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Kaplan turbines

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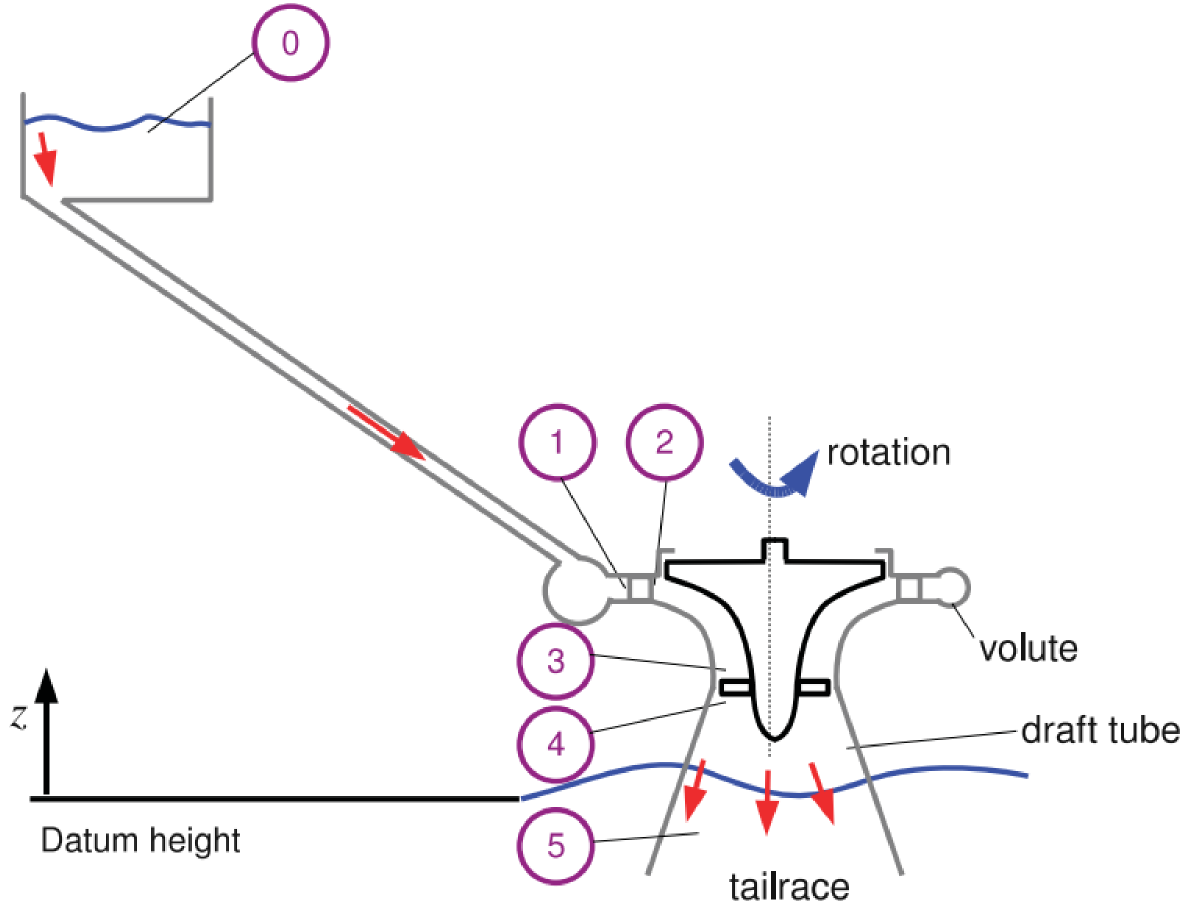
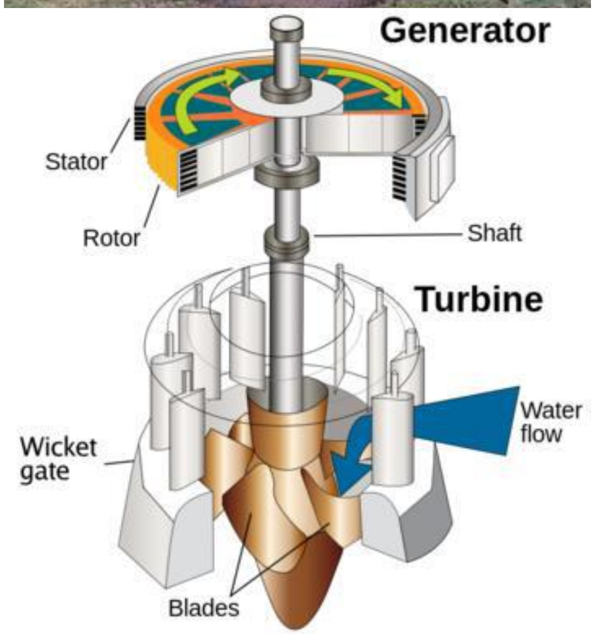
Kaplan turbine Flow Analysis

