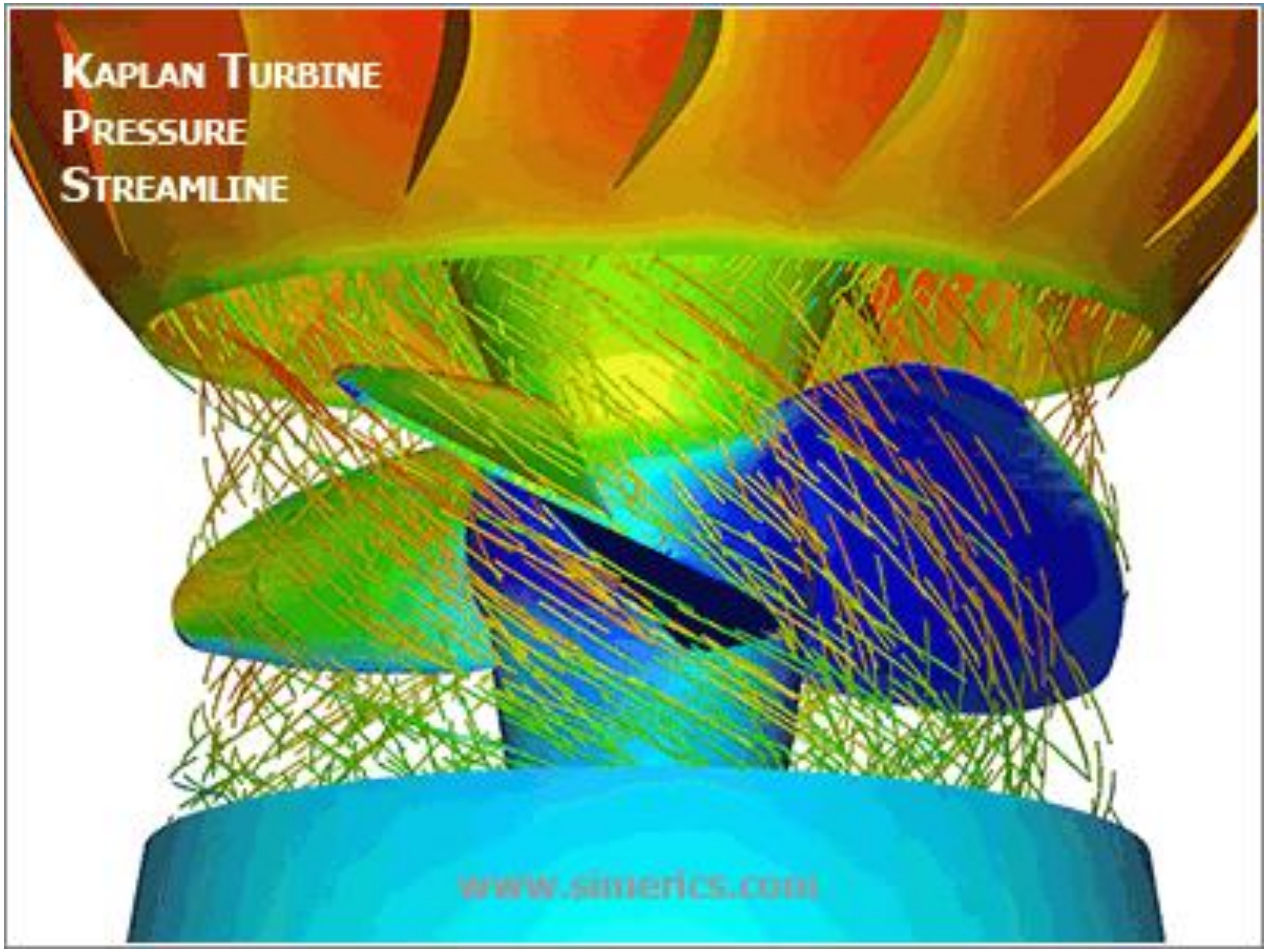
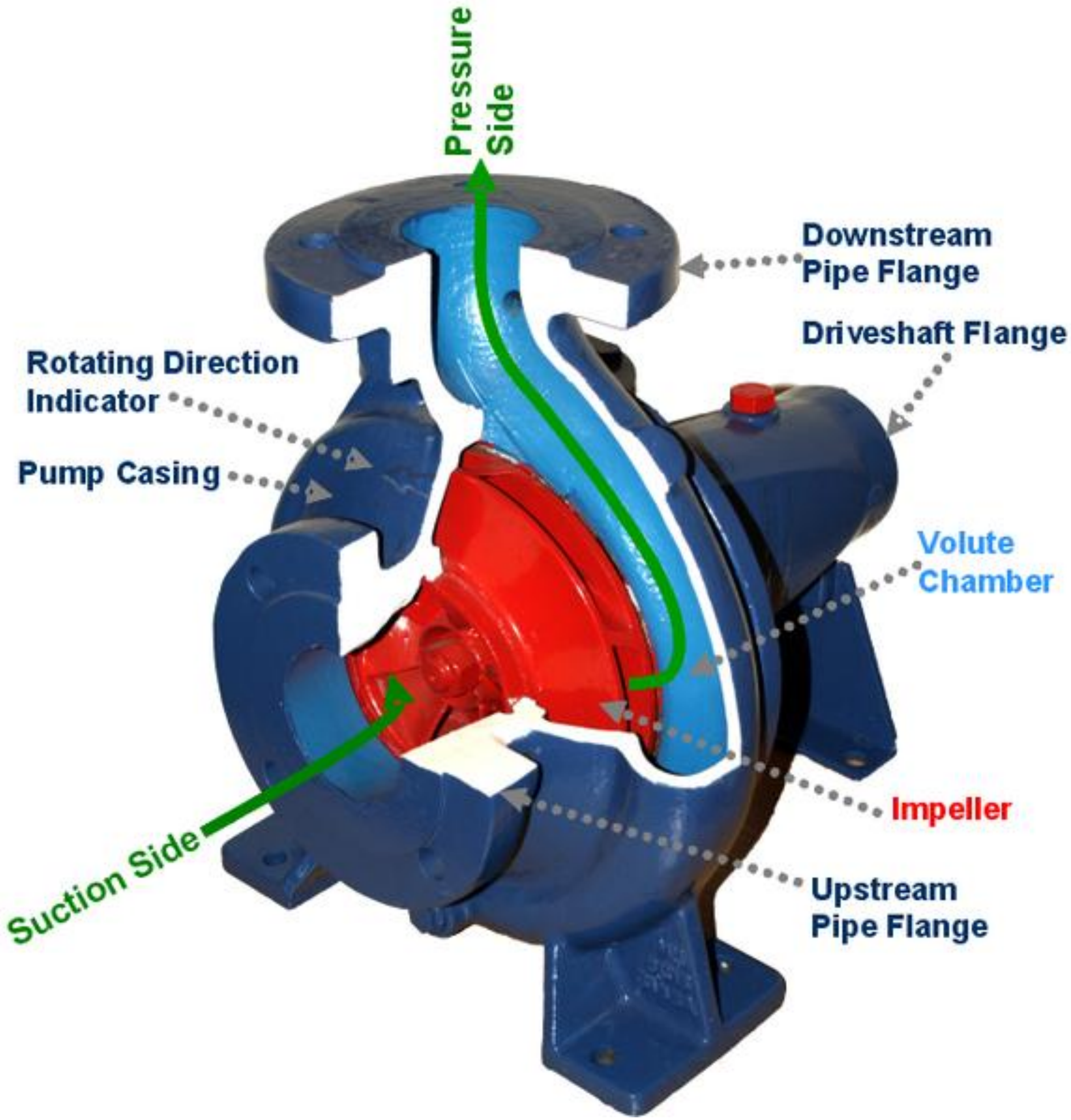
J. Szantyr – Lecture No.3: Rotors and Guide Vanes in Turbomachinery

