Power Plants
A1M15ENY

Lecture No. 10

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Fuel mass flow: $\dot{m}_{fuel}=\frac{P_{net}}{\eta_{net}.Q_i^r}=\frac{1000}{0,33.15}=202\ \mathrm{kg.s^{-1}}$

Sulfur mass flow: $\dot{m}_S=\dot{m}_{fuel}.S^r=202.0,02=4,04\ \mathrm{kg.s^{-1}}$

Ash mass flow: $\dot{m}_A=\dot{m}_A.A^r=202.0,15=30,3\ \mathrm{kg.s^{-1}}$

Fuel water mass flow: $\dot{m}_W=\dot{m}_W.W^r=202.0,30=60,6\ \mathrm{kg.s^{-1}}$

Carbon mass flow: $\dot{m}_C=\dot{m}_W.\left(1-W^r-A^r-S^r\right)=202.\left(1-0,3-0,15-0,02\right)=107,1\ \mathrm{kg.s^{-1}}$

Volume flow of inlet air:

$$\dot{V}_a=\lambda.\dot{V}_{adt}=\lambda.\frac{22,4}{0,21}.\left(\frac{\dot{m}_C}{12}+\frac{\dot{m}_S}{32}\right)=1,3.\frac{22,4}{0,21}.\left(\frac{107,1}{12}+\frac{4,04}{32}\right)=1,3.965=1255\ \mathrm{m^3.s^{-1}}$$

what means: $\frac{1255}{202}=6,21\ \mathrm{m^3/kg\ fuel}$

Dry flue gas flow without excess of air:

$$\dot{V}_{fgdt}=\frac{22,3}{12}.\dot{m}_C+\frac{21,9}{32}.\dot{m}_S+0,79.\dot{V}_{adt}=199,02+2,76+762,35=964\ \mathrm{m^3.s^{-1}}$$

Dry flue gas flow with considered excess of air:

$$\dot{V}_{fgd}=\frac{22,3}{12}.\dot{m}_C+\frac{21,9}{32}.\dot{m}_S+0,79.\dot{V}_{adt}+\left(\lambda-1\right).\dot{V}_{adt}=199,02+2,76+762,35+\left(1,3-1\right).965=1253,6\ \mathrm{m^3.s^{-1}}$$

Excess of oxygen in dry flue gas at outlet from boiler behind FGF is:

$$\omega_{O_2(FGF)}=0,21.\left(\lambda-1\right)\frac{\dot{V}_{adt}}{\dot{V}_{fgd}}=0,21.\left(1,3-1\right).\frac{965}{1253,6}=4,8\ \%$$

Wet flue gas flow with considered excess of air:

$$\dot{V}_{fg}=\dot{V}_{fgd}+1,24.\dot{m}_W=1253,6+1,24.60,6=1328,7\ \mathrm{m^3.s^{-1}}$$

Fly ash filtering:

$$\dot{m}_{ash(F)} = X_{fly\_ash}.\left(1-O_c\right).\dot{m}_A = 0,9.\left(1-0,999\right).30,3 = 0,0272 \text{ kg.s}^{-1}$$

FGD filtering:

Amount of substance of the sulfur:

$$\dot{n}_S = \frac{\eta_{FGD}.\dot{m}_S}{M_S} = \frac{0,95.4,04}{0,032} = 199,9 \text{ mol.s}^{-1}$$

Molar masses of reactants and products:

$$M_{CaCO_3} = \left(40+12+3.16\right)/1000 = 0,1 \text{ kg.mol}^{-1}$$
$$M_{CaSO_4.2H_2O} = \left(40+32+4.16+2.2+2.16\right)/1000 = 0,152 \text{ kg.mol}^{-1}$$
$$M_{CO_2} = \left(12+2.16\right)/1000 = 0,044 \text{ kg.mol}^{-1}$$

For total reaction balance must be

$$\dot{n}_S = \dot{n}_{CaCO_3} = \dot{n}_{CaSO_4.2H_2O} = 2.\dot{n}_{O_2}$$

Necessary theoretic mass of limestone: $\dot{m}_{CaCO_3} = \dot{n}_{CaCO_3}.M_{CaCO_3} = 199,9.0,1 = 19,99 \text{ kg.s}^{-1}$

Mass of produced gypsum: $\dot{m}_{CaSO_4.2H_2O} = \dot{n}_{CaSO_4.2H_2O}.M_{CaSO_4.2H_2O} = 199,9.0,152 = 30,38 \text{ kg.s}^{-1}$

Mass of sulfurous oxide: $\dot{m}_{SO_2(ODS)} = \left(1-\eta_{FGD}\right)\dot{n}_S.M_{SO_2} = \left(1-0,95\right).199,9.0,032 = 0,319 \text{ kg.s}^{-1}$

Dry flue gas change at the FGD output:

The initial amount of *dry flue gas* is reduced by: $\Delta\dot{V}_{SO_2} = -\eta_{FGD}.\frac{21,9}{32}.\dot{m}_S = -0,95.\frac{21,9}{32}.4,04 = -2,63 \text{ m}^3.\text{s}^{-1}$

Increment of CO_2: $\Delta\dot{V}_{CO_2} = +22,3.10^{-3}.\dot{n}_{CO_2} = +22,3.10^{-3}.199 = +4,45 \text{ m}^3.\text{s}^{-1}$

Increment of the rest of oxidation air : $\Delta\dot{V}_a = +22,4.10^{-3}.\dot{n}_{O_2}.\frac{0,79}{0,21} = +\frac{22,3.10^{-3}}{2}.199,9 = +8,42 \text{ m}^3.\text{s}^{-1}$

Total balance: $\dot{V}_{ad(FGD)} = \dot{V}_{sns} + \Delta\dot{V}_{SO_2} + \Delta\dot{V}_{CO_2} + \Delta\dot{V}_a = 1253,6-2,63+4,45+8,42=1263,8 \text{ m}^3.\text{s}^{-1}$

Oxygen excess in dry flue gas at FGD output:

$$\omega_{O_2(FGD)} = 0,21.(\lambda - 1)\frac{\dot{V}_{adt}}{\dot{V}_{fgd(FGD)}} = 0,21.(1,3-1).\frac{965}{1263,8} = 4,8\ \%$$

Mass of carbon dioxide:

$$\dot{m}_{CO_2} = \dot{m}_C.\frac{M_{CO_2}}{M_C} + \dot{n}_{C(FGD)}.M_{CO_2} = 107,1.\frac{0,044}{0,012} + 199,9.0,044 = 392,7 + 8,79 = 401,5\ \text{kg.s}^{-1}$$

1.A SOLID PARTICLES EMISSION LIMITS

$$c_{SPref} = c_{SP}.\frac{0,21-\omega_{O_2ref}}{0,21-\omega_{O_2(FGD)}} = \frac{\dot{m}_{fly\_ash(F)}}{\dot{V}_{fgd(FGD)}}.\frac{0,21-\omega_{O_2ref}}{0,21-\omega_{O_2(FGD)}} = \frac{0,0272}{1263,8}.\frac{0,21-0,06}{0,21-0,048} = 19,9\ \text{mg.m}^{-3}$$

Is in compliance with emission limits

1.B SO_2 EMISSION LIMITS

$$c_{SO_2ref} = c_{SO_2}.\frac{0,21-\omega_{O_2ref}}{0,21-\omega_{O_2(FGD)}} = \frac{\dot{m}_{SO_2(FGD)}}{\dot{V}_{fgd(FGD)}}.\frac{0,21-\omega_{O_2ref}}{0,21-\omega_{O_2(FGD)}} = \frac{0,319}{1263,8}.\frac{0,21-0,06}{0,21-0,048} = 233,7\ \text{mg.m}^{-3}$$

Not in compliance with SO_2 emission limits, too high sulfur content in the fuel for proposed FGD plant

2 MASS FLOWS

$$\dot{m}_{fuel} = 202\ \text{kg.s}^{-1}$$

$$\dot{m}_{CaCO_3} = 19{,}99\ \text{kg.s}^{-1}$$

$$\dot{m}_{CaSO_4.2H_2O} = 30{,}38\ \text{kg.s}^{-1}$$

$$\dot{m}_{CO_2} = 401,5\ \text{kg.s}^{-1}$$

3 CO_2 ALLOWANCES

Per one generated and sold MWh has to be paid:

$$E_{CO_2/MWh} = \frac{3600.\dot{m}_{CO_2}}{P_{net}}.E_{CO_2/t} = \frac{3600.401,5}{1000}.25 = 1445,4\ \text{kg.MWh}^{-1}\ .\ 0,025\ \text{EUR.kg}^{-1} = 36,1\ \text{EUR.MWh}^{-1}$$

If the power plant buys the allowances, it will mean a ratio

$$\frac{E_{CO_2/MWh}}{V_{/MWh}} = \frac{36,1}{60} = 60\ \%$$

from generated electricity turnover

Pumps

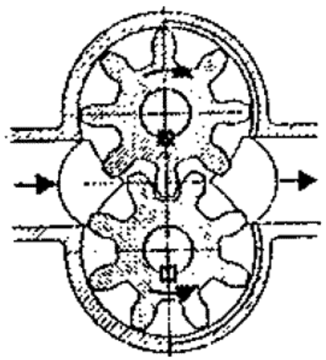
Pump power inlet:

$$P = \frac{Q.\rho}{\eta}\left(H.g + \frac{p}{\rho} + \frac{c^2}{2}\right) = \frac{Q.\rho}{\eta}.Y$$

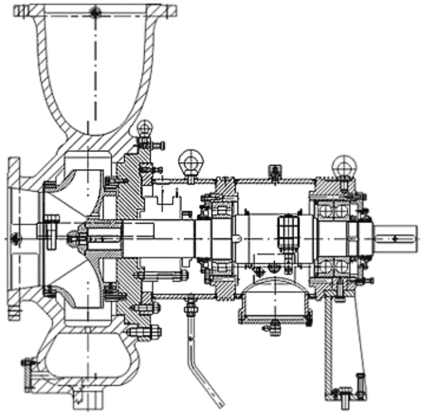
Y [J.kg^{-1}] per kg energy

Classification of pumps:

- **hydrostatic** – pressure and potential energy dominates $Y \approx H.g + \frac{p}{\rho}$
- **hydrodynamic** – kinetic energy dominates $Y \approx \frac{c^2}{2}$

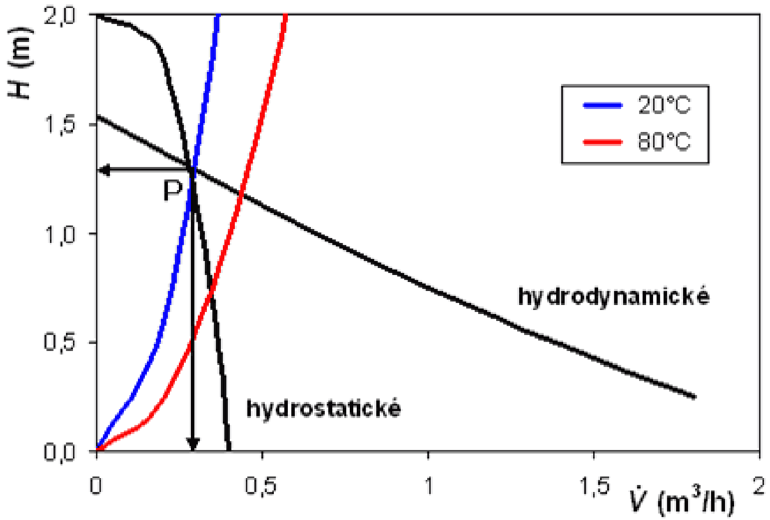


Hydrostatic
gear pump



Hydrodynamic
process pump

Q-H characteristics:



Fans

Transporting overpressure:

- less than 1 kPa – low pressure
- 1 to 3 kPa - medium pressure
- 3 to 10 kPa – high pressure

For higher overpressures – blowers and compressors

Overall fan efficiency:

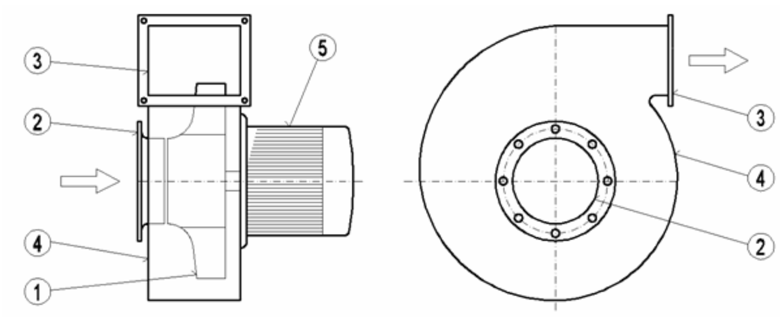
$\eta = \eta_m . \eta_i$ η_m mechanical efficiency (bearing friction, vibrations) $\eta_i = \dfrac{\Delta p_c}{\Delta p_c + \Delta p_z}$ inner efficiency (pressure losses of polytropic change)

Fan power input on the shaft:

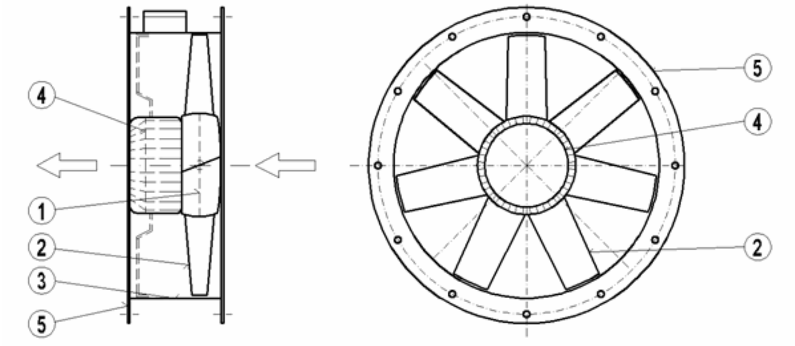
$$P_{mech} = \frac{\dot{V} . \Delta p_c}{\eta}$$

Fan classification:

Radial



Axial

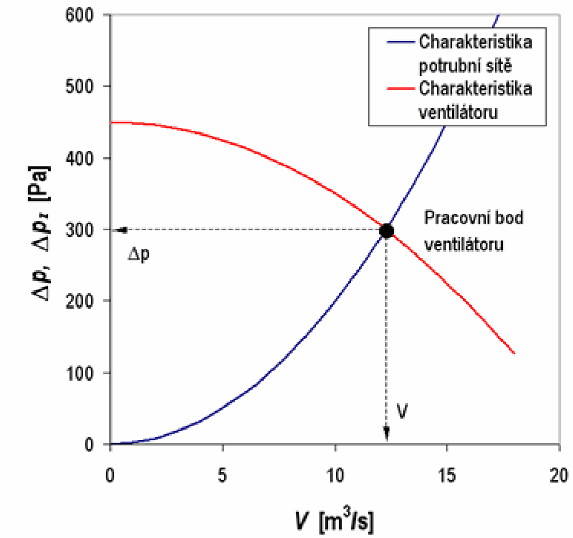


Fans

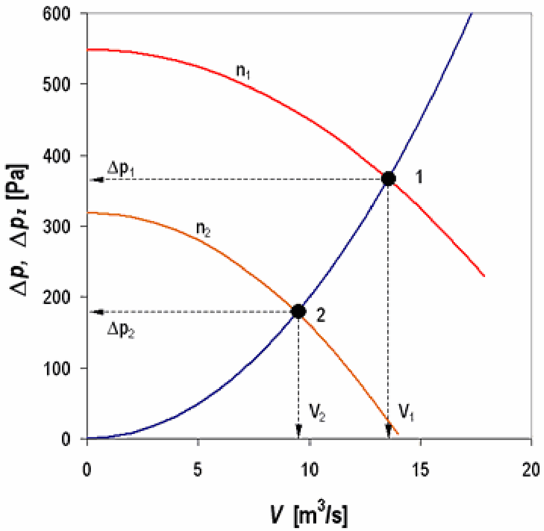
Fan operating characteristics:

Operating overpressure dependency on the volume flow at constant speed

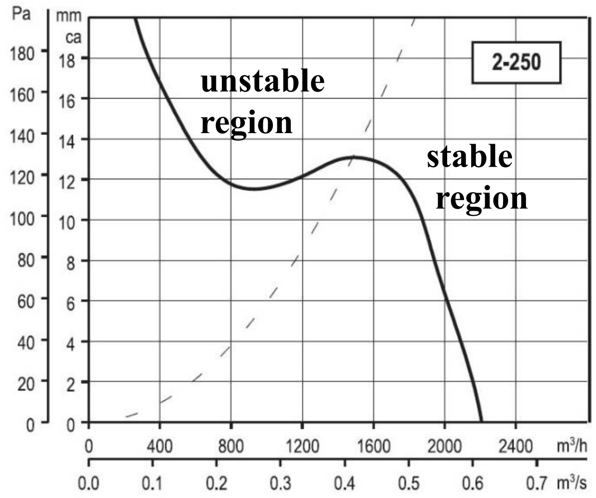
Operating point is located in the intersection of operating and pipe network characteristics



Characteristics typical for a radial fan



Change of speed influence on the new operating characteristics



Characteristic typical for an axial fan

Parametr	a) změna otáček při ρ = konst.	b) změna hustoty při n = konst.
Objemový průtok vzduchu V [m³/s]	$V_2 = V_1 \dfrac{n_2}{n_1}$	$V_2 = V_1$
Celkový dopravní tlak Δp [Pa]	$\Delta p_2 = \Delta p_1 \left(\dfrac{n_2}{n_1}\right)^2$	$\Delta p_2 = \Delta p_1 \dfrac{\rho_2}{\rho_1}$
Výkon ventilátoru P [W]	$P_2 = P_1 \left(\dfrac{n_2}{n_1}\right)^3$	$P_2 = P_1 \dfrac{\rho_2}{\rho_1}$

TPP in the Czech Republic

Power Plant	Installed Power [MW]	units	Year of start of operation	fuel	operator
Prunéřov II	1 050	5	1981 - 82	lignite	ČEZ
Počerady	1 000	5	1970 - 77	lignite	ČEZ
Chvaletice	800	4	1977 - 78	lignite	ČEZ
Dětmarovice	800	4	1975 - 76	hard coal	ČEZ
Tušimice II	800	4	1974 - 75	lignite	ČEZ
Mělník III	500	1	1981	lignite	ČEZ
Prunéřov I	440	4	1967 - 68	lignite	ČEZ
Vřesová	370	2	1996	natural gas, energo-gas	Sokolovská uhelná
Opatovice	363	6	1960 - 97	lignite	Elektrárny Opatovice
Mělník I	352	6	1961 - 95	lignite	ENERGOTRANS
Kladno - Dubská	306	4	1976 - 99	lignite, hard coal, biomass	Alpiq Generation (CZ)
Ostrava-Kunčice	254	11	1957 - 2000	hard coal, furnace gas	Arcelor Mittal
Komořany	239	8	1959 - 98	lignite, natural gas	United Energy
Mělník II	220	2	1971	lignite	ČEZ
Ledvice 2	220	2	1967	lignite	ČEZ
Vřesová (teplárna)	220	4	1967 - 91	lignite, natural gas	Sokolovská uhelná
Tisová I	184	4	1959 - 60	lignite, biomass	ČEZ
Třebovice	174	2	1961	hard coal, LFO	Dalkia
Litvínov T200	166	8	1942 - 55	lignite	Unipetrol
Poříčí	165	3	1957	lignite, hard coal, biomass	ČEZ

Hydroelectric Plants

Principle:

- Transformation of flowing water kinetic energy into turbine rotation energy
- Turbine is on the identical shaft with generator

Advantages:

- renewable source
- no air pollution, emissions
- independent on fuel or substances transportation
- peaking power source (start up to full power ~ 100 s)
- minimum of service and maintenance, remote control
- low variable costs, minimum investment risk
- can be integrated into water management system
- high lifespan

Disadvantages:

- significant initial costs, time of construction and payback period (~ 15 years)
- large flooded area need
- dependency on stable water flow

Share in the Czech Republic:

- circa 10% of electrical energy consumption

Hydroelectric Plants

By principle of water energy accumulation (swelling mechanism):

- **run-of-the-river /weirs/** (the head is made by weir)
- **derivation** (the head is made by derivation)
- **conventional /dams/** (utilizing a dam wall to make the head)
- **pumped-storage** (utilizing water pumped from lower to upper reservoir)
- **tide** (utilizing a difference between high and low tide)

By head (pressure value):

- **low pressure** (head up to 20 m)
- **medium pressure** (head 20 – 100 m)
- **high pressure** (head above 100 m)

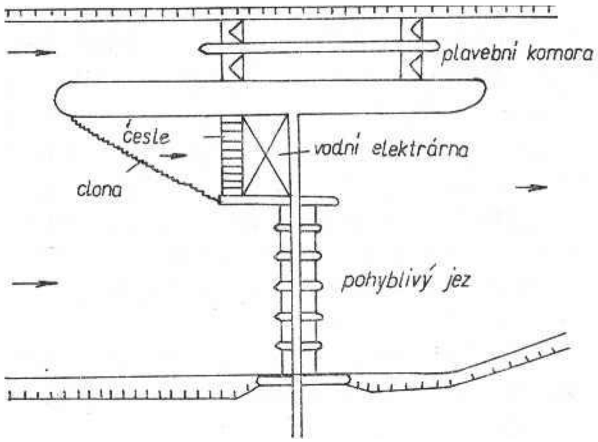
By duty in the load diagram:

- **basic** (rated power exploitation $\tau \approx$ more than 6 t. hrs/year)
- **semi-peak** (rated power exploitation $\tau \approx$ 2-4,5 t. hrs/year)
- **peak** (rated power exploitation $\tau \approx$ 0,7-1,5 t. hrs/year)

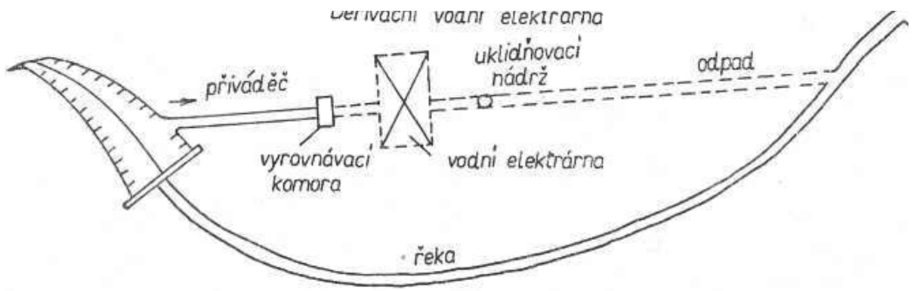
By power output:

- **small hydropower** (up to 10 MW Europe / 30 MW US / 50 MW Canada)
- **of greater output** (above 10 MW)

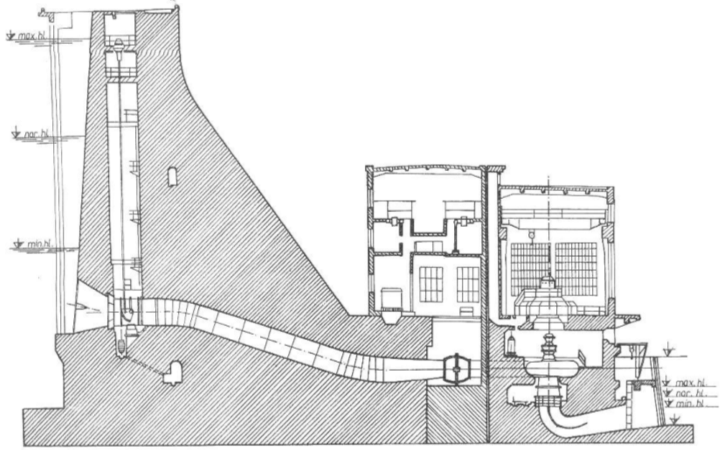
Hydroelectric Plants



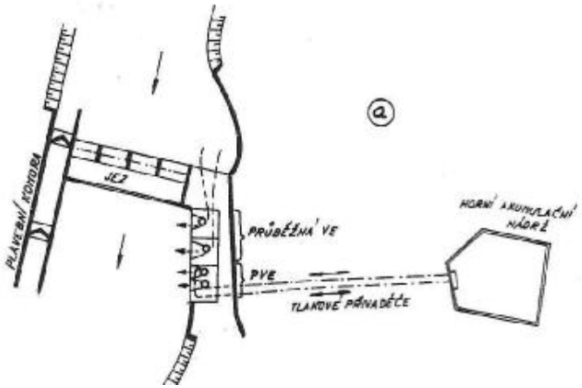
run-of-the-river (weir) HPP



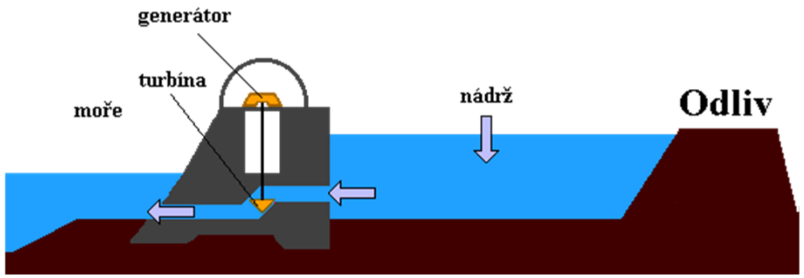
derivation HPP



dam HPP



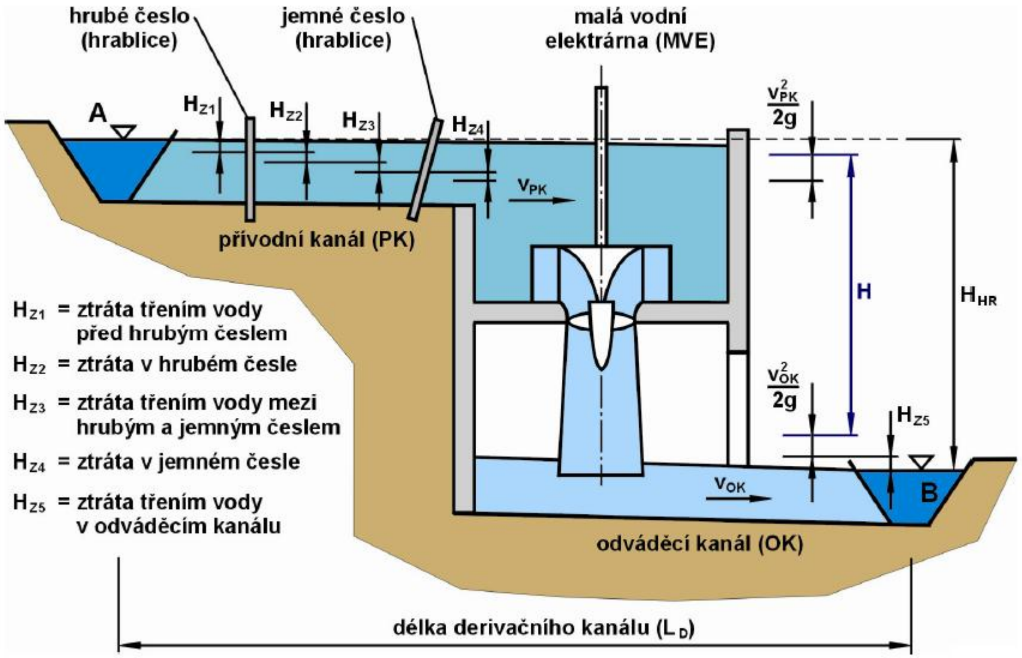
pumped-storage HPP



tide HPP

Basic Terminology and Relations

Gross head H_{HR}:
Gross (brutto) head [m] is layer difference between upper level of inflow and bottom level of outflow