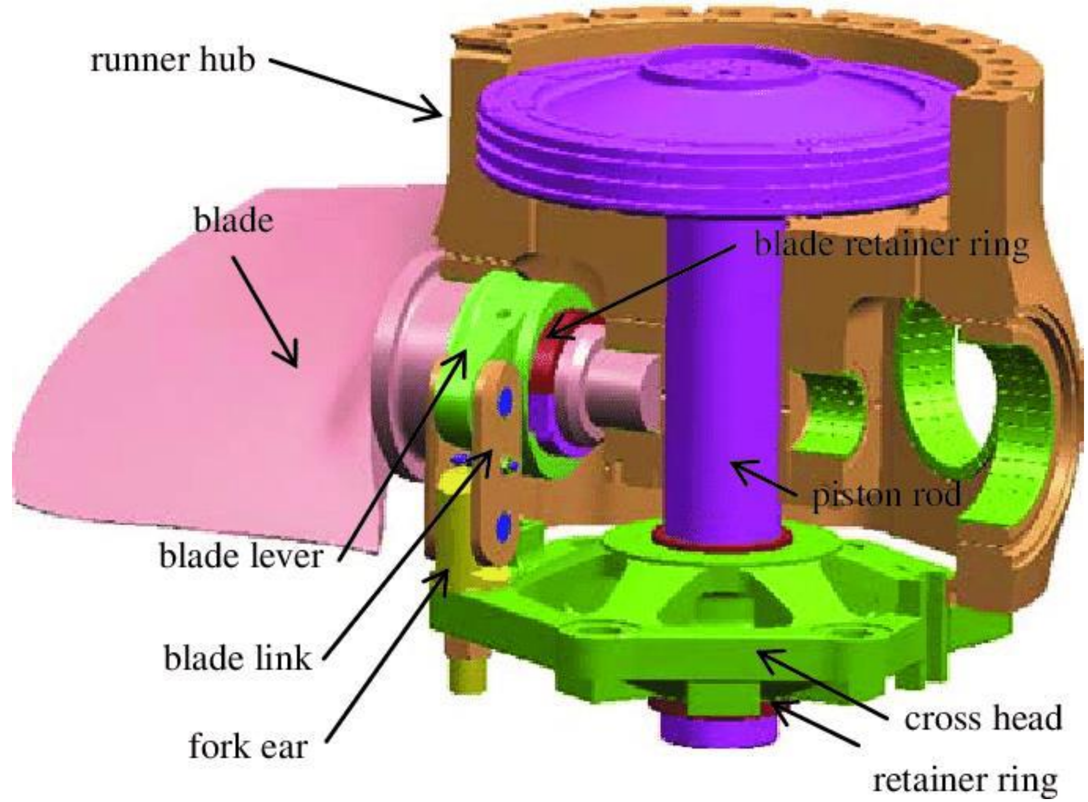
In comparison to the analysis of the Francis turbine an extra analysis station is

needed (station 3) between the guide vane exit and the runner inlet as the flow is

turned through 90◦ before entering the turbine blades.




Generator
Stator
Rotor
Shaft
Turbine
Water flow
Wicket gate
Blades


runner hub
blade
blade retainer ring
piston rod
blade lever
blade link
fork ear
cross head
retainer ring

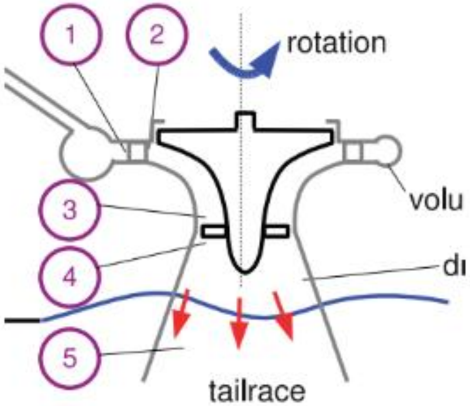
KAPLAN TURBINES – THE ANALYSIS APPROACH

1. From station 2, the guide vane exit, to station 3, the runner inlet one can either use conservation of angular momentum if guide vanes are radial. In some Kaplan turbines the guide vanes are orientated in the axial direction in which case station 2 = station 3. The velocity triangles are then drawn to get relative velocities at rotor inlet.

2.At 4, runner exit, draw the velocity triangle to obtain relative and absolute exit velocities.

3. The Euler equation gives the power

$$P = \dot{m}\omega\left(r_3 V_{3\theta 3} - r_4 V_{4\theta}\right)$$



4. Calculate losses. In this example component losses will be considered. Component head losses will be obtained by working through the stations from 0 to 4. This gives the turbine efficiency.

5. Finally the draft tube, station 4 to station 5 is analysed, this calculation will give us the overall efficiency of the site

EXAMPLE

Consider a Kaplan Turbine with the following characteristics:

Consider a Kaplan Turbine with the following characteristics:

Guide Vane Exit:
radial flow, $r_2 = 2\ m$, blade height, $b_2 = 1.0\ m$,
blockage, $t=0.08$, absolute flow angle to radial $\alpha_2 = 75^\circ$

Runner:
Mean radii at station 3 and 4 are the same: $r_{3m} = r_{4m} = 0,85\ m$, blade height $b_3 = 0.7\ m$, designed for zero exit absolute swirl, $V_{4\theta} = 0$,
rotational speed $N=300\ o/min$

Draft tube
Reduces axial velocity to $V_5 = 5\ m/s$

Flow rate Q = 36 m3 / s

Calculate the runner relative and absolute flow angles and velocities at inlet and exit. Also calculate the power output.

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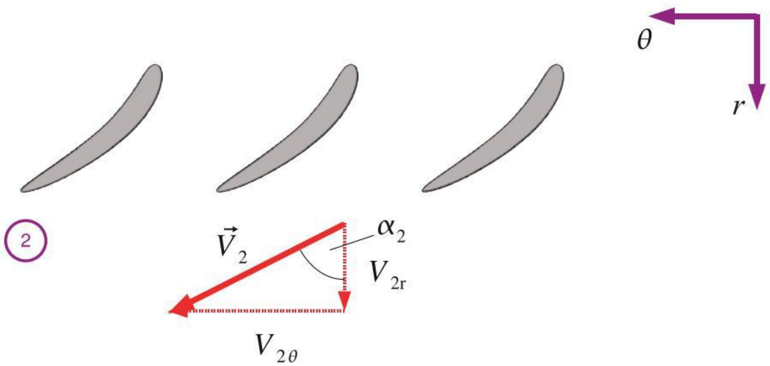
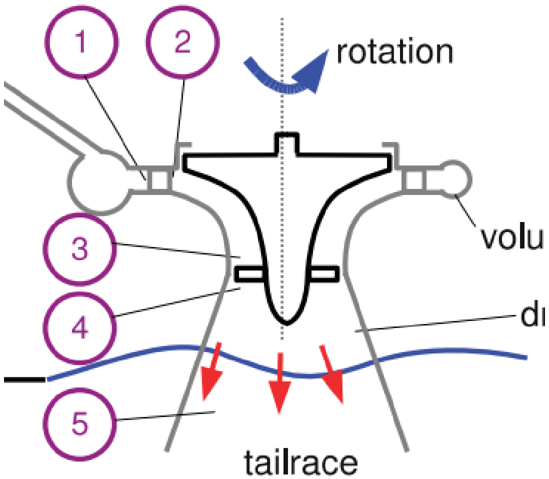
SOLUTION