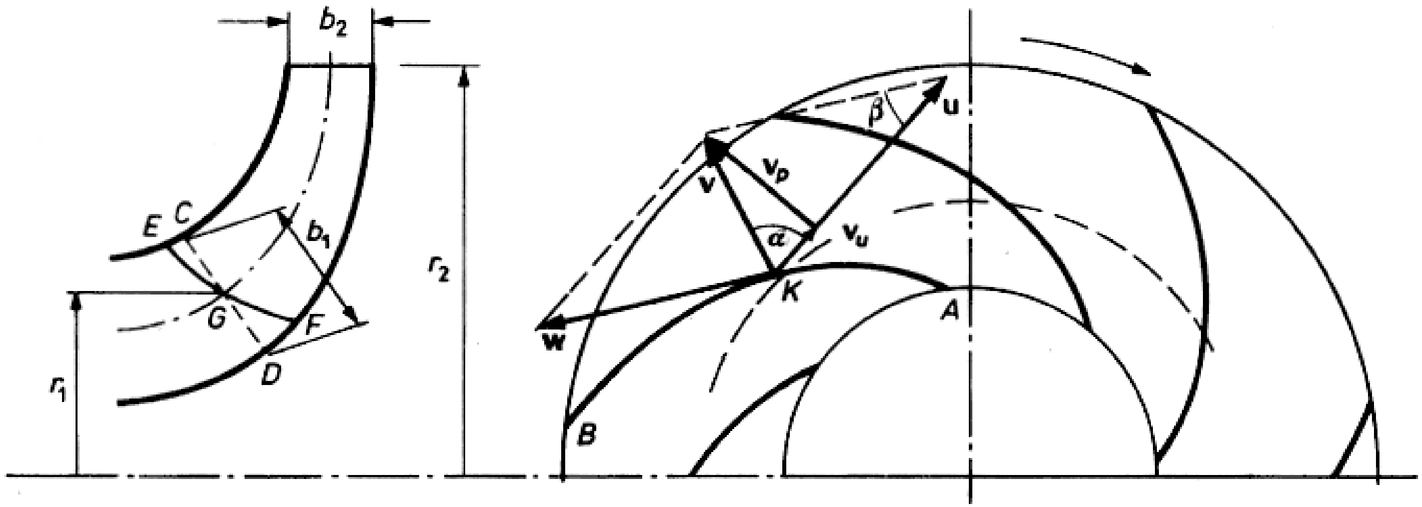


Viktor Kaplan
1876 - 1934



Centrifugal pump

Scheme of the flow through a centrifugal pump



v_u – projection of the absolute velocity onto the direction of convective velocity, v_p – meridional velocity

Pump efficiency

Change of the pressure head is the main contribution to the pump head:

$$\Delta h_p = \frac{p_2 - p_1}{\rho g}$$

Change of pressure head is connected to the useful pump power:

$$N_u = \rho \cdot g \cdot Q \cdot \Delta h_p$$

Power delivered to the pump N is higher than the useful power because of the losses, which may be divided into hydraulic, volumetric and mechanical losses. The combined influence of losses is reflected by the pump efficiency, which may be presented as a multiple of the hydraulic, volumetric and mechanical efficiencies:

$$\eta = \frac{N_u}{N} = \eta_h \cdot \eta_v \cdot \eta_m$$

The hydraulic losses are generated by friction of the liquid against the rotor and casing, together with the internal friction in the liquid.

$$\eta_h = \frac{\Delta h_p}{\Delta h_p + h_p} = \frac{\Delta h_p}{H_t}$$

where the theoretical head of a pump with finite number of blades is (Euler formula):

$$H_t = \frac{\Omega}{g}\left(r_2 v_{2u} - r_1 v_{1u}\right)$$

The volumetric losses result from the backflow existing between the rotor and the casing. Due to this backflow the real volumetric flow through the rotor is higher than the pump capacity.