Net head H:
Net (exploitable) head [m] is a gross head reduced by hydraulic losses in the inflow channel and friction losses in outlet channel

$$H = H_{HR} - \sum_i H_{zi} - \frac{v_{OK}^2}{2.g} + \frac{v_{PK}^2}{2.g} =$$

$$= H_{HR} - H_{\Delta} \qquad \text{(in steady state)}$$

Actual turbine power output:

$$P_t = \eta_o.\eta_t.\eta_m.\rho.Q_t.H.g = \eta_o.\eta_t.\eta_m.\rho.Q_t.Y$$

η_o [-] volume efficiency

η_t [-] hydraulic turbine efficiency

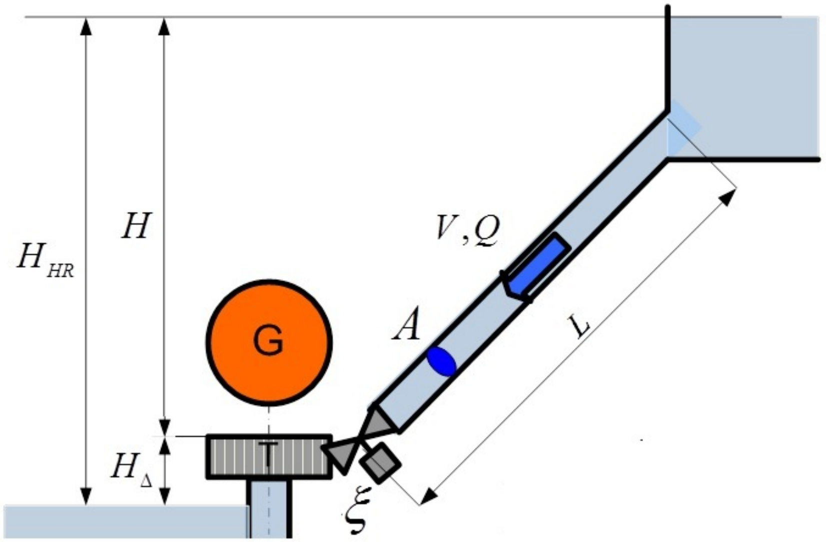
η_m [-] mechanical turbine efficiency

Q_t [m^3.s^{-1}] water flow $\qquad \dot{m}_t = Q_t.\rho$

Y [J.kg^{-1}] per kg energy $\qquad Y = H.g$

Dynamic Turbine Model

Principle:



Symbols:

Symbol of quantity	units	name
A	[m^2]	Penstock cross-section
L	[m]	Penstock length
V, v	[m/s], [p.u.]	Velocity of liquid flow
ξ	[p.u.]	Valve position /gain/ (0-closed,1-open)
P_t , p_t	[W], [p.u.]	Turbine power output
H, h	[m], [p.u.]	Hydraulic (net) head
ρ	[m^3/kg]	Volume density
K_{PQ} , K_{Pv} , K_{vH}		Proportional constant
s		Laplace operator
M	[Nm]	Torque
Q_t, q_t	[kg/s] [m^3/s]	Mass flow, volume flow
T_w	[s]	Water time constant

Dynamic Turbine Model

Turbine power output formula in p.u.:

$$P_t = \eta_o.\eta_t.\eta_m.\rho.Q_t.H.g = \eta.\rho.Q_t.H.g = K_{PQ}.Q_t.H$$

Dividing by base (nominal) quantities we obtain:

$$\frac{P_t}{P_{tN}} = \frac{P_t}{\eta.\rho.Q_{tN}.H_N.g} = \frac{Q_t}{Q_{tN}}.\frac{H}{H_N}$$ and thus $$p_t = q_t.h \quad [\text{p.u.}]$$

or:

$$P_t = \eta.\rho.A.V_t.H.g = K_{Pv}.V_t.H$$ similarly $$p_t = v_t.h \quad [\text{p.u.}]$$

Velocity of the flow formula in p.u.:

Dependent on gain (valve position) and on head:

$$V_t = \xi.\sqrt{2.g.H} = K_{vH}.\xi.\sqrt{H}$$ and thus $$v_t = \xi.\sqrt{h} \quad [\text{p.u.}]$$

$$h = \left(\frac{v_t}{\xi}\right)^2 \quad [\text{p.u.}]$$

Result: $p_t = v_t.h = \xi.h^{\frac{3}{2}} \quad [\text{p.u.}]$ a $q_t = v_t \quad [\text{p.u.}]$

Dynamic Turbine Model

Linear model:

Simplification: stiff penstock, non-compressible water, can be used only for small changes

From forces' equality:

$$\rho LA.\frac{dV_t}{dt} = \underbrace{\overbrace{(H_{HR} - H - H_\Delta)\rho g}^{p} A}_{F} \qquad \frac{dV_t}{dt} = -\Delta H.\frac{g}{L}$$

in p.u.: $$\frac{dv_t}{dt} = \frac{d\Delta v_t}{dt} = -\Delta h.\frac{g.V_{tN}}{L.H_N} = -\frac{\Delta h}{T_{wN}}$$ where $$T_{wN} = \frac{V_N.L}{g.H_N} = \frac{L.Q_N}{g.A.H_N}$$

Where ***T_{wN}*** is time constant, expressing duration of water acceleration from zero to the nominal speed (flow) ***V_N*** (***Q_N***) at nominal head ***H_N***. Typically ***T_{wN}* = 0,5 – 4 s.**

With operator use: $$\Delta v_t = -\frac{\Delta h}{T_{wN}.s}$$

Expression of differentials:

$$\Delta v_t = \left(\frac{\partial v_t}{\partial \xi}\right)_0 .\Delta\xi + \left(\frac{\partial v_t}{\partial h}\right)_0 .\Delta h \qquad \Delta p_t = \left(\frac{\partial p_t}{\partial v_t}\right)_0 .\Delta v_t + \left(\frac{\partial p_t}{\partial h}\right)_0 .\Delta h$$

Dynamic Turbine Model

thus:

$$\Delta\xi = \frac{\Delta \mathrm{v}_t - \left(\frac{\partial v_t}{\partial h}\right)_0 .\Delta \mathrm{h}}{\left(\frac{\partial v_t}{\partial \xi}\right)_0} = \frac{\left(1 + T_{wN}.s.\left(\frac{\partial v_t}{\partial h}\right)_0\right)}{\left(\frac{\partial v_t}{\partial \xi}\right)_0}.\Delta \mathrm{v}_t$$

and

$$\Delta \mathrm{p}_t = \left(\left(\frac{\partial p_t}{\partial v_t}\right)_0 - T_{wN}.s.\left(\frac{\partial p_t}{\partial h}\right)_0\right).\Delta \mathrm{v}_t$$

Partial derivatives expressions:

$$\left(\frac{\partial v_t}{\partial h}\right)_0 = \left(\frac{\partial\left(\xi.\sqrt{h}\right)}{\partial h}\right)_0 = \frac{\xi_0}{2.\sqrt{h_0}}$$

$$\left(\frac{\partial v_t}{\partial \xi}\right)_0 = \left(\frac{\partial\left(\xi.\sqrt{h}\right)}{\partial \xi}\right)_0 = \sqrt{h_0}$$

$$\left(\frac{\partial p_t}{\partial v_t}\right)_0 = \left(\frac{\partial\left(v_t.h\right)}{\partial v_t}\right)_0 = h_0$$

$$\left(\frac{\partial p_t}{\partial h}\right)_0 = \left(\frac{\partial\left(v_t.h\right)}{\partial h}\right)_0 = v_{t0} = \xi_0.\sqrt{h_0}$$

Dynamic Turbine Model

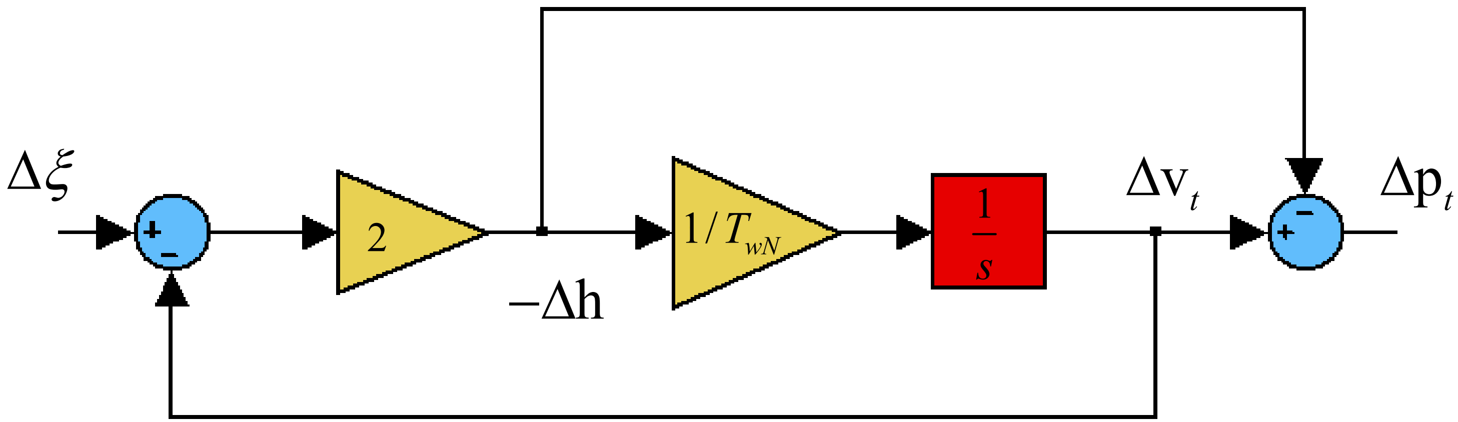
Resulting transfer function between turbine power output and gain:

$$\frac{\Delta \mathrm{p}_t}{\Delta \xi} = \frac{\left(\left(\frac{\partial p_t}{\partial v_t}\right)_0 - T_{wN}.s.\left(\frac{\partial p_t}{\partial h}\right)_0\right).\left(\frac{\partial v_t}{\partial \xi}\right)_0}{1 + T_{wN}.s.\left(\frac{\partial v_t}{\partial h}\right)_0} = \frac{\left(h_0 - T_{wN}.s.\xi_0.\sqrt{h_0}\right).\sqrt{h_0}}{1 + T_{wN}.s.\frac{\xi_0}{2.\sqrt{h_0}}}$$

If the turbine is operating close to nominal parameters, i.e. $h_0 \to 1$ and $\xi_0 \to 1$

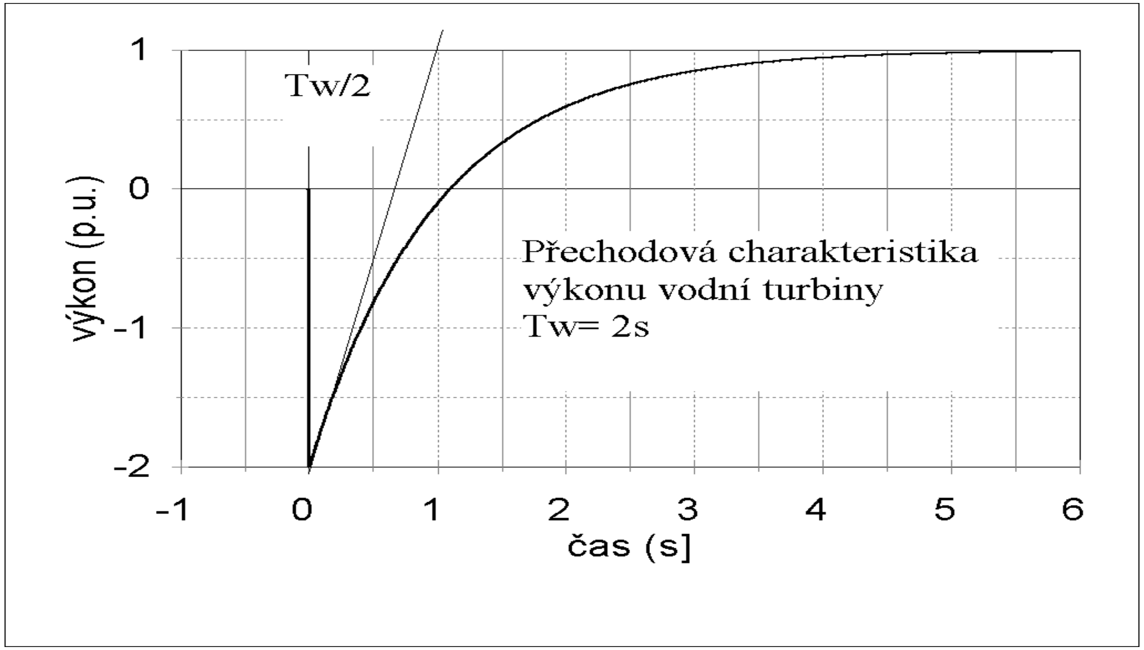
The transfer function will be: $\frac{\Delta \mathrm{p}_t}{\Delta \xi} = \frac{1 - T_{wN}.s}{1 + 0,5.T_{wN}.s}$