The strategy is to apply the analysis steps outlined earlier.

In a Kaplan turbine the flow through the guide vanes may be be radial or axial so the geometry must be carefully inspected.



Velocity triangle at exit of guide vanes



Guide Vane Exit to Runner Inlet. At guide vane exit (station 2), the flow is radial. Apply continuity:

$$Q = 2\pi r_2 b_2 (1 - t) V_{2r}$$

$$V_{2r} = \frac{36}{2\pi \times 2.0 \times 1.0 \times (1 - 0.08)} = 3.11\, m/s$$

From velocity triangle

$$V_{2\theta} = V_{2r} \tan \alpha_2 = 3.11 \times \tan 50^\circ = 3.71\, m/s$$

$$V_2 = \sqrt{(V_{2r})^2 + (V_{2\theta})^2} = \sqrt{3.11^2 + 3.71^2} = 4.84\, m/s$$

Since there are no blades between station 2 and station 3, angular momentum is conserved, so:

$$r_{3m} V_{3\theta} = r_2 V_{2\theta} \implies V_{3\theta} = \frac{r_2 V_{2\theta}}{r_{3m}} = \frac{2.0 \times 3.71}{0.85} = 8.73\, m/s$$

At station 3 the flow is axial. The continuity equation is : $Q = 2\pi r_{3m} b_3 V_{3x},$

$$V_{3x} = \frac{36}{2\pi \times 0.85 \times 0.7} = 9.63\, m/s \quad \text{Axial velocity at 3}$$

From velocity triangle :

$$V_3 = \sqrt{(V_{3x})^2 + (V_{3\theta})^2} = \sqrt{9.63^2 + 8.73^2} = 13.00m/s$$

$$\alpha_3 = \tan^{-1}\left(\frac{V_{3\theta}}{V_{3x}}\right) = \tan^{-1}\left(\frac{8.73}{9.63}\right) = 42.2°$$

The blade blade speed U_3 is given by:

$$\omega r_{3m} = \frac{2\pi N}{60} r_{3m} = \frac{2\pi \times 300}{60} 0.85 = 26.70\ m/s$$

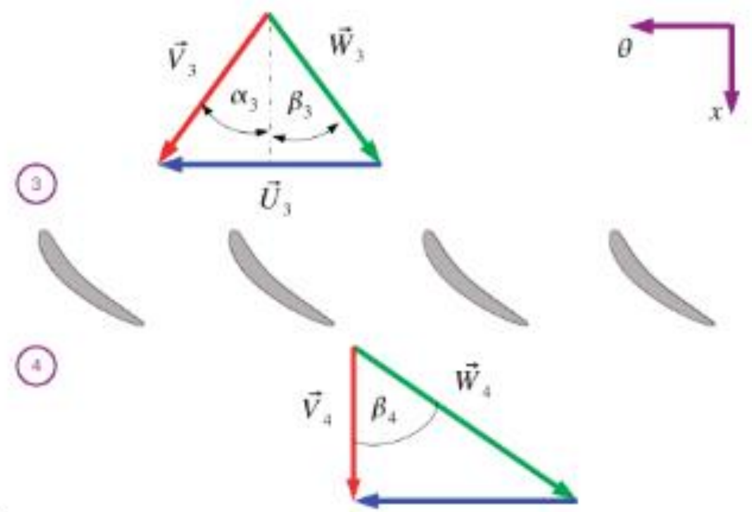
From the velocity triangles in the tangential direction:

$$V_{3\theta} = U + W_{3\theta} \implies W_{3\theta} = V_{3\theta} - \omega r_{3m}$$

$$8.73 - 26.70 = -17.97\ m/s$$

Now we have all the ingredients to work out the relative flow angle at inlet to the runner:

$$\beta_3 = \tan^{-1}\left(\frac{W_{3\theta}}{V_{3x}}\right) = \tan^{-1}\left(\frac{-17.97}{9.63}\right) = -61.8°$$



Velocity triangle for Kaplan runner

Across Runner

At the runner exit: $V_{4\theta} = 0.\ V_{4x} = V_{3x}$ as $r_{4m} = r_{3m}$

From velocity triangle at 4:

$$W_{4\theta} = -\omega r_{4m} = -\omega r_{3m} = -27.60\ m/s$$

$$\beta_4 = \tan^{-1}\left(\frac{W_{4\theta}}{V_{4x}}\right) = \tan^{-1}\left(\frac{-27.60}{9.63}\right) = -70.2^\circ$$

$$W_4 = \sqrt{W_{4x}^2 + W_{4\theta}^2} = \sqrt{17.97^2 + 27.60^2} = 28.38 m/s$$

Euler equation $P = \dot{m}\omega\left(r_3 V_{3\theta 3} - r_4 V_{4\theta}\right)$

$V_{4\theta} = 0 \qquad \rho = 1000 kg/m^3$

$$P = 36 \times 1000 \times 26.70 \times 8.73 = 8391 kW$$

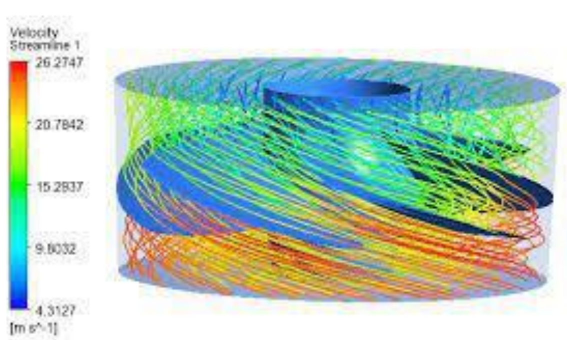
This is the power produced by the turbine.

