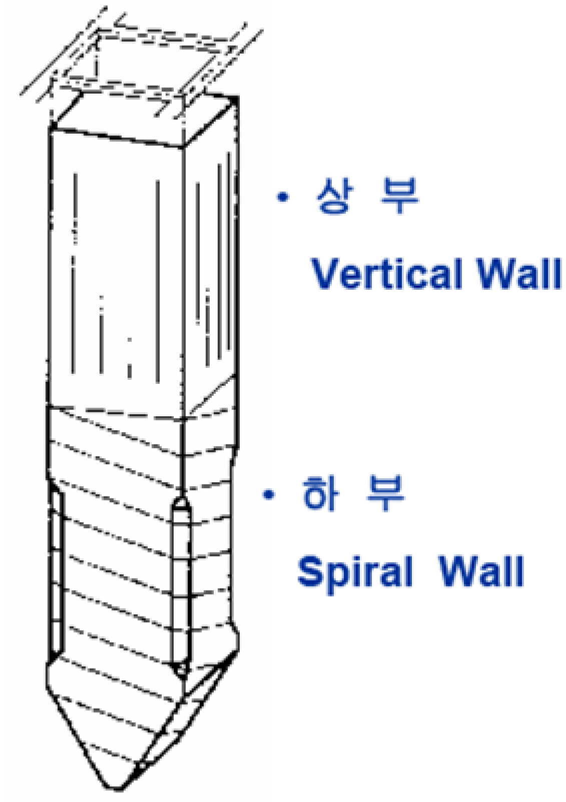


Fig. 4.41 Inside wall temperatures of a heated plain tube (Franke et al. 1993)

4.3.5.2 Design of the Evaporator (Radiation)

The number *n* of the welded parallel tubes depends on the mass flow density *Φ* required for cooling at the partial load *q*, the inner tube diameter *d*, and the steam flow *m* (Strauß 200



Evaporator 설계 :

항 목	단위	표준 500MW	후속기 500MW
Spiral wall tube O.D. / Spacing	mm	38 / 54	38 / 54
Spiral wall tube angle	Degree	14.2086	13.95
Vertical wall tube O.D. / Spacing	mm	31.8 / 55	34,38 / 56
No of Vertical tube / Spiral tube		4	4

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$$\beta = \arcsin\,(b/P_F)$$

The helix angle increases with the boiler size.

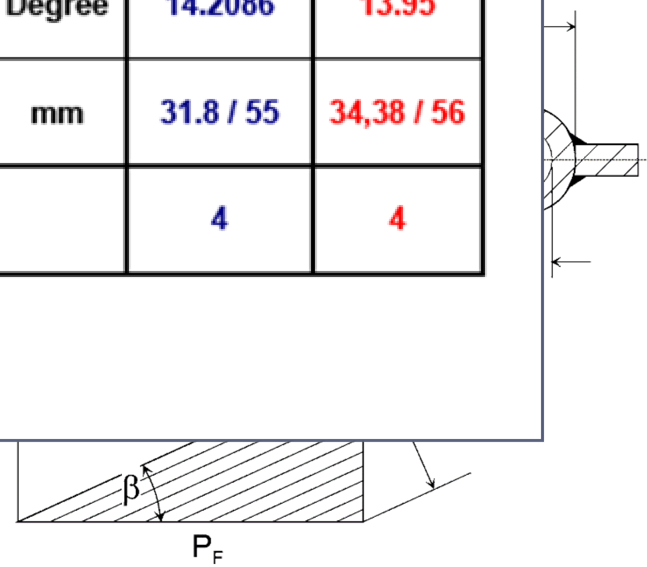


Fig. 4.42 Schematic drawing of the helical winding (Doleˇzal 1990)

4.3.5.2 Design of the Evaporator

In the upper area of the steam generator begins, the helical winding transforms into vertical tubing. Because vertical tubing is more economical than helical winding, it should be designed to begin at the lowest possible furnace height. The helical winding and vertical tubing are joined by clevises. In the vertical tubing, the mass flow density of the working fluid, then in a vapour state, is diminished by increasing the number of tubes by a factor of 3~4. If the transition to vertical tubing is carried out at a furnace height which is too low, it is possible that, with high gas-side heat flux densities and low mass flow densities of the cooling fluid, excessive tube wall temperatures arise in the vertical tubes. In contrast, when the helical winding is too high, non-uniform heating can have stronger effects due to the longer tubes of the helix, thus also causing the tube wall temperatures to exceed the allowable limit (Kefer et al. 1990).

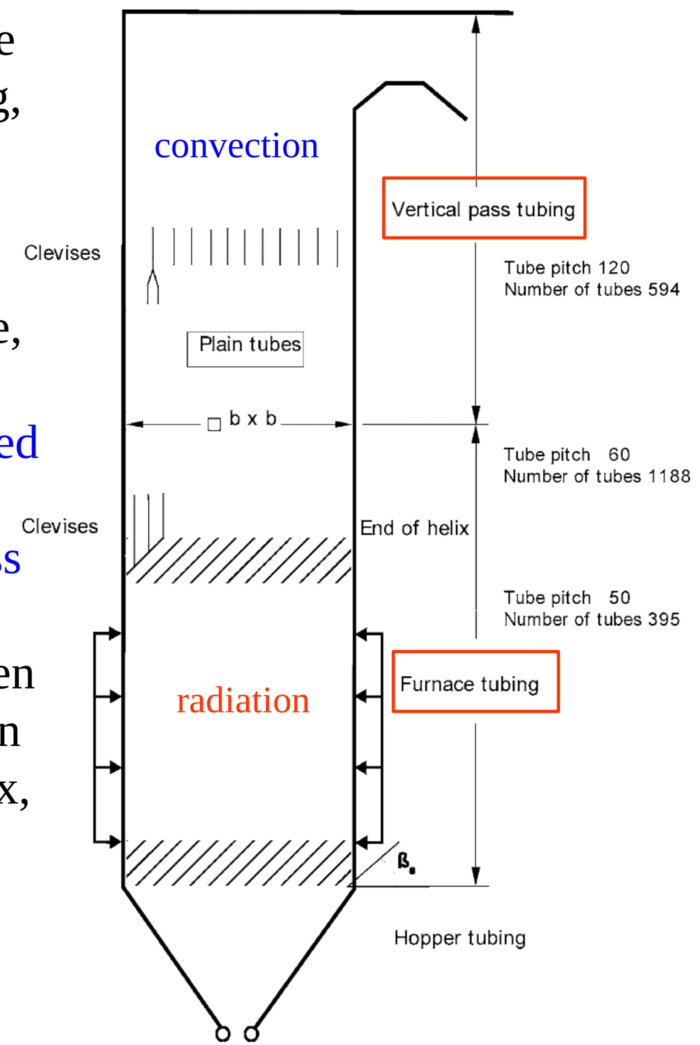


Fig. 4.43 Wall tubing of a single-pass boiler with helical winding in the furnace section (Source: Alstom Power)

4.3.5.3 Evaporators with Vertical Internally Rifled Tubes

Internally rifled evaporator tubes allow lower densities of the water/steam mass flow at the same heat flux density, owing to the more intensive heat transfer from the inner tube wall to the working fluid, so that the evaporator tubes can also be mounted vertically in the furnace (see Fig. 4.44).

The low mass flow densities of internally rifled tubes allow sufficient cooling at low minimum capacities, without causing high pressure losses due to high mass flow densities at full load. In comparison to inclined tubes, vertical mounting avoids segregation processes. The minimum load of the steam generator can be lowered from 35–40 to 20%.

A lower minimum load could decrease the number of start-ups and shutdowns, which would have a positive effect as regards both the fatigue of elements and the fuel consumption, because start-up and shutdown losses are avoided

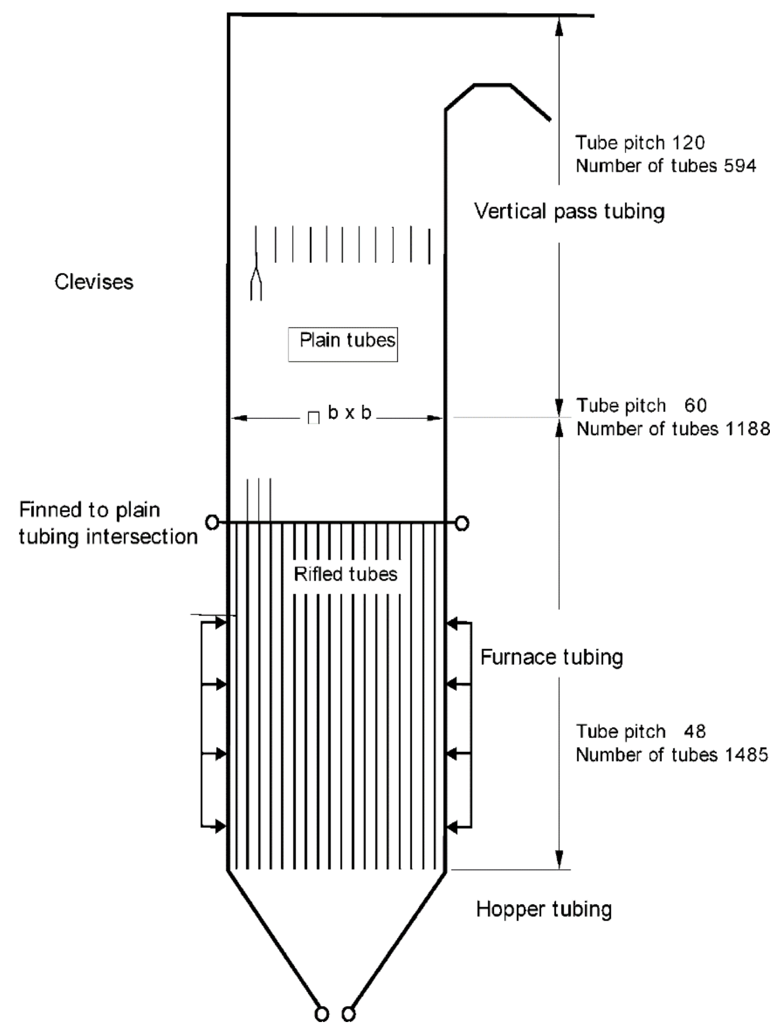


Fig. 4.44 Wall tubing of a single-pass boiler with vertical tubes in the furnace section (Source: Alstom Power)

4.3.5.3 Evaporators with Vertical Internally Rifled Tubes

If the geodetic pressure loss predominates, the additional heating leads to higher mass flow densities. Due to the increased steam formation, the geodetic pressure loss of a tube with constant mass flow diminishes, because the steam column becomes lighter. The decrease of the geodetic pressure drop is higher than the rise of the friction loss.

The pressure loss being given, however, the mass flow through the additionally heated tube rises (natural-circulation characteristic, see Fig. 4.45). The impact of the extra heating on the steam temperatures at the evaporator outlet is minimised by the self-regulating effect. This can be an advantage for the application of higher steam conditions, since the difference between the fluid temperature in the evaporator and the allowable material temperature may be smaller (Franke et al.1993, 1995; Wittchow 1995). On the other hand, the counterbalance of the heating by the helically wound tubes does not apply.

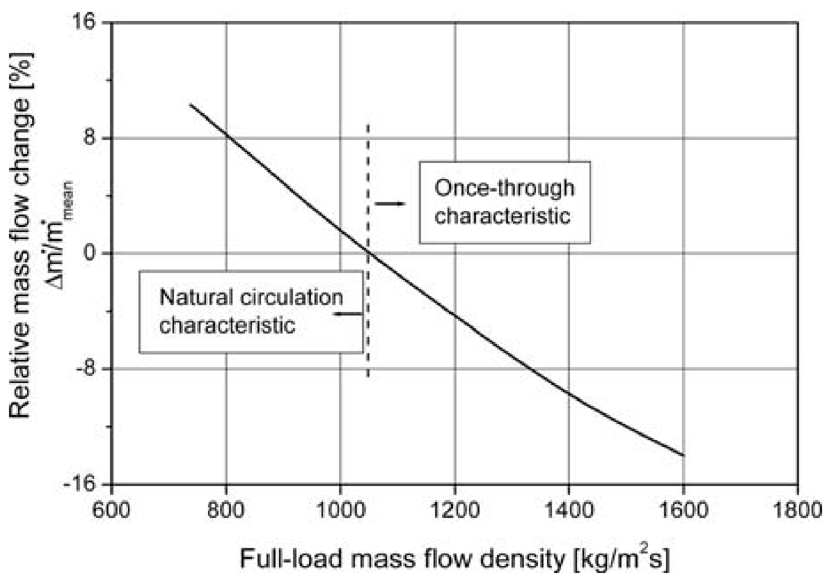


Fig. 4.45 Throughput characteristic of a tube with 25% extra heating (Wittchow 1995)

4.3.5.4 Evaporator Stability

Different operating modes of and uneven fuel flows to the burners of a burner group cause asymmetric firing conditions and non-uniform heat fluxes to the furnace walls. The helical winding still ensures a good heating balance because each of the parallel tubes runs along all four walls of the furnace(Franke et al. 1993).