$$\eta_v = \frac{Q}{Q_w} = \frac{Q}{Q+Q_v} = \frac{\rho g H_t \left(Q_w - Q_v\right)}{\rho g Q_w H_t} = \frac{N_w - N_v}{N_w}$$

Now the formula for the hydraulic efficiency may be written as:

$$\eta_h = \frac{\rho \cdot g \cdot Q \cdot \Delta h_p}{\rho \cdot g \cdot \left(Q_w - Q_v\right) \cdot H_t} = \frac{N_u}{N_w - N_v}$$

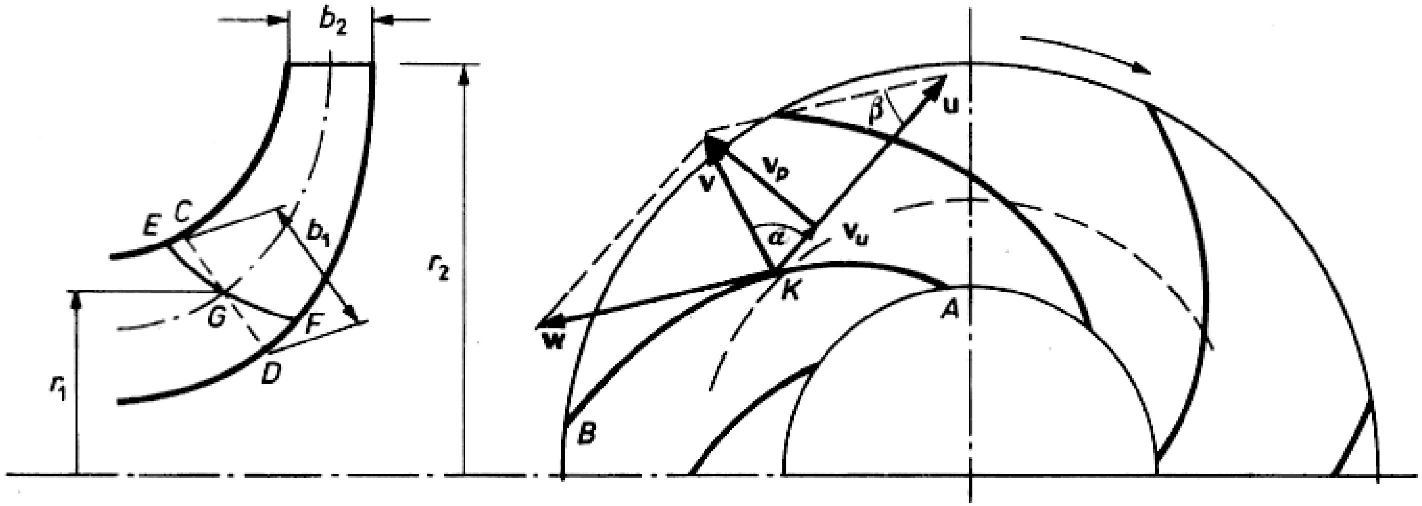
The mechanical losses result from friction in the bearings and seals, together with the friction of the liquid against the external rotor ring.

$$\eta_m = \frac{N_w}{N_w + N_m} = \frac{N_w}{N}$$

Finally:

$$\eta = \frac{N_u}{N} = \frac{N_u}{N_w - N_v} \cdot \frac{N_w - N_v}{N_w} \cdot \frac{N_w}{N} = \eta_h \cdot \eta_v \cdot \eta_m$$

Kinematics of flow through the radial pump rotor

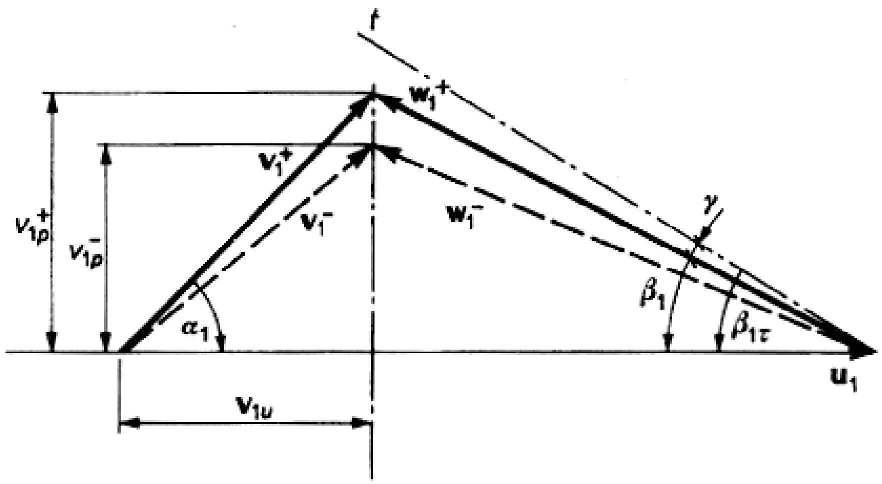


Meridional velocity v_p at a given point is the projection of the absolute velocity onto the axial plane passing through this point. The meridional velocity at the rotor inlet is determined by:

$$v_{1p}^{+} = \frac{Q_w}{S_1} = \frac{Q}{2 \cdot \pi \cdot r_1 \cdot b_1 \cdot \eta_v \cdot \psi_1}$$

where ψ_1 - blockage coefficient of the inlet cross-section

Knowing the meridional velocity, the velocity υ_{1u} and convective velocity, the velocity triangle at inlet may be constructed. υ_{1u} is in general equal zero at inlet to the first stage, unless there is a system of guide vanes before the inlet. At inlet to the next stages this velocity is equal to the outlet velocity of the preceding stage.