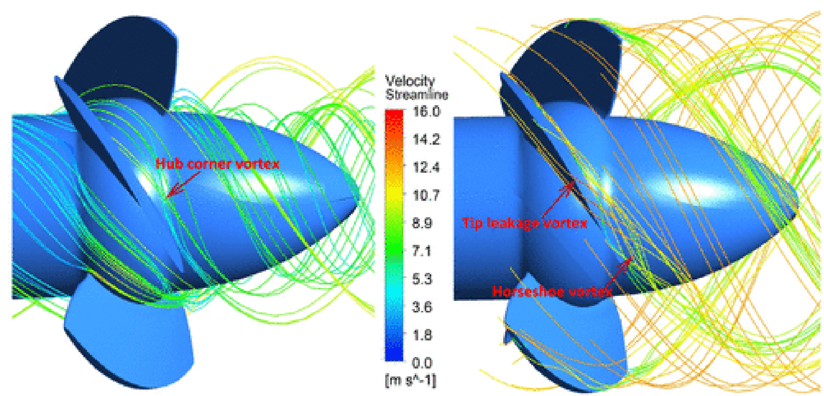
Loss Estimation

One way losses may be calculated is to assume that across any component the total head loss is proportional to the velocity head at exit:

$$\Delta H = k\left(\frac{V^2}{2g}\right)_{exit}$$

This is a concept borrowed from minor losses in pipe flow calculations, this is still an empirical method as one has to specify an appropriate value of k for each component. For runners the relative dynamic head is used the and the cause of most fluid friction is the the relative motion between the runner surfaces and the mainstream flow.





(a) (b)

EXAMPLE

With the Kaplan turbine from the previous example the loss coefficients may be taken as ***k* = 0.05** for the guide vanes and bend and ***k* = 0.06** for the runner. Ignoring pipe friction estimate the turbine efficiency.

Solution:

O => 1, for this example ignore the pipe loss. Though it is relatively easy to calculate.

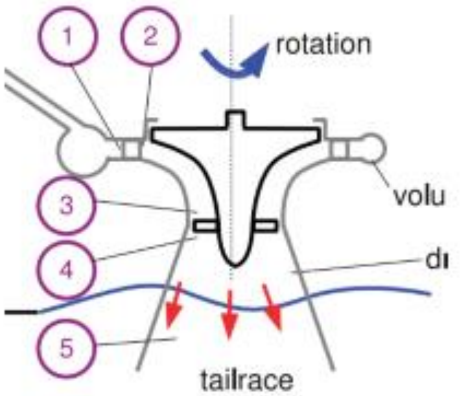
1 => 2, gor guide vanes , $k = 0.05$: $\Delta H = 0.05(V_2^2)/2g = 0.05(4.84^2)/2g = 0.06\ m$

2 => 3, *k=0,05* for the bend from radial to axial flow,

$$\Delta H = 0.05(V_3^2)/2g = 0.05(13.0^2)/2g = 0{,}43\ m$$

3 => 4, for the runner use the relative exit head.with *k=0,06*

$$\Delta H = 0.06(W_4^2)/2g = 0.06(28.38^2)/2g = 2.46\ m$$



From station 1 to station 4, across the whole turbine, the ideal total head drop for the actual power is given by: :

$$\dot{m}g\Delta H_{ideal} = P_{actual} \implies \Delta H_{ideal} = \frac{P_{actual}}{\dot{m}g\Delta H_{ideal}}$$

$$\Delta H_{ideal} = \frac{8391 \times 10^3}{36 \times 1000 \times 9.81} = 23.76\ m$$

Which is the total head required **if there were no losses in the turbine.** The actual total head dropacross the turbine,

$$H_1 - H_4 = \Delta H_{ideal} + \sum \Delta H_{losses}$$

$$H1 - H4 = 23.76 + (0.06 + 0.43 + 2.46) = 26.71\ m$$

This is the required total head drop across the turbine alone to produce the desired flow rate. **The turbine efficiency is:**

$$\eta_T = \frac{\text{Actual Power}}{\dot{m}g(H_1 - H_4)} = \frac{8391 \times 10^3}{36 \times 1000 \times 9.81 \times 26.71} = 0.89$$

