Power Plants
A1M15ENY

Lecture No. 8

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Without reheat:

$p_2 = 12$ MPa
$T_2 = 530$ °C

$\Downarrow$

$i_2 = 3429$ kJ.kg^{-1}
$s_2 = 6,591$ kJ.K^{-1}.kg^{-1}

Knowing *s* to the next process:

$s_5 = 6,591$ kJ.K^{-1}.kg^{-1}
$p_5 = 3,5$ kPa

$\Downarrow$

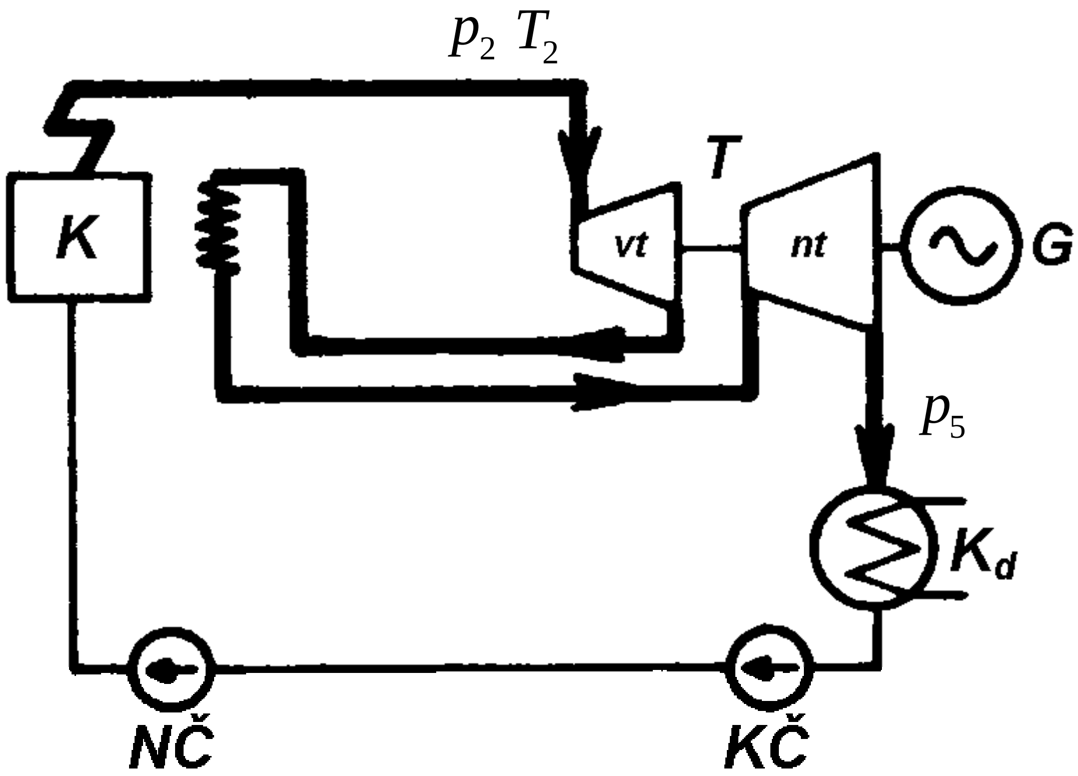
$T_5 = 27$ °C
$i_5 = 1971$ kJ.kg^{-1}

Knowing *T* to the next process:

$T_1 = 27$ °C
$x = 0$ (condensation)

$\Downarrow$

$i_1 = 111,8$ kJ.kg^{-1}
$s_1 = 0,3906$ kJ.K^{-1}.kg^{-1}



Total efficiency is thus:

$$\eta = \frac{i_2 - i_5 - \left(p_2 - p_1\right).v}{i_2 - i_1 - \left(p_2 - p_1\right).v} =$$

$$= \frac{3429 - 1971 - 1,2.10^4.0,001}{3429 - 111,8 - 1,2.10^4.0,001} =$$

$$= \frac{1446}{3305} = 43,7\%$$

With reheat:

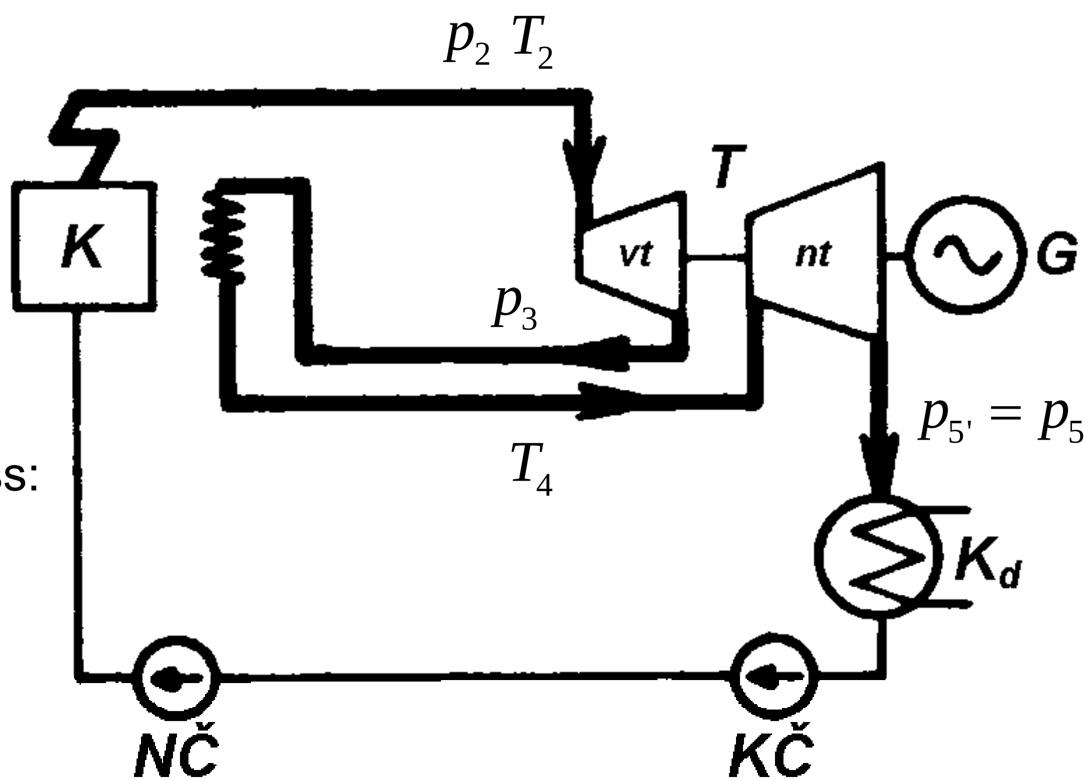
$$p_3 = 2,3 \text{ MPa}$$
$$s_3 = 6,591 \text{ kJ.K}^{-1}\text{.kg}^{-1}$$
$$\Downarrow$$
$$i_3 = 2959 \text{ kJ.kg}^{-1}$$
$$T_3 = 277 \text{ °C}$$

Knowing p,T to the next process:

$$p_4 = 2,3 \text{ MPa}$$
$$T_4 = 480 \text{ °C}$$
$$\Downarrow$$
$$i_4 = 3420 \text{ kJ.kg}^{-1}$$
$$s_4 = 7,308 \text{ kJ.K}^{-1}\text{.kg}^{-1}$$

Knowing *s* to the next process:

$$s_{5'} = 7,308 \text{ kJ.K}^{-1}\text{.kg}^{-1}$$
$$p_{5'} = 3,5 \text{ kPa}$$
$$\Downarrow$$
$$i_{5'} = 2186 \text{ kJ.kg}^{-1}$$
$$T_{5'} = 27 \text{ °C}$$