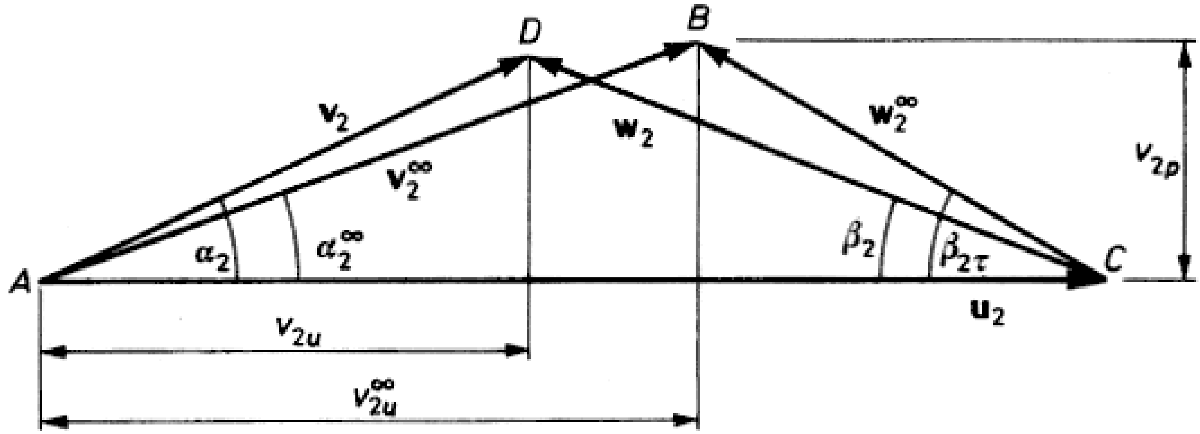
Symbol „+” determines the values for the rotor with real number of blades, while the symbol „-” – determines the values for the rotor with infinite number of infinitely thin blades.

The relative inlet velocity, together with the angles α_1 and β_1 may be determined from the velocity triangle. The angle of attack γ is typically assumed in the range of 3 to 8 degrees. It should be noted that the shape of the velocity triangle depends only on the pump capacity Q and on the angular velocity Ω.

The meridional velocity at outlet is determined by the formula:

$$v_{2p} = \frac{Q_w}{S_2} = \frac{Q}{2 \cdot \pi \cdot r_2 \cdot b_2 \cdot \eta_v \cdot \psi_2}$$

The velocity v_{2u} may be determined from the Euler formula for a given hydraulic head.

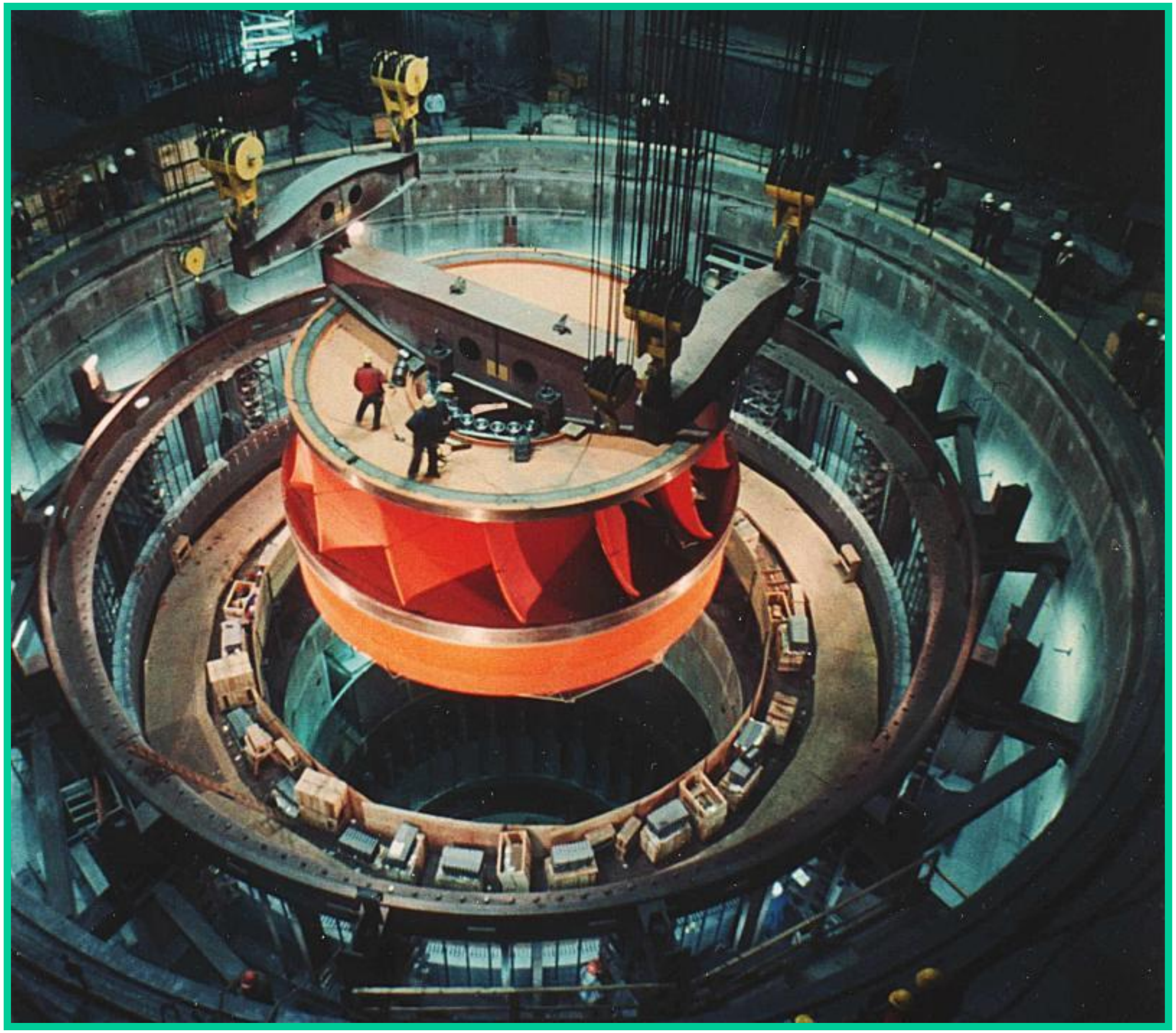


Because of the finite number of the rotor blades, the theoretical velocity traingle ACD is corrected to ABC by means of the empirical formula of Stodola:

$$v_{2u}^{\infty} - v_{2u} = \frac{\pi}{z} \cdot u_2 \sin \beta_{1\tau}$$

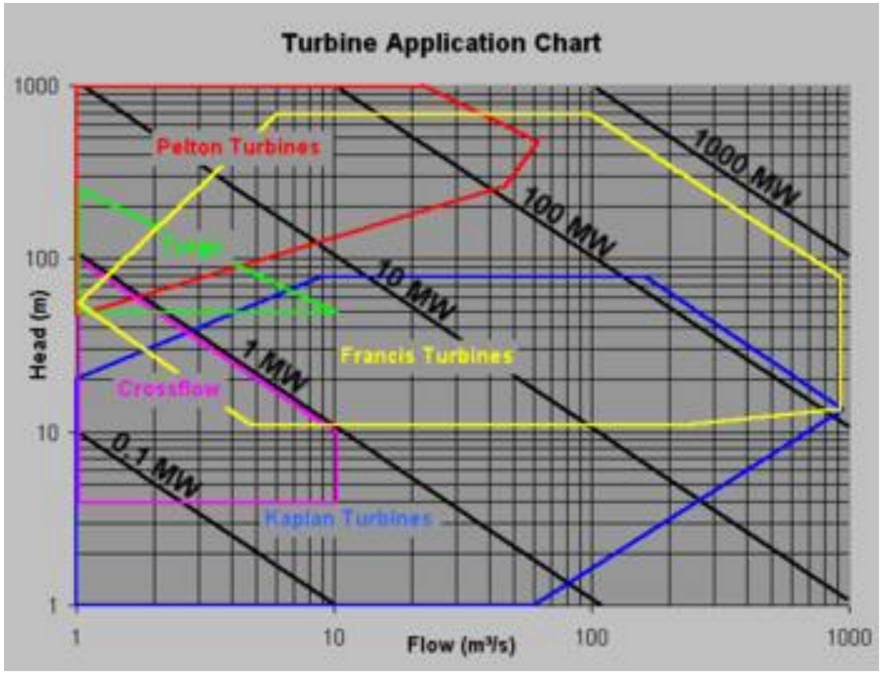
Definition of the inlet and outlet velocity triangles enables an approximate drawing of the geometry of the rotor blade.

Radial-axial water turbine (Francis turbine)