Corresponding model:



Dynamic Turbine Model

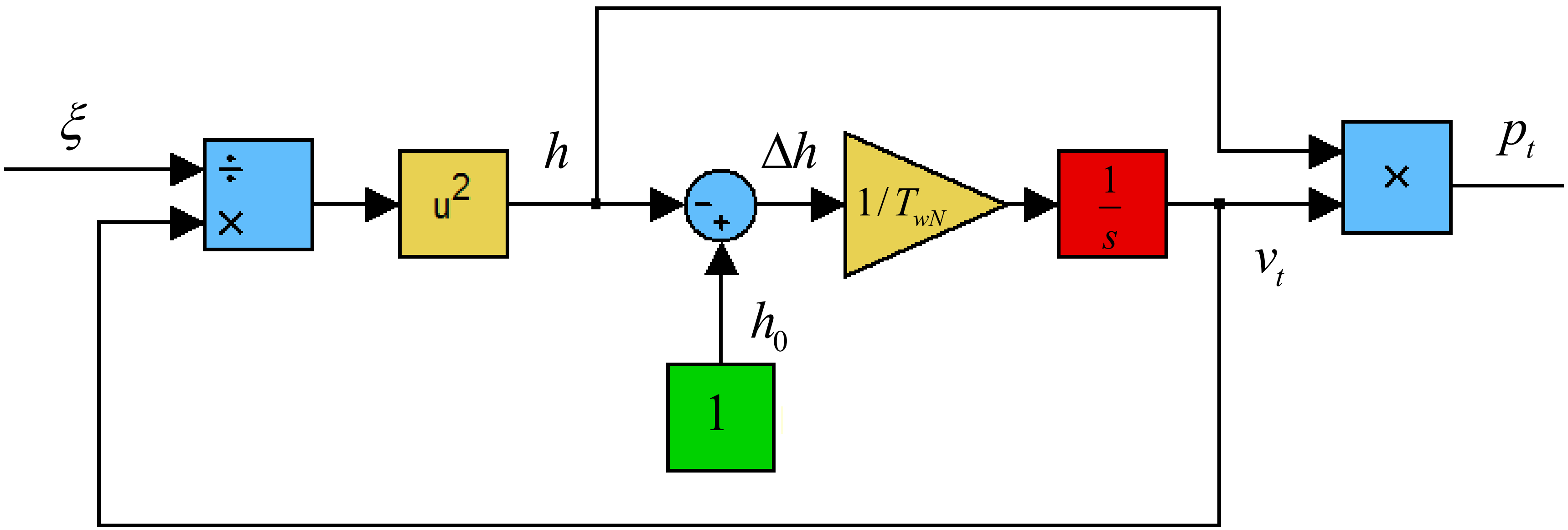
Important character of the turbine:
Initially, after fast valve opening, turbine power output is decreasing (**reverse change!**) and then is normally increasing.
Flow velocity in the penstock remains constant at initial phase due to water inertia. (the water then is accelerating).
Unlike the turbine pressure drops immediately and causes power drop.



Dynamic Turbine Model

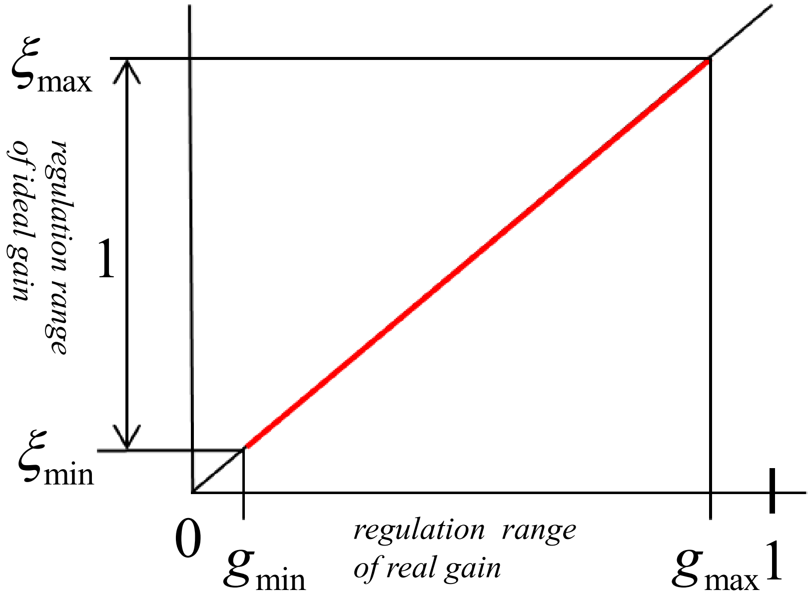
<u>Non-linear model:</u>

Simplification: stiff penstock, non-compressible water, speed and head are constant
application: for a wide range of flow



Dynamic Turbine Model

Real gain influence:



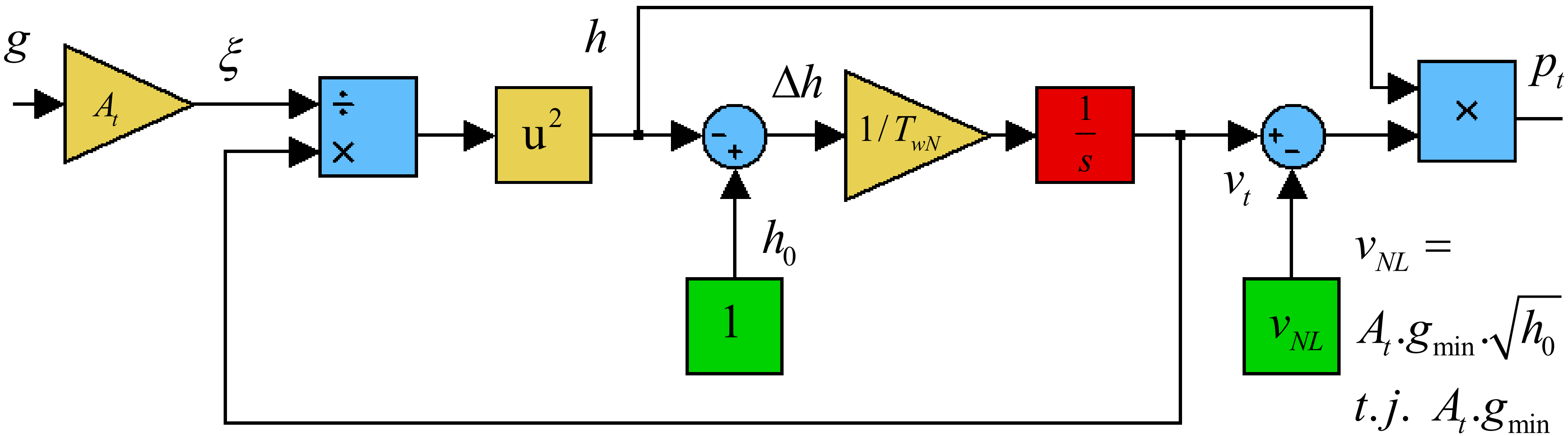
In fact the lower limit of real regulation range is greater than zero. The lower limit has to ensure flow at rated speed under no-load v_{NL} (q_{NL}).

Relation between an ideal and real valve position is thus:

$$\xi = A_t.g$$

where A_t is turbine gain constant

$$A_t = \frac{1}{g_{\max} - g_{\min}}$$



$v_{NL} =$

$A_t.g_{\min}.\sqrt{h_0}$

$t.j.\ A_t.g_{\min}$

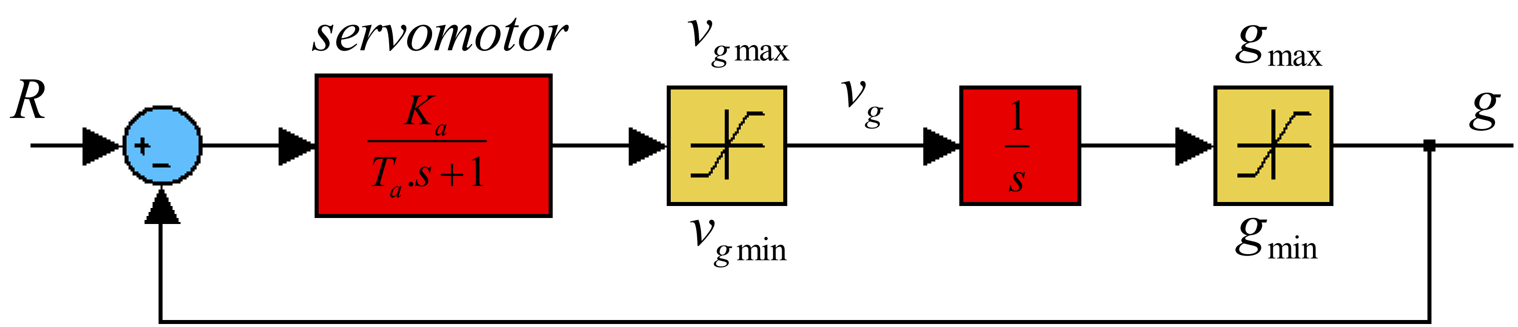
Valve Regulation Model

Valve opening mechanism:
Servomotor is opened/closed by a valve, an opening velocity is limited $v_{g\,\min} \div v_{g\,\max}$

