Power Plants
A1M15ENY

Lecture No. 7

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MECHANICAL AND THERMAL PART OF THE POWER PLANTS

Basics of Thermodynamics

<u>Internal energy:</u>

is a total kinetic and potential energy of the all chaotically moving particles of a system. It excludes all kinds of kinetic and potentional energy influencing a system as a whole (external forces, body move etc.).

Acording to equipartition theorem, internal energy per one molecule of ideal gas:

single-atom:	two-atom:	three and more:
$u = \frac{3}{2}kT$	$u = \frac{5}{2}kT$	$u = 3kT$

per kg of gas:

$u = \frac{3}{2}rT$	$u = \frac{5}{2}rT$	$u = 3rT$

k Boltzmann constant, r individual gas constant $r = \frac{N_A.k}{M}$

Internal energy change in a closed thermodynamic system is caused either by heat or work exchange: $du = \delta Q - \delta A$

(1st law of thermodynamics)

Basics of Thermodynamics

<u>Entropy:</u>

is a measure of disorder (position and velocity of particles) of a system, „measure of probability of a state“). For an ideally ordered system (=crystal grid at 0 K) is s=0. Entropy is decreasing with rising temperature and decreasing pressure. During irreversible processes (i.e. friction, throttle etc.) is entropy rising.

$$ds = \frac{dq}{T}$$

Total entropy of an isolated system cannot decrease

(2^{nd} law of thermodynamics)

<u>Entalpy:</u>

Is total energy of a thermodynamic system = sum of internal (u) and mechanical (p.V, where p=pressure, V=volume) system energy. Expression in per kg units:

$$di = du + d\left(p.v\right) = du + p.dv + v.dp = dq + v.dp$$

<u>Work:</u>

$$da = p.dv$$

Equation of State for Gasses

Equation of state for an ideal gas:

$$p.v = r.T$$

where

Gas pressure: p [Pa]

Thermodynamic temperature: T [K]

Specific volume: v $[\mathrm{m^3.kg^{-1}}]$

Individual gas constant: r $[\mathrm{kJ.K^{-1}.kg^{-1}}]$

Mayer's relation:

Relation between specific heats at constant volume and pressure

$$c_P = c_V + r$$

State Processes

Basic process types present in thermal cycles :

<u>Isobaric process:</u>

i.e. feedwater heating, superheating and reheating of steam, generally:

$$dq\big|_{p=konst} = du + p.dv = i\big|_{p=konst} \Rightarrow q\big|_{p=konst} = i_2 - i_1$$

$$da\big|_{p=konst} = dq - du \Rightarrow a\big|_{p=konst} = i_2 - i_1 + u_1 - u_2$$

Considering equation of state for ideal gas:

$$dq\big|_{p=konst} = c_V dT + r.dT = c_P.dT \Rightarrow q\big|_{p=konst} = c_P.\left(T_2 - T_1\right)$$

$$da\big|_{p=konst} = p.dv = r.dT \Rightarrow a\big|_{p=konst} = p.\left(v_2 - v_1\right) = r.\left(T_2 - T_1\right)$$

<u>Isochoric process:</u>

i.e. start up with closed valves, generally:

$$dq\big|_{v=konst} = du + p.dv = du \Rightarrow q\big|_{v=konst} = u_2 - u_1$$

$$da\big|_{v=konst} = dq - du = p.dv = 0$$

Considering equation of state for ideal gas:

$$dq\big|_{v=konst} = c_V dT \Rightarrow q\big|_{p=konst} = c_V.\left(T_2 - T_1\right)$$

State Processes

Isotermal process:

i.e. slow compression/expansion, generally:

$$dq\big|_{T=konst} = Tds \Rightarrow q\big|_{T=konst} = T.\left(s_2 - s_1\right)$$

$$da\big|_{T=konst} = dq - du \Rightarrow a\big|_{T=konst} = T.\left(s_2 - s_1\right) + u_1 - u_2$$

If an internal structure of matter doesn't change:

$$da\big|_{T=konst} = dq\big|_{T=konst} \Rightarrow a\big|_{T=konst} = T.\left(s_2 - s_1\right)$$

Considering equation of state for ideal gas:

$$dq\big|_{T=konst} = da\big|_{T=konst} = \frac{rT}{v}.dv \Rightarrow q\big|_{T=konst} = a\big|_{T=konst} = rT.\ln\frac{v_2}{v_1} = rT.\ln\frac{p_1}{p_2}$$

Adiabatic process (isentropic):

No heat exchange with surroundings, i.e. an ideal adiabatic expansion in turbine, generally:

$$dq\big|_{s=konst} = Tds = 0 \Rightarrow q\big|_{s=konst} = 0$$

$$da\big|_{s=konst} = dq - du = -du \Rightarrow a\big|_{s=konst} = u_1 - u_2$$

Considering equation of state for ideal gas:

$$c_p.\frac{dv}{v} + c_V.\frac{dp}{p} = \frac{dv}{v} + \kappa.\frac{dp}{p} \Rightarrow p.v^{\kappa} = konst \quad \text{a} \quad a\big|_{s=konst} = \frac{r}{\kappa - 1}\left(T_1 - T_2\right)$$

State Processes

Polytropic process:

Heat exchanged with surroundings is corresponding to temperature change, i.e. real polytropic expansion in a turbine, „something between“ isothermal and adiabatic process:

$$p.v^n = konst$$

Formulas hereinabove, which are not reflecting equation of state for gases are valid GERERALLY, for all kind of matters including real matters and all their phases

Reversible process:

The system can be returned to its original state by reversing of order of every single step

Irreversible process:

An original state cannot be achieved within identical process performed in reversed order of every single step

Mollier i-s diagram:

State quantities for water - liquid / saturated / vapor region

Thermal Cycles

Thermal cycle is a sequence of processes which is finishing with an identical state as the original state of the system

If the changes are *reversible*, then the whole cycle is *reversible* and can be shown as a closed loop in T-s diagram

Real cycles are *irreversible* mostly thanks to:

- flow velocity changes
- working medium weight change
- irreversible phase changes