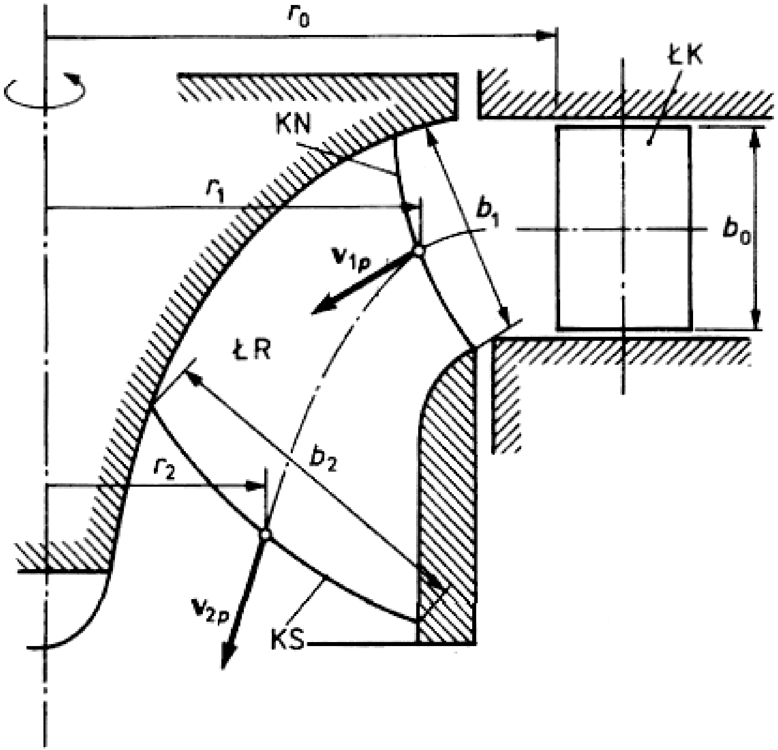


James Francis
1815 - 1892

←Guide vane setting of a Francis turbine at high and low flow rate



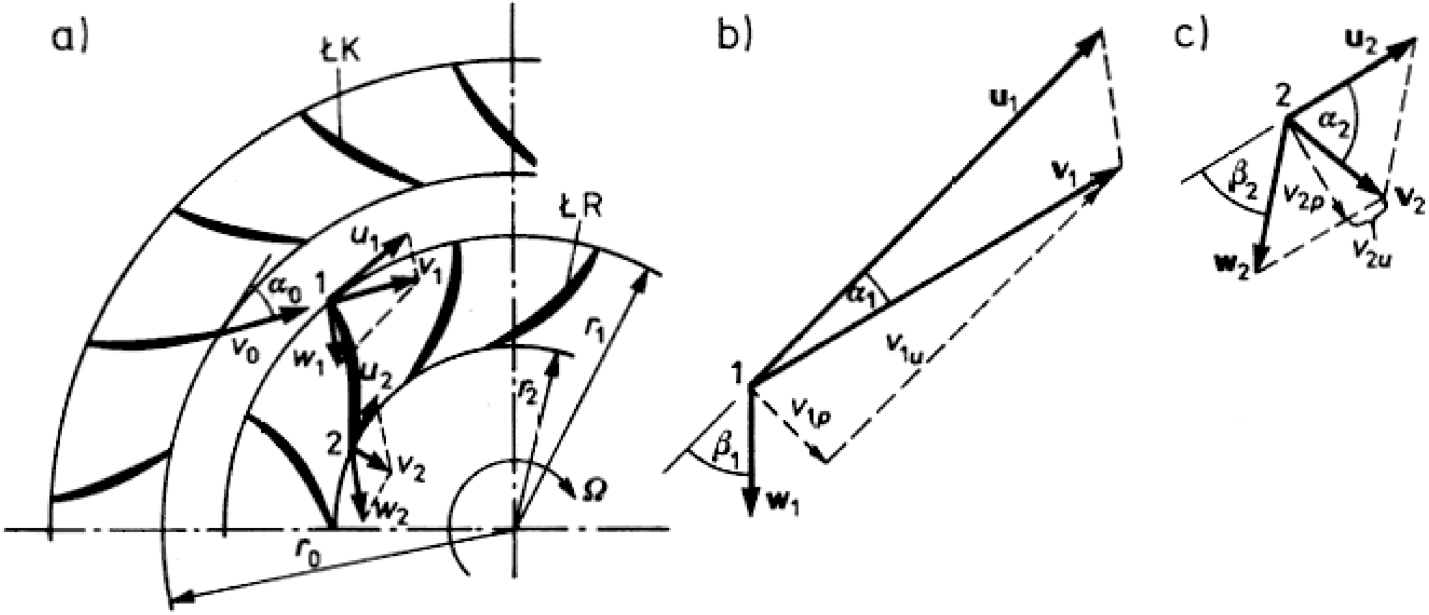
Flow through the radial-axial (Francis) turbine



From the kinematical point of view the Francis turbine is the reverse of the centrifugal pump. The stream of water flowing through the guide vanes acquires certain moment of momentum, which is then reduced almost to zero during the flow through the rotor.

The turbine operates with the highest efficiency when the inflow to the rotor is shock-free (angle of attack equal zero), and outflow from the rotor is purely axial, i.e. there is:

$$v_{2u} = 0$$



Water leaves the guide vanes at an angle $\alpha_0 \approx \alpha_1$ with velocity:

$$v_0 = \frac{Q}{2 \cdot \pi \cdot r_0 \cdot b_0 \cdot \sin \alpha_0}$$

The circumferential component of this velocity at the rotor inlet is:

$$v_{1u} = \frac{r_0}{r_1} \cdot v_0 \cdot \cos \alpha_0$$

The meridional velocity at inlet is: $v_{1p} = \dfrac{Q}{2 \cdot \pi \cdot r_1 \cdot b_1}$

On the basis of the above data and knowing the convective velocity, the velocity triangle at inlet may be defined, what enables determination of the relative velocity and its angle β_1 , which is also the inlet angle of the blade. At the rotor outlet there is:

$$v_{2p} = \frac{Q}{2 \cdot \pi \cdot r_2 \cdot b_2} \qquad w_2 = \frac{v_{2p}}{\sin \beta_2} \qquad u_2 = \Omega \cdot r_2 \qquad v_{2u} \approx 0$$

This defines the velocity triangle at outlet.

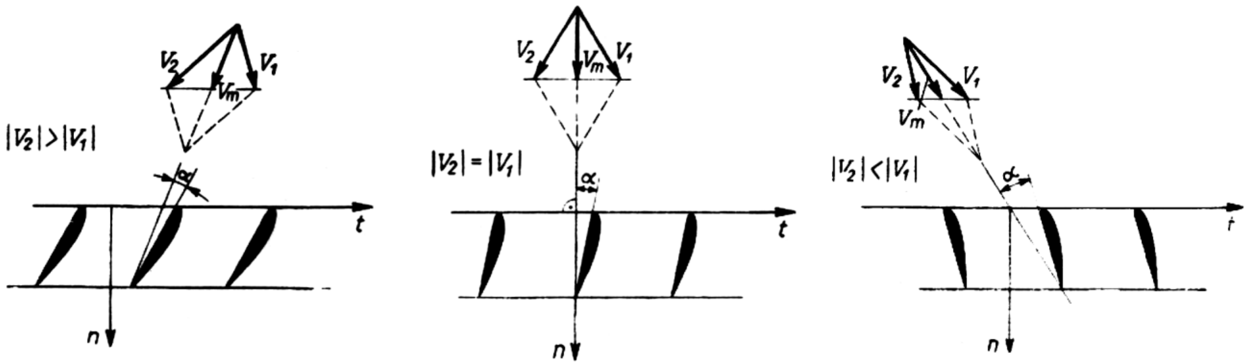
The Euler formula may be transformed to:

$$H = \frac{v_1^2 - v_2^2}{2g} + \frac{u_1^2 - u_2^2}{2g} - \frac{w_1^2 - w_2^2}{2g}$$

Analysis of this formula connects the available hydraulic head with the appropriate type of a turbine (axial or radial-axial).

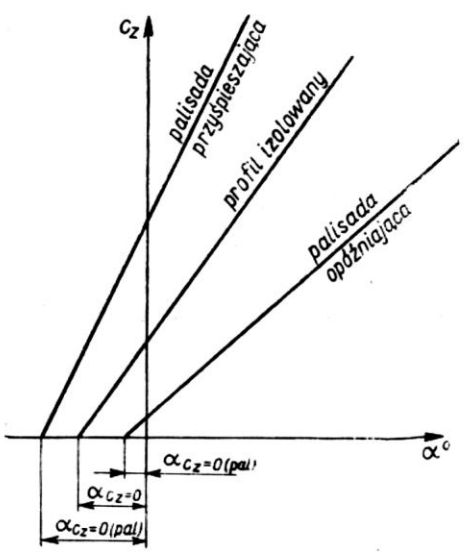
Palisades of profiles

Lifting foils, forming the turbine and pump rotors, interact with each other, changing their hydrodynamic characteristics. This effect may be demonstrated in the so called palisade of profiles.