A valve can be opened in range $g_{\min} \div g_{\max}$

For opening velocity we obtain

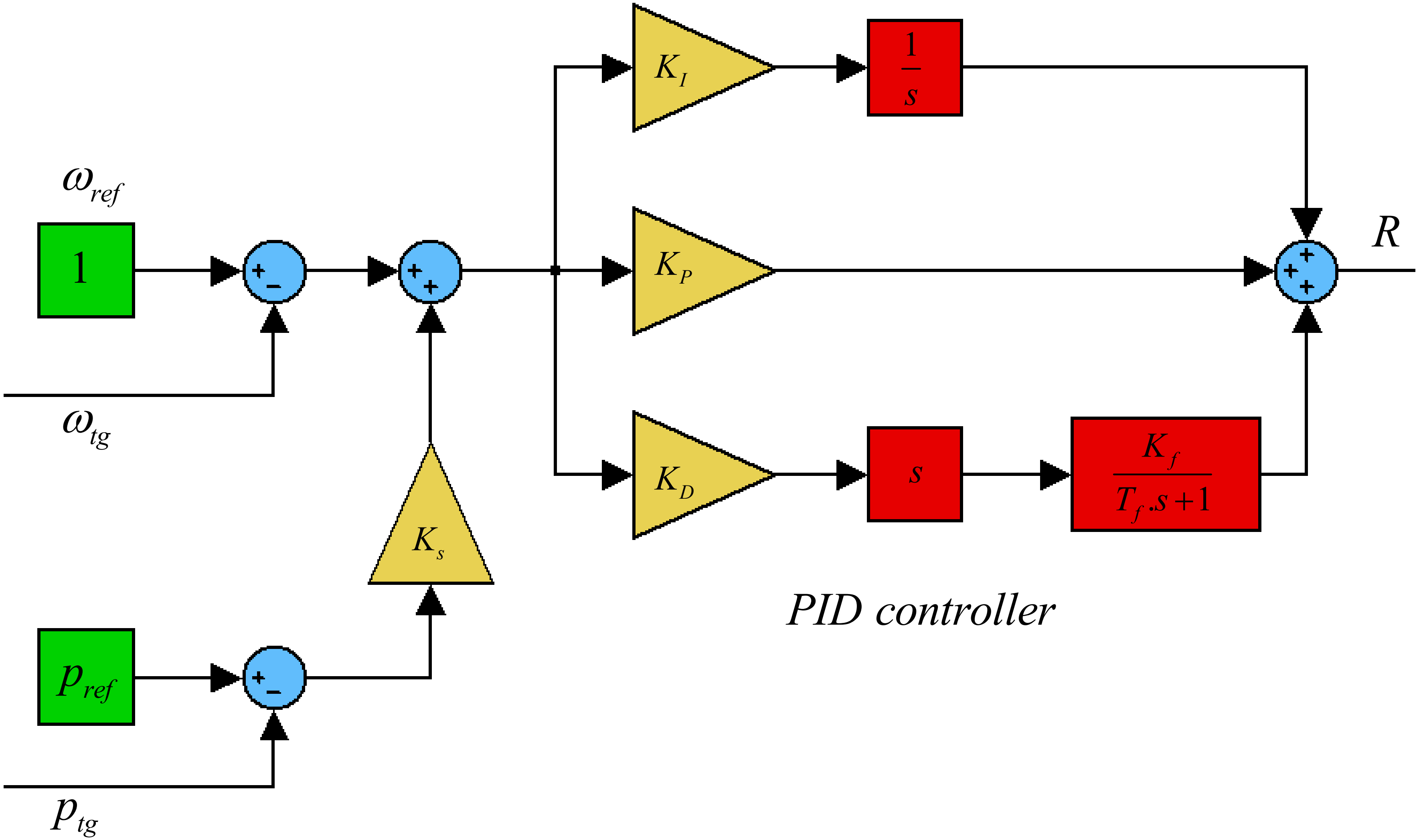
$$\frac{dg}{dt} = v_g$$



K_a [-] servomotor gain

T_a [s] servomotor time constant

Dynamic Model of Turbine Control



PID controller

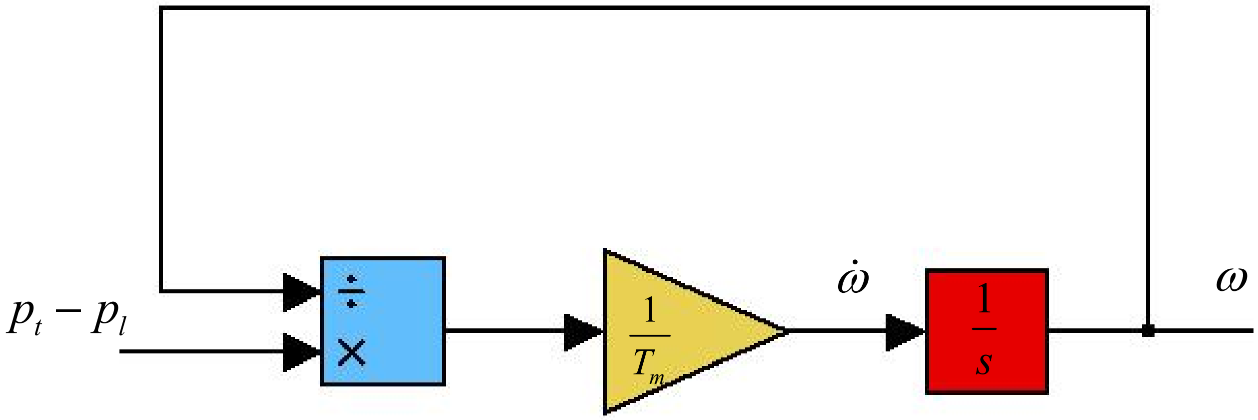
Dynamic Generator Model

For a rotating machine in p.u.:

$$p_t - p_l = \frac{J.\Omega_n^2}{S_n}.\dot{\omega}.\omega = \frac{GD^2.\Omega_n^2}{4.S_n}.\dot{\omega}.\omega = T_m.\dot{\omega}.\omega$$

If $\omega \approx 1$ we can simplify to

$$p_t - p_l = T_m.\dot{\omega}$$



p_l [-] load active power

T_m [s] mechanical time constant of turbo-generator set

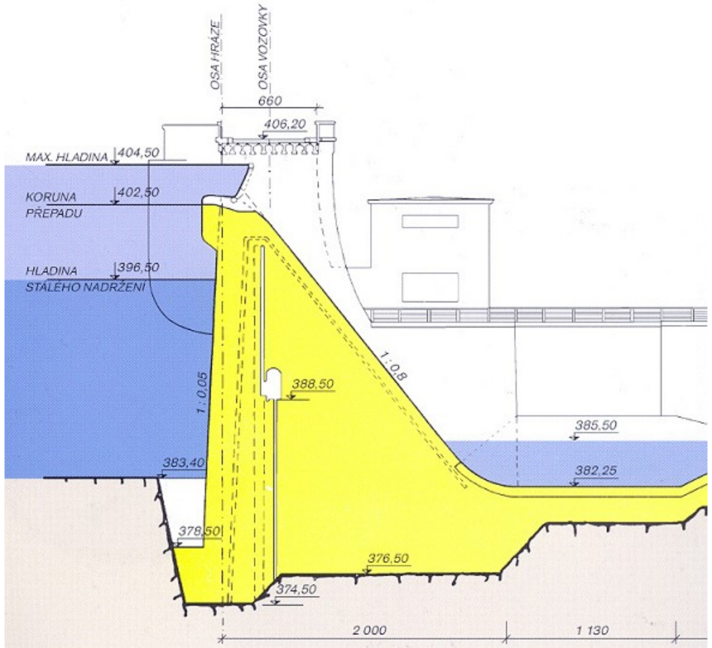
Hydroelectric Plants

HPP components:

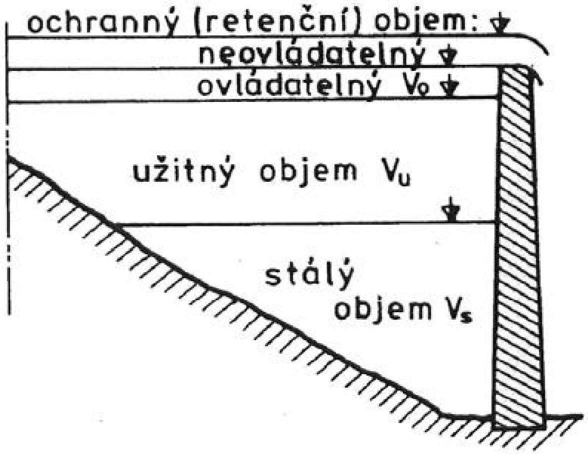
- **inflow**
- **penstock**
- **surge tower (surge tank)**
- **penstock to the turbine**
- **spiral case**
- **hydraulic turbine**
- **draft tube**
- **outflow (tailrace)**

HPP Elements

Dam:

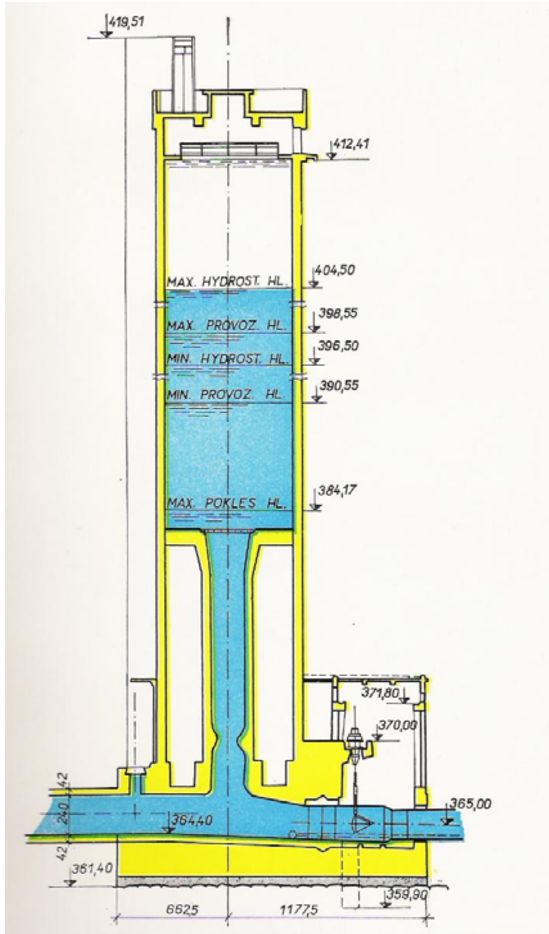


Křižanovice I Dam (HPP Práčov)



Surge tower:

Surge tanks are used for dissipate the water hammer pressure or dynamic transients on turbine-generator set in case of high head HPPs.

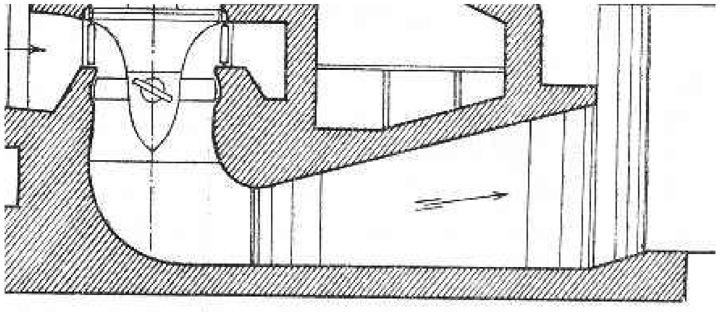


Vyrovnávací komora v přivaděči k VE Práčov

HPP Elements

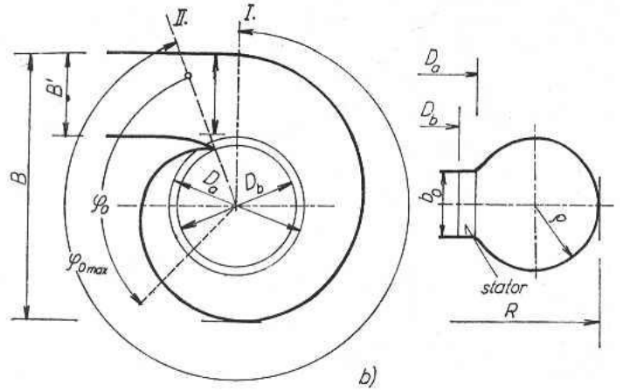
Draft tube:

Divergent piping leads under tail race water level. Water flow in the tube is generating underpressure on lower parts of the turbine blades (reaction turbines only). Relative speed is thus increasing along the blade of the turbine rotary wheel. Total absolute outflow speed is therefore reduced and more energy is transferred to the turbine. By means of this effect, the turbine can exploit spare head coming from turbine and tail race height difference

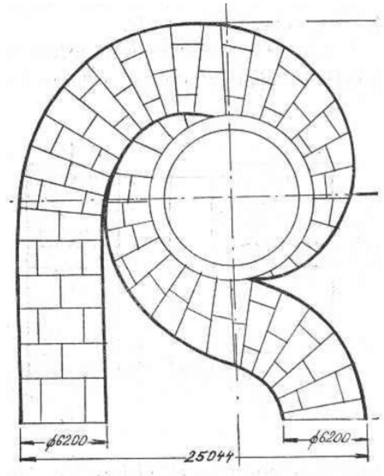


Spiral case:

a housing designed to provide uniform water intake around the entire circumference of the distributor blades.



single spiral case



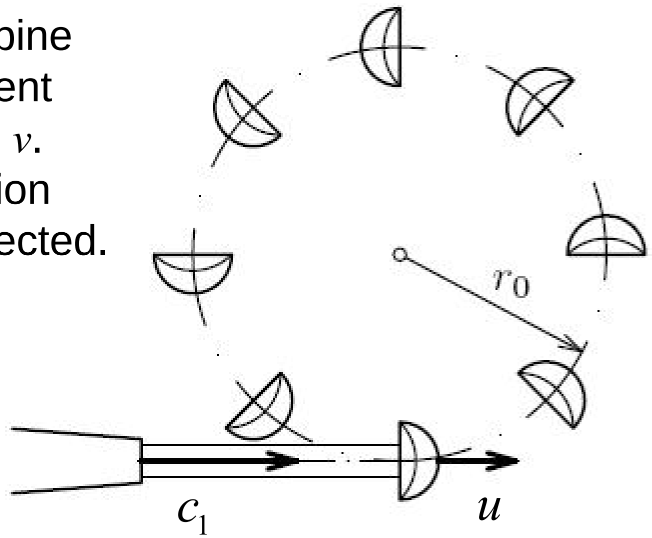
double spiral case

Water Turbines - Theory

<u>Idealized Pelton Turbine:</u>

Water is flowing into turbine with velocity c_1. A rotating turbine blade is moving with perimeter velocity u, relative movement of water towards a blade is expressed with relative speed v. The relative speed changes from input to output its direction by 180°. Water flow Q is constant and friction losses neglected. From relative speeds balance, the blade reaction force is:

$$2.Q.(c_1 - u) = R$$



Turbine power output:

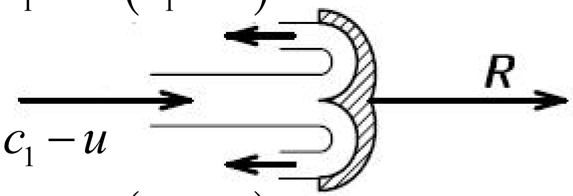
$$P = R.u = 2.Q.(c_1 - u).u$$

Power output maximum:

$$\frac{dP}{du} = 0 \Rightarrow u_{max} = \frac{c_1}{2}$$

Relative blade speed:

$$v_2 = -v_1 = -(c_1 - u)$$



$$v_1 = c_1 - u$$

$$v_2 = -v_1 = -(c_1 - u)$$

Absolute outflow velocity:

$$c_2 = v_2 + u = -(c_1 - u) + u = -(c_1 - 2.u)$$

Hydraulic efficiency:

$$\eta_t = 1 - \frac{P_{ztr}}{P_{přiv}} = 1 - \frac{(c_1 - 2.u)^2}{c_1^2}$$

a pro $c_2\big|_{P=P_{max}} = -\left(c_1 - 2.\frac{c_1}{2}\right) = 0$ je $\eta_t = 1$

Water Turbines - Theory

Real phenomenon description is more complicated: depending on inflow and outflow angle, the velocities must be added as vectors, the velocity on perimeter is not constant due to real shape of the blade, relative speed on rotary wheel is changing with variable pressure

<u>Euler turbine equation, general formula:</u>

$$Y = c_{1(u1)}.u_1 - c_{2(u2)}.u_2$$

Adding relative energy losses increment we obtain

$$Y_{id} = c_{1(u1)}.u_1 - c_{2(u2)}.u_2 + Y_{ztr}$$

<u>Hydraulic efficiency is thus:</u>

$$\eta_t = \frac{Y}{Y_{id}}$$

Water Turbines - Theory

Classification by the kinetic water energy way of transformation:

- **impulse**
- **reaction**

Impulse turbines:

Water inputs to a *distributor*, where its pressure energy is turning to kinetic energy. The outflowing water is optimally directed for input to the opposite positioned blades in *runner* thanks to good shaping of the distributor blade. Water inflows to the runner at atmospheric pressure. Water pressure is thus constant along the runner path and *relative velocity towards the runner changes only in direction, but absolute value remains constant*. After transferring majority of its kinetic energy, the water flows out with residual velocity.