Figure 4.46 shows the correlation between pressure loss and steam mass flow with the heating as a parameter. While the characteristics of tubes filled with a water flow correspond to a second-order parabola, tubes which are filled by a flowing two-phase mixture, i.e. boiling water and steam, give a third-order curve. An unstable flow occurs if the curve has a saddle-like behaviour, the consequence of which can be that three different mass flows evolve for the same pressure gradient. If a mass flow has a lower rate than needed for cooling the tubes, the effect can be damage to the tubes. The stability of steam generators and measures to raise the stability are dealt with in detail in Dolezal (1990). <-과제: 조사할것

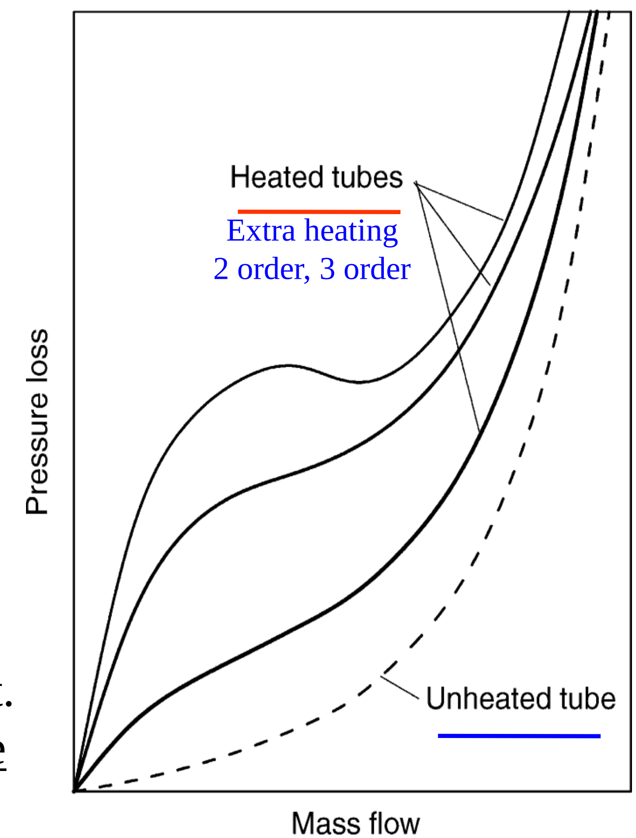


Fig. 4.46 Characteristic curves of the evaporator (Baehr 1985)

4.3.5.5 Design of the Convective Heating Surfaces

The units in the flue gas path following the furnace are the convective heating surfaces of the superheater, the reheater and the economiser. In contrast to the heat transfer to the evaporator surfaces by radiation, transfer by convection applies heat to the whole tube circumference, which is why the heating surface banks are smaller for the same temperature difference. The heating surface dimensions being decided previously, the heat transfer depends on the flue gas velocity and the driving temperature difference. The distance of the tube, however, is limited by the increasing pressure loss on the flue gas side and by possible fouling due to fly ash deposits (Stultz and Kitto 1992).

Superheater and Reheater

Heating surfaces used for superheaters and reheaters can be hanging or horizontal tube bundles. In Germany, where the single-pass construction is commonly built, only horizontal, drainable heating surfaces are used. In boilers in two-pass construction, hanging heating surfaces are often used in the cross-pass for super or reheating. The distance between the tubes, the so-called tube pitch, depends on the flue gas temperature and the flue gas dust concentration (ash content of the coal). With the decreasing temperature, the tube pitch narrows in the direction of the flue gas flow. With low ash contents of the coal, it is possible to use smaller tube pitches and hence to build a more compact steam generator.

4.3.5.5 Design of the Convective Heating Surfaces

Two-pass boilers of the US type often have a hanging plate-type superheater which can be used in areas of high temperatures of around 1,400*C*. With these heating surfaces, the predominant method of heat transfer is radiation. The tubes, wound closely to each other in a plane, form a plate, with large distances, of more than 1m, bet. the plates. Such plate superheaters are insensitive to ash deposits.

Figure 4.47 shows the tube pitches as a function of the flue gas temperature for US-type steam generators. In the case of single-pass or tower boilers, the upward-diminishing transverse pitch enables the dropping through of ash deposits that have come off.

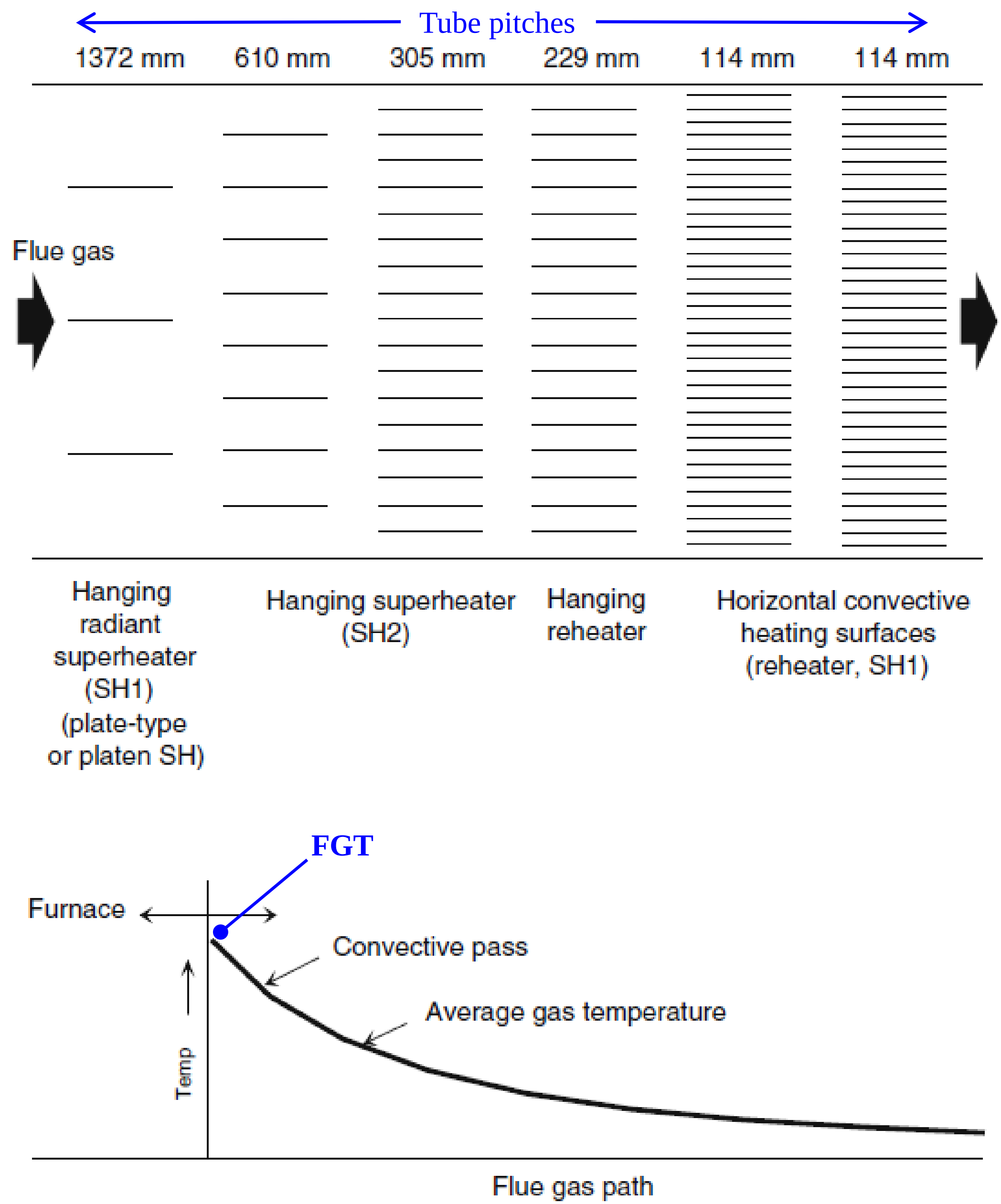


Fig. 4.47 Heating surface divisions in US constructions (Stultz and Kitto 1992)

4.3.5.5 Design of the Convective Heating Surfaces

Design and construction therefore have to guarantee an even flow and to counteract the impacts of an imbalanced heating. Temperature discrepancies between individual tubes are balanced out by dividing the superheater system into several stages, combining and mixing all single-tube steam flows in one stage and then re-establishing the division in the following stage into single-tube steam flows. Large steam generators usually have superheaters divided up into several piping runs connected in parallel. By crossing the piping runs between the superheater stages, it is possible to counteract uneven heating (see Fig. 4.48).

With this construction the steam flows in the runs change their position in the flue gas pass from one outside to the other or, in the case of four piping runs, from outside to inside and vice versa (Strauß 2006; Baehr 1985). For the control and limitation of live steam and reheater steam temperatures, attemperation is commonly applied. High-pressure feed water (HP feed water) is injected before or after the last superheater or reheater stage in attemperators.

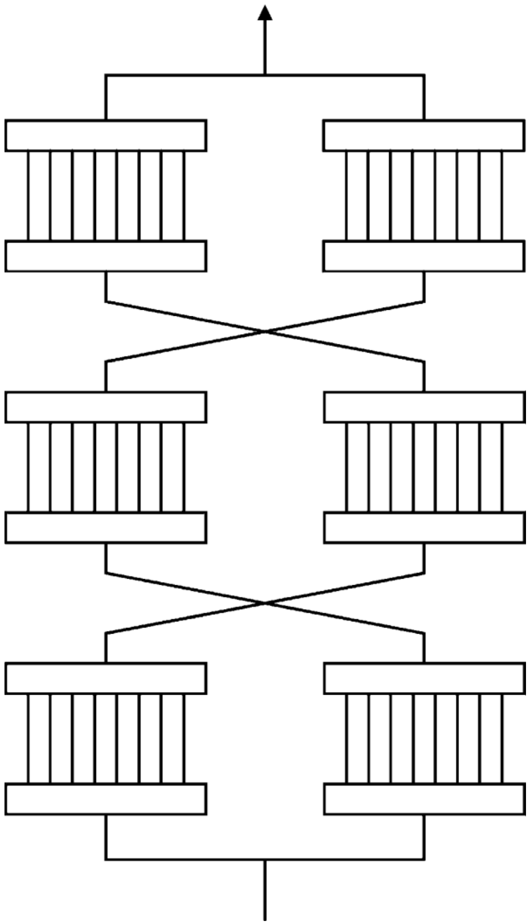


Fig. 4.48 Crossing of multistage superheaters

4.3.5.5 Design of the Convective Heating Surfaces

Falling high pressures and reheater steam temperatures in partial-load conditions diminish the mean temperature of the heat addition and hence the thermal efficiency. The live steam and reheater steam temperature should therefore be constant throughout the whole load range. The design and location of the heating surfaces determine the temperature characteristics as a function of the load for each superheater and reheater. Heating surfaces in areas of high temperatures, above about 1,*200°C, take up heat predominantly* by radiation, and heating surfaces in areas of low temperatures, mainly by convection (Strauß 2006).

With the output diminishing, the radiant heating surfaces in the furnace take up relatively more heat (radiation characteristic) whereas the heat share of the convective heating surfaces decreases (convection characteristic). Care should be taken that, for superheating and reheating convection surfaces, both convection and radiation characteristics are incorporated into the design, in order to achieve a constant steam temperature throughout the output range (see Fig. 4.49)

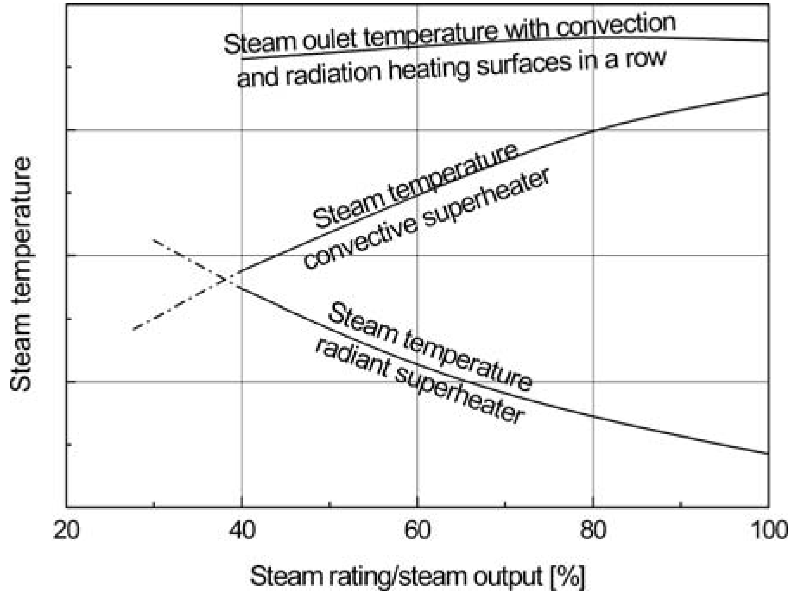
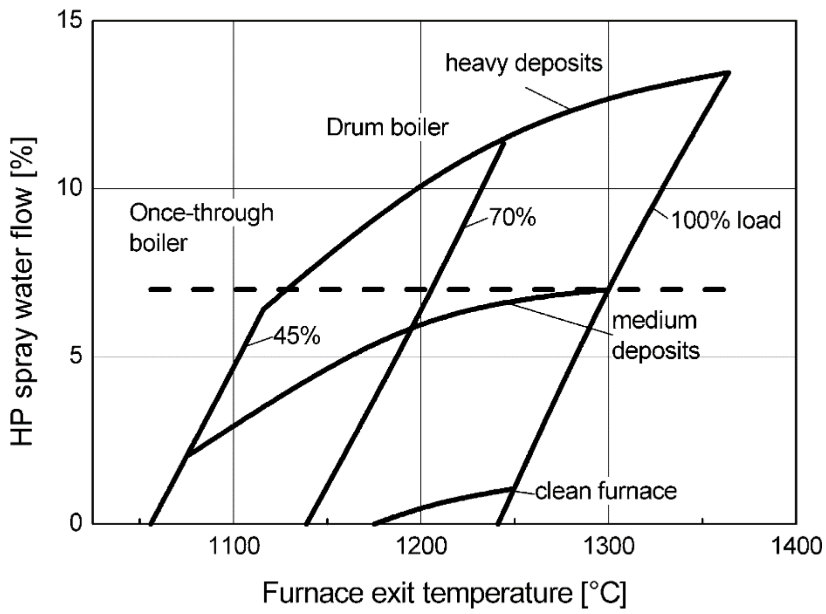


Fig. 4.49 Characteristics of radiation and convection heating surfaces

4.3.5.5 Design of the Convective Heating Surfaces

Firing conditions and heat flux distributions deviating from the normal state evolve through the fouling of the heating surfaces. Ash and slag deposits on the furnace evaporator walls move heat to the convective heating surfaces. The radiation heat fraction taken up by the furnace walls and the evaporator decreases. Figure 4.50 shows, for once-through and circulation steam generators, the impacts of fouling on the spray water mass flow in relation to the output-dependent flue gas temperature at the furnace exit (Wittchow 1982).