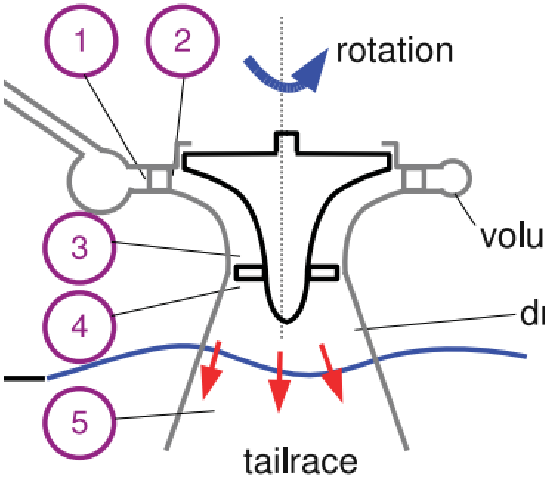
DRAFT TUBE ANALYSIS

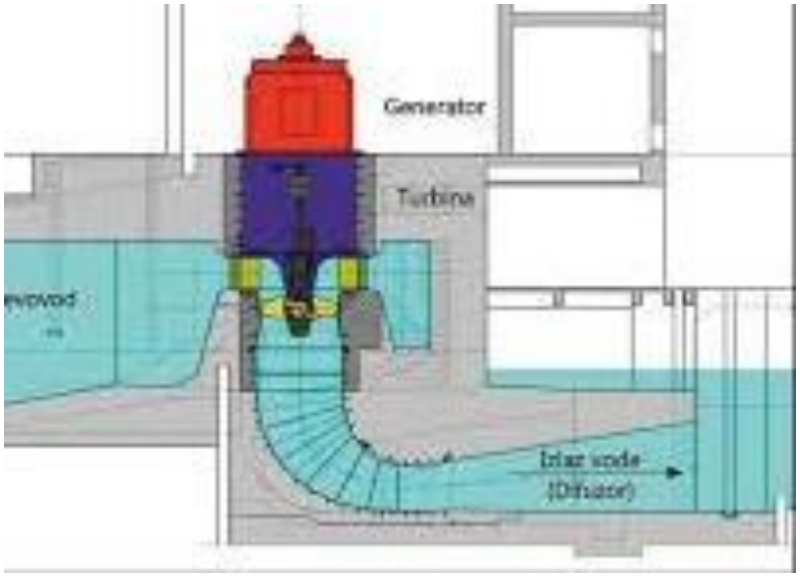
To repeat:
The draft tube is a conical diffuser with around 7◦ divergence which reduces the exit kinetic energy in the departing fluid and therefore increases the efficiency of the machine as a whole.

In hydraulic machines two efficiencies may be quoted:

"Turbine efficiency" excludes draft tube and inlet pipe losses from the definition of available head and gives an indication as to the "goodness" of the turbine runner.

"System efficiency" includes the losses in inlet pipes and draft tube and is of crucial interest to those designing or operating particular hydro-electric power plants.





Example: Using the Kaplan turbine in the previous example. Estimate the system efficiency if the draft tube loss coefficient is given by k = 0.2.

SOLUTION

Između tačaka 4 i 5 difuzor snižava aksijalnu brzinu i stoga izgubljenu kinetičku energiju na izlazu. Ukupni pad u tačci 5 je dat sa:

$$H_5 = \frac{p_5}{\rho g} + \frac{V_5^2}{2g} + z_5$$

Kako je $z_5 = -h_5$, $p_5 = \rho g h_5$ pritisak u tačci 5 mora biti jednak pritisku donje vode (pritisku u rijeci)

$$H_5 = \frac{p_5}{\rho g} + \frac{V_5^2}{2g} + z_5 = \frac{V_5^2}{2g} = \frac{5^2}{2 \times 9.81} = 1.27\ m$$

$\Delta H_{4-5} = H_4 - H_5$, so $H_4 = H_5 + \Delta H_{4-5}$ and $k = 0.2$

Gubitci u difuzoru su ➡ $\Delta H_{4-5} = k\left(\frac{V_4^2 - V_5^2}{2g}\right) = 0.2 \times \left(\frac{9.63^2 - 5^2}{2 \times 9.81}\right) = 0.69\ m$

$H_4 = 1.27 + 0{,}69 = 1{,}96\ m$, tako da je ulazni pad za turbinu $H_1 = (H_1 - H_4) + H_4 = 26{,}71 + 1{,}96 =$ ***28,67 m***.

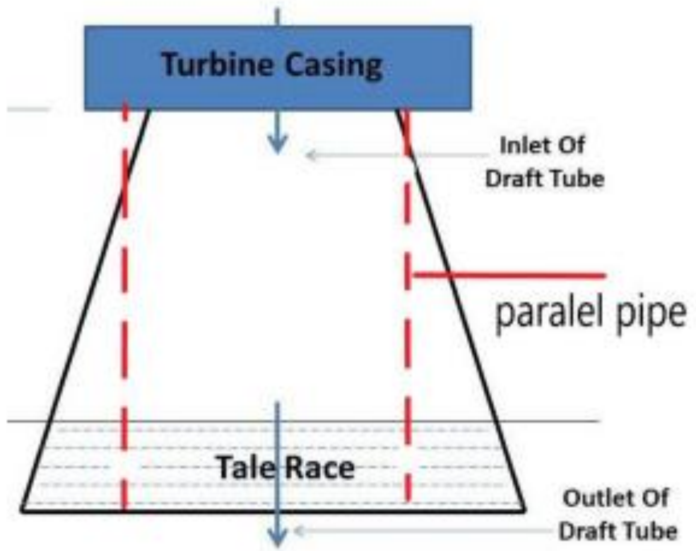
Ovo je ukupni pad potreban da bi turbina isporučila stvarnu snagu. Pa je ***ukupna efikasnost sistema data sa:***

$$\eta_S = \frac{\text{Actual Power}}{\dot{m} g H} = \frac{8391 \times 10^3}{36 \times 1000 \times 9.81 \times 28.67} = 0.83$$

EFECTS OF DRAFT TUBE

To make clearer the influence of the draft tube, a calculation is carried out to illustrate what would happen if it was replaced with a parallel pipe, $V_5 = V_4 = 9{,}63\ m/s$ *and* and assuming that there is no head loss associated with the parallel pipe.
Consider what happens to the head and efficiency when there is a parallel pipe

$$H_4 = H_5 = \frac{V_5^2}{2g} = \frac{9.63^2}{2 \times 9.81} = 4.73m$$



This makes the inlet head to the turbine :

$$H_1 = (H_1 - H_4) + H_4 = 26.71 + 4.73 = 31.44\,m.$$

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EFECTS OF DRAFT TUBE

For the same power output the inlet head would need to be raised from *28,67 to 31,43 m.*

The changed overall efficiency for the turbine is given by:

$$\eta_S = \frac{\text{Actual Power}}{\dot{m}gH} = \frac{8391 \times 10^3}{36 \times 1000 \times 9.81 \times 31.44} = 0.76$$

So there is a **substantial reduction in efficiency**, the difference in cost between a parallel pipe and a diffuser is negligible so a draft tube is well worth having!