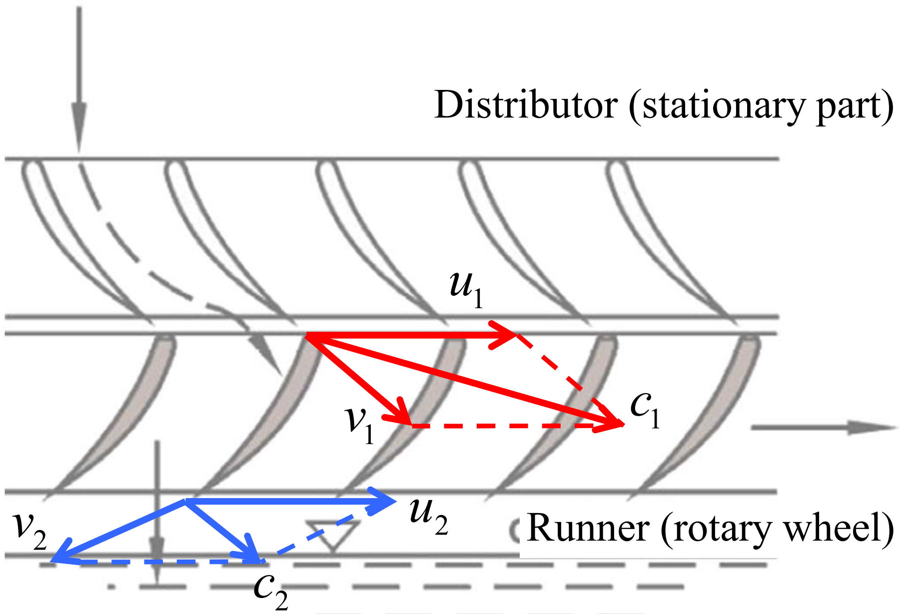
Reaction turbines:

Water inputs to a *distributor*, where its pressure energy is turning to kinetic energy. Thanks to employing of a draft tube, there is pressure difference between input and output of the rotary blades. This pressure difference accelerates the water in runner and thus the absolute value of relative velocity is growing. This causes reduction of absolute outflow speed and The turbine is able to exploit the turbine and tail race height difference.

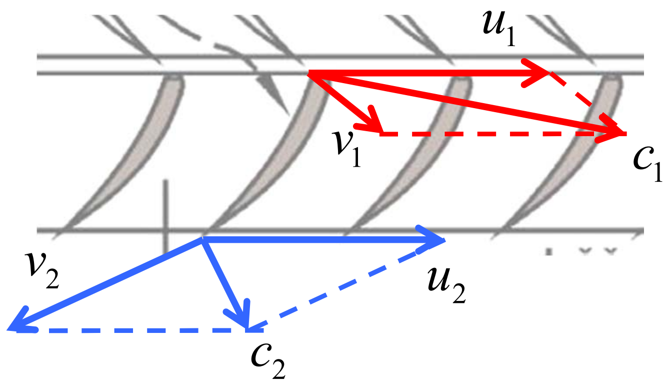
Rem. Impulse and reaction turbines classification is thus principally identical as in the case of steam turbines. However, water is almost uncompressible and the phenomenon is described by continuity equation

Water Turbines - Theory

<u>Impulse turbine:</u>



<u>Reaction turbine:</u>



c_1 $[\mathrm{m.s^{-1}}]$ absolute inflow water velocity

v_1 $[\mathrm{m.s^{-1}}]$ relative inflow water velocity

u $[\mathrm{m.s^{-1}}]$ velocity on perimeter

c_2 $[\mathrm{m.s^{-1}}]$ absolute outflow water velocity

v_2 $[\mathrm{m.s^{-1}}]$ relative outflow water velocity

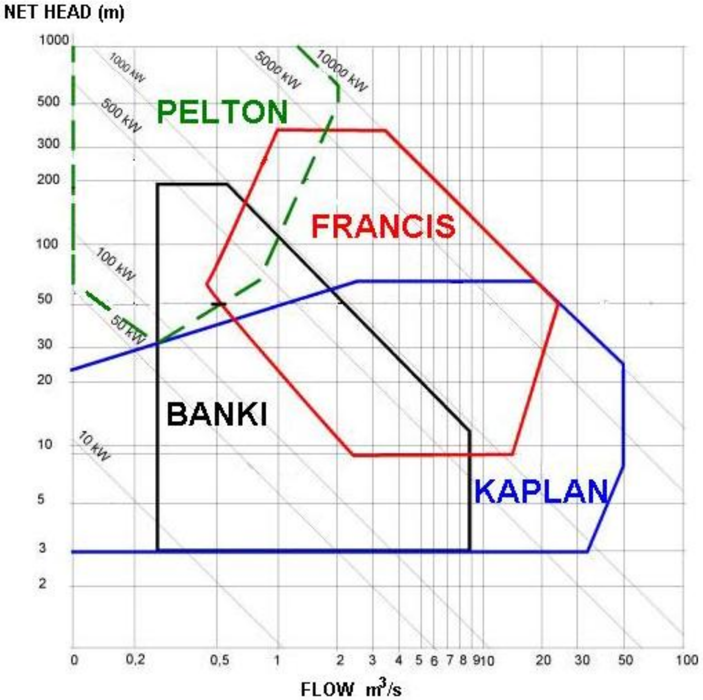
Rem. "relative" means related to runner position

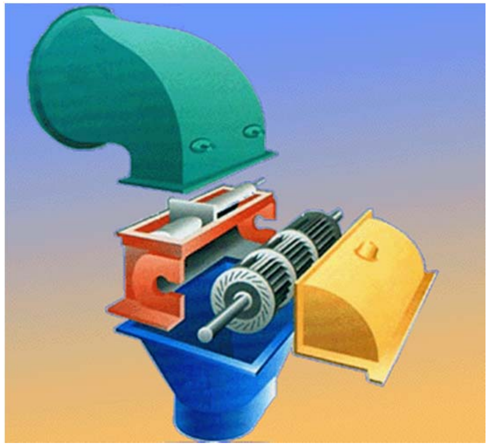
Water Turbines - Construction

The most common turbine types

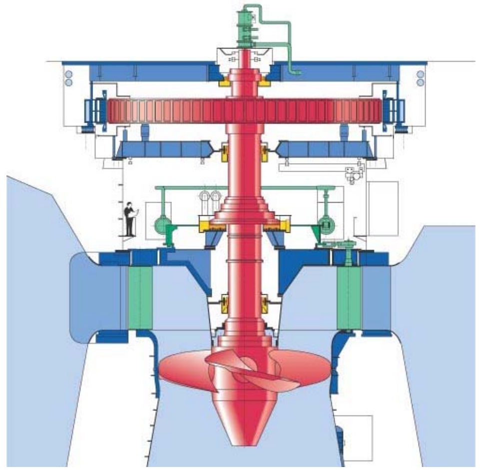
- **Banki (Cross-flow)**
- **Kaplan**
- **Francis**
- **Pelton**

Turbine selection by net head and flow rate:

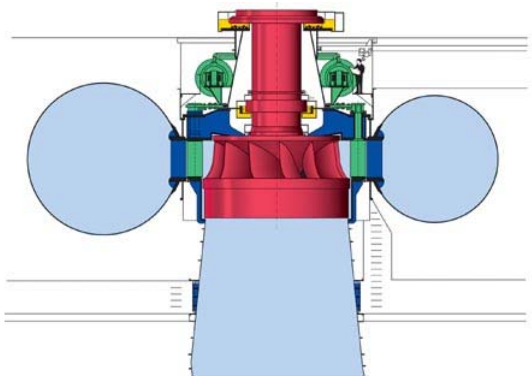




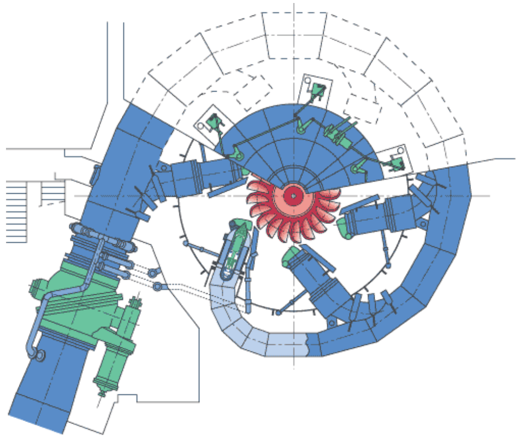
Banki turbine



Kaplan turbine



Francis turbine

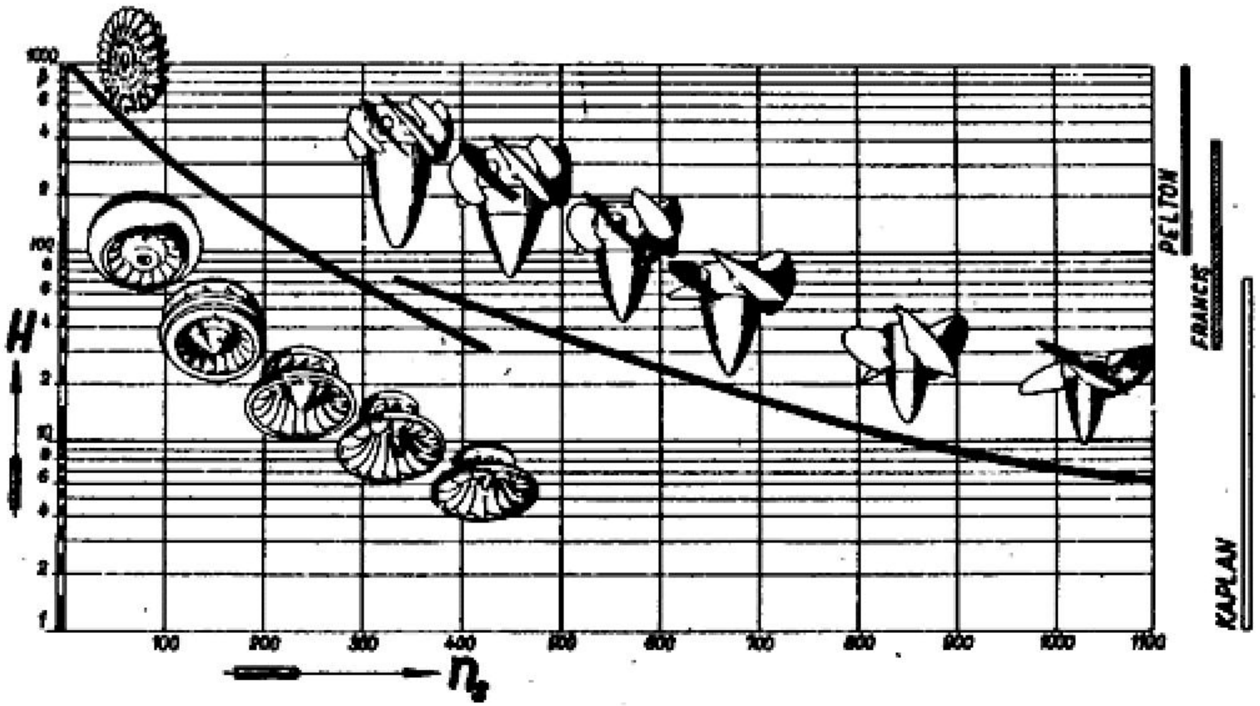


Pelton turbine

Water Turbines - Construction

<u>Turbine selection by speed and head range:</u>

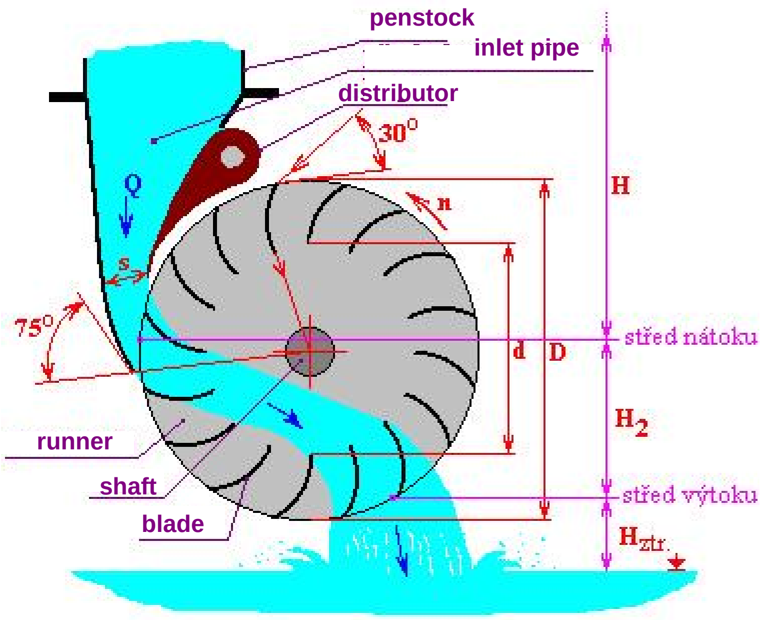
Turbine type	arrangement	n_s
Pelton	with one nozzle	4 - 35
	with two nozzles	17 - 50
	with four nozzles	24 - 70
Francis	low speed	80 - 120
	standard	120 - 220
	high speed	220 - 350
	express	350 - 450
Kaplan		450 - 1000