Once-through steam generators adapt to these changes by shifting the liquid-vapour phase transition point in the direction of the evaporator end, so that the superheater area becomes smaller. The greater convective heat flux fraction is balanced out by the altogether smaller effective superheating surface, while the steam temperatures and the spray water flows remain nearly constant, independent of the fouling state in the steady-state condition.



4.3.5.5 Design of the Convective Heating Surfaces

Maintaining Constant Reheater Temperatures

As in the case of superheating in high-pressure zones, heating surfaces with convection and radiation characteristics should be utilised in order to keep constant reheater steam temperatures. Reheating does not involve the balancing influence on the live steam temperature by migrating vaporisation and superheating zones in the furnace wall of a once-through steam generator.

The operating regime of a steam generator – fixed or sliding pressure – can have an influence, however, on the necessary temperature rise. In fixed-pressure operation, the reheater must be supplied with relatively more heat because the reheater inlet steam temperature drops as the output decreases. But in sliding-pressure operation, the reheater inlet temperature is nearly independent of the output.

A relatively simple method to control and limit the reheater outlet temperature is to spray feed water between two subsequent reheat surfaces at a pressure similar to the exit steam pressure of the high-pressure turbine. In this case, the reheater is designed to be larger for full load and its steam exit temperature is limited to the allowable temperature by spray water admixing. When output diminishes, the necessary spray water flow decreases as well.

Reheater spraying for temperature control, however, has the consequence of a loss in efficiency, because the high-pressure zone of the steam generator is bypassed, and only steam at the reheater pressure is produced and exploited. The heating of the spray water by mixing at a low reheater pressure results in a lower temperature of heat addition.

4.3.5.5 Design of the Convective Heating Surfaces

Economiser

The economiser (sometimes shortened to “eco”), or feed water preheater, is a steam generator’s penultimate fireside heating surface and at the same time its first heating surface on the steam side. The entrance temperature of the feed water is 250◦C for the reference power plant, while the flue gases are cooled from about 450 to about 350◦C. In once-through steam generators, the last part of preheating before boiling starts occurs in the evaporator, avoiding premature vaporisation in the economiser.
In circulation steam generators, the preheated feed water, for the same reason, is fed into the evaporator drum before the boiling stage.
As a consequence of the small temperature difference between the two working media, the economiser needs a very large heat exchange surface. The raw material utilised for the economiser is usually unalloyed steel. Plain tubes are used as a rule. External fins improve the fireside heat transfer if they are kept free from ash deposits (Stultz and Kitto 1992).

4.3.5.6 Air Preheater

The air preheater transfers flue gas heat from the lower flue gas temperature region to the combustion air. This low-temperature heat transfer diminishes the necessary fuel energy on one hand and, on the other, influences the ignition and the combustion course of the firing by higher temperatures of the combustion air.

The use of regenerative feed water preheating to raise the cycle efficiency requires combustion air preheating, as the medium, water, cannot be used to make use of the flue gas waste heat, because of the higher temperatures. Air preheating raises the combustion temperature (in the furnace) and, owing to the higher temperature drop between flue gas and steam, makes it possible to use smaller heating surfaces.

In the air preheater of the reference power plant, the flue gas cools down from the temperature after the economiser of 350◦C to a temperature of 130◦C, which lies above the acid dew point of the flue gas. In the counterflow, the combustion air of about 45◦C, after being preheated by a steam air heater, is heated up to the combustion temperature of 310◦C. Low outlet flue gas temperatures minimise the flue gas energy losses of the steam generator. The acid dew point of the flue gas sets the low-temperature limit, as temperatures below this point would result in corrosion and fouling. Measures to increase the efficiency by limiting the flue gas energy losses make use of existing design reserves, but may apply restrictions on the coal feedstock (see also Sect. 4.4.2.2).

4.3.6 Design of the Flue Gas Cleaning Units and the Auxiliaries

4.3.6.1 Design of the Flue Gas Cleaning Units

The allowable emission standards require installations for dust removal, nitrogen control and desulphurisation. The boundary conditions are the pre-determined flue gas mass flow and the necessary removal efficiencies, which are determined by the dust content, the sulphur dioxide and nitrogen oxide concentrations in the raw gas and the respective emission standards. The techniques of emission reduction are dealt with in Chap. 5 in the context of firing technology.

4.3.6.2 Design of the Auxiliaries

The forced-draught fan supplies the burners with the air mass flow required for combustion (determined during the design). Booster fans produce the rise in pressure of the primary air necessary to surmount the additional resistance in the mills, classifiers, pulverised coal supply pipes and the burners. The pressure losses of the secondary air range around 70 mbar; those of the primary air are about 160 mbar.

The power demand of the induced-draught (ID) fans for transporting the flue gases depends on the flue gas mass flow and on the pressure drop along the flue gas path. The furnace is operated with some mbar of underpressure. The pressure drop along the flue gas path before the ID fan, which transports the total flue gas, amounts to 40–50 mbar at the rated power of the plant, depending on the fireside flow resistance.

4.4 Possibilities for Efficiency Increases in the Development of a Steam Power Plant

One solution for the reduction of CO_2 emissions from power plants fired with fossil fuels is to increase their efficiency. All fossil fuels have a content of carbon, either higher or lower. Coal, among them, is a fuel with a comparatively high carbon content and at the same time the fuel with the highest percentage use worldwide in power production.

Research and development is currently being conducted, aimed at reducing CO_2 emissions by increasing the efficiencies of all the units in a power plant discussed in this book. The possibilities in this respect are distinguished for stationary operation as follows:

- Increases in the thermal efficiency of the cycle
- Measures to minimise the losses
- Measures to reduce the auxiliary power requirements

The stated efficiency rates usually refer to the rated power. However, the efficiency of the plant in part-load operation and the losses at start-up and shutdown should be taken into account as well.

4.4.1 Increases in Thermal Efficiencies

Improvements of thermal cycles aim at attaining a high mean temperature of the heat addition and a low mean temperature of the heat extraction.
High mean temperatures of the heat addition and therefore high thermal efficiencies are achieved by

- increasing the live steam conditions (temperature and pressure),
- single or double reheating,
- regenerative feed water preheating,
- reducing reheater spraying and lowering mean temperatures of the heat dissipation and
- low exhaust steam temperatures in the condenser.

The conversion processes associated with losses are presented in Fig. 4.2.

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

High mean temperatures of the heat addition contribute to a high thermal efficiency. They can be achieved by a high pressure in the high-pressure steam generator (HP steam generator), by a high live steam temperature, by regenerative feed water preheating and by reheating to high reheater temperatures.

A higher live steam pressure entails correspondingly high boiling water temperatures, which raise the heat input temperatures to a higher mean level, with the outlet temperature remaining the same, thus increasing the thermal efficiency. Lower live steam pressures and hence lower boiling water temperatures decrease the mean temperature of heat addition and the efficiency. However, higher pressures require more power for the feed water pump. Further pressure increases give diminishingly greater thermal efficiencies, which are eventually cancelled out, and then exceeded by, the efficiency losses due to the increased feed water pump power requirements.

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

With increasing pressure, the live steam conditions shift towards smaller entropies. Correspondingly, the exhaust steam conditions also shift to lower steam and higher water contents (see Fig. 4.51). However, for technical reasons, the so-called exhaust moisture $(1-x_4)$ must not exceed values of about 0.1. With an excessively high exhaust moisture, droplet impact occurs in the last stages of the turbine, which leads to erosion of the final-stage blades. The prescribed exhaust moisture limits the choice of the live steam pressure for a simple steam-generating plant without reheating or makes it necessary to install reheating (Baehr and Kabelac 2006). Since the reheated steam after expansion has a higher entropy with a higher steam quality, damage of turbine blades through droplet impact is less likely.

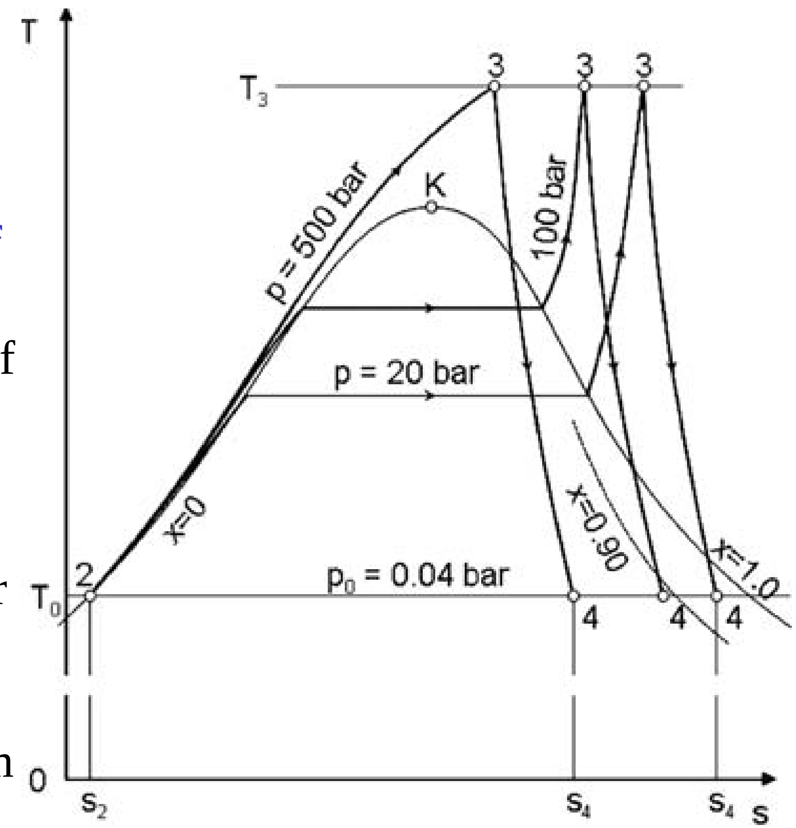


Fig. 4.51 Pressure influence on the exhaust steam conditions (Baehr 2006)

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

Higher live steam and reheater outlet steam temperatures also result in higher mean temperatures of the heat input, and thus in a higher thermal efficiency. Figure 4.52 shows the influence of pressure and temperature on the efficiency of the cycle, given as the relative heat rate gain. For the temperature range of up to 600◦C, a rule of thumb is an increase of the net efficiency of 0.02% (absolute) per degree of temperature increase (with the live steam temperature equalling the reheater temperature). In the range fro
per degree of temperature increase. Th
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In the pressure range up to 250 bar, a r
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an improvement of about 0.008 bar. W
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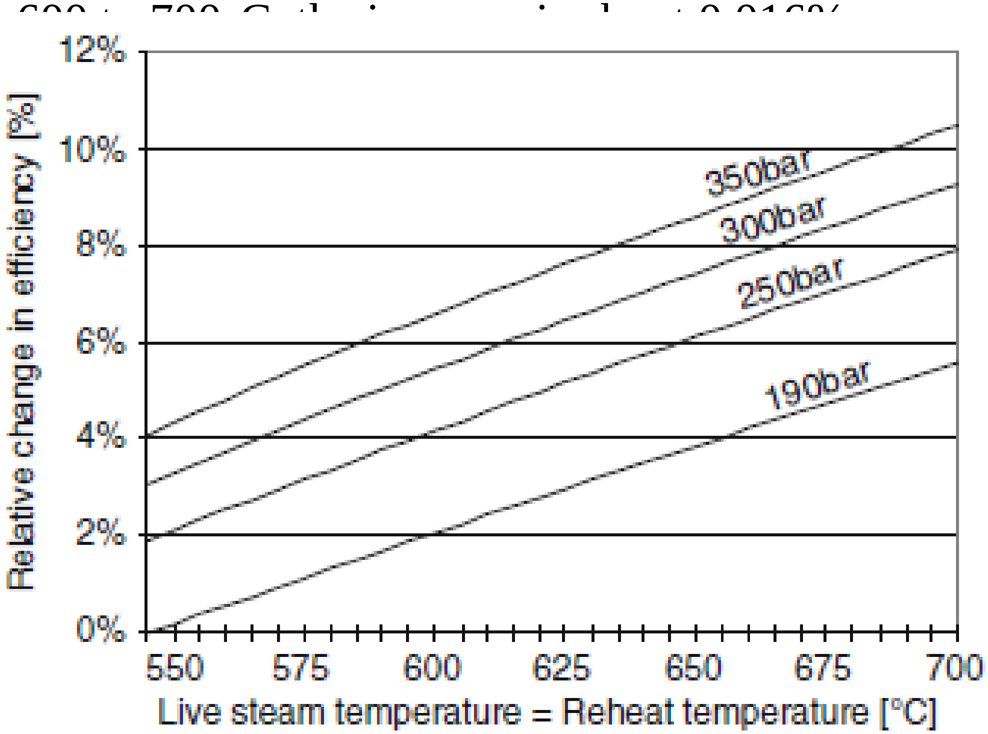

Fig. 4.52 Influence of live steam pressure and temperature on heat rate

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

High live steam pressures and temperatures are limited by the available construction material. Reheating raises the mean temperature of the heat input (Fig. 4.53) since the mean reheating temperature is higher than that of the simple steam process. For the reference power plant with conventional steam conditions (190 bar, 530◦C, 530◦C (turbine inlet)) the mean heat input temperature in the high-pressure part of the steam generator lies at $T_{m'}$=364◦C, while the medium temperature of heat addition in the reheater is $T_{m''}$= 430◦C, resulting in an overall mean temperature of heat addition T_m= 376◦C. For a power plant with advanced steam conditions (285 bar, 600◦C, 620◦C) the medium temperature of heat addition lies at T_m= 415◦C, while in the high-pressure part of the steam generator the mean temperature is $T_{m'}$= 400◦C and in the reheater $T_{m''}$= 470◦C.