Real cycle is idealized by reversible c. by cycle comparison factor

<u>Cycle comparison factor:</u>

$$\nu = \frac{\eta_{t0}}{\eta_{tp}}$$

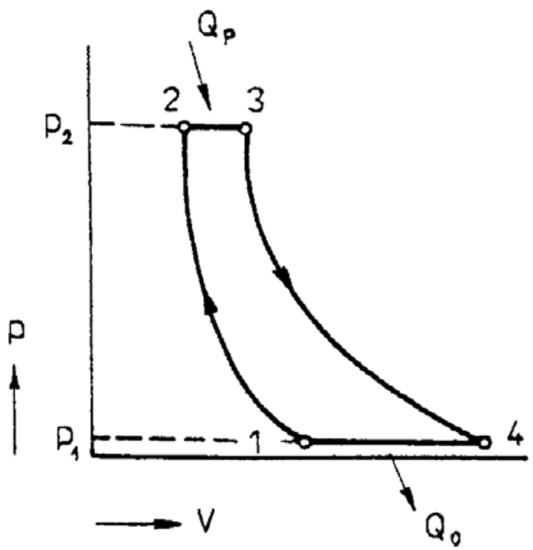
η_{t0} Real cycle efficiency

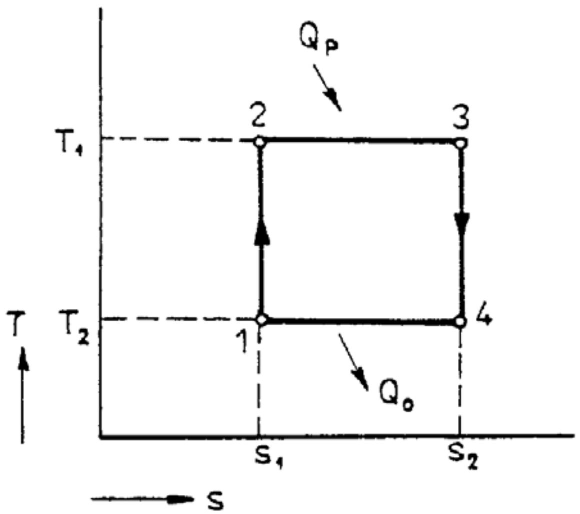
η_{tp} Ideal cycle efficiency

Carnot Cycle

- The highest efficiency
- For ideal gases only (can be performed approximately in saturated region)
- 4 reversible processes of working medium:

Isothermal expansion (heater body temperature T1) [2→3] {heating in boiler}.
Heat q_p is transferred into the system.
Adiabatic expansion (temperature decrease from T1 to T2) [3→4] {turbine}
Isothermal compression (cooling body temperature T2) [4→1] {condenser}
Heat q_o is transferred out of the system.
Adiabatic compression (between T1 and T2) [1→2] {compression in feedwater pump (compressor)}



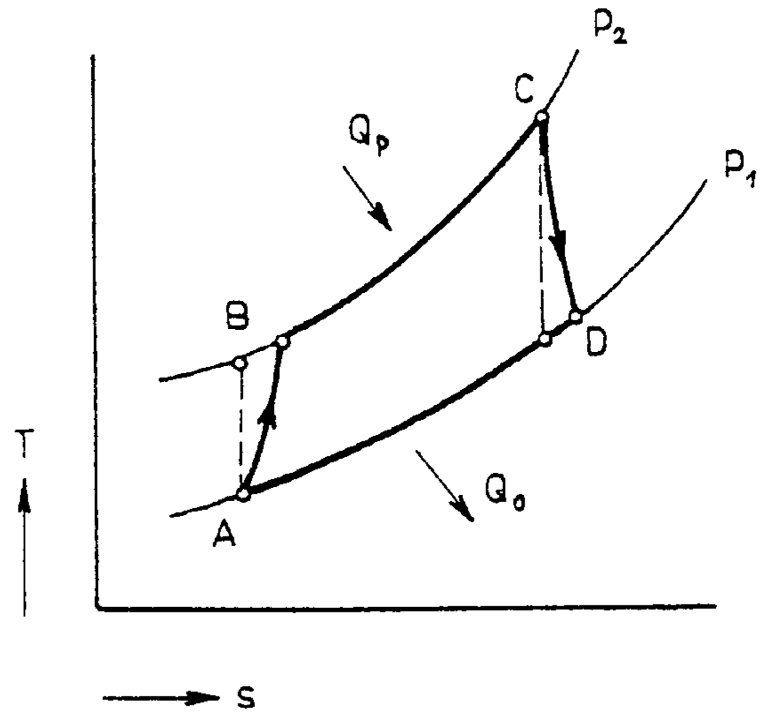


Efficiency:

$$\eta = \frac{q_p - q_o}{q_p} = 1 - \frac{T_2}{T_1}$$

Joule (Brayton) Cycle

A→B ... **gas adiabatic compression** {compressor}
B→C ... **isobaric heat inlet** {combustion chamber}
C→D ... **adiabatic gas expansion** {gas turbine}
D→A ... **isobaric heat outlet** {for closed cycles in heat exchanger, for open cycles to atmosphere}



Heat inlet: $q_p = i_C - i_B$

Heat outlet: $q_o = i_D - i_A$

Work: $a = (i_C - i_B) - (i_D - i_A)$

Compress ratio:

$$\varepsilon = \frac{v_A}{v_B}$$

Efficiency:

$$\eta = \frac{q_p - q_o}{q_p} = 1 - \frac{T_D}{T_C} =$$

$$= 1 - \left(\frac{p_{AD}}{p_{BC}}\right)^{\frac{\kappa-1}{\kappa}} = 1 - \frac{1}{\varepsilon^{\kappa-1}}$$

Clausius-Rankine Cycle

Employing phase changes of liquid and vapor

$\Rightarrow$ total condensation of steam to vapor

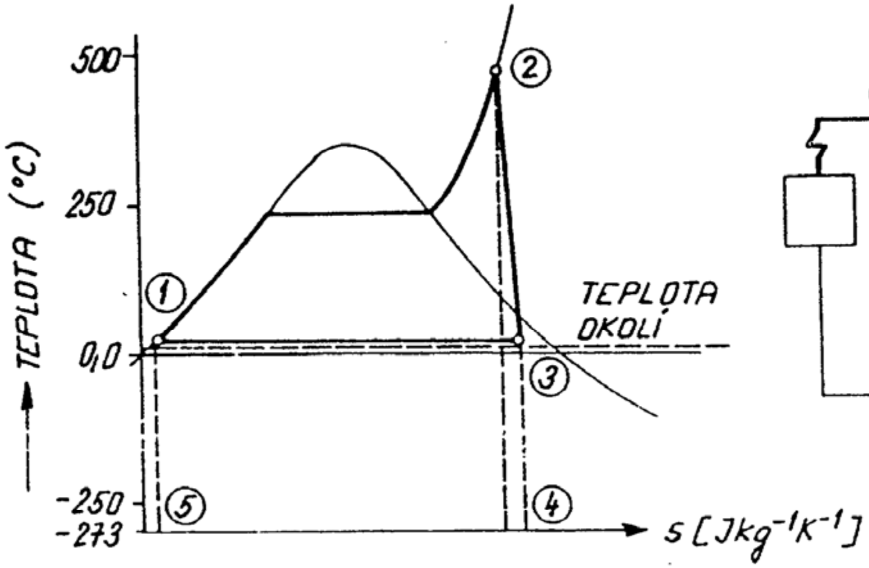
A feedwater pump power input is significantly lower than power input of a compressor !