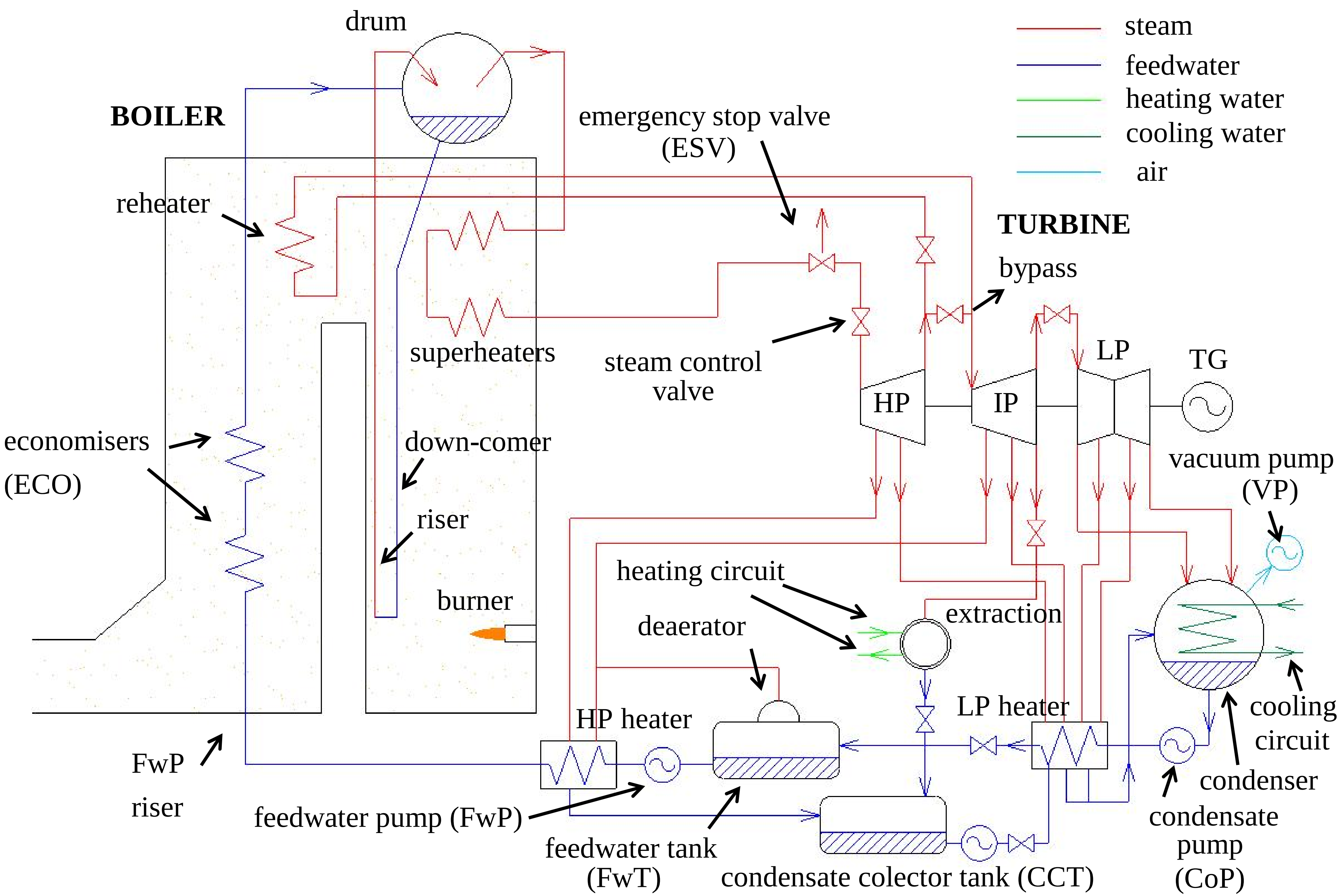
Total efficiency is thus:

$$\eta = \frac{i_2 - i_{5'} + i_4 - i_3 - \left(p_2 - p_1\right).v}{i_2 - i_1 + i_4 - i_3 - \left(p_2 - p_1\right).v} =$$

$$= \frac{3429 - 2186 + 3420 - 2959 - 12}{3429 - 111,8 + 3420 - 2959 - 12} =$$

$$= \frac{1701}{3766} = 45,2\%$$

Steam and Feedwater Circuit



Turbines

By working medium:

- **Gaseous** (gaseous or liquid fuels, in. temperature 600-1400°C, out. temperature 450-600°C)
- **Steam** (superheated steam, in. temperature 400-650°C, out. temperature 28-42°C)
- **For wet steam** (in NPP, in. temperature 300°C, out. temperature as previous)

By output steam pressure:

- **Back pressure turbines** (out. pressure 0,11-0,6 MPa)
- **Condensing turbines** (out. pressure 20-40 kPa)

By steam extraction:

- **Turbine with unregulated extraction** (for feedwater conditioning, gland steam etc.)
- **Turbine with regulated extraction** (instead of above mentioned extractions one or more additional for heating system purposes)

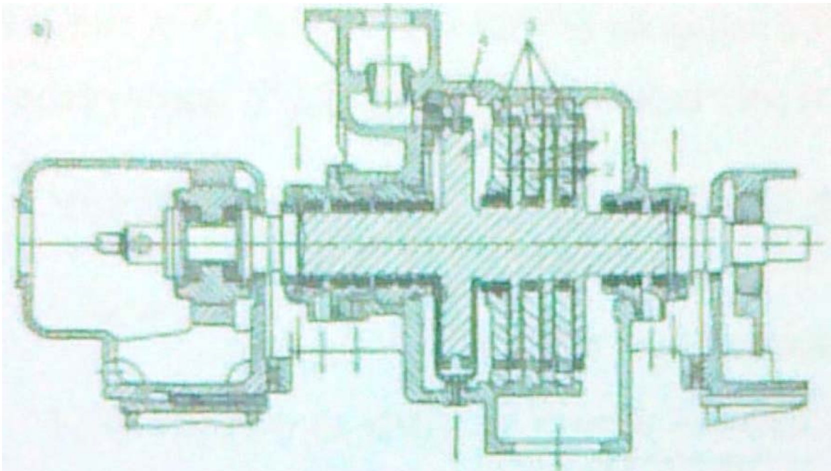
According to number of parts:

- **Single part** (lower power ratings)
- **Multi part** (high p. - HP, intermediate p. - IP, low p. - LP parts)

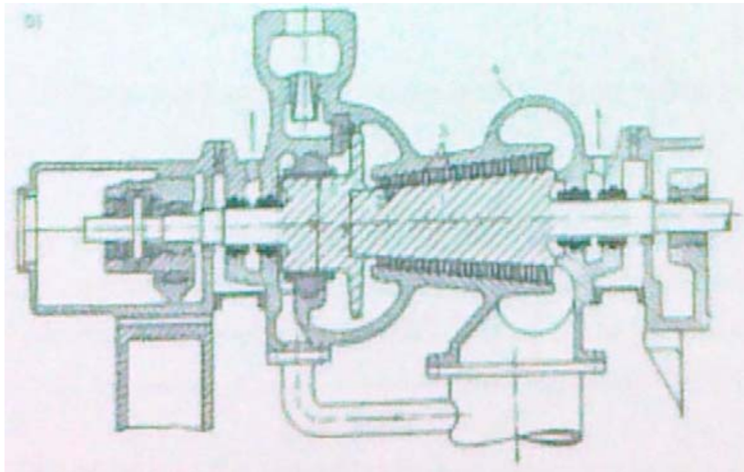
Turbines

<u>By principle of operation:</u>

- **Impulse** (whole enthalpy drop of the stage is totally changed into kinetic energy in the stator nozzles, steam pressure is the same at input and output of stage rotor moving blades)
- **Reaction** (a part of enthalpy drop of the stage is additionally changed into kinetic energy in the rotor moving blades)