There is a **drawback to having a draft tube,** consider the turbine exit pressure for both the parallel pipe and the draft tube.

The total head at station 4 is given by:

$$H_4 = \frac{p_4}{\rho g} + \frac{V_4^2}{2g} + z_4 = h_4 + \frac{V_4^2}{2g} + z_4$$

h_4 static pressure at 4

$$h_4 = H_4 - \frac{V_4^2}{2g} - z_4$$

For the sake of example say that the turbine exit is 1 m above the tail race, that is $z_4 = 1.0\ m$, $H_4 = 1{,}96\ m$ i $V_4 = 9{,}63\ m$ so static pressure at 4 :

$$h_4 = 1.96 - \frac{9.63^2}{2 \times 9.81} - 1.0 = -3.77\ m$$

The negative sign indicates that the pressure is **below atmospheric pressure**. Expressing this pressure in *Pa* rather than *m* of fluid yields

$$p_4 = h_4 \rho g = -3.77 \times 1000 \times 9.81 = -36.9 kPa \approx -0.4\ bar$$

Without the draft tube so with a cylindrical pipe at exit.

$$H_4 = 4.72\,m \text{ and } V_4 = 9.63\,m/s$$

Static pressure at 4:

$$h_4 = 4.72 - \frac{9.63^2}{2 \times 9.81} - 1.0 = -1.00\,m$$

Or: $$p_4 = h_4\rho g = -1.00 \times 1000 \times 9.81 = -9.8kPa \approx -0.1\,bar$$

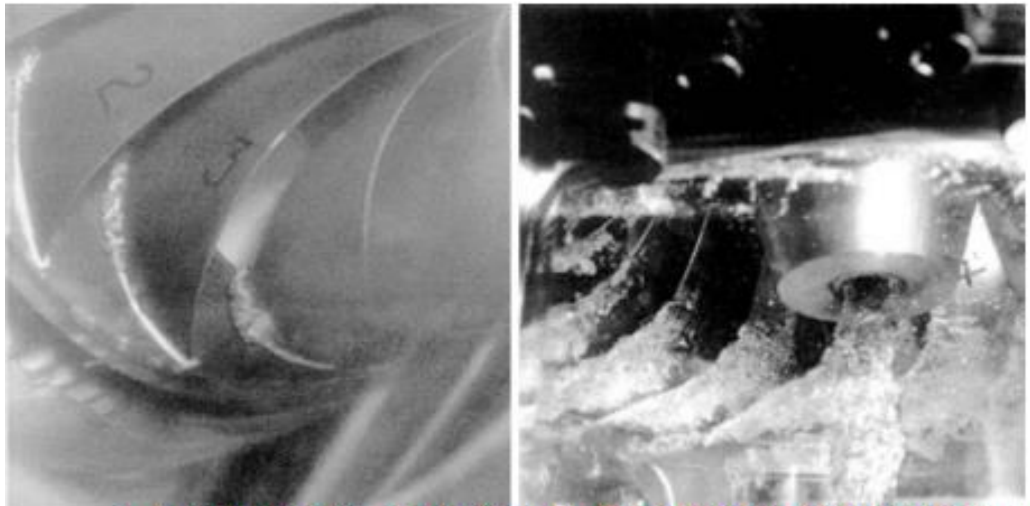
which is considerably greater.

The problem with excessively low pressures is that they can lead to a phenomenon known as **cavitation.**

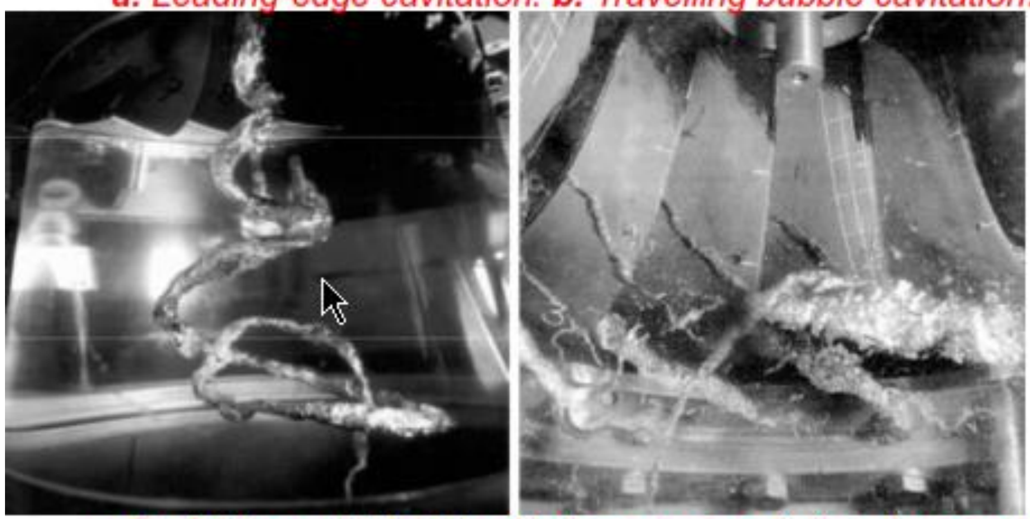
Cavitation occurs if the absolute pressure falls below the saturated vapour pressure, p_v, for the water temperature.