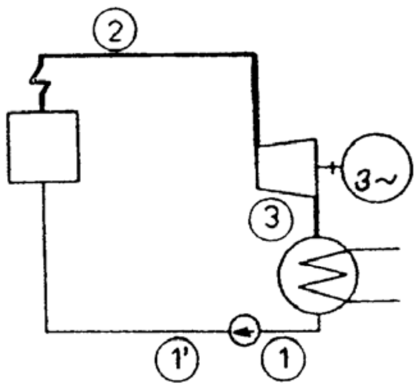
1... Liquid compression to working pressure {pump}

1→2 ... <u>heat transfer</u> *Qp* {boiler} = heating to boiling point + isothermal vaporization + superheating of steam

2→3 ... <u>adiabatic expansion</u> {steam turbine}

3→4 ... <u>isothermal condensation</u> {condenser} , heat outlet *Q**o***





Heat inlet: $q_p = i_2 - i_1$

Heat outlet: $q_o = i_3 - i_1$

Work of pump: $a = \left(p_2 - p_1\right).v$

<u>Efficiency:</u>

$$\eta = \frac{q_p - q_o}{q_p} =$$

$$= \frac{i_2 - i_3 - \left(p_2 - p_1\right).v}{i_2 - i_1 - \left(p_2 - p_1\right).v}$$

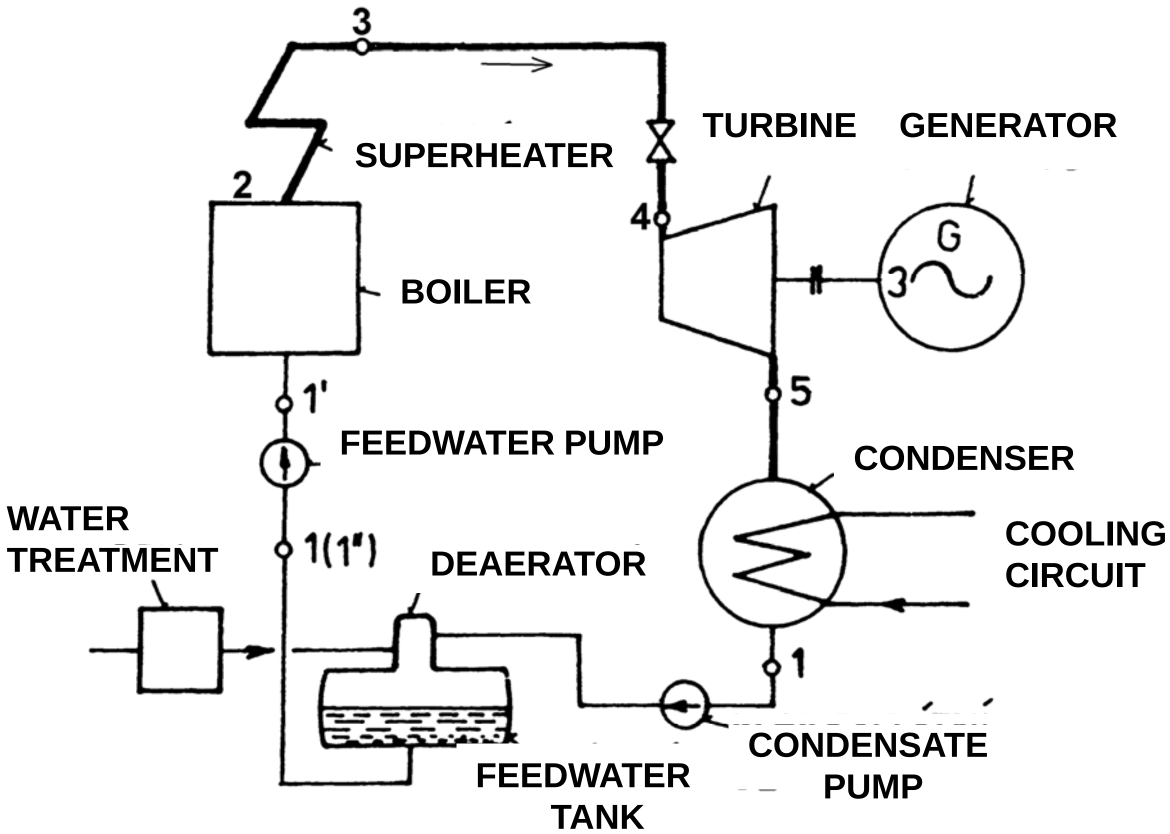
Power Plant with Condensing Turbine

<u>Turbine type:</u> condensing – condensing of steam exhaust from turbine in **condenser**

<u>Condensate:</u> 25 – 40˚C

<u>Feedwater:</u> > 104˚C [de-aeration (air, oxygen) → corrosion !] ⇒

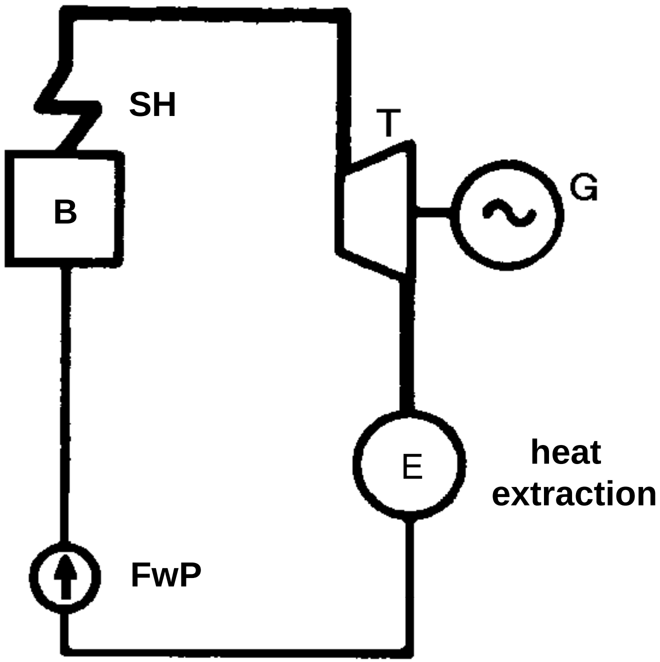
~ 200˚C = condensate + additional water to cover losses in the steam – water main circuit



