# HYDRAULIC TURBOMACHINES
# Exercises 8 - Pumped Storage Power Stations

**Pumped storage power plant – Bajina Basta**

The Bajina Basta pumped storage power plant is located on the Drina river, about 150km south-west from Belgrade, near the border between Serbia and Bosnia-Herzegovina. The power plant was originally commissioned in 1966. At that time, the project featured two pump-turbines with a nominal power of 281 MW in pumping mode and of 294 MW in turbine mode. The outline of the power plant and a cut-view of one power unit are given respectively in Figure 1 and 2.

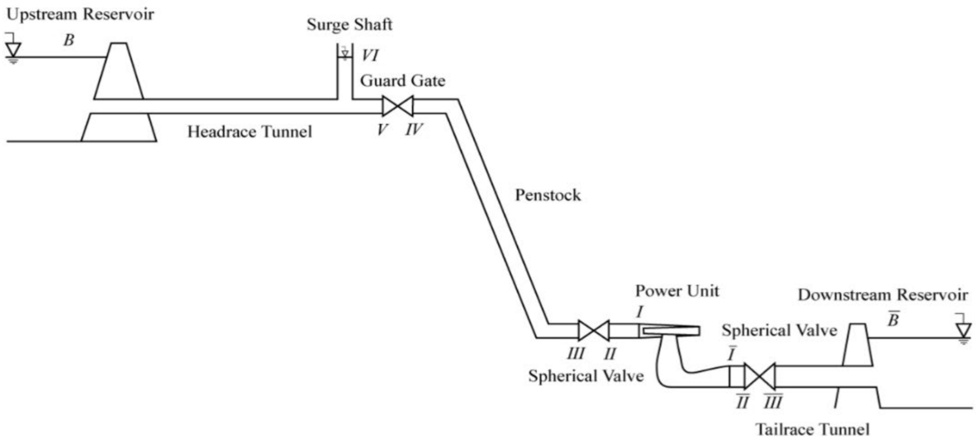



*Figure 1 – Outline of the power station*

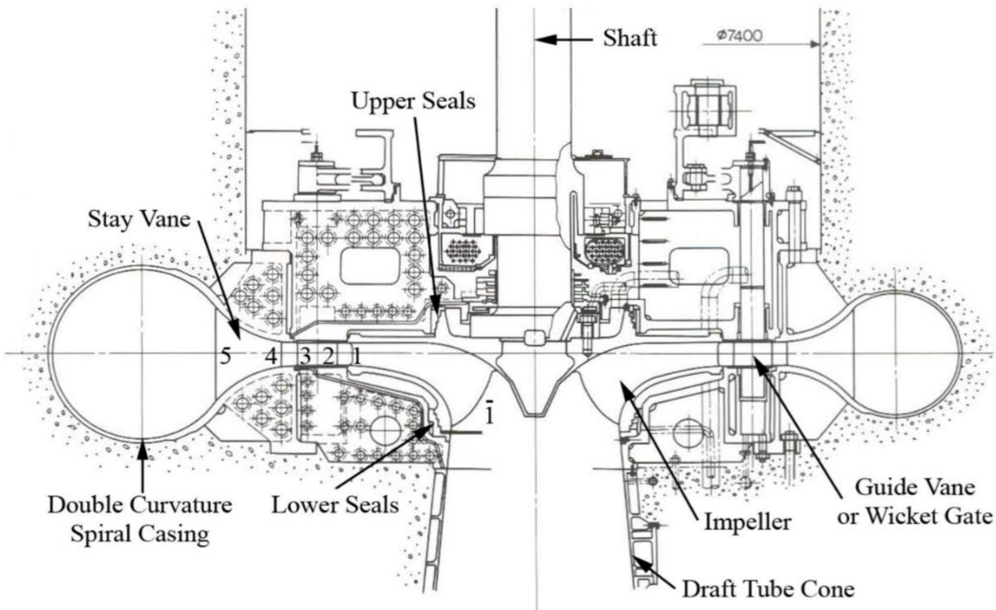



*Figure 2 – Outline of the power unit*

In Table 1, the main operating parameters of the power units are listed. The power P is the electrical power generated/consumed. To simplify, neglect the electrical losses in the generator/motor, so you can consider P as the output power in turbine mode, respectively the input power in pumping mode.

*Table 1 – Operating parameters of the power unit*

| **Pompe** | minimum | nominal | maximum |
|---|---|---|---|
| E (J/kg) | 5214 | 5904 | 6093 |
| H (m) | 531,7 | 602 | 621,3 |
| Q ($m^3/s$) | 50,8 | 41,8 | 36,7 |
| P (MW) | | 281,0 | 310,0 |
| $N_{QE}$ (-) | 0,069 | $\sigma_{min}$ (-) | 0,104 |
| $n_q$ | 23 | $\sigma_{max}$ (-) | 0,159 |
| ν (-) | 0,14 | | |

| **Turbine** | minimum | nominal | maximum |
|---|---|---|---|
| E (J/kg) | 4874 | 5434 | 5885 |
| H (m) | 497,0 | 554,1 | 600,1 |
| Q ($m^3/s$) | 57,0 | 61,8 | 60,5 |
| P (MW) | 243,0 | 294,0 | 315,0 |
| $N_{QE}$ (-) | 0,089 | $\sigma_{min}$ (-) | 0,108 |
| $n_q$ | 30 | $\sigma_{max}