Inlet velocity - V_1

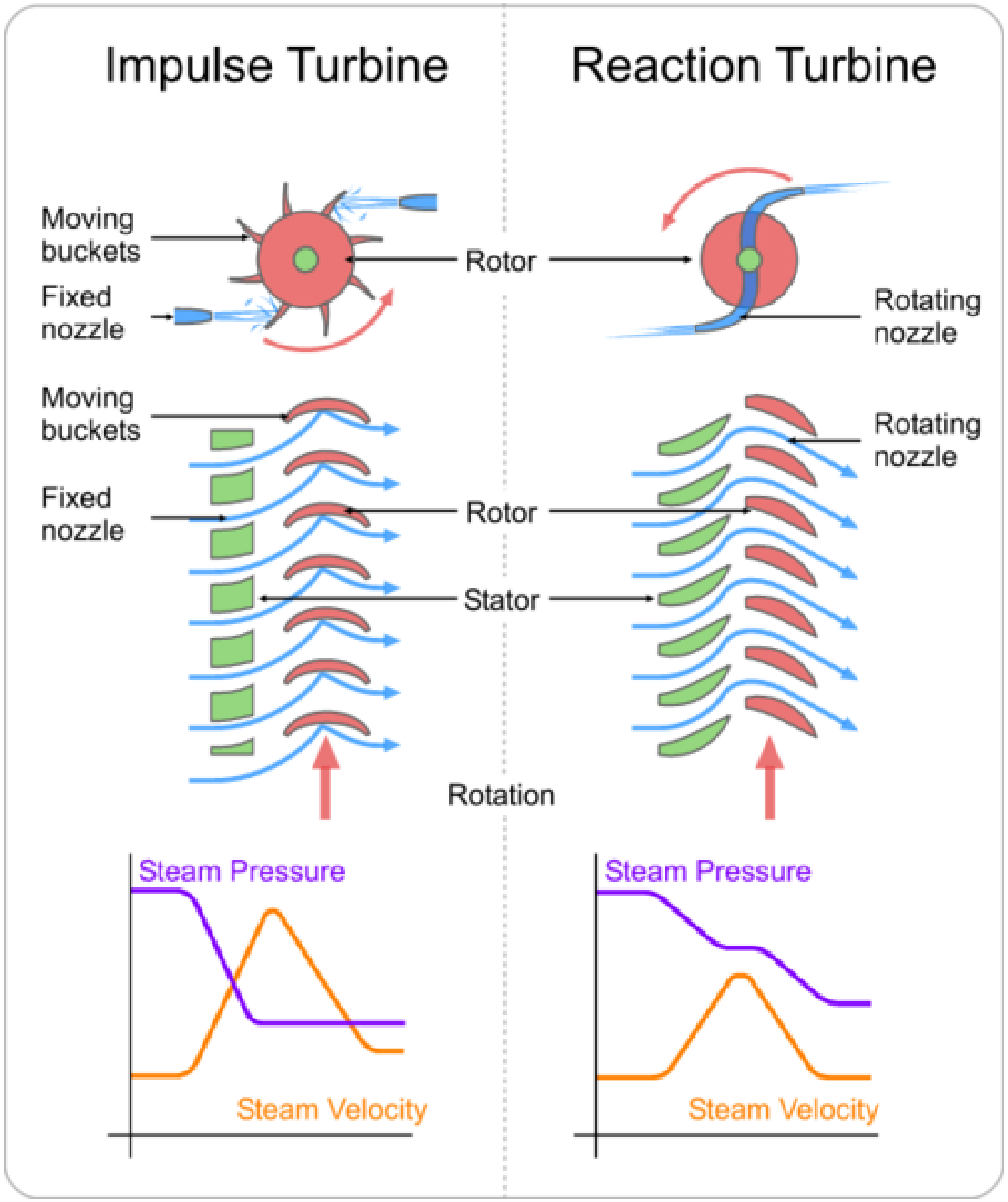
Outlet velocity- V_2



Accelerating palisade – velocity at outlet is higher than the velocity at inlet (reaction turbines)