Power Plant with Back Pressure Turbine

<u>Turbine type:</u> back pressure – turbine steam can be utilized for district heat
<u>Disadvantage:</u> *Generator electric power output is in direct proportion to generated heat into the district heat network*

$$\frac{Q}{P_{el}} = const$$



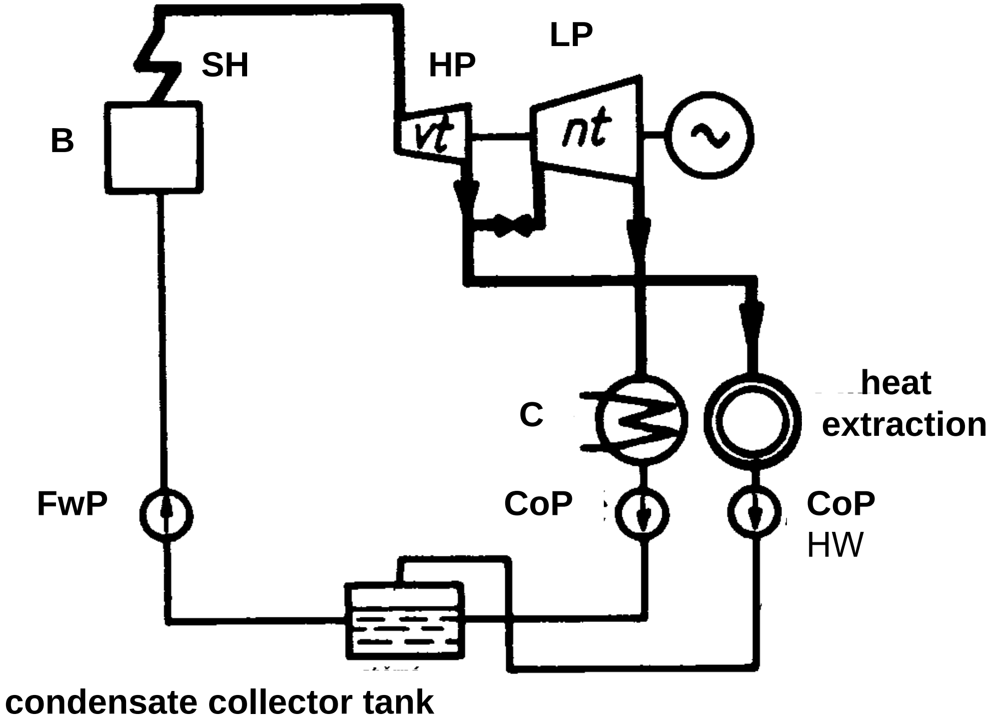
A steam from back pressure part which is utilized in heat exchangers is condensing there under similar conditions as in condensers. Unlike in condensers, the heat is transferred to heat water and thus a condensate temperature is in this case higher (about 100°C)

Back pressure turbine running at rated parameters is thus the most efficient device thanks to latent heat of phase transition (steam-water) utilization

Power Plant with Extraction Condensing Turbine

Combined heat and power generation, CHP (condensing power plant + heating plant) ⇒ turbine with two main bleeders (extraction, exhaust)
Power plant with heating water system

Advantage: *Electric power can be regulated independently up to transfer into heat system in wide range (dependent on high pressure /HP/ part power rating)*



Steam Reheating

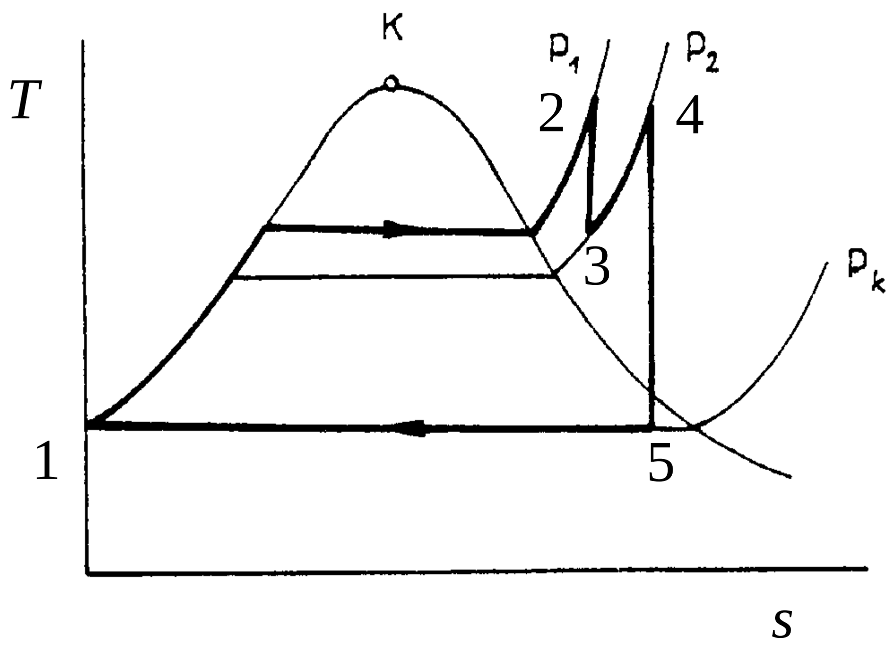
Partial steam expansion in turbine HP part – steam reheating in reheater – finishing the expansion in turbine LP part

<u>*Advantages:*</u>

significantly increasing efficiency (single step reheat: adding 5-7 %, second step: adding 1-2 %)

increasing steam dryness at the end of expansion (i.e. on the last turbine blades) $\Rightarrow$ thermodynamics efficiency increase

lesser risk of turbine blades erosion (water drops on the last turbine blades at LP part)