In other words when the pressure drops too low the water starts to boil even at the ambient temperature encountered by hydraulic machines. Essentially bubbles of vapour will be formed.

As the pressure rises again the bubbles collapse suddenly and very high instantaneous pressures can be created (p > 500bar) which can cause significant erosion of blade surfaces and lower machine performance.



a. Leading-edge cavitation. b. Travelling bubble cavitation.



a. Draft tube swirl. b. Inter blade vortex cavitation. Image source

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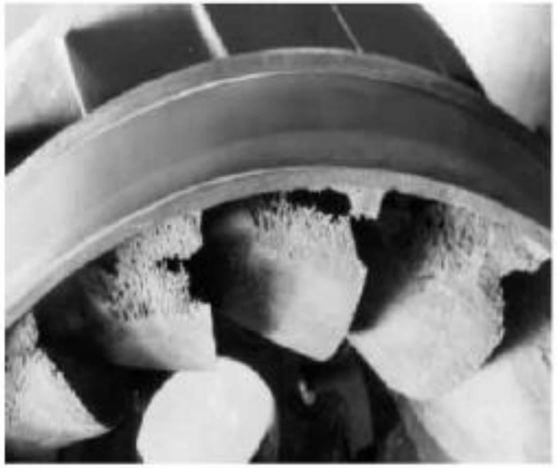


Fig. 14. Picture of actual cavitation damage runner [Chrisopher, 1994].

Oštećenja radnih kola usljed djelovanja kavitacije

In practise dissolved air starts to come out of solution of the fluid first and so some margin between the saturated vapour pressure and the pressure the turbine operates at is required.

This is normally done by using an empirical parameter known as Thoma's parameter:

$$\sigma = \frac{p - p_v}{\Delta H}$$

Ehere : p Fluid static pressure , p_v s the saturated vapour pressure and , and ΔH is the head drop across the machine.

Charts of Thoma's parameter derived from practical experience are available.

The problem of cavitation can be alleviated somewhat by reducing z4, that is lowering the turbine closer to the river level or even below it.