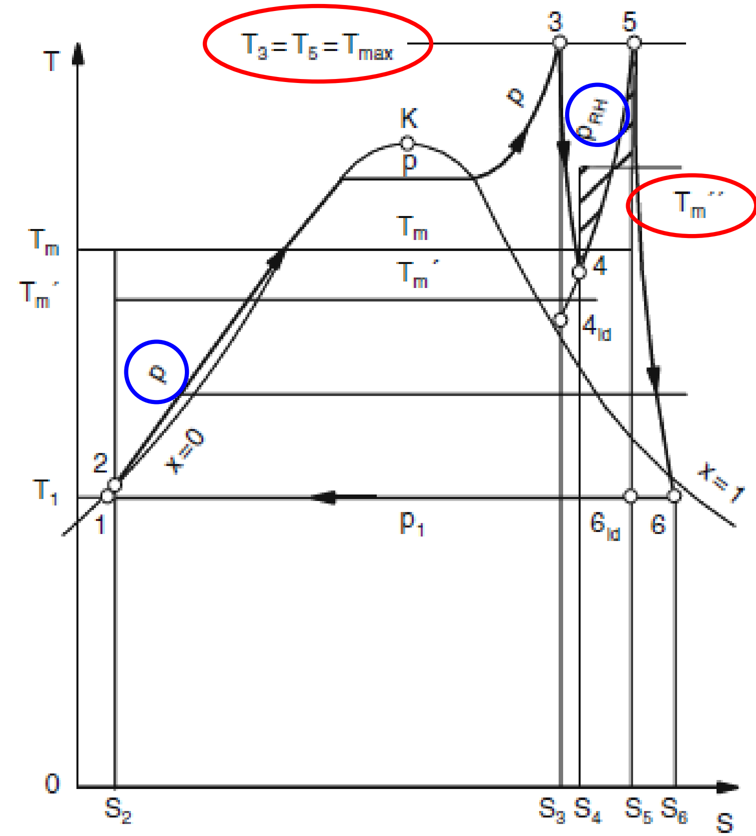

Fig. 4.53 Changes of state in the process with reheating (Baehr and Kabelac 2006)

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

In designing a power plant, optimum pressure ranges arise both in single and in double reheating. The optimum pressure depends on the live steam pressure. As a rule of thumb, the ratio between live steam and reheater pressure in modern power plants is between 5 and 6. The optimum can be explained with Fig. 4.53. Reheating results in a maximum increase of efficiency, if the cold reheat temperature T4, which is a function of the reheat pressure, is at the level of the medium temperature of heat addition tm in the high-pressure part of the steam generator. In this case, reheating increases the medium temperature of heat addition of the steam generator. If the cold reheat temperature is lower, at least part of the heat addition in the reheater results in lower efficiencies. Additionally, a reheat pressure that is too low can result in superheated steam at the turbine exit and thereby increase the medium temperature of heat dissipation.

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

Figure 4.54 shows the optimum of a double reheating regime in the form of equidistant efficiency curves. Deviations from the optimum pressures entail a deterioration of the efficiency. The optimum depends on the chosen live steam pressure (Rukes et al. 1994).

Assuming conventional steam conditions–such as those of the reference power plant for instance–introducing double reheating raises the net efficiency rate by up to 2%. Higher live steam pressures increase, while higher live steam temperatures decrease the gain in efficiency (Adrian et al. 1986). For a power plant with steam conditions of 280 bar, 585◦C (live steam), 600◦C (reheat steam), double reheating raises the net efficiency by 0.7% (Kotschenreuther et al. 1993). Double reheating can have a disadvantageous effect on the operating regime; an allowable load change rate between 2 and 4% per minute was reported.

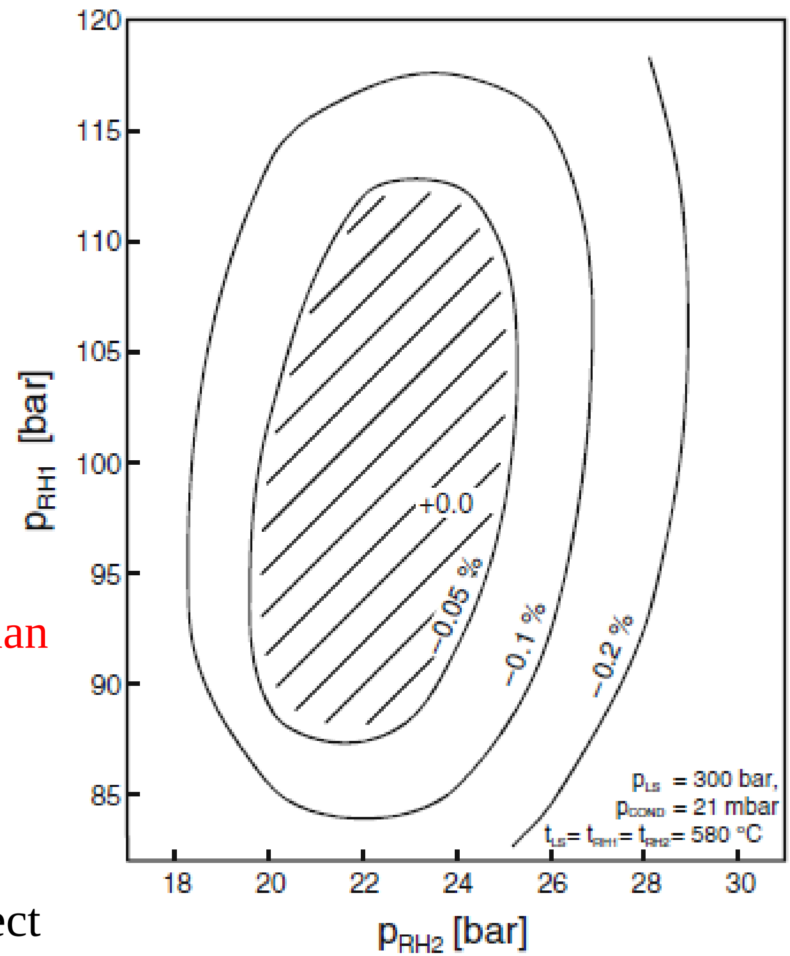

Fig. 4.54 Equidistant efficiency curves with the deviation from the optimum net efficiency as a function of the reheater pressures with double reheating (Kjaer 1990)

4.4.1.1 Increasing the Live Steam and Reheater Steam Conditions, Single or Double Reheating and Reheater Spraying

As reported in Sect. 4.3.5.5, controlling the reheater temperature by a spray attemperator diminishes the efficiency, because the high-pressure range of the steam generator is bypassed by doing so, and steam is produced at a low pressure and temperature. Figure 4.55 shows the influence on the efficiency of the reheater attemperator mass flow (Baehr 1985). In the case of the reference power plant, the spraying mass flow at full load is about 0.9% of the feed water mass flow. New power plant designs limit the temperature-controlling spraying mass flow to 0.2% of the feed water mass flow (Breuer et al. 1995)

(500MW; 1520t/h, 20t/h)

The measures described above have an effect only on the thermal and on the turbine efficiency, but not on the energetic steam generator efficiency. They are included in the exergetic steam generator efficiency rate, though (see Sect. 3.2).

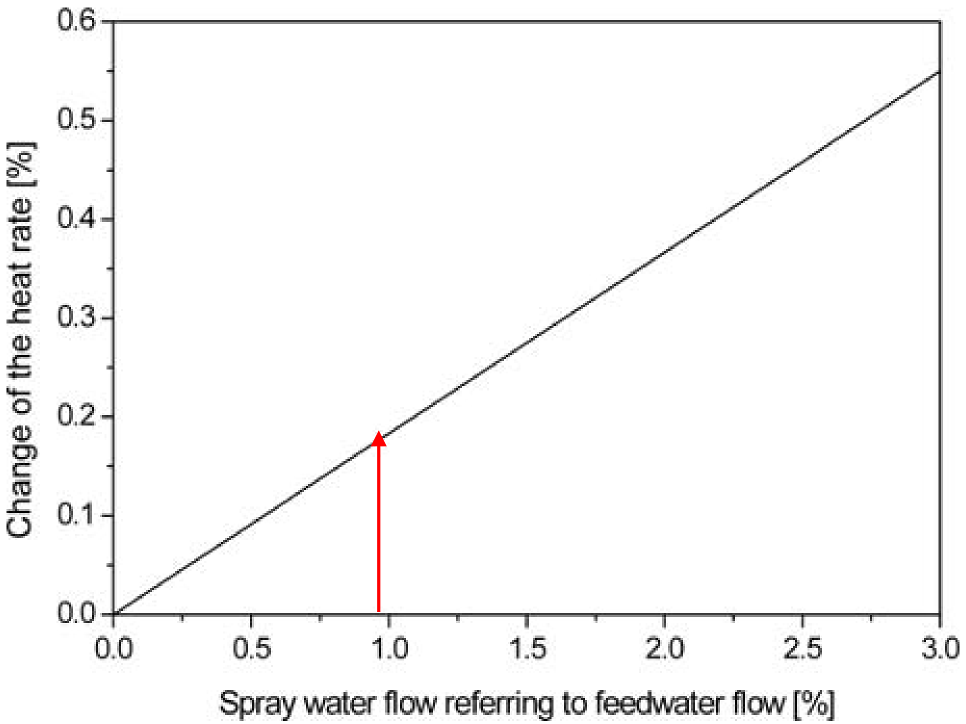

Fig. 4.55 Influence on the efficiency of reheater spraying (Baehr 1985)

4.4.1.2 Influence of Feed Water Preheating

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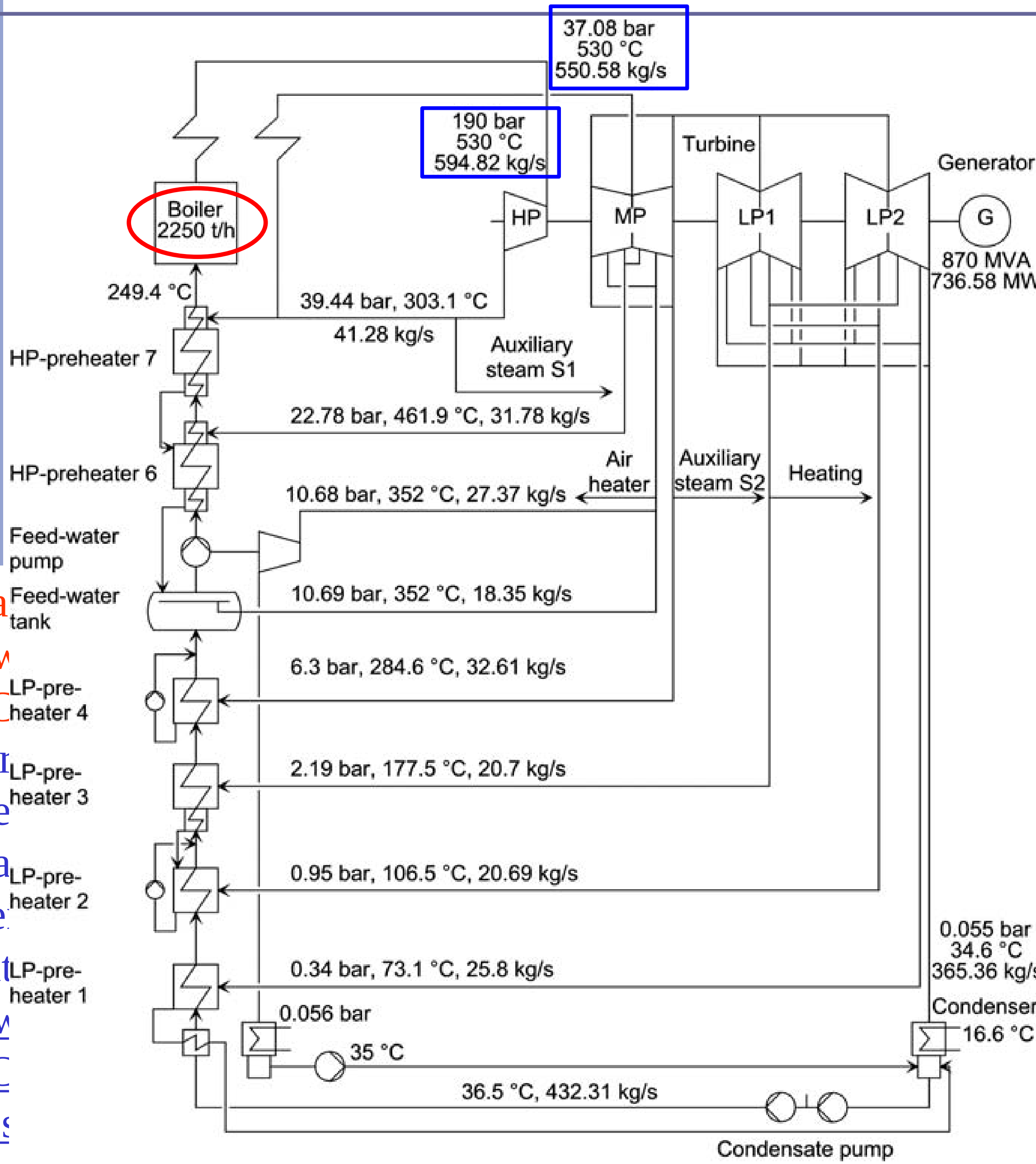