Heat inlet: $q_p = i_2 - i_1 + i_4 - i_3$

Heat outlet: $q_o = i_5 - i_1$

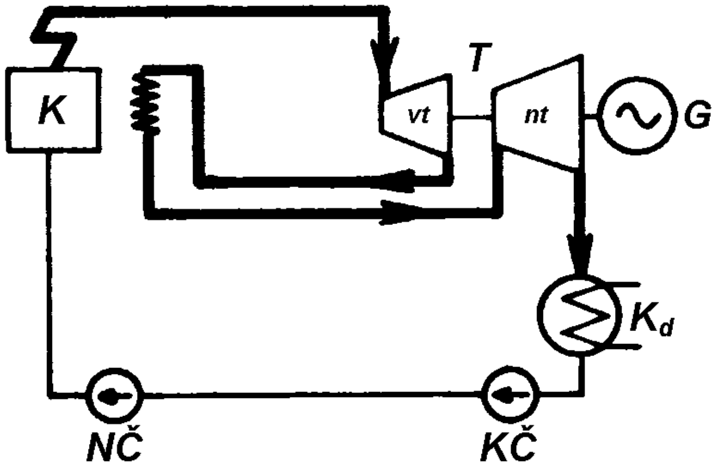
Work of pump: $a = (p_2 - p_1).v$

<u>Efficiency:</u>

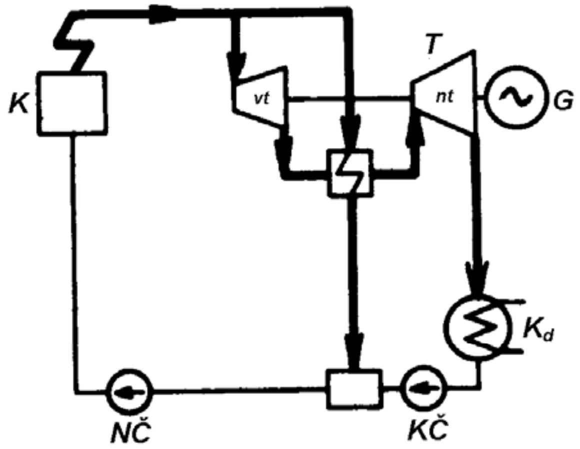
$$\eta = \frac{i_2 - i_5 + i_4 - i_3 - (p_2 - p_1).v}{i_2 - i_1 + i_4 - i_3 - (p_2 - p_1).v}$$

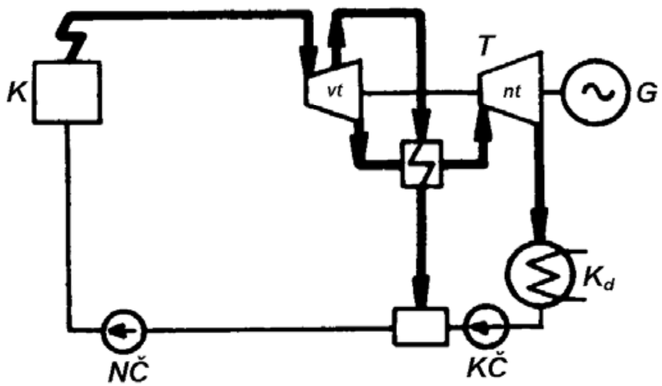
Steam Reheating

Reheating of team by flue gases from boiler:

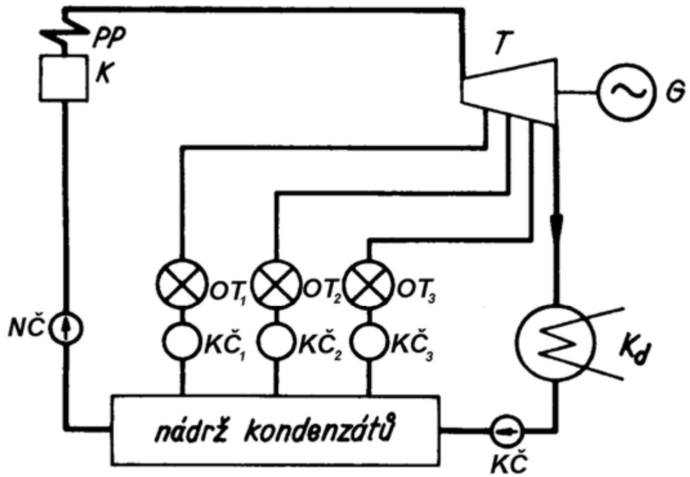


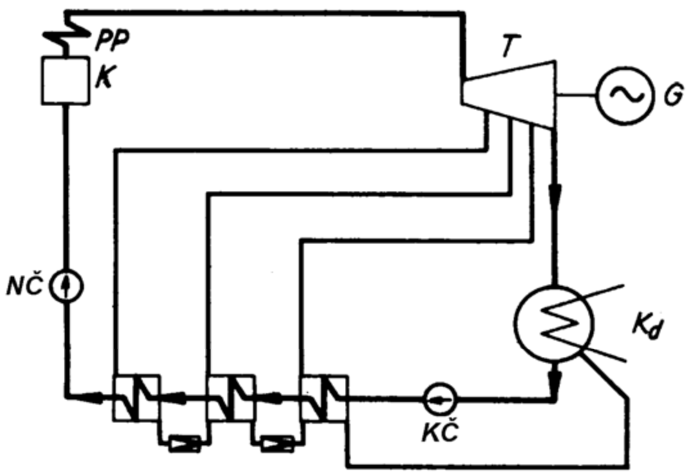
Steam reheat by admission or extraction steam:





Feedwater Conditioning





Non-regulated extraction (bleeders) are leaded out from several positions inside the turbine (different steam parameters). This kind of steam extraction is proportional to the turbine actual power output.

The extracted steam latent heat is then utilized for condensate and feedwater heating.

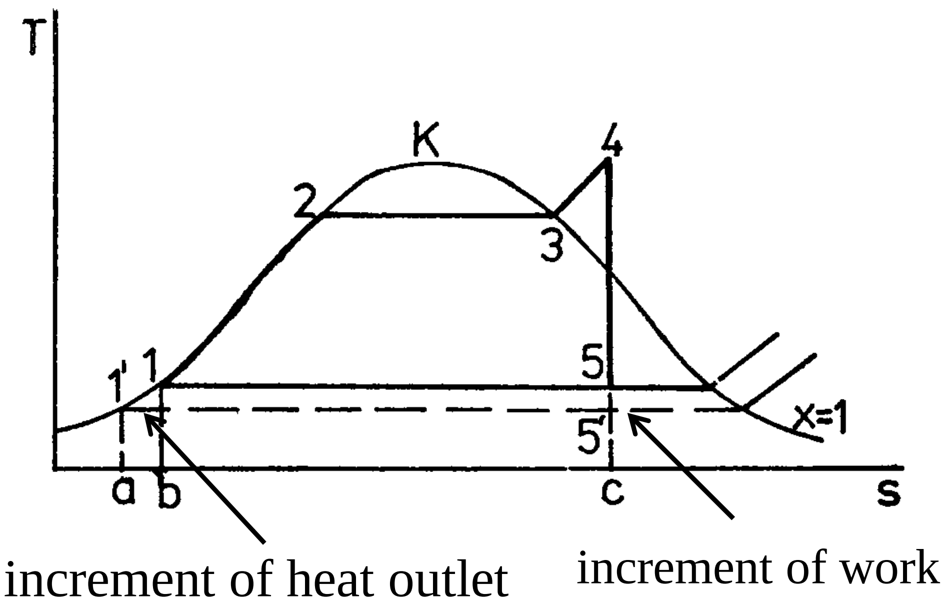
The effect is an efficiency increase. Max. number of bleeders/exchangers: 8 to 10 (for more we obtain an opposite effect, the efficiency is decreasing).