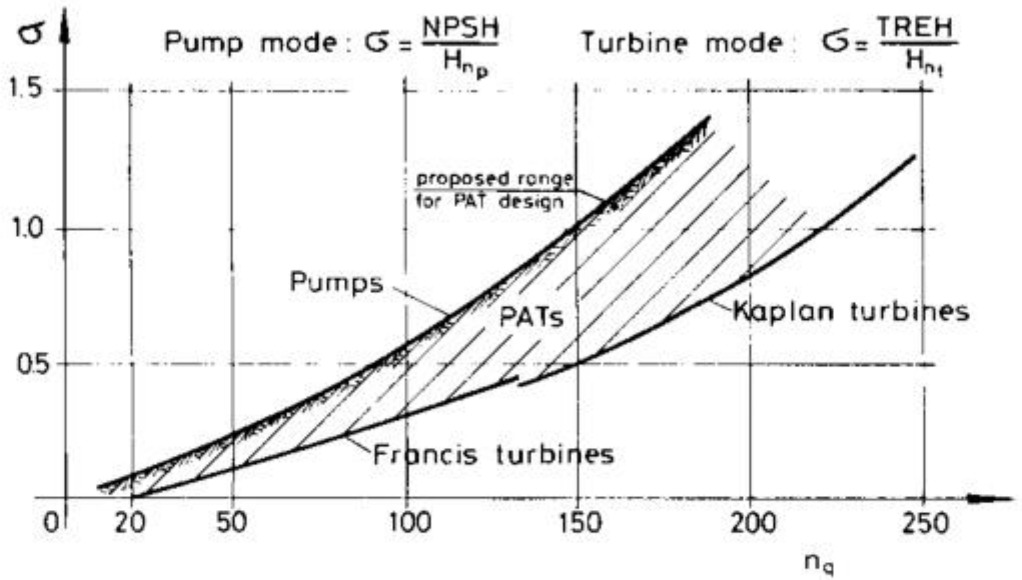
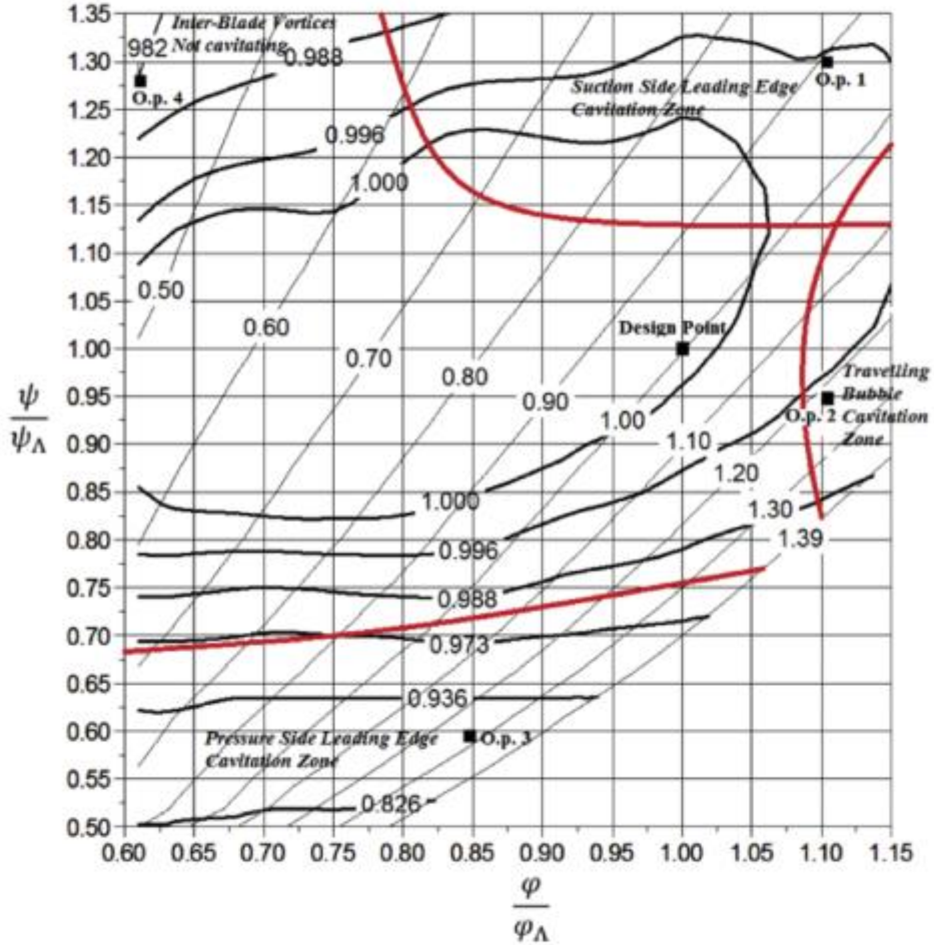
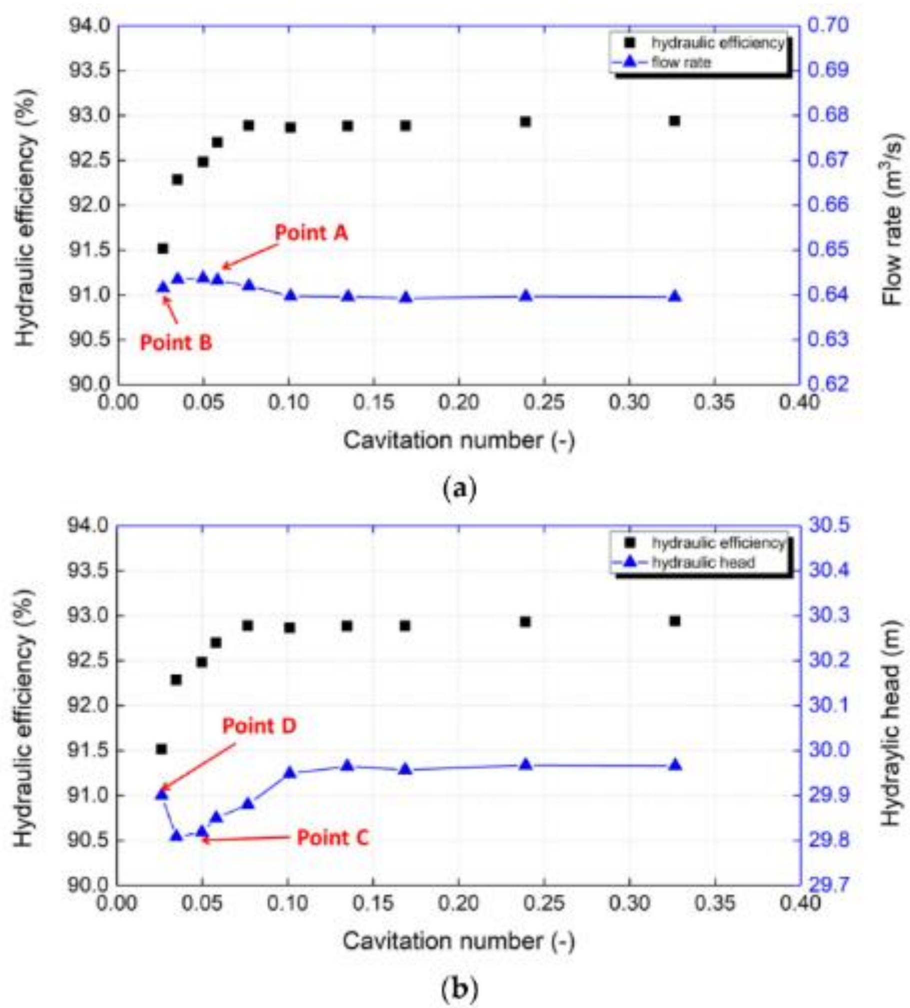


FIGURE 3.32 :
Cavitation for pumps, turbines and PATs expressed by the Thoma number Sigma versus specific speed nq (only valid if operated at bep) (source: R.K. Turton: Principles of Turbomachinery)


Inter-Blade Vortices Not cavitating
O.p. 4
0.988
0.996
1.000
Suction Side Leading Edge Cavitation Zone
O.p. 1
0.50
0.60
0.70
0.80
0.90
1.00
1.10
1.20
1.30
1.39
Design Point
Travelling Bubble Cavitation Zone
O.p. 2
0.973
0.936
Pressure Side Leading Edge Cavitation Zone
O.p. 3
0.826
ψ/ψ_Λ
φ/φ_Λ


hydraulic efficiency
flow rate
Point A
Point B
Hydraulic efficiency (%)
Flow rate (m^3/s)
Cavitation number (-)
(a)
hydraulic efficiency
hydraulic head
Point D
Point C
Hydraulic efficiency (%)
Hydraylic head (m)
Cavitation number (-)
(b)

How to solve or reduce cavitation influence??

Problem with cavitation could be solved (ot try) on different ways :

- Decreasing *z4* , , that is lowering the turbine closer to the river level or even below it.
- Choice of material blade quality, cavitation resistance
- Turbine parts hydraulic desig optimization .
- Avoid exploitation in cavitation zone.