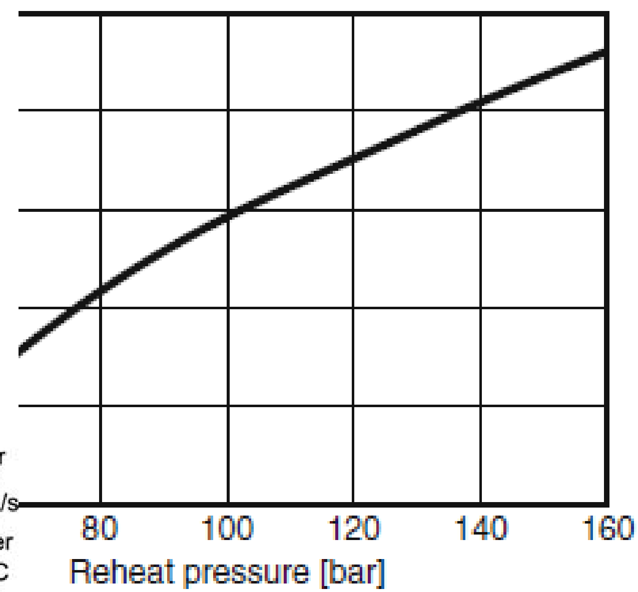

ed water temperature as a function of
ressure (Rukes et al. 1994)

4.4.1.2 Influence of Fe

The feed water heating temperature can be further raised by inserting an additional preheater, heated by extraction steam from the high-pressure section of the turbine. Such additional extraction from the HP turbine section uncouples the reheater pressure and feed water outlet temperature.

Figure 4.57 shows a heat flow diagram, with stages, where the feed water is preheated to 300◦C.

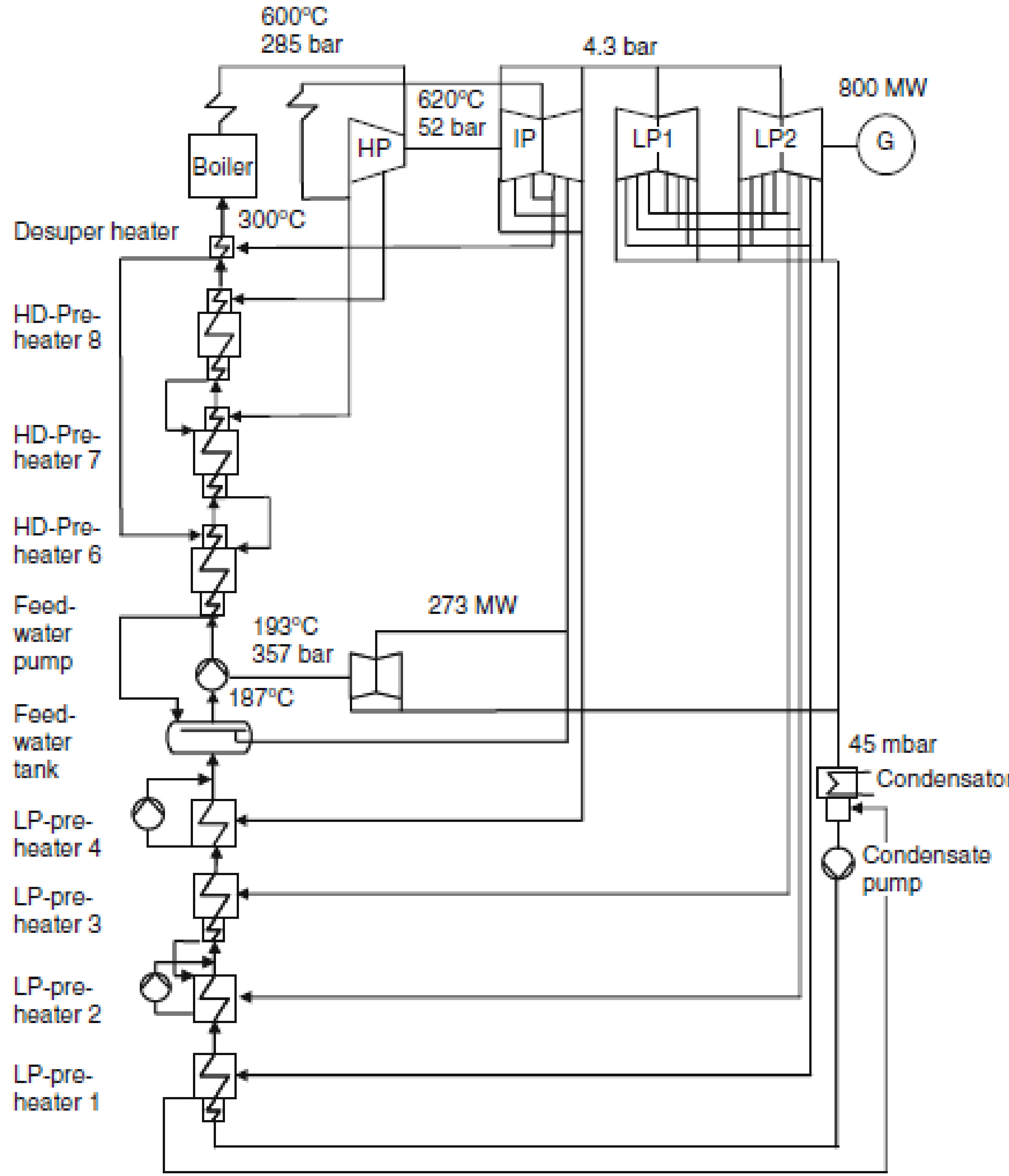

Fig. 4.57 Heat flow diagram of a thermal power plant with advanced steam conditions and ninestage feed water heating (data from Tremmel and Hartmann 2004)

4.4.1.2 Influence of Feed Water Preheating

The rise of the feed water outlet temperature comes up against limiting factors with regard to the steam generator design. It is imperative to prevent boiling in the economiser in order to avoid flow instabilities and to ensure a steady charge of the evaporator tubes. For this reason, the economiser must be designed to be smaller for higher outlet temperatures of the regenerative feed water heating.

Increasing feed water temperatures entering the steam generator make the transferable flue gas heat in the economiser decrease, which can then be used only to preheat the combustion air.

Figure 4.58 shows the impact of an increase in the feed water temperature – a relative decrease of the heat rate, which is dependent on the pressure level (Klebes 2007).

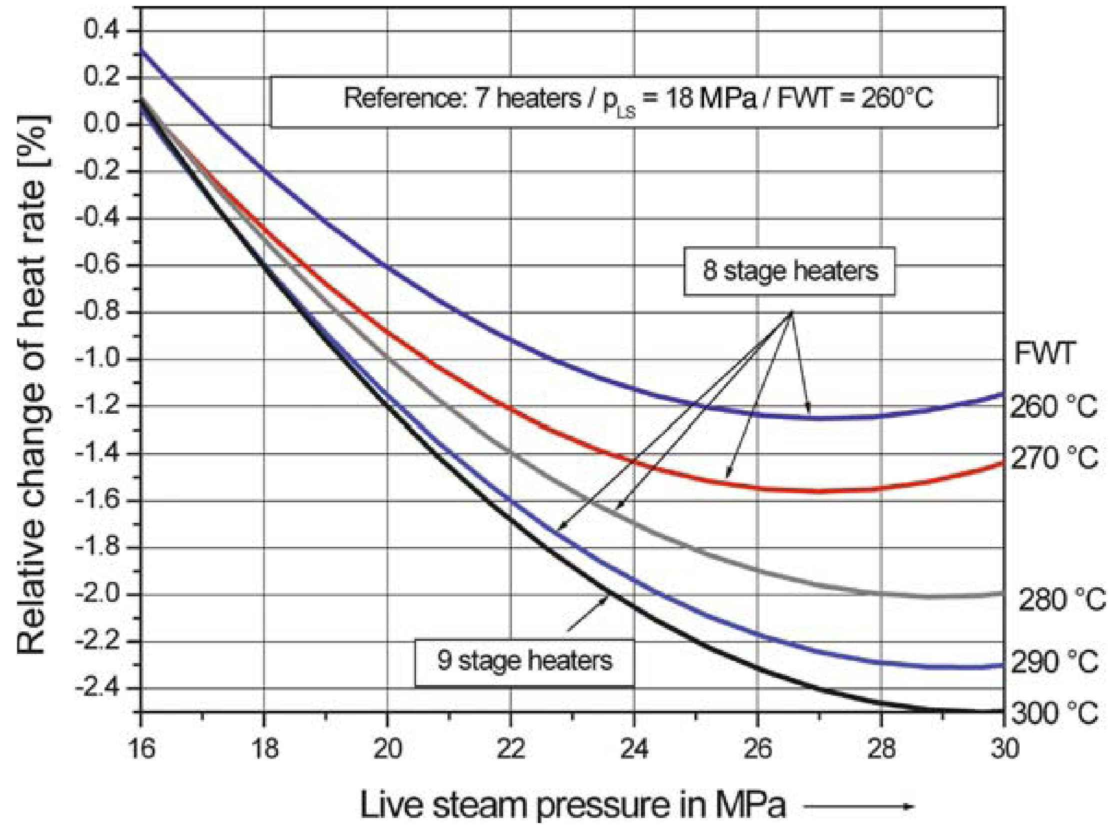

Fig. 4.58 Effect of the live steam pressure and the feed water temperature on the heat rate (Klebes 2007)

4.4.1.2 Influence of Feed Water Preheating

In designing a power plant, after the feed water heating outlet temperature is defined, further optimisation is only possible within the feed water heating chain.

The design should, in this process, provide for the smallest possible temperature difference between the heating medium. By increasing the number of heaters while keeping the same outlet temperature, smaller temperature rises for the individual stages result. This helps to achieve a better adaptation of the temperatures of the heat-absorbing to the heatdissipating heat transfer medium – water flow and extraction steam flows – and thus to minimise the exergy losses. The improvement in efficiency of each additional stage, as shown in Fig. 4.59, is positive but decreasing, so that a point is reached where installation of yet another stage cannot be justified economically (Eichholtz et al. 1994).

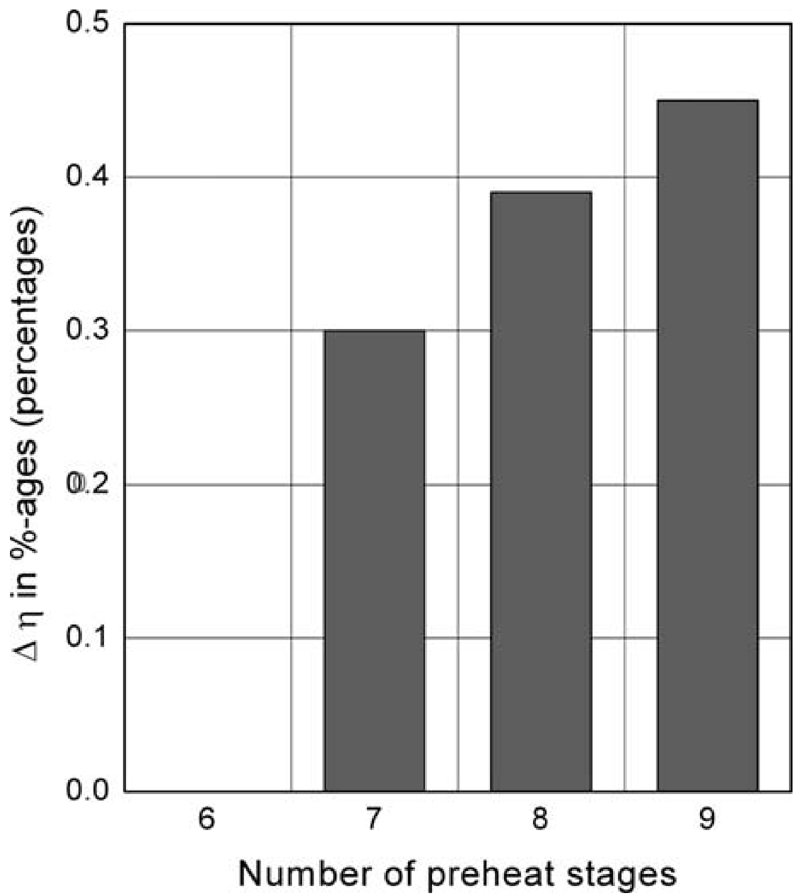

Fig. 4.59 Influence of the number of stages on the net efficiency, at constant outlet temperature (Eichholtz et al. 1994)

Coal Quality Evaluation System (C-Quens) for Pulverized Coal Fired Power Plant

December, 2012

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Gifu University
Graduate School of Engineering
Environmental and Renewable Energy Division