Impulse turbine example



Reaction turbine example

Turbines

Functioning:

Each single turbine stage is consisting of:

- **Stationary blades** i.e. solid grid of blades (a part of stator) = a set of parallel nozzles, which are converting steam pressure energy to kinetic energy at minimum losses
- **Rotary blades** i.e. set of blades (a part of rotor), where steam kinetic energy is converted into rotation energy of turbine body

Velocity of the steam leaving the nozzle:

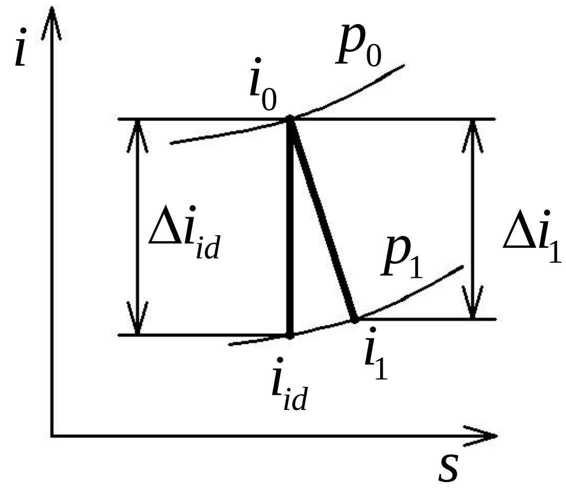
Generally:

$$dq = di + da + d\left(\tfrac{1}{2}c^2\right)$$

For adiabatic expansion:

$$0 = di + d\left(\tfrac{1}{2}c^2\right) \quad \text{a} \quad c_1 = \sqrt{c_0^2 + 2.\left(i_0 - i_1\right)}$$

Real process is not exactly adiabatic, entropy is rising (edge losses, friction losses on stationary and rotary blades, flow direction change etc.)



Turbines

Output flow velocity c_0=0, put into equation of state:

$$c_1 = \sqrt{2.(i_0 - i_1)} = \sqrt{2.c_P.(T_0 - T_1)} = \sqrt{\frac{2\cdot\kappa}{\kappa - 1}.r.T_0.\left(1 - \frac{T_1}{T_0}\right)} =$$

$$= \sqrt{\frac{2\cdot\kappa}{\kappa - 1}\cdot p_0 \cdot v_0 \cdot \left[1 - \left(\frac{p_1}{p_0}\right)^{\frac{\kappa-1}{\kappa}}\right]}$$

Saint Vénant-Wantzel equation

Maximum flow velocity of ideal gas is thus into vacuum p_1=0:

$$c_{1\max} = \sqrt{\frac{2\cdot\kappa}{\kappa - 1}\cdot p_0 \cdot v_0} = \sqrt{\frac{2\cdot\kappa}{\kappa - 1}.r.T_0}$$

Pressure ratio: $\beta = \frac{p_1}{p_0}$

If we define mass flow density as:

$$\frac{\dot{m}}{A} = \rho.c_1 = \frac{1}{v_0}\cdot\beta^{\frac{1}{\kappa}}\cdot\sqrt{\frac{2\cdot\kappa}{\kappa - 1}\cdot p_0 \cdot v_0 \cdot \left[1 - \beta^{\frac{\kappa-1}{\kappa}}\right]}$$

[kg.s^{-1}.m^{-2}]

Turbines

Maximum mass flow density occurs at

$$\beta_k = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa\text{-}1}} = \frac{p_k}{p_0} \rightarrow \text{critical pressure}$$

Mass flow density is not any more growing along with further output pressure decrease.

Other critical parameters:

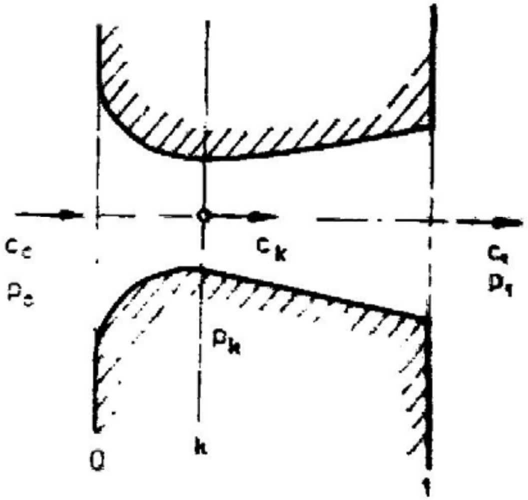
$$T_k = T_0 \cdot \frac{2}{\kappa+1} \qquad c_k = \sqrt{\frac{2\cdot\kappa}{\kappa+1}\cdot p_0 \cdot v_0} = \sqrt{\frac{2\cdot\kappa}{\kappa+1}\cdot r \cdot T_0}$$

Speed of sound for ideal gas:

$$a = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} = \sqrt{\kappa \cdot r \cdot T}$$ and thus $$a\big|_{T=T_k} = \sqrt{\kappa\cdot r\cdot T_k} = \sqrt{\kappa \cdot p_0 \cdot v_0 \cdot \beta_k^{\frac{\kappa-1}{\kappa}}} = c_k$$

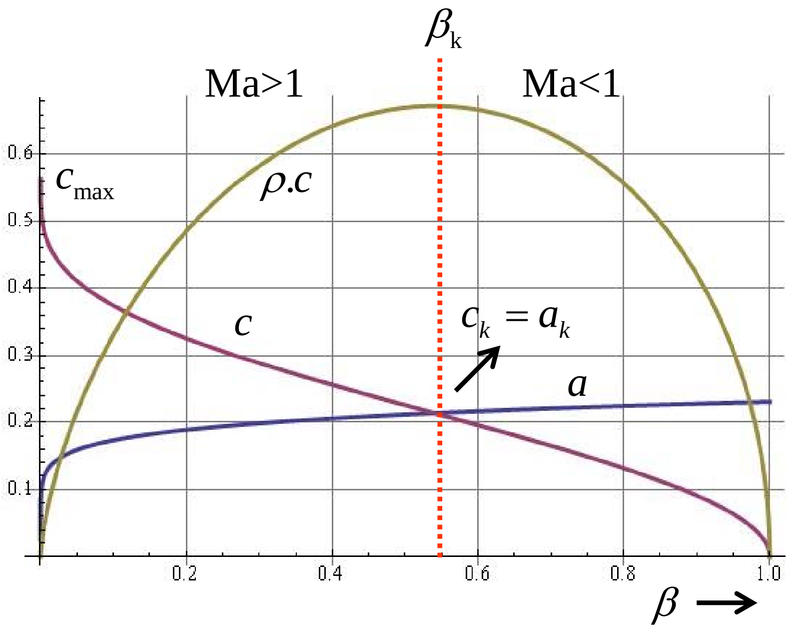
In a convergent nozzle, there is no outlet velocity rise after exceeding critical parameters – choked flow. A part of pressure energy is converted to whirling
=> to be able to rise the velocity we must use convergent-divergent *de Laval nozzle*.