The other effect is a steam flow reduction done step by step **(⇒ possible reduction of LP part size and/or greater initial steam flow ⇒ turbine power output increase !)**

Condenser Pressure Reduction

Condenser steam temperature and pressure reduction increases q_o, but total efficiency is growing

<u>Possible solutions how to reduce pressure:</u>
Cooling water temperature decrease **(most advantageous !)**
Cooling water flow increase **(disadv.: power input of pumps is rising ⇒ auxiliary rise)**
Heat exchanging surface extension **(disadv.: higher investment costs)**



Heat inlet: $q_p = i_4 - i_{1'}$

Heat outlet: $q_o = i_{5'} - i_{1'}$

<u>Efficiency:</u>

$$\eta = \frac{q_p - q_o}{q_p} =$$

$$= \frac{i_4 - i_{5'} - (p_2 - p_1).v}{i_4 - i_{1'} - (p_2 - p_1).v}$$

Admission Steam Parameters Increase

Means a suitable combination of either steam pressure or temperature increase:

***Increasing admission steam* PRESSURE**

Limitation:

Steam wetness increase at the end of expansion ⇒ the efficiency is getting down. Max. max. tolerable steam wetness is circa: 12 až 14 %.

Natural circulation limits: boilers with natural circulation: max. steam pressure17 MPa; boilers with forced circulation 24 MPa !

Increasing admission steam TEMPERATURE

Increases thermal cycle efficiency

Increases thermodynamic turbine efficiency (steam wetness decrease at the end of expansion!)

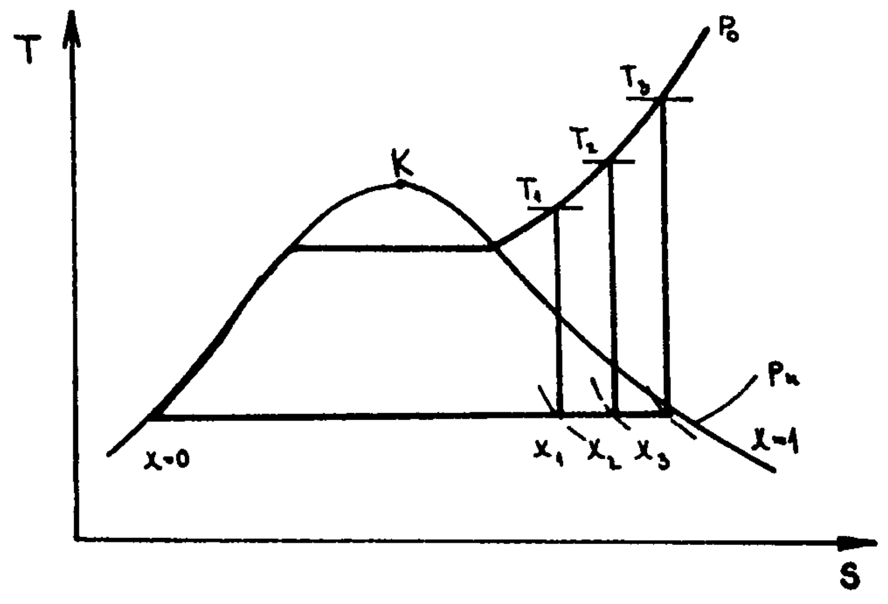
Limitation :

Construction material properties ⇒ Constructions of feritic-pearlitic steels: max. steam temp. 535 ºC. For higher temperatures: austenitic steels

⇒ ***Highest temperatures:***

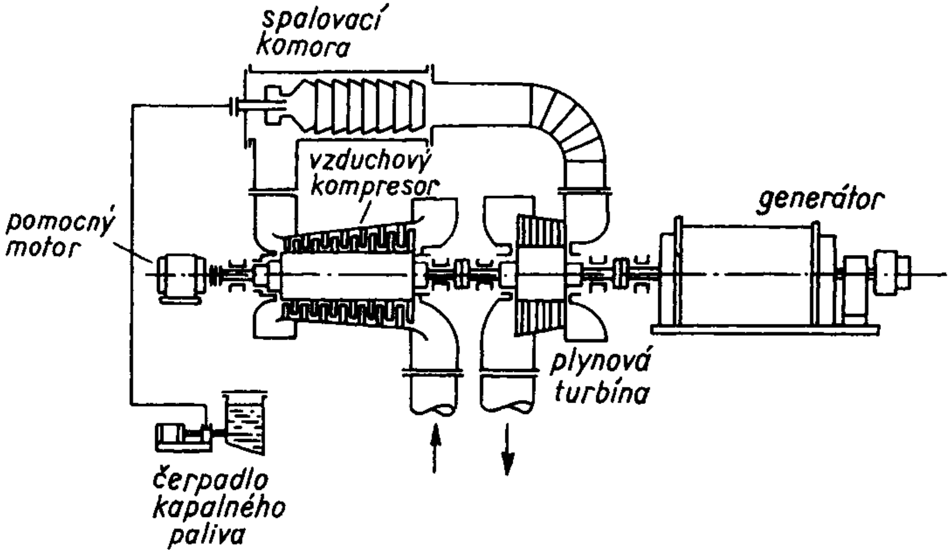
For units 110 a 200 MW: 535 - 545 ºC, for 500 MW: 545 ºC.