INTRODUCTION

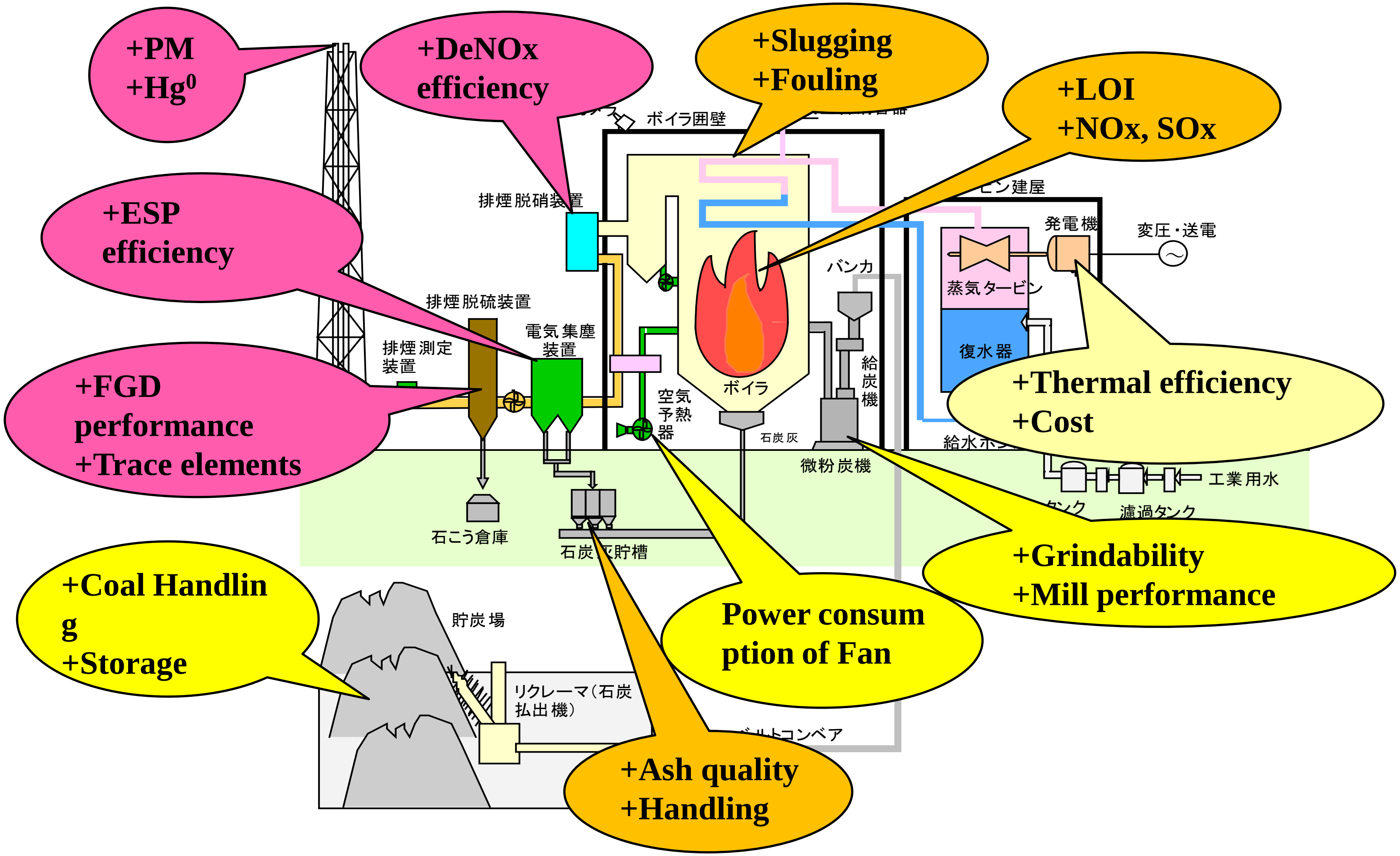
1. Coal quality directly influences plant performance and power generation costs.

2. To generate a stable electricity supply at a minimum cost, it is important to accurately gauge the effect of coal quality on the performance of pulverized coal fired power plants.

As the solution

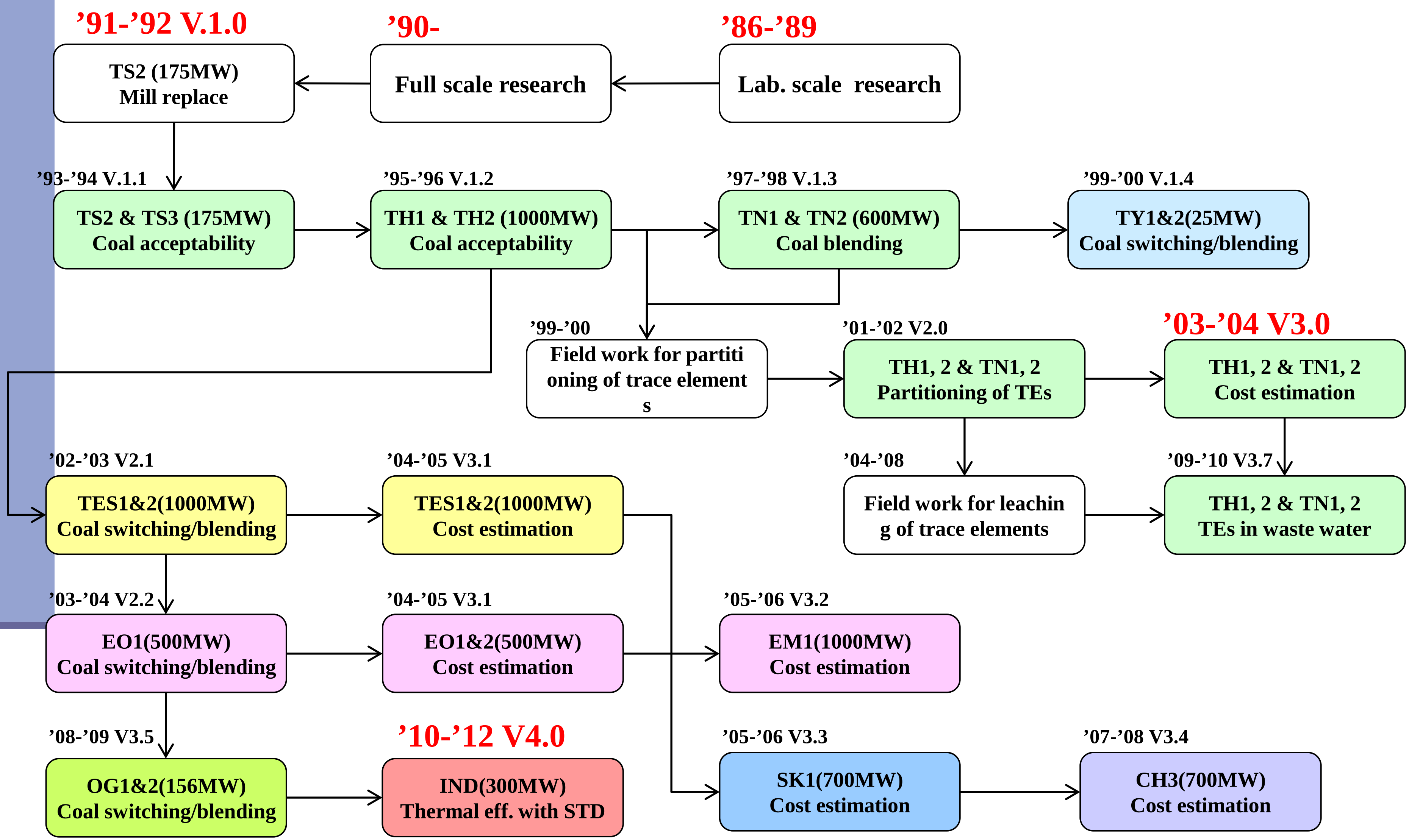
"*Coal Quality Evaluation System* (C-Quens)" can quantitatively estimate the impact of coal quality on the performance of power plants by the coal properties and the plant specifications.

Effects of Coal Quality on Power Plants



The C-Quens is the comprehensive model that can evaluate the influence on the entire pulverized coal fired power station.

History of C-Quens Development



Input and Output

Input data

Coal Analysis Data
- Proximate analysis
- Ultimate analysis
- HGI
- Ash composition

Plant conditions
- Boiler specification
- Operation conditions
- Plant components
- Baseline data

Coal Impact Models

1. Boiler efficiency
2. Heat loss
3. Auxiliary power consumpt.
   Fans, Mills, ESPs, etc.
4. Carbon burnout
5. Environmental performance
   NOx/SOx/CO_2/Hg
6. Ash composition
7. Chemical consumption
8. Thermal efficiency
9. Coal acceptability
10. Leaching of TEs

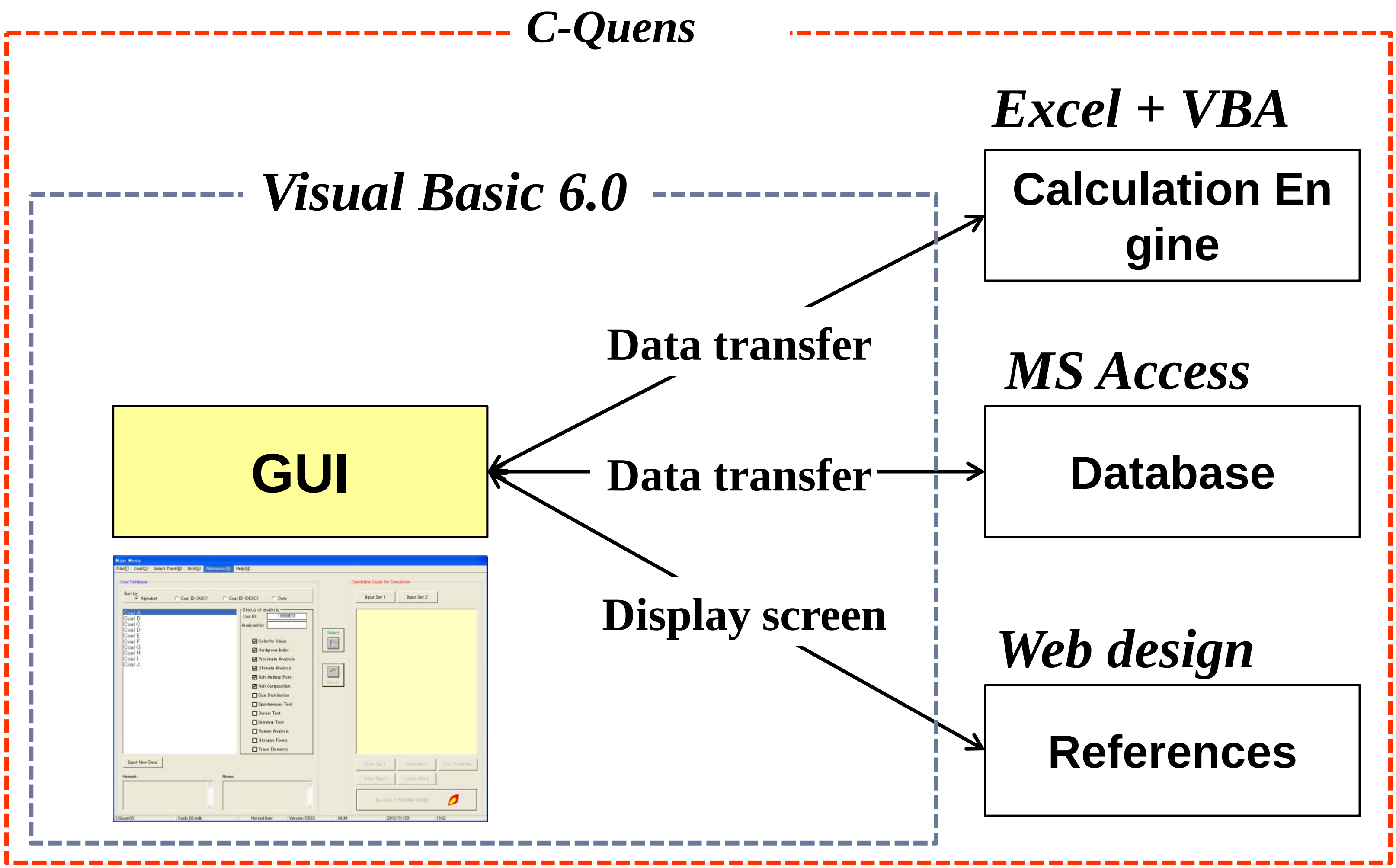
OUTPUT 1 — Plant Performance

OUTPUT 2 — Coal Acceptability

OUTPUT 3 — Cost Estimation

OUTPUT 4 — Coal Blending

Overall representation of the C-Quens



Calculation methodologies are “No black box”!