Banki (Cross-Flow) Turbine

Descriptrion:

- **impulse** turbine
- A.G.M. Mitchel (1903), F. Ossenberger, D. Banki in 1918
- Water inlet from penstock pipe is of circular cross-section
- The last stage in front of the turbine water flows through special inlet pipe which is adapting the flow from circular to rectangular cross section, the pipe is terminated with distributor (regulation)
- Kinetic energy of the water is transferred in two places
- At first blade crossing transfers the water flow circa **79**%, at second circa **21**% of total turbine power
- Full exploitation of the head H**,** partially H_2„ height difference between runner end and water level H_{ztr} is lost completely



H	active head
H_2	head in wheel
H_{ztr}	head above water level (loss)
D	outer wheel diameter
d	blade endings diameter

Utilization

- low flow rates and small and intermediate heads
- cost effective solution

Francis Turbine

Description:

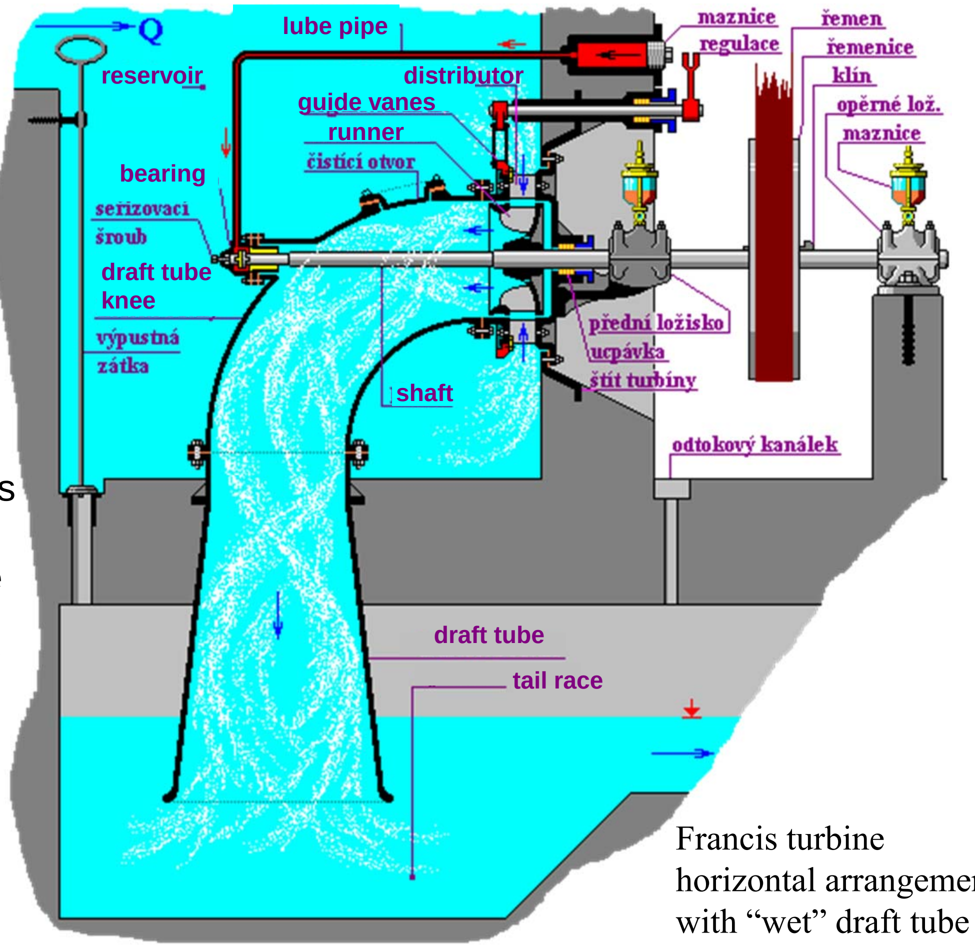
- **reaction** turbine
- J. B. Francis (1848)
- *draft tube effect* allows whole head exploitation even if the runner is located high above tail race water level
- Two modifications: vertical and horizontal

Horizontal arrangement

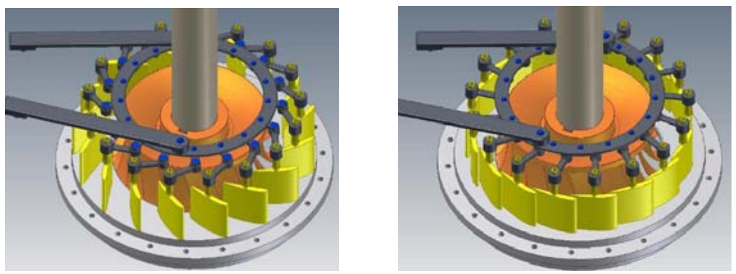
- Turbine is placed inside the wall of the water reservoir
- Water gets from reservoir into the guide vanes around the whole turbine circumference
- The turbine in the reservoir wall is high above lower water level -> *knee of draft tube*
- A knee can be led inside the reservoir – *"wet" draft tube*
- or turbine hall – *"dry" draft tube*

Vertical arrangement

- Turbine is located at the bottom of the water reservoir
- The shaft is continuing vertically to the turbine hall – no risk of flooding
- Water gets from reservoir into the guide vanes around the whole turbine circumference
- Water outlet is realized by means of the draft tube, which generates underpressure on outflowing side of the runner