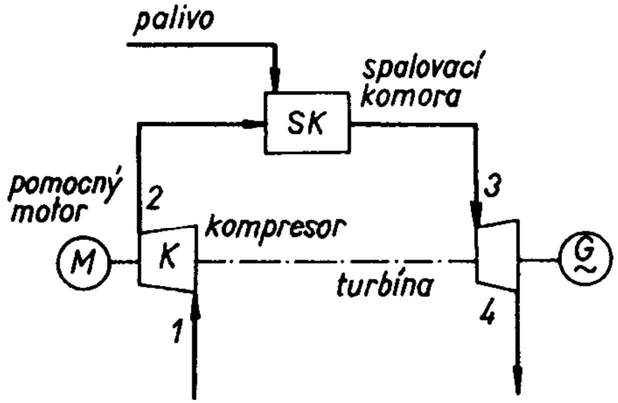
More – supercritical boilers

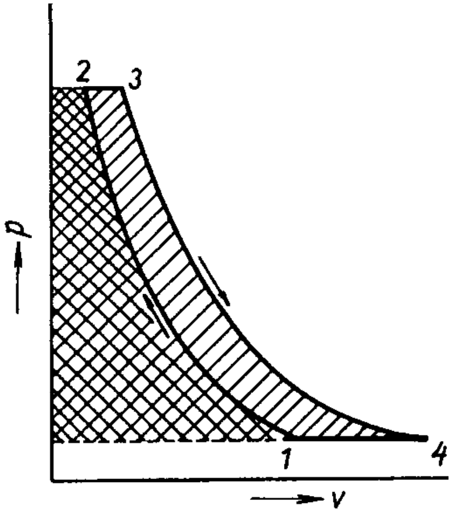


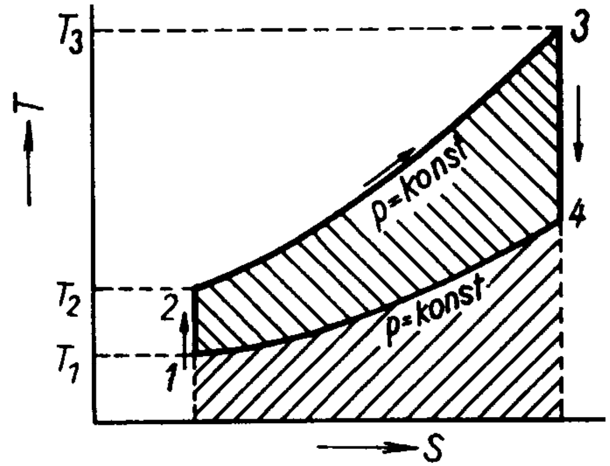
Gas Cycle

Combustion turbine









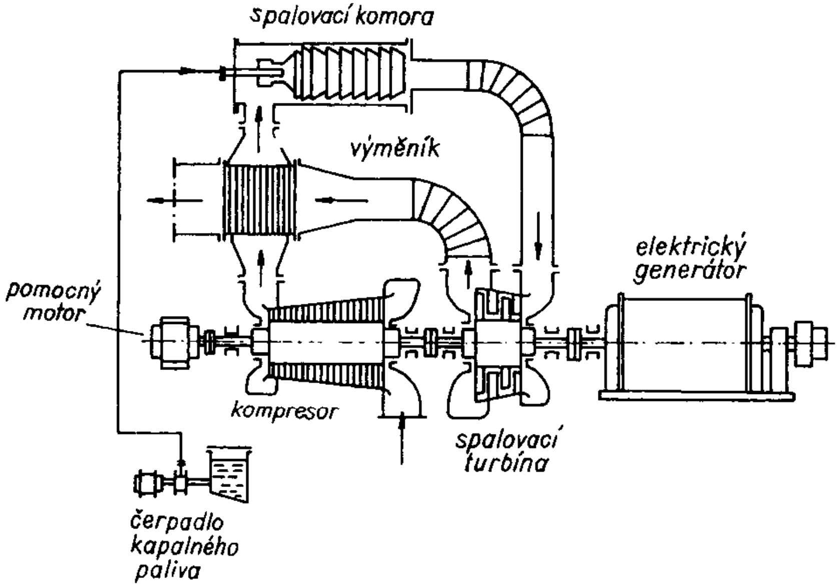
Gas Cycle

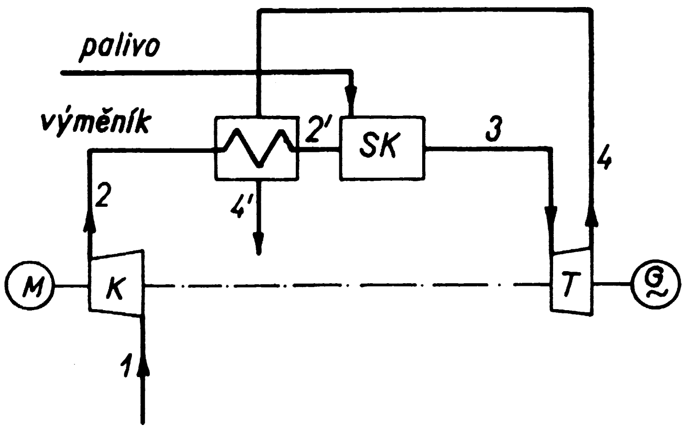
<u>Advantages and disadvantages:</u>

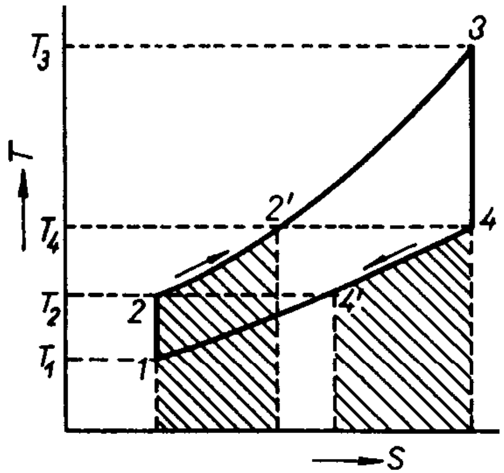
- fast start up and shut down (good power elasticity)
- low investment costs (low needs on building materials)
- good efficiency (25 to 35 %)
- high reliability
- compact machine arrangement, small site area requirements
- expensive fuel (gas, oil)
- material quality demanding
- high compressor power input: up to 70 % of turbine rated power !
- high flue gas temperature ($\Rightarrow$ decrease thermodynamic efficiency): Flue gas temperature: > 1500 ºC + air (cooling) $\rightarrow$ mixture temperature at turbine inlet: 600 to 800 ºC.

Regeneration by Exhaust Gases

Exhaust gases are heating inlet gases in front of the entrance to the combustion chamber







Combined Cycle Gas Turbines

Temperatures:

Gas cycles:

Heat inlet: 600 to 800 ºC

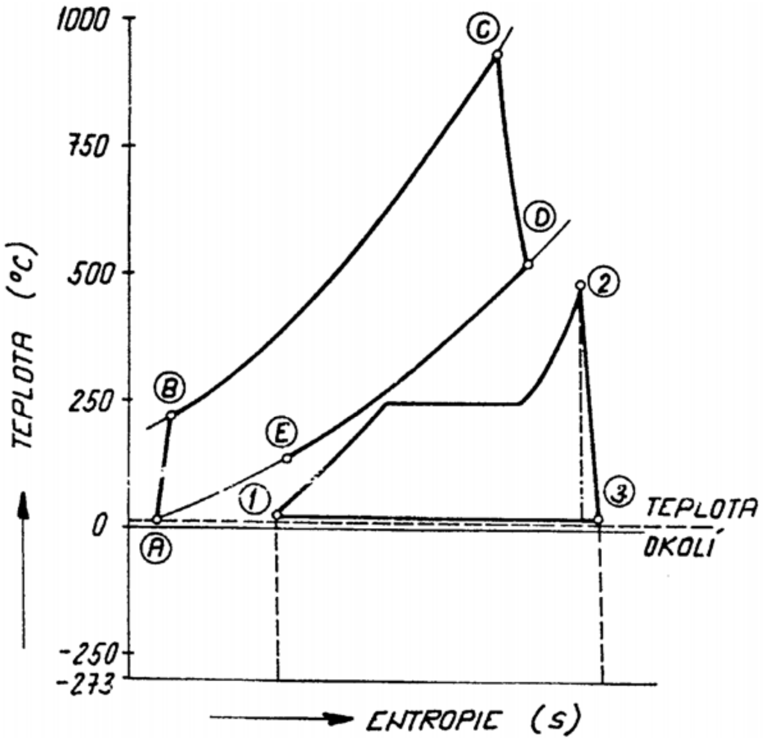
Heat outlet: high temperature ⇒ low cycle efficiency