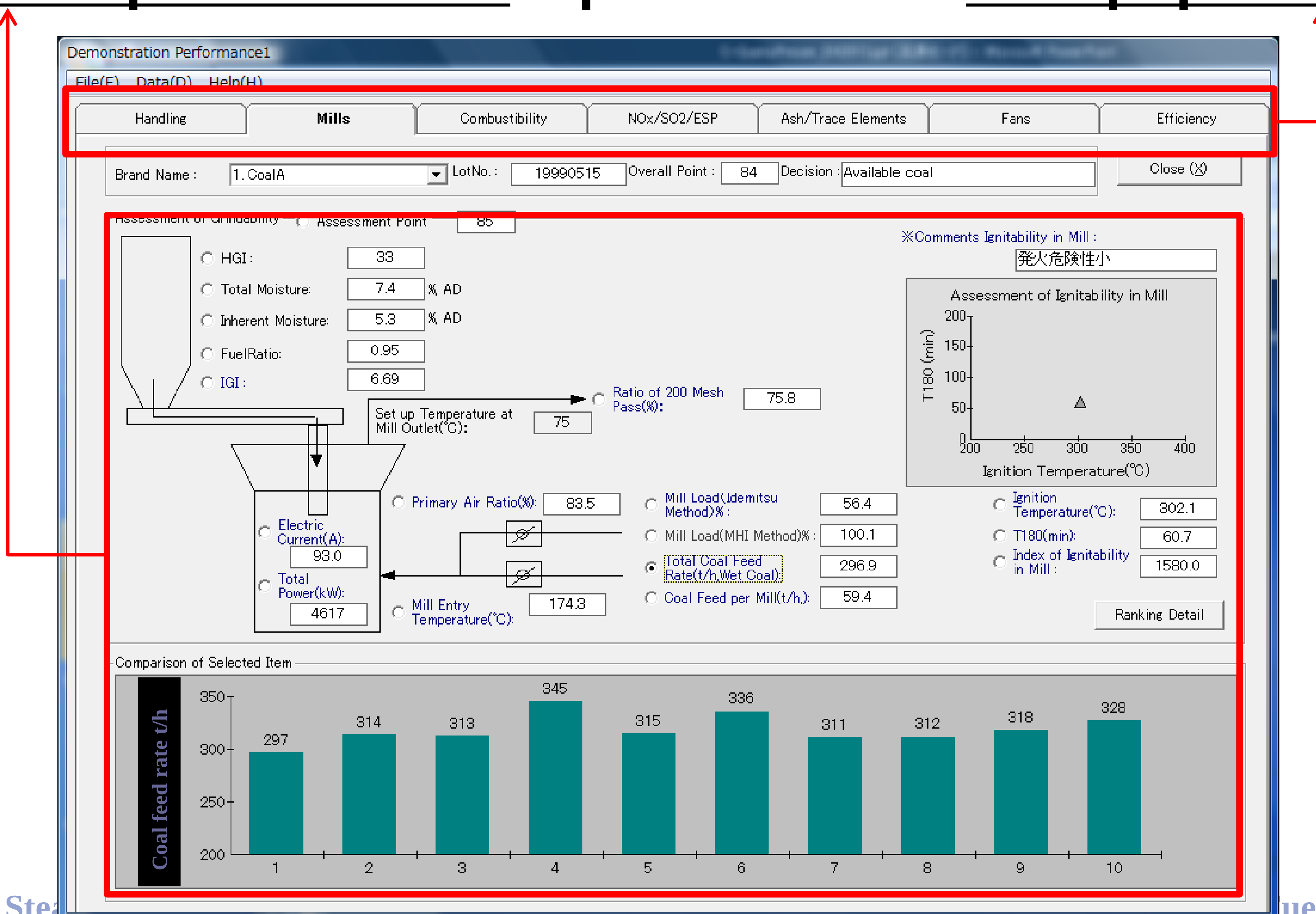
Coal Impact Models

Simple and accurately indices were found by coal research, which are used in estimation of process performances.

INDEX for estimation	Related coal properties
Coal handling	Moisture, Ash, Particle diameter
Spontaneous combustion	Moisture, Oxygen, Heat value
Grinding	C, HGI
Carbon burnout	H/C, O/C, Heat value
NOx emission	N, O/C, Volatile matter at T_{eq}
Ash deposition	Ash composition and combustion performances
Hg emission	Hg, carbon burnout

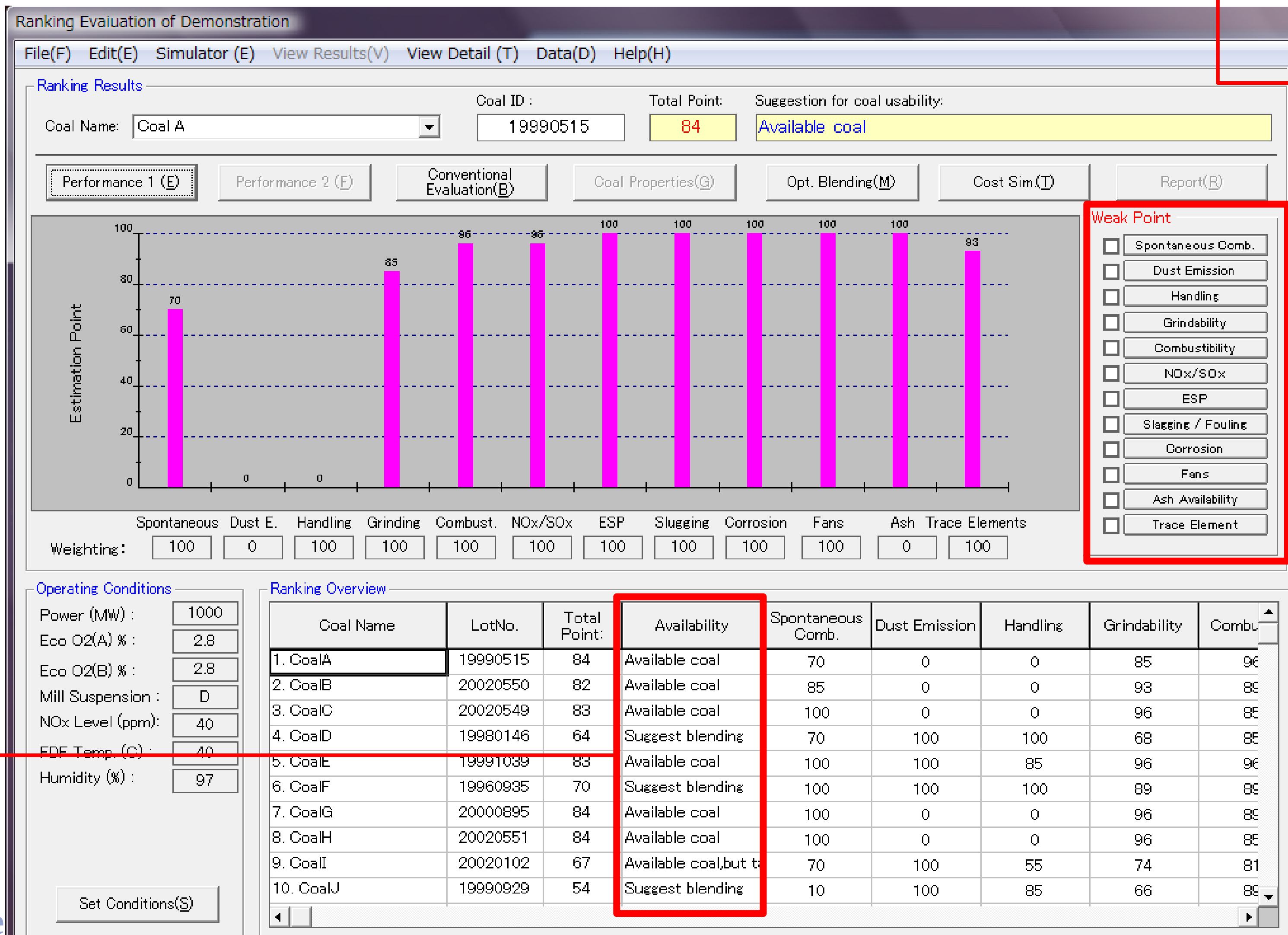
Output 1 –Plant Performances

Plant performance is predicted for all equipments.



Output 2 –Coal Acceptability

<u>The acceptability</u> are evaluated by <u>12 performance</u>.



Output 3 −Cost simulation

Power generation costs (JPY/kWh) are predicted for any coal

Cost Estimation for DEMO version

File(F) Help(H)

Coal : 1. CoalA ID No : 19990515 Overall Point : 84 Decision Available coal Bar Graph (D) Close(X)

Baseline Data

Unit: 千円 Term: 年間 Operating Ratio: 81.4

(A)	Gross Erectricity	MWh	7130640		
	Auxiliary Erectricity		356700.9962		
	Net Erectricity		6773939.004		
	Auxiliary electricity ratio	%	5		
	Operating Ratio		81.4		
	Gross thermal efficiency		43.17869968	Cost Estimation	(¥/kWh)
	Net thermal efficiency		41.01976469	Gross	Net
(B)	Cost 1	¥	499600	0.0701	0.0738
	Cost 2		2656000	0.3725	0.3921
	Cost 3		16280750	2.2832	2.4034
	Cost 4		12748450	1.7878	1.882
	Cost 5		766045	0.1074	0.1131
	Tax 1		597909.7241	0.0839	0.0883
	Tax 2		11222052.93	1.5738	1.6567
	Tax 3		1222248.046	0.1714	0.1804
	Total	¥	44770807.65	6.4501	6.7898

List of Cost Estimation

Sort(S)

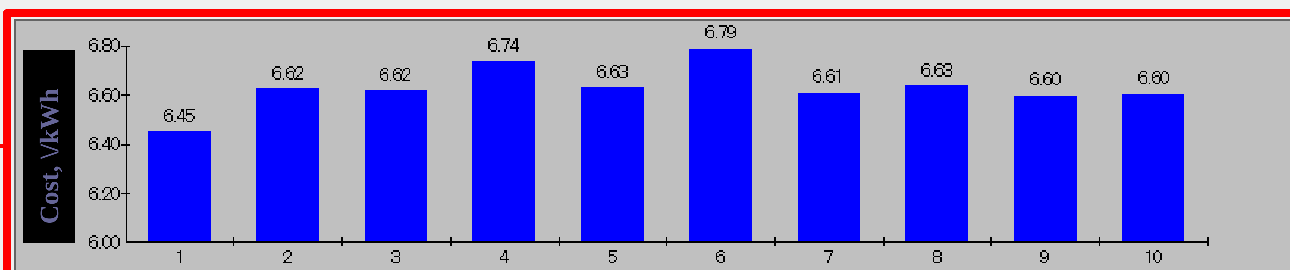
	Coal Name	Cost (¥/kWh)	Fuel Cost(¥)	Running Cost (¥)
1	1. CoalA	6.4501	766045	16280750
2	9. CoalI	6.596	766045	16280750
3	10. CoalJ	6.6035	766045	16280750
4	7. CoalG	6.607	766045	16280750
5	3. CoalC	6.6176	766045	16280750
6	2. CoalB	6.6217	766045	16280750
7	5. CoalE	6.6285	766045	16280750
8	8. CoalH	6.6343	766045	16280750
9	4. CoalD	6.7379	766045	16280750
10	6. CoalF	6.7879	766045	16280750
11				
12				
13				
14				
15				
16				
17				
18				
19				
20				

Bar Graph

Operation Cost

Fuel Cost

Erectricity Cost (Net)



Cost Merit by Coal Switching

The difference in cost of coal F and A is 0.34 JPY/kWh. Consequently, the difference in power generation cost per month between the two coals is

(0.34 JPY/kWh) · (10^6 kWh) ·(24 h) · (30 days) · (0.80) = 244,800,000 JPY/M,

where 0.80 is a utilization rate of the full load (1000 MW).

Cost change by coal switching is high enough.

Output 4 –Design of Coal Blending

Optimum blending ratios are suggested.

Demonstration Table of Combinations of Base coal and Blending coals

File(F) Edit(E) Data(D)

This table reasonable blending ratios. "○"means a good combination at any blending ratio. "△" means fair caombination with limited blending ratios. "×"means bad combination.
Press one button to see the details in the blend performances.

Close (X)

More number of ○ and △ means a better coal for fitness.

	CoalA	CoalB	CoalC	CoalD	CoalE	CoalF	CoalG	CoalH	CoalI		Good Coals	Moderate Coals	Total
CoalA	○	○	○	○	○	○	○	○	○	–	9	0	9
CoalB	○	○	○	○	○	○	○	○	○	–	9	0	9
CoalC	○	○	○	○	○	○	○	○	○	–	9	0	9
CoalD	○	○	○	×	○	○	○	○	○	–	8	0	8
CoalE	○	○	○	○	○	○	○	○	○	–	9	0	9
CoalF	○	○	○	○	○	×	○	○	○	–	8	0	8
CoalG	○	○	○	○	○	○	○	○	○	–	9	0	9
CoalH	○	○	○	○	○	○	○	○	○	–	9	0	9
CoalI	○	○	○	○	○	○	○	○	△	–	8	1	9
	–	–	–	–	–	–	–	–	–	–	0	0	0

Recent Application

The C-Quens has evolved to meet the ever changing needs of coal users, providers, and constructors.

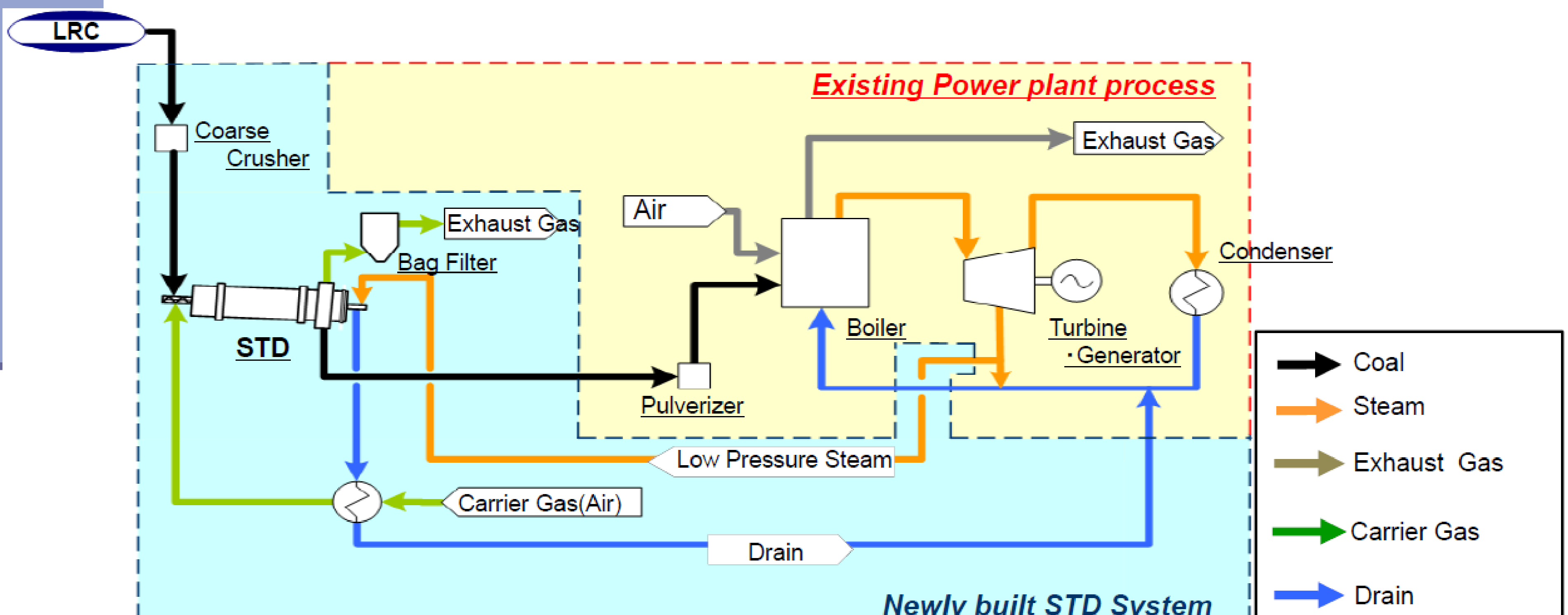
Recently, for a constructor, *the advantage of steam tube dryers (STDs)* was demonstrated in a 315 MWe plant using low rank coal in Indonesia.