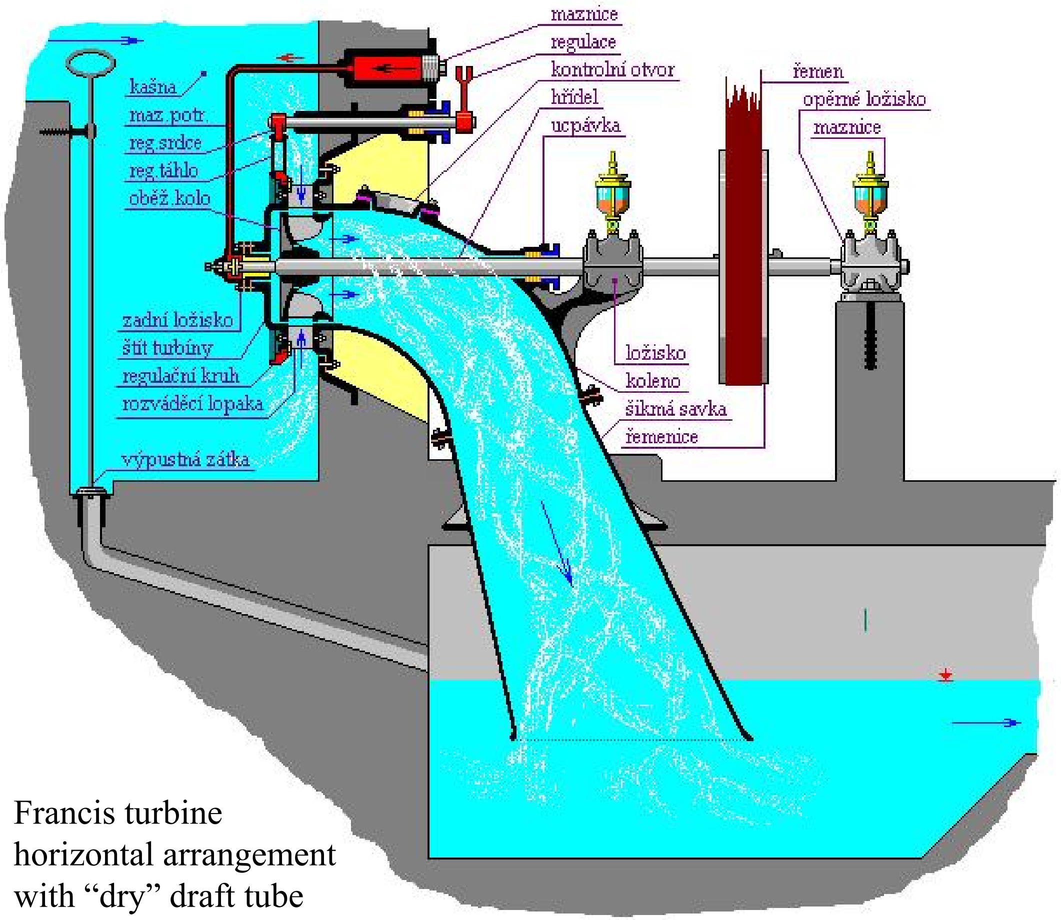


Francis turbine horizontal arrangement with "wet" draft tube

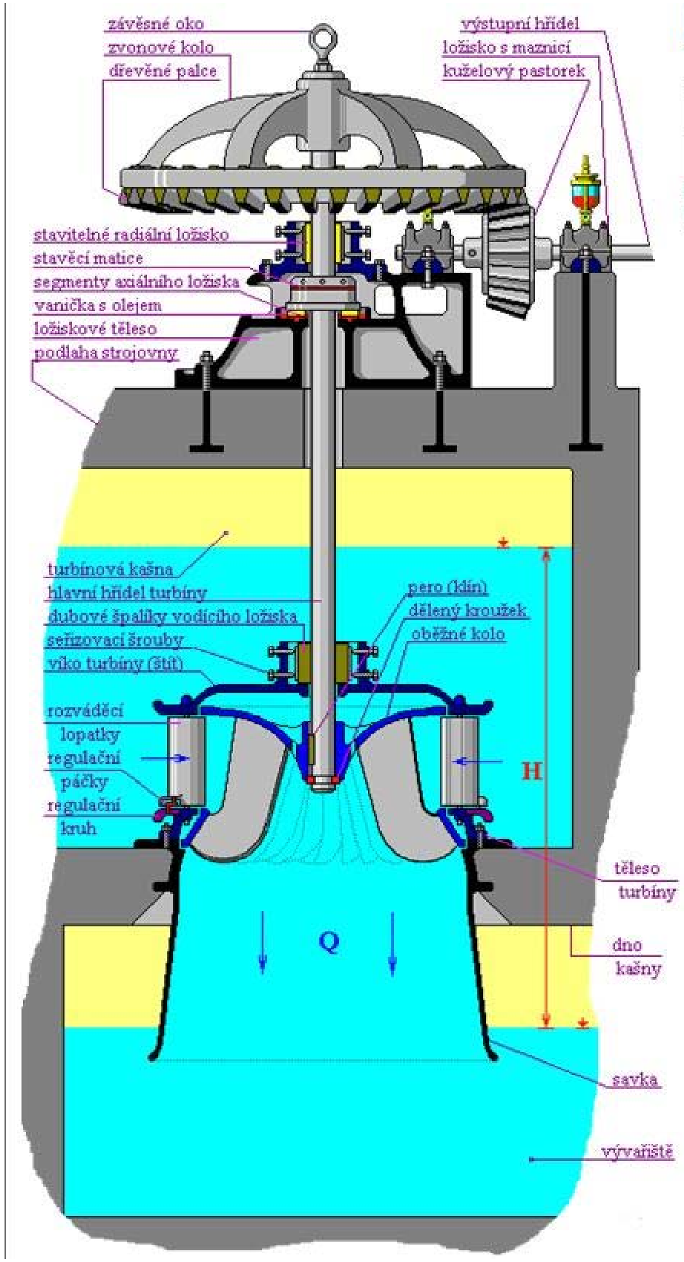


Guide vanes of Francis turbine

Francis Turbine



Francis turbine
horizontal arrangement
with "dry" draft tube



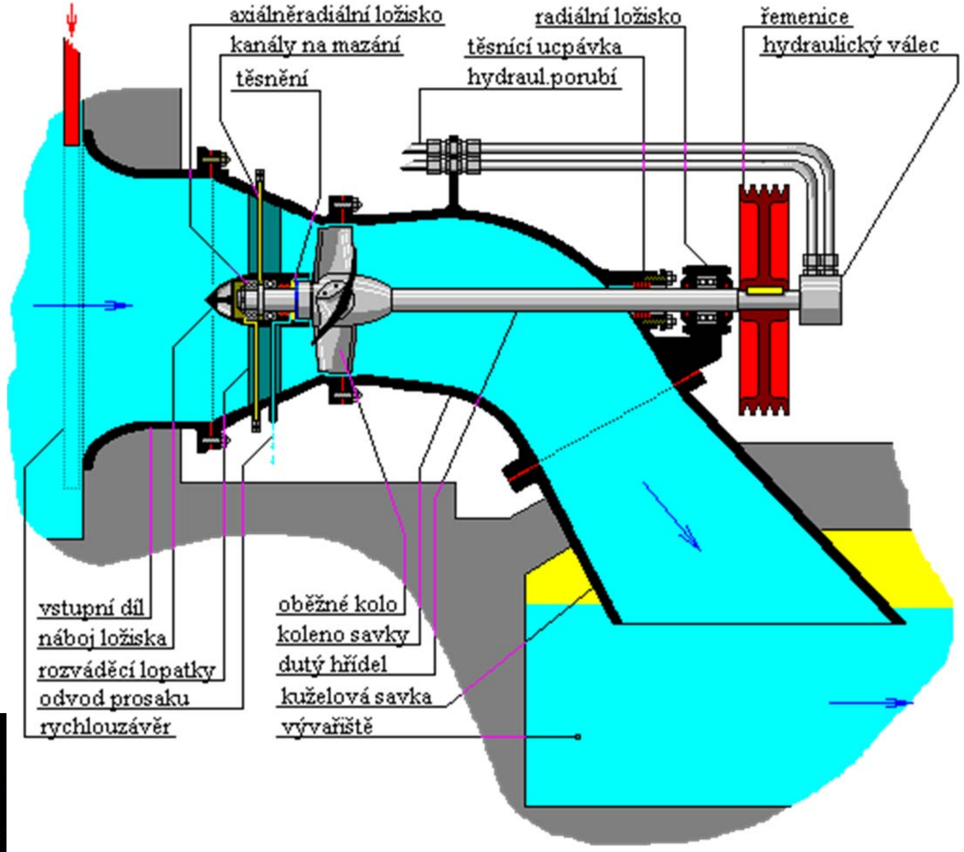
Francis turbine, vertical arrangement

<u>Utilization</u>

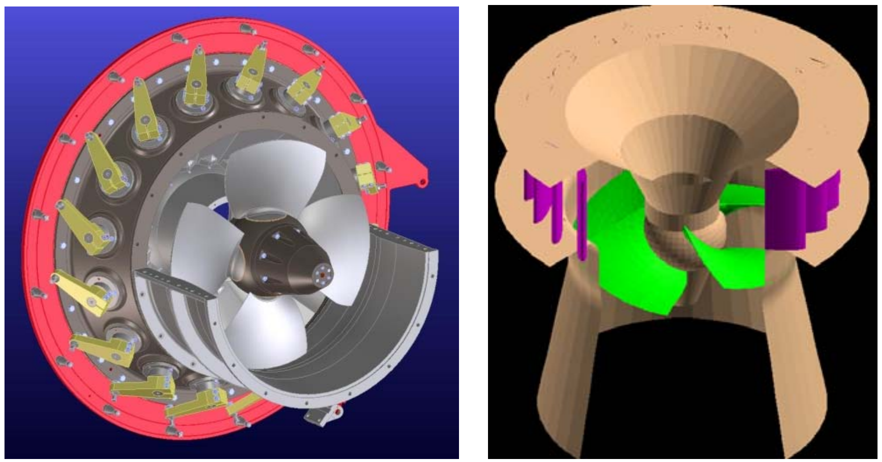
- For intermediate heads and greater flow rates
- Francis reversible turbine – used in pumped-storages
- efficiency 75 – 90%

Kaplan Turbine

<u>Description:</u>
- **Reaction** turbine
- V. Kaplan (1912-13)
- upgraded Francis turbine, runner is propeller-shaped with *adjustable blades*
- typical efficiency above 90% in wide range of flow rates (up to 95%)
- the highest rated speeds among all turbine types
- **Kaplan-S:** guide vanes + rotary blades regulation
- **Semi-Kaplan:** runner rotary blades regulation only



Semi-Kaplan type turbine

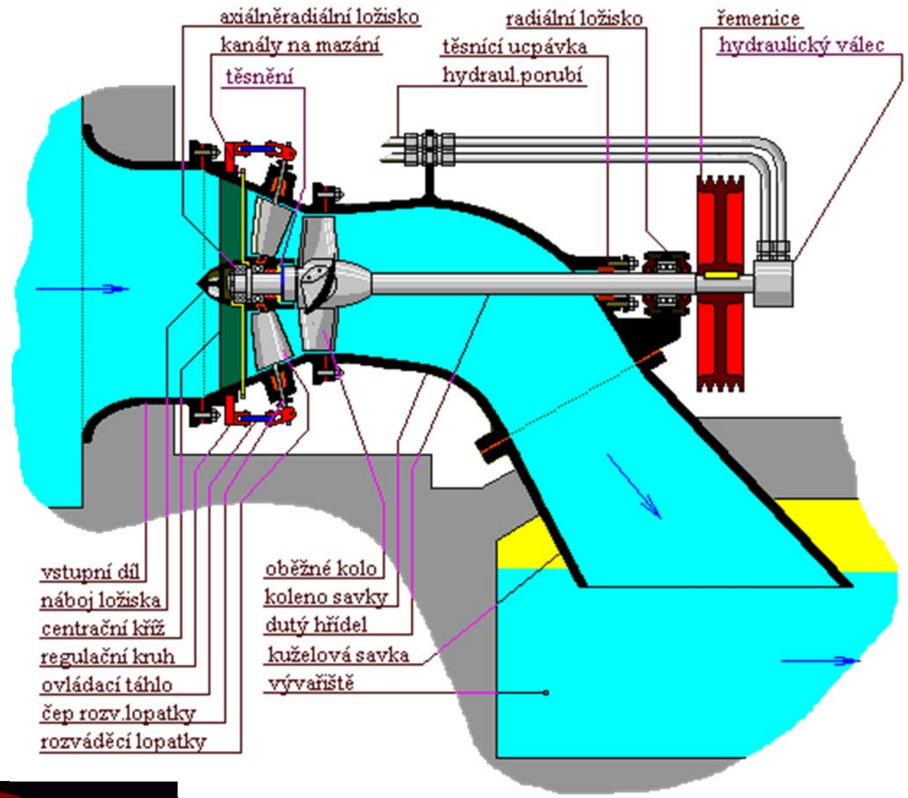


Regulation elements of Kaplan turbine

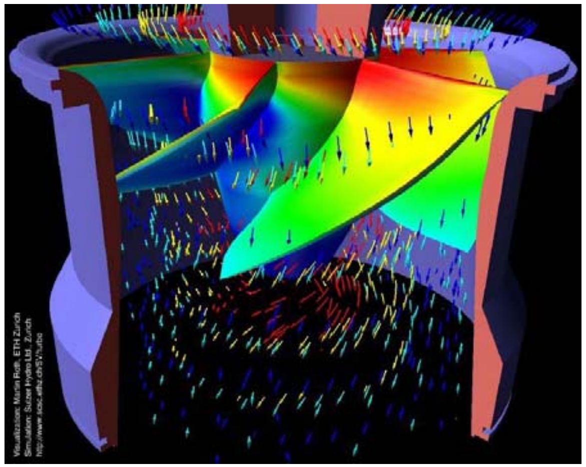
Kaplan Turbine

<u>Utilization:</u>

- water flows with variable flow rate
  (double regulation advantage)
- greater investment costs and more expensive maintenance
  (double regulation disadvantage)
- for low heads and greater flow rates



Kaplan-S type turbine



Pressures and velocities on Kaplan and Francis turbine

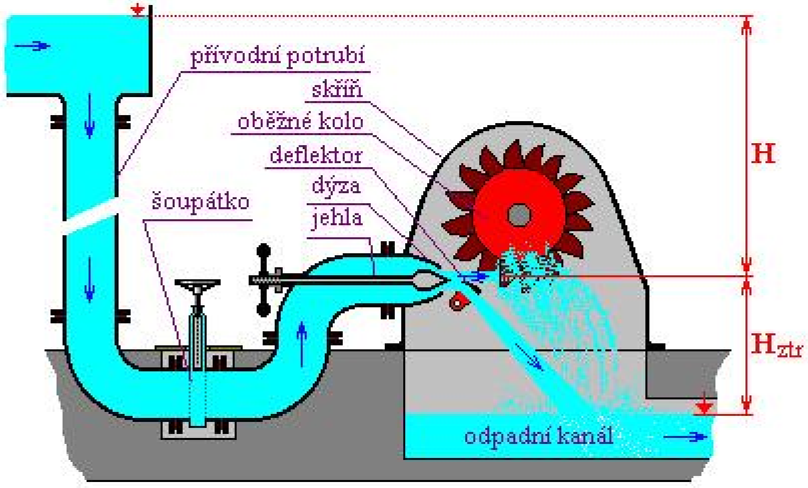
Pelton Turbine

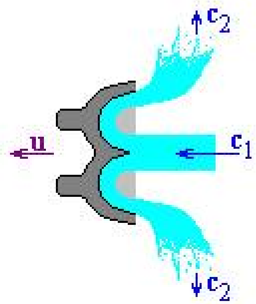
Description:

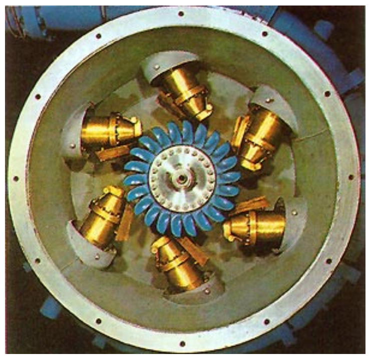
- **impulse** turbine
- L. A. Pelton **(**1880**)**
- water inlet from penstock pipe is of circular cross-section which distributes water to one or more nozzles
- water from the nozzle is flowing with a velocity of c_1, water is getting tangentially to the runner equipped with buckets (special spoon-shaped blades)
- the edge in the middle of the bucket divides water jet into two halves and spoon shape changes water flow to the opposite direction
- Thanks to a mutual concurrence of water velocity on the bucket surface and a velocity at runner circumference u, the water is leaving the buckets on their outer side with minimal residual speed c_2
- efficiency of small turbines is **80 - 85%**, for greater power outputs **85 - 95%**
- fully exploited head is depicted as H, height difference H_{ztr} is representing losses

Utilization:

- for small water flow rates at highest head levels
- without cavitation problems, sand abrasion endurance
- low head levels are resulting to low rpm, gearbox is thus needed







Nozzles of Pelton turbine

Cavitation

Cavitation phenomenon:

Cavitation is a process of generating “bubbles” in water (water starts to boil) under local pressure decrease, followed by their implosion. Pressure drop might be caused by local speed increase (hydrodynamic cavitation), eventually by passing an intensive acoustic wave in the periods of dilution (acoustic cavitation). Cavitation bubble is at initial phase filled by vacuum, later by vapor from nearby liquid or, if dissolved, by gases from the liquid. After pressure increase, the bubble is collapsing accompanied by a transient wave with *damaging effect on surrounding material*.

Affecting predominantly **reaction turbines**, mostly present in runner, causing material damage by stress corrosion. Cavitation might be accompanied by *local heating, acoustic effects, vibrations and luminescence*. The cavities are imploding with high speed (up to 300 $m \cdot s^{-1}$) within the bubble terminating process. This implosion causes pressure waves and hydrodynamic transients reaching up to 10^3 MPa are responsible for noise and “corrosion”.

Damaging effects of cavitation

HPP in the Czech Republic

Název	Vodní tok	Rok	Typ	Výkon [MW]	Výroba [GWh/rok]	Počet soustrojí /typ turbín	Celková hltnost [m^3/s]	Spád [m]
Dlouhé Stráně	Divoká Desná	1996	PŠ	650	*)	2/FR	137,2	510,7-547,5
Dalešice	Jihlava	1978	PŠ	450	*)	4/FR	600	60,5-90,7
Orlík	Vltava	1961-2	Š	364	400	4/K	600	44-70,5
Slapy	Vltava	1954-55	Š	144	280	3/K	300	27-56
Lipno I	Vltava	1959	Š	120	168	2/F	92	148-162
Štěchovice II	Vltava	1947/1996	PŠ	45	*)	1/FR	27	209,8-219,5
Kamýk	Vltava	1961	Š	40	76	4/K	360	10,5-15,5
Štěchovice I	Vltava	1943-4	PR	22,5	89,6	2/K	150	14,5-20
Střekov	Labe	1936	PR	19,5	80	3/K	300	6-8,7
Vranov	Dyje	1934	Š	18,9	30	3/F	45	25-42
Vrané	Vltava	1936	PR	13,9	57,3	2/K	180	8-11
Vír I	Svratka	1958	Š	12	15,5	2/F	24	max. 64
Nechranice	Ohře	1968	PR	12	55	2/F	32	17-44
Křižanovice – Práčov	Chrudimka	1952	Š	8,9	14	1/F	12	88,5
Fláje – Meziboří	Flájský potok – Divoký potok	1961	Š	7,6	0,9	2/F	3,6	236,7-261,2

Š – špičková, PŠ – pološpičková, PR – průtočná

F – Francisova turbína, K – Kaplanova turbína, FR – Francisova reverzní turbína