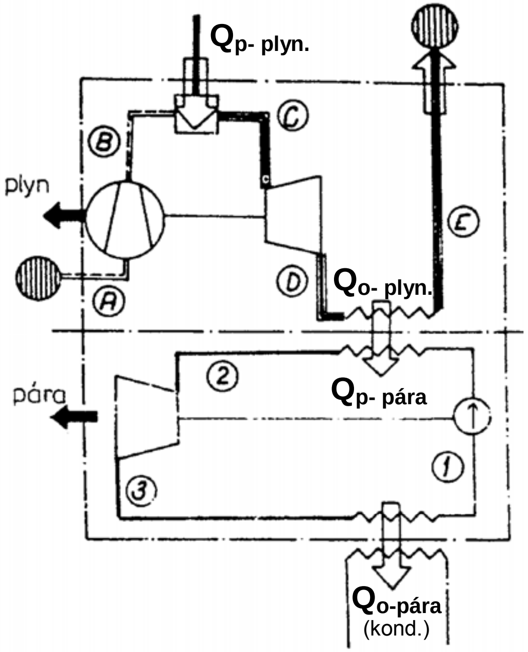
Steam cycles:

Heat inlet: 250 to 350 ºC (max. 650 ºC),

Heat outlet: cca 30 ºC

⇒ ***Combination:***





Combined Cycle Gas Turbines

<u>**Advantages:**</u>

<u>**Higher efficiency**</u>**:**

power plant with gas cycle: ηel = 0,28 - 0,38

power plant with steam cycle: ηel = 0,28 - 0,42

power plant with CCGT: ηel = 0,42 - 0,58

Compressor work of gas cycle is decreasing. Thanks to steam condensation significant volume decrease

$\Rightarrow$ feedwater pump power input is several per cent of turbine rated power (unlike in the case of gas cycle - 2/3 of gross turbine power output !)

<u>**Disadvantages:**</u>

Combustion of high quality fuel (gas, oil)

Cheng Cycle

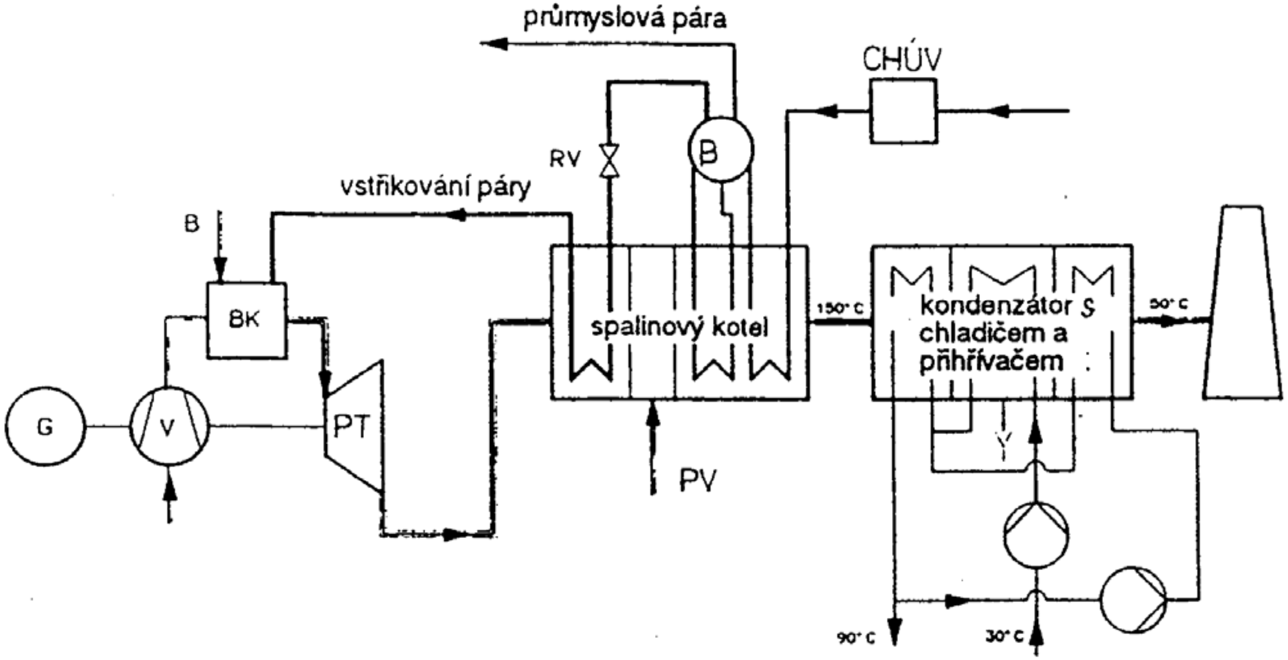
Steam and gas turbine integration into one machine

⇒ steam (generated in the boiler) is injected to the turbine combustion chamber

⇒ low investment costs

⇒ low combustion temperature and NOx emissions (thanks to the injections)

Lesser dependency heat / power generation (in case of heat stations)



B - fuel inlet, BK - combustion chamber, G - generator, PT - gas turbine, V - compressor, PV - external heating, CHÚV - water treatment plant, B - drum, RV - reduction valve

Addenum to 7th Lecture

Thermal power plant cycle uses one steam reheat. Pressure $p_2 = 12\ \mathrm{MPa}$, temperature $T_2 = 530\ °\mathrm{C}$, pressure $p_5 = 3{,}5\ \mathrm{kPa}$. Reheat is performed at pressure $p_3 = 2{,}3\ \mathrm{MPa}$ to the temperature $T_4 = 480\ °\mathrm{C}$. Compare the efficiency of this cycle and a cycle without reheat and compute the incremental efficiency!