Mach number: $\mathrm{Ma} = \frac{c}{a}$ → Ma>1 supersonic flow; ← Ma<1 subsonic flow



De Laval nozzle

Turbines



Values βk:

β_k	Plyn
0,487	ideal single-atom gas
0,528	ideal two-atom gas
0,540	ideal three-atom gas
0,53	air
0,55	superheated steam
0,58	saturated steam

In real nozzle we must take into account friction and whirling losses, so the output velocity is lower than adiabatic, which can be expressed by factor:

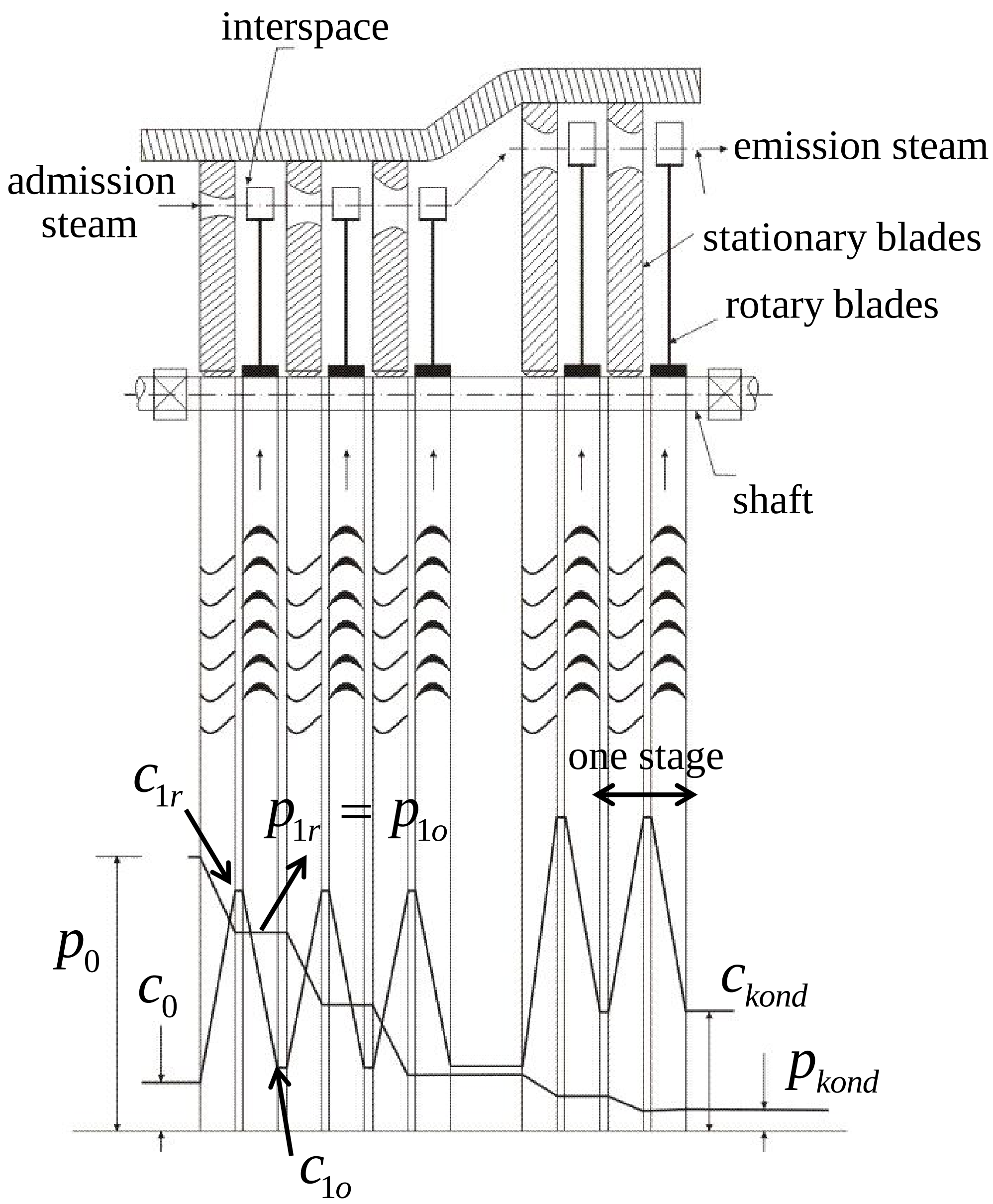
$$\varphi = \frac{c_1}{c_{id}}$$

and nozzle efficiency:

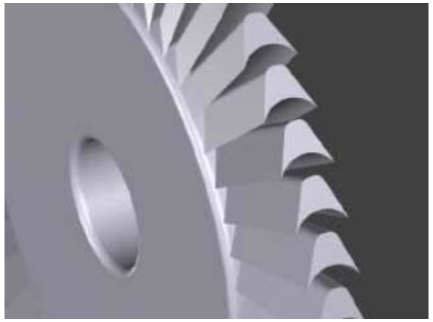
$$\eta = \varphi^2 = \frac{i_0 - i_1}{i_0 - i_{id}}$$

Turbines

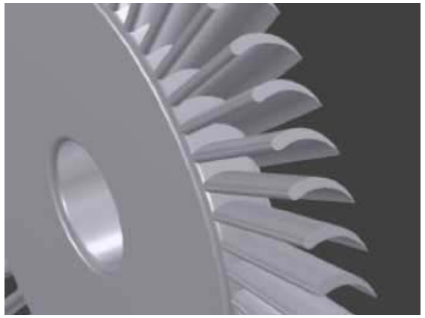
Impulse turbine:



Blades:

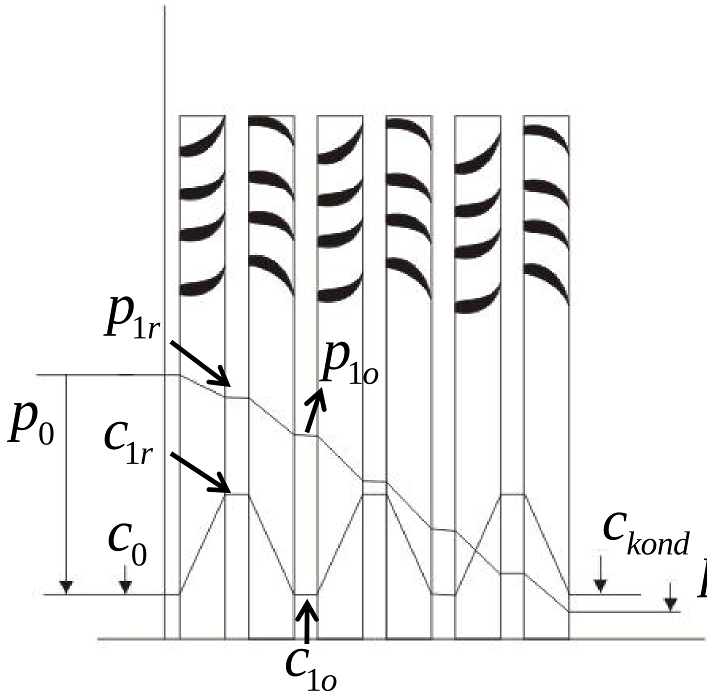


impulse turbine



reaction turbine

Reaction turbine:



Turbines

Degree of reaction:

In the case of reaction turbine:

$$R = \frac{i_0 - i_{1r}}{i_0 - i_1} = \frac{isentropic\ heat\ drop\ in\ rotor}{isentropic\ heat\ drop\ in\ stage}$$

Shaft torque and stage power output:

Working medium leaving the nozzles of stationary blades at velocity c_{1r} and entering the rotary blades, where the kinetic energy is converted to the shaft torque. Thus force arising at the perimeter of wheel with rotary blades:

$$F_o = \dot{m}.\left(c_{1r} - c_{1o}\right)$$

and the shaft torque:

$$M_o = F_o, r = \dot{m}.\left(c_{1r} - c_{1o}\right).r$$

stage power output:

$$P_o = M_o.\omega = \dot{m}.\left(c_{1r}.u_{o1} - c_{1o}.u_{o2}\right)$$

where u_{o1}, u_{o2} are corresponding perimeter velocities

Turbine Oil and Gland Steam System

<u>Oil:</u>

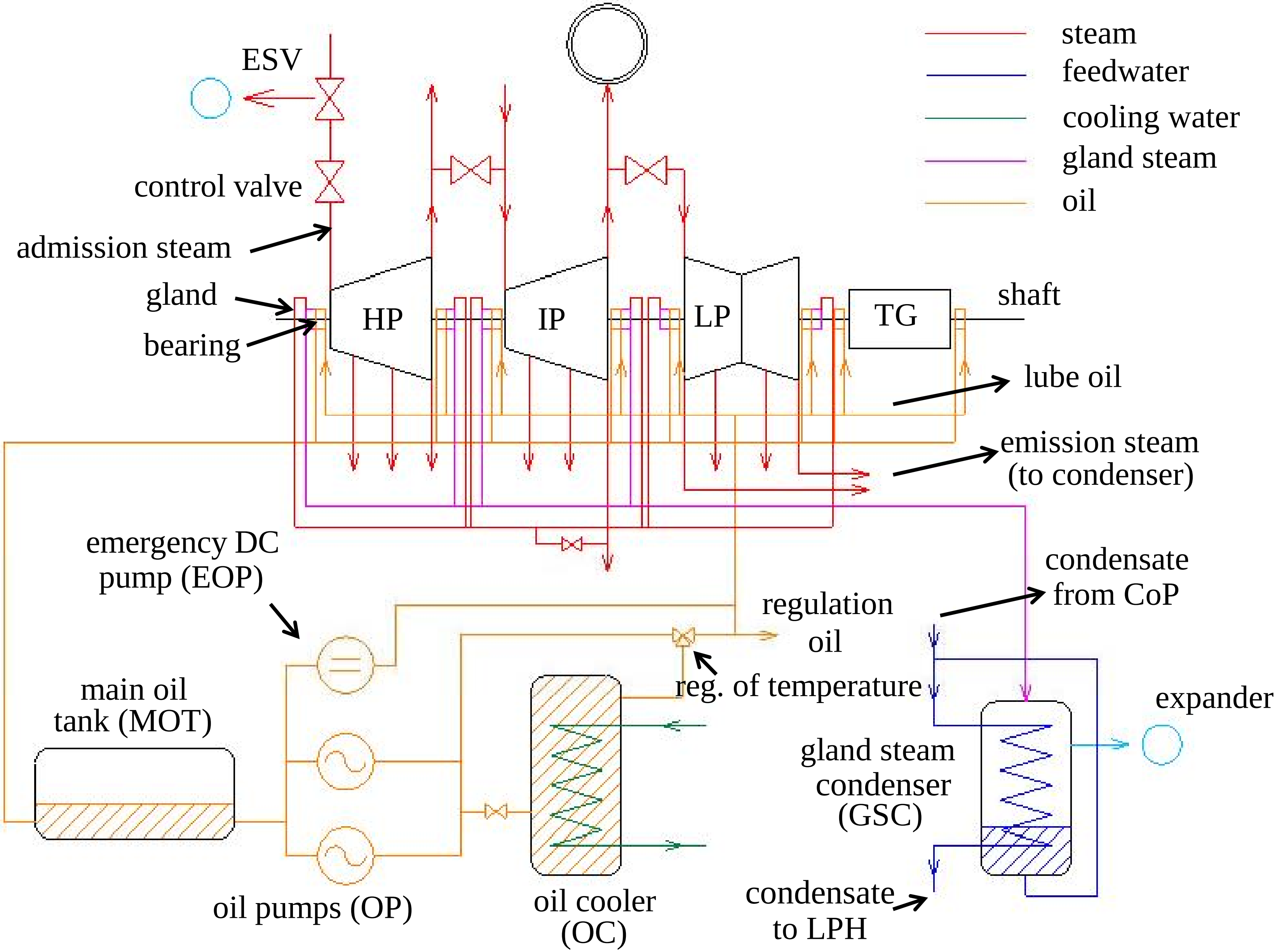
Two basic functions:

- **Lube oil** is used for machine bearing cooling and lubrication
- **Regulation oil** is used in turbine electro-hydraulic (EH) regulation systems – regulation quantity is oil pressure
  - Primary oil – turbine speed
  - Secondary oil – valve opening position
  - Oil for emergency stop valve
  - Regulating oil – turbine elements control

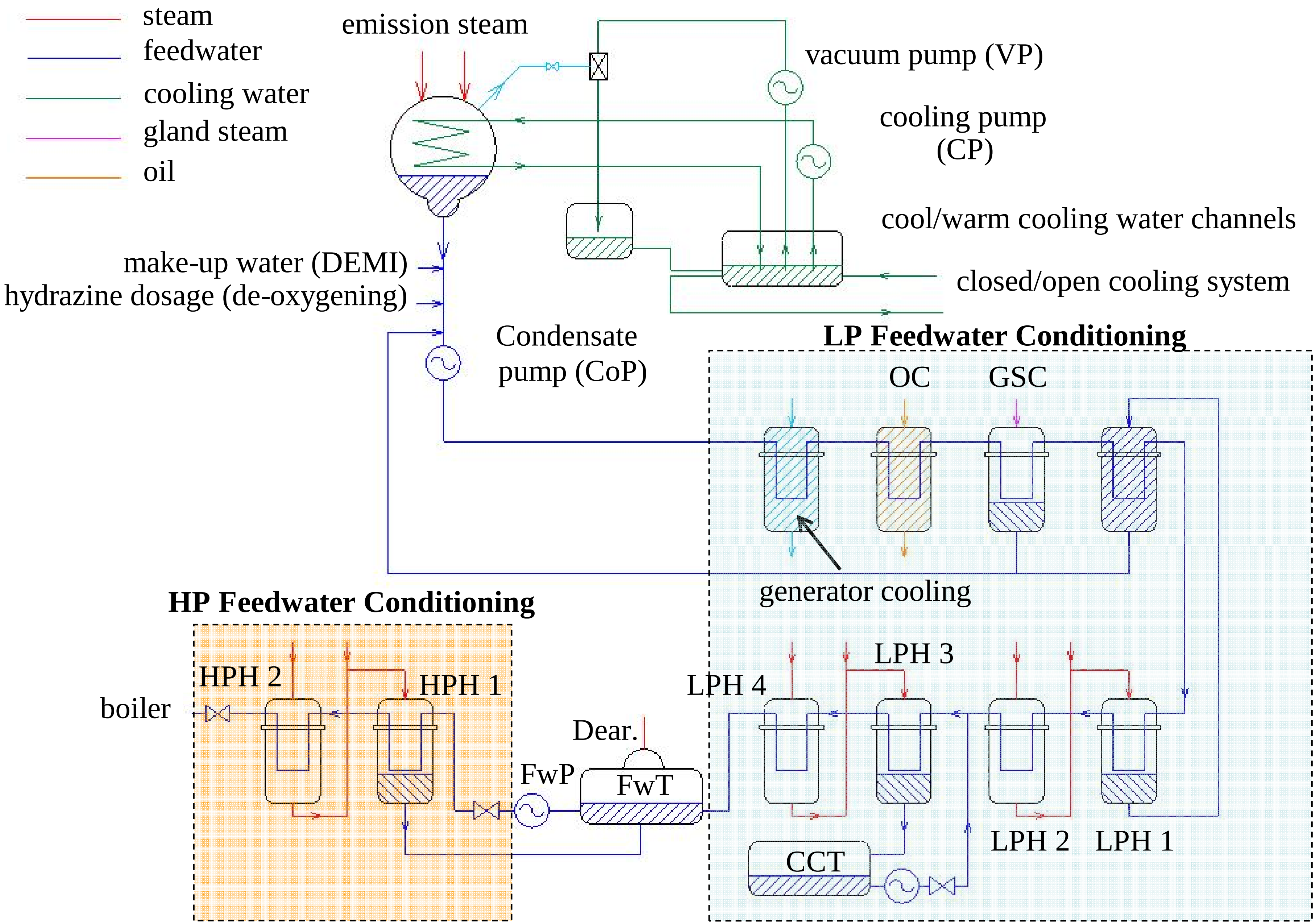
<u>Gland steam:</u>

- Turbine cannot be ideally sealed. The air is getting into the underpressure parts of turbine, unlike the steam is leaking from the overpressure parts
- To avoid this effect every part of machine set is equipped with labyrinth seals supplied with IP gland steam (circa ~2 MPa)
- Gland steam gets into (or out from) the labyrinth steam trap to slow down the leakage in the space between rotor and stator