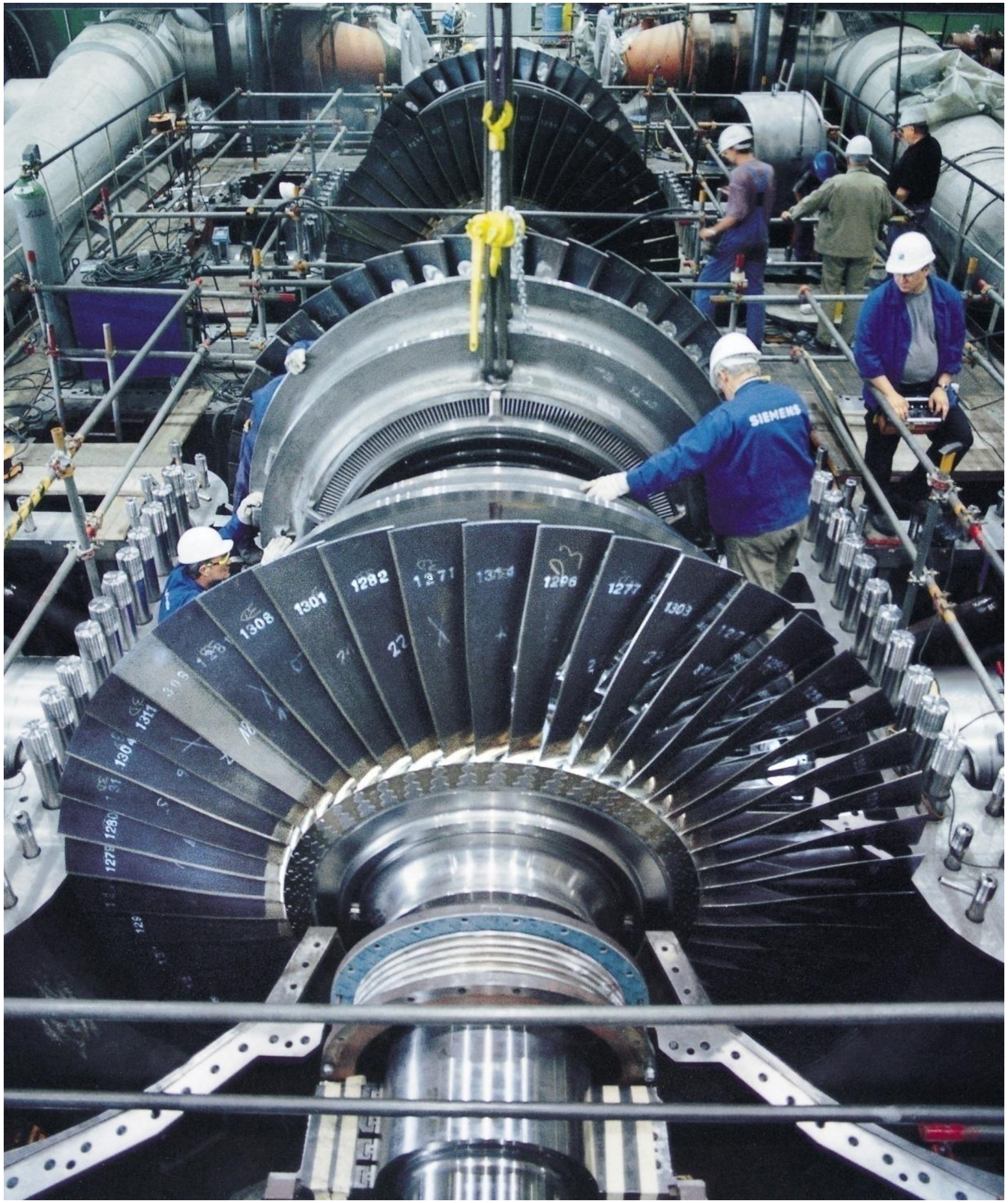
Neutral palisades – velocity modules at inlet and outlet are identical (impulse turbines)

Decelerating palisades – velocity at outlet is smaller than at inlet (pumps)

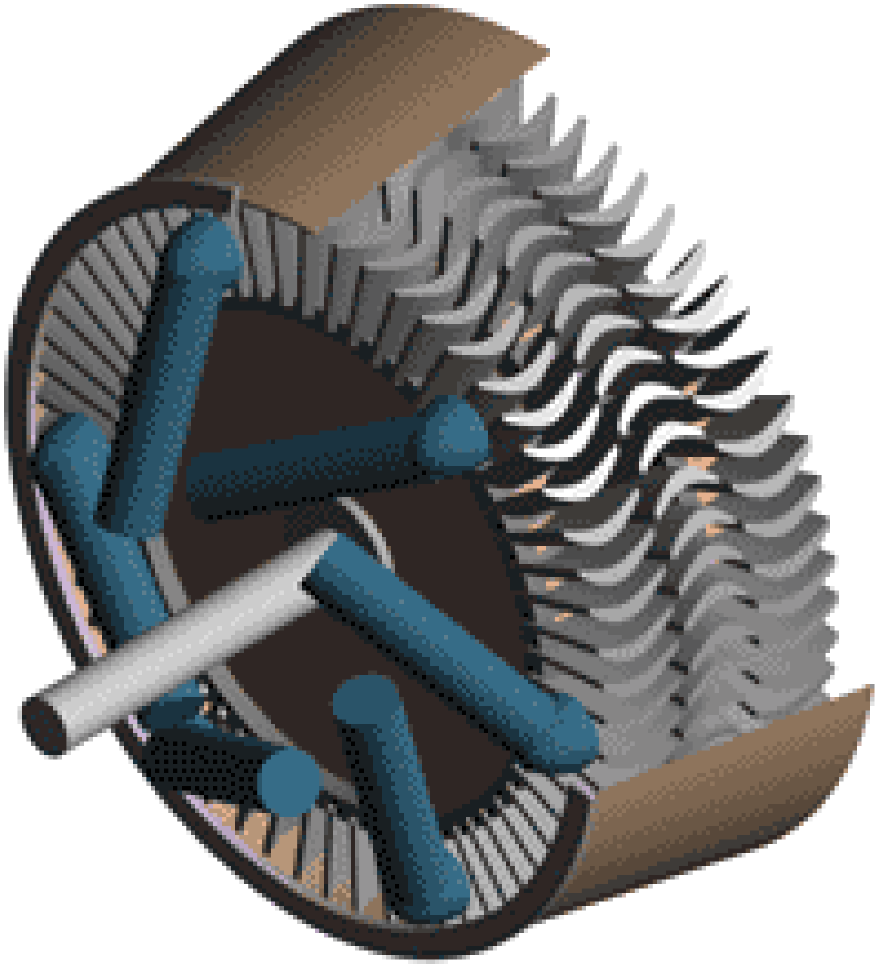


In the impulse turbine the expansion of the medium takes place only in the guide vanes

In the reaction turbine part of the expansion process takes place in the guide vanes and the remaining part – in the rotor. Proportion of the expansion in the rotor to the total expansion defines the degree of reaction of the turbine.



Assembly of the reaction steam turbine



Mutual interaction of two rows of rotor blades with the row of guide vanes located betweeen them