For another coal user, a leaching model for trace elements was developed to manage the quality of waste water coming from landfill.

Feasibility studies for STD

STDs installation to PC plants using low rank coal are expected:

- **to improve thermal efficiency**
- **to reduce mills and fans power consumptions**
- **to reduce CO_2 emission.**

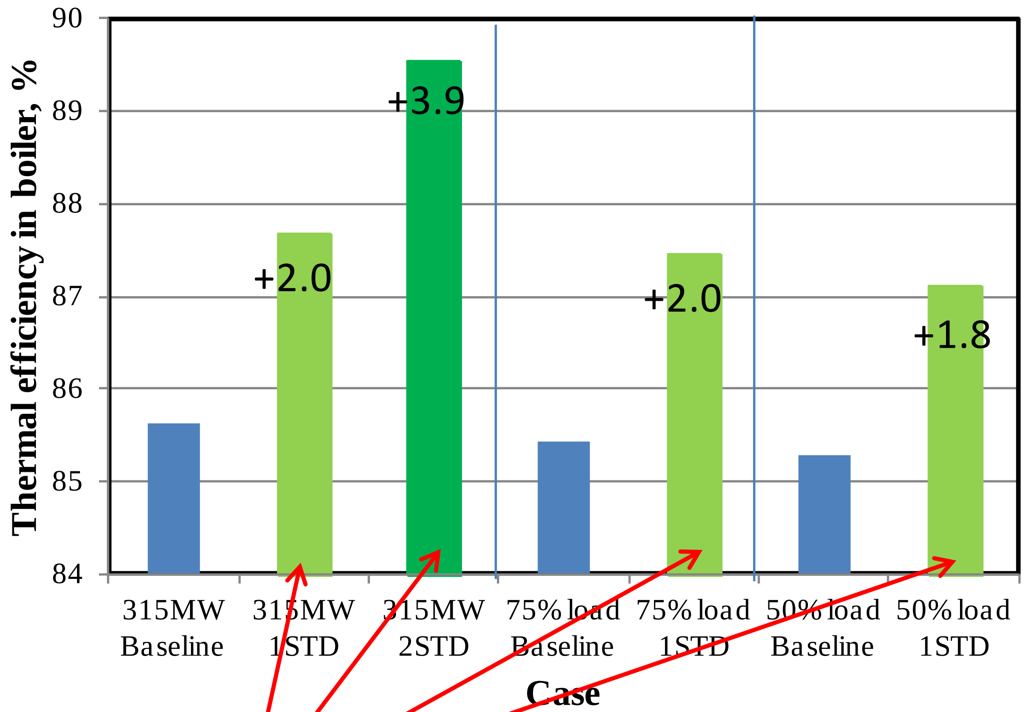


Simulation Conditions

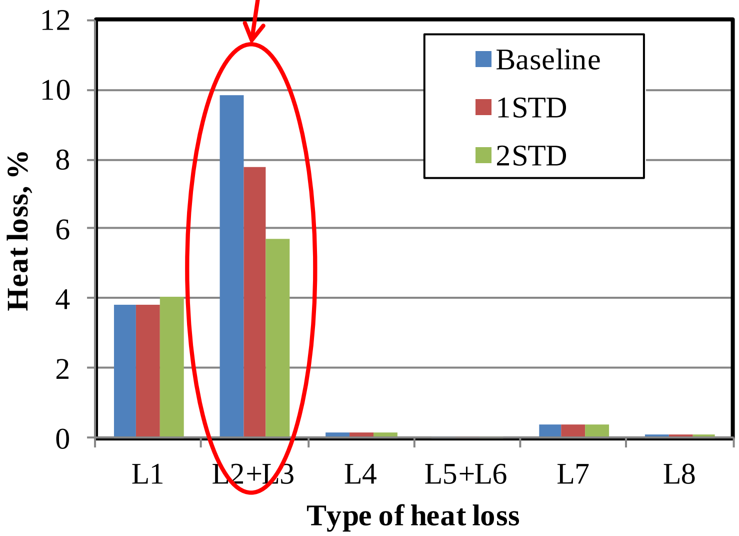
- ☐ **Fixed net electric power (302, 214, 142 MW)**
- ☐ **One or two STD units**

	Unit	Baseline	1 unit STD	2 unit STD	75% load 1 STD	50% load 1 STD
Net power	MW	302.1	302.1	302.1	214.2	141.7
Total moist.	%	35.0	25.0	11.4	25.0	25.0
Steam temp.	℃	–	342	342	340	322
Steam flow	t/h	–	30.8	60.4	22.0	15.5
Steam press.	MPa	–	0.862	0.862	0.619	0.438
Steam temp. at STD exit	℃	–	172.2	172.2	158.8	145.7

Improvement of Boiler Efficiency

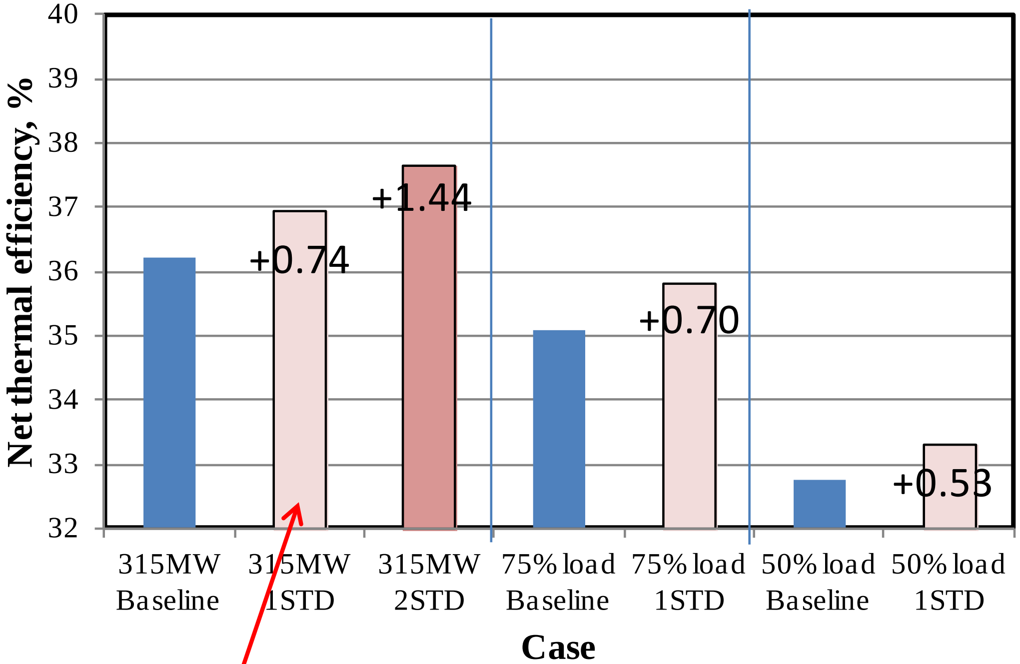


Heat loss by H_2O is a key parameter to increase boiler efficiency.

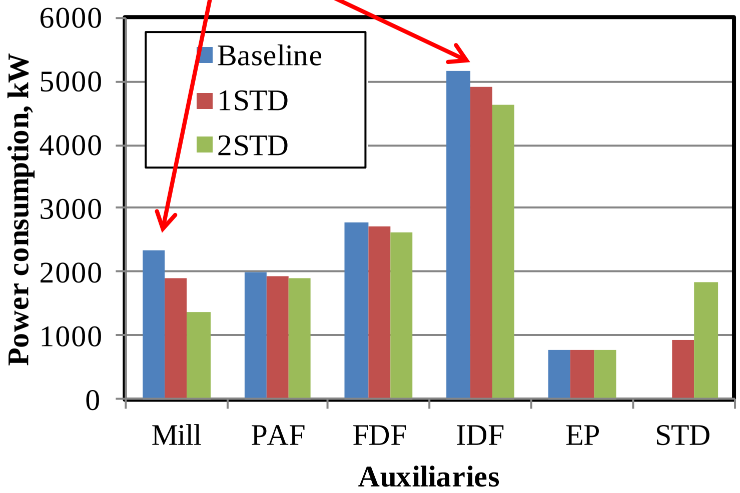


STDs are increased boiler efficiency in any cases.

Improvement of Net Thermal Efficiency

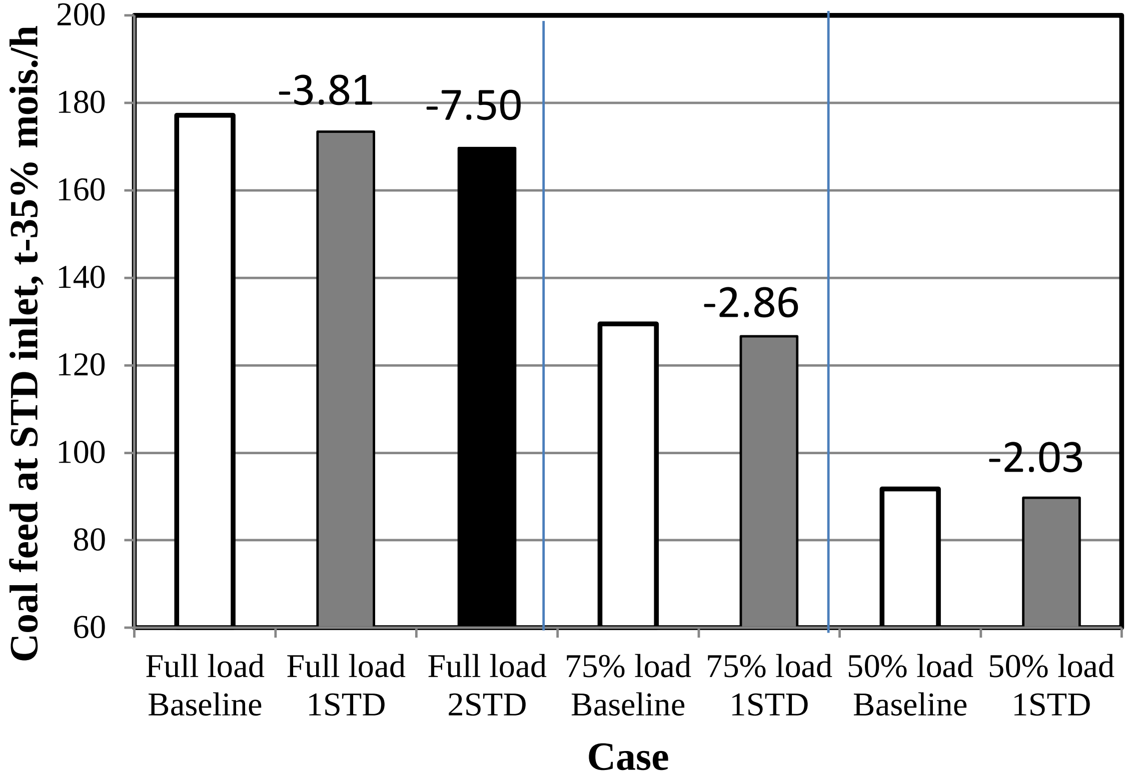


Decrease in power consumption of mills and IDF fans is strongly contributed to improve net thermal efficiency.



Net thermal efficiency of +0.74 point is estimated by STD installation.

Saving of fuel consumption



In a 315 MW power plant, 3.81 t-coal/h can be save by 1 unit STD addition, which is corresponded to 26,508 t-coal/year.

Summary

- The C-Quens was developed to quantitatively estimate the impact of coal quality on pulverized coal fired power plants.
- The software package has various functions including evaluation of coal acceptability, estimation of plant performance, cost estimation of power ge neration, and optimum design of coal blending.
- The C-Quens has the empirical equations using original coal indices, whi ch were developed by many years of research on the relation between coal properties and numerous actual power plant data.
- The C-Quens is available for the power stations of electric companies, th e utility divisions of manufacturing companies, coal suppliers, and constru ctors.

Installation Procedure of C-Quens

STEP 1: Process Survey

+Investigation of the process specifications and conditions

+Investigation of actual process data

+Coal analysis

STEP 2: Modeling

+Simulator building of C-Quens

+Verifying between simulation results and actual results

STEP 3: C-Quens installation

+C-Quens installation on user PCs (Windows)

+Feasibility studies based on user needs

+Reporting