Turbine Oil and Gland Steam System



Condensation and Water Conditioning



Boilers

By fuel type:

- **Solid** (coal, coke, biomass, waste)
- **Gas** (NG, CNG, LPG)
- **Liquid** (LFO, HFO)

By output steam pressure:

- **Low pressure** (pressure up to1,6 MPa)
- **Intermediate pressure** (pressure 1,6-5 MPa)
- **High pressure** (pressure 5-13 MPa)
- **Extra high pressure** (pressure 13-22,5 MPa)
- **Supercritical** (pressure over 22,5 MPa)

By evaporator type:

- **Boilers with natural circulation**
- **Boilers with forced circulation** (La Mont)
- **Drumless** (supercritical - Sulzer, Benson)

} boilers with low water content

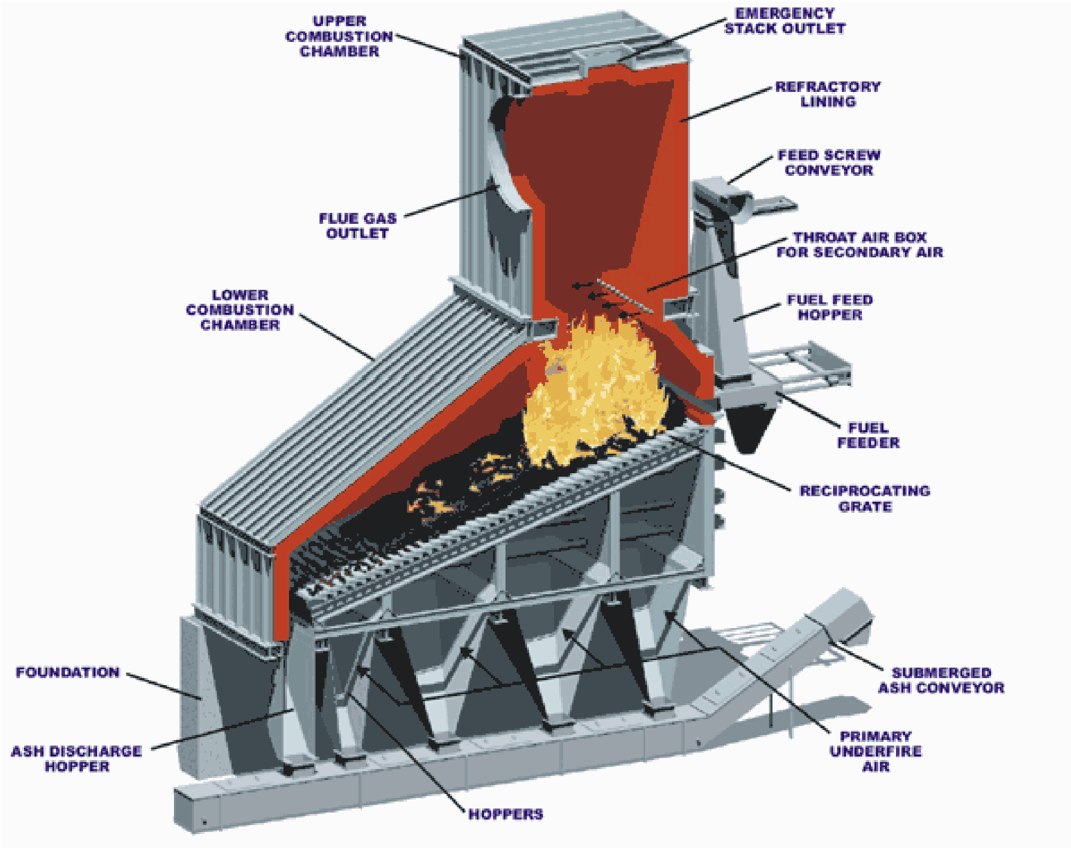
- **Shell boilers** (heat transfer through corrugated iron)
- **Fire tube boilers** (flue gases are flowing inside tubes)

} boilers with high water content

Boilers

By combustion device principle:

- **Grate** (lower power outputs)
- **Pulverized** (dry bottom – evaporators on walls, wet bottom – evaporators on walls + bottom)
- **Fluidized bed** (less sensitive to fuel change, desulphurization inside the boiler)



Main parts:

- The walls are surrounding the combustion chamber
- Grate with fuel feed hopper, barrier, slag weir and ash hoppers
- Primary air inlet from below

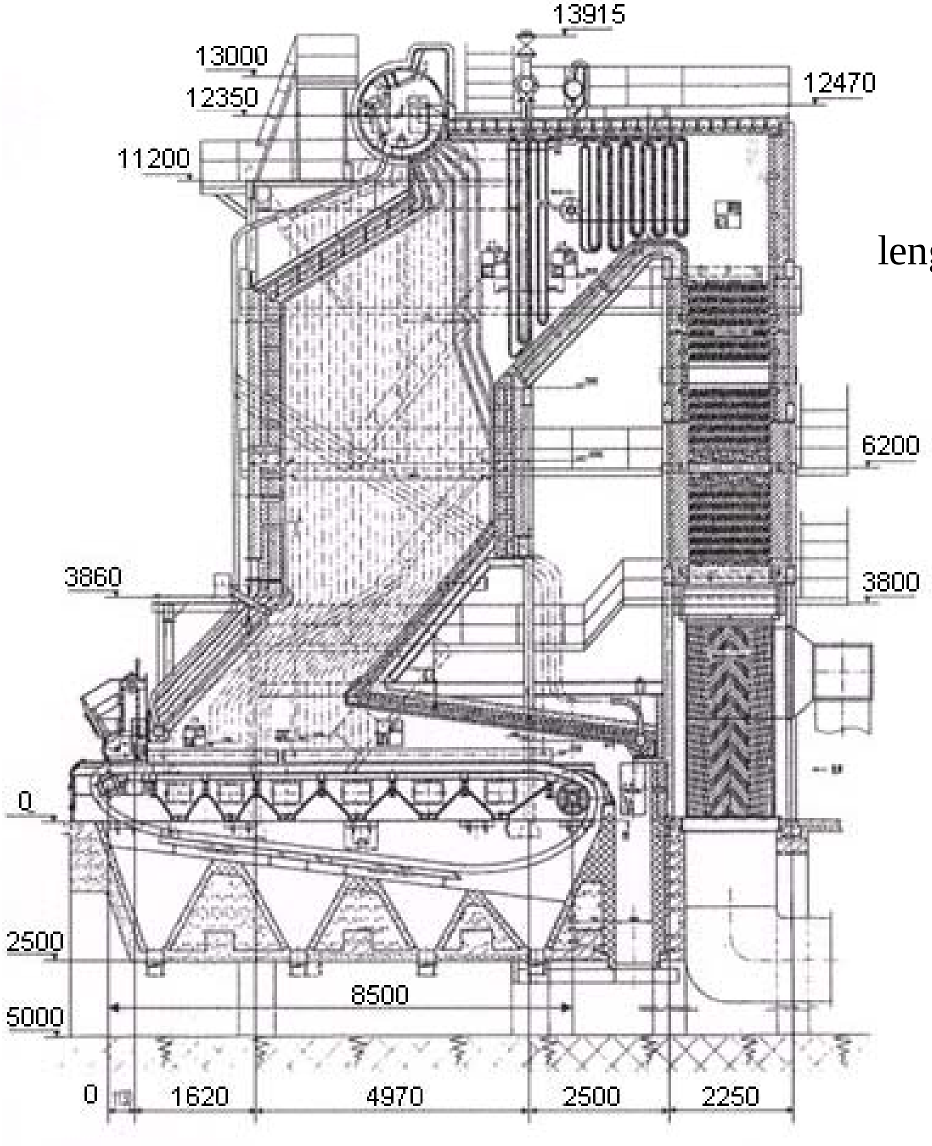
Main grate parts:

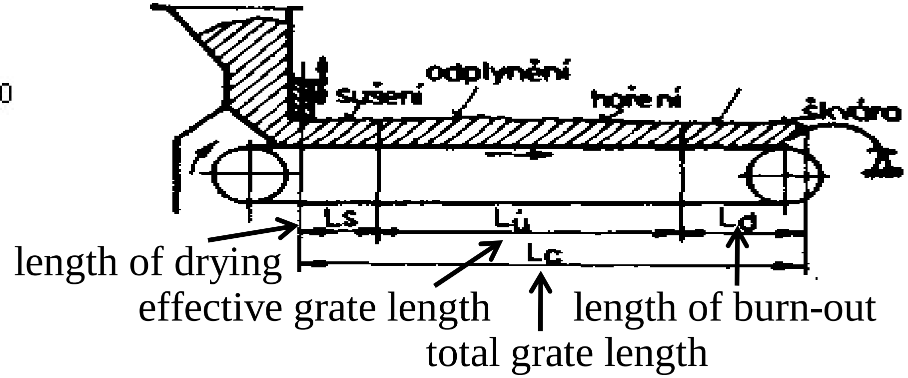
- Supporting construction
- Grate mover (travelling grates)

Travelling grate – suitable for energetic applications

Boilers

Grate boilers:





Gross grate area:

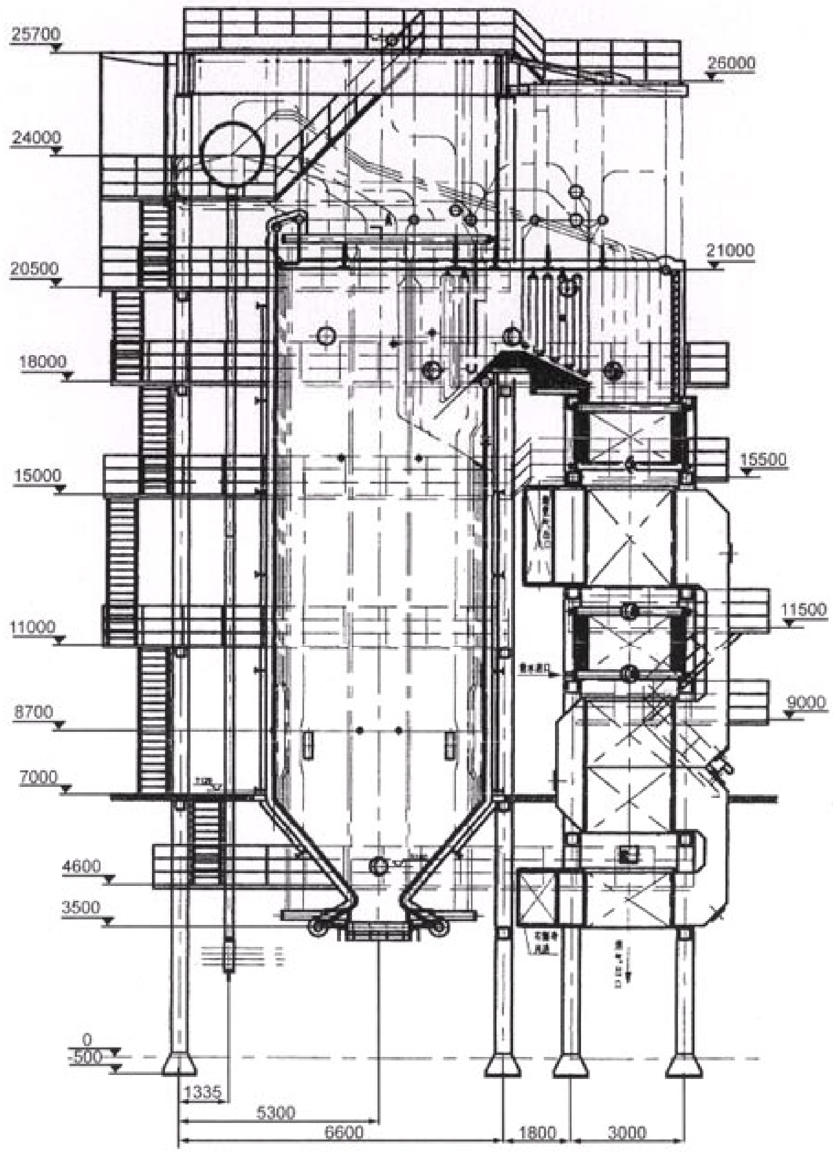
$$S_r = \frac{\dot{m}_{pal} \cdot Q_n}{\bar{q}_r} = a \cdot L \; [\mathrm{m}^2]$$

$\dot{m}_{pal}$ [kg.s^{-1}]	fuel mass flow
Q_n [MJ.kg^{-1}]	fuel calorific value
$\bar{q}_r$ [MW.m^{-2}]	average rated heat otput of grate $\approx$ 0,7-1,6 MW.m^{-2}
a, L [m]	grate width, length

Typical burning time for grate boilers: tens of minutes

Boilers

Pulverized boilers:



The boilers are equipped with hammer mills for combustion lignite or black coal in pulverized form. Boilers have wide power output regulation range without employing stabilization (LFO). Pulverized boilers are constructed from rating 50 t/h.

Typical burning time for pulverized boilers:

1 - 3 s

Boilers

Fluidized bed boilers:

Technology is based on properties of solid/fluid mixture. Combustion air inlet is at chamber bottom and blows air into solid fuel particles. Resulting fluid mixture has large reaction surface with relatively high particle speeds. Thus there is achieved very intensive burning in the bed. Solid or liquid waste can be combusted this way (crushed or milled at the same grain size). Fluid combustion is suitable for waste of high sulfur content, because combustion products can be separated by adding lime or limestone directly into the waste. Not applicable for sintering waste.

Classification of fluidized beds:

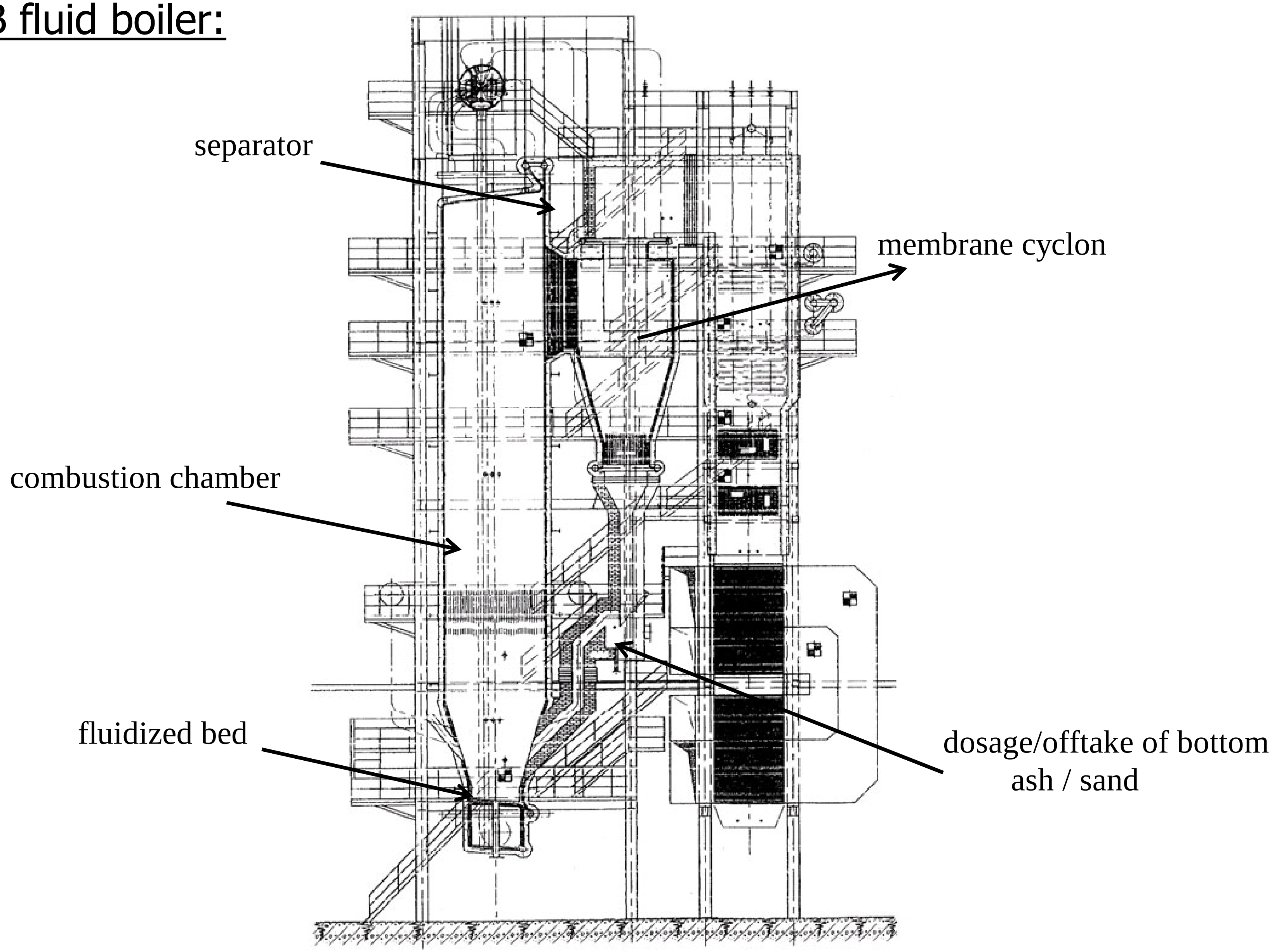
- **Bubbling Fluidized Bed – BFB**
- **Circulating Fluidized Bed - CFB**

CFB meets nowadays requirements on emissions and is the best choice for lignite fired power plants of higher ratings

CFB fluid boiler

Boilers

CFB fluid boiler:



Boilers – Hydraulic Part

Pressure losses:

Calculation of pressures inside boiler is necessary for final dimensioning of feedwater pump (resp. turbine). Total pressure losses consist of:

$$\Delta p = \Delta p_t + \Delta p_m \pm \Delta p_h \pm \Delta p_d$$

- Δp_t – pipe friction
- Δp_m – local resistances (inflow, outflow, knees...)
- Δp_h – hydrostatic pressure
- Δp_d – dynamic flow pressure

Water:

$$\Delta p_t = \lambda . \frac{l}{d_{ekv}} . \frac{c^2}{2} . \rho$$

λ – friction factor

Espec. for laminar flow:

$$\lambda = \frac{64}{\mathrm{Re}}$$

Re – Reynolds n.

$$\mathrm{Re} = \frac{c.d}{\nu}$$

ν – kinematic viscosity

Steam:

$$\rho = \frac{p}{r.T} \qquad c = \frac{1}{\rho} . \frac{\dot{m}}{A} = \frac{4.\dot{m}}{\pi . d^2 . \rho} \qquad \Delta p_t = p_1 - \sqrt{p_1 - \frac{16}{\pi^2} . \lambda . \frac{\dot{m}^2}{d^5} . \frac{r.T}{\rho} . l}$$

Evaporator:

Complex model, approximated by cubic equation:

$$\Delta p_t = \frac{A}{\dot{Q}} . \dot{m}^3 - B . \dot{m}^3 + C . \dot{Q} . \dot{m}$$

A, B, C konstanty

Boilers – Hydraulic Part

Local resistances:

$$\Delta p_m = \sum \zeta_m \cdot \frac{c^2}{2} \cdot \rho$$

↑ local resistances factor

Hydrostatic pressure:

$$\Delta p_h = \Delta h . \rho . g$$

Dynamic pressure:

$$\Delta p_d = \frac{\rho_1 . c_1^2}{2} - \frac{\rho_2 . c_2^2}{2}$$

Natural circulation, circulation ratio:

Pressure losses exposed by water/steam flow in evaporator are a function of velocity

$$\Delta p_{zc} = \Delta p_t + \Delta p_m = f_{zc}\left(c\right)$$

Static overpressure in evaporator is caused by hydrostatic pressure difference

$$\Delta p_{sop} = h . \rho_{down-comer} . g - h . \rho_{riser} . g = f_{sop}\left(c\right)$$

For natural circulation dimensioning must be found flow velocity c^*, corresponding to equality between losses and overpressure

$$f_{zc}\left(c^*\right) = f_{sop}\left(c^*\right)$$

Circulation ratio:

$$CR = \frac{\dot{m}_z}{\dot{m}_v}$$

$\dot{m}_z$ → water passing into riser

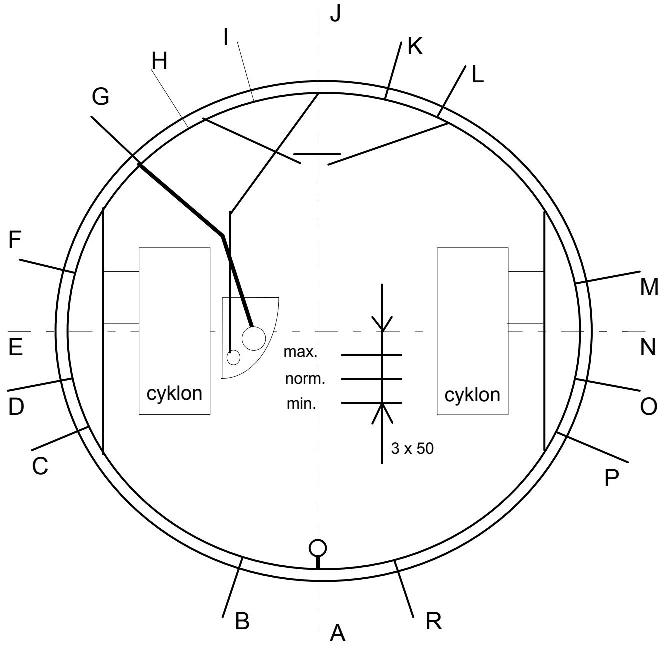
$\dot{m}_v$ ← steam generated in evaporator

Typical value for boilers of higher ratings is 6-8.

Boilers – Hydraulic Part

Steam drum:

Boiler drum is a thick-walled pressure vessel (wall thickness about 10 cm) With diameter circa 1 m, located in upper part of front pass of boiler. Down-comers and risers outlets are located in underneath, superheater outlets in upper parts. *Boiler water level* is an interface for two water states. This level must be kept at constant value circa in the middle of drum pressure vessel. No steam must enter into down-comers and no water into superheaters!



A - odvodnění bubnu
B,R - zavodňovací trubky
C,D - várnice pravé strany
E,F - várnice přední stěny
G - napájení bubnu
H - pojišťovací ventily
I,K,L - parovody k PK 1
J - alkalizace
M,N - várnice levé strany
O,P - várnice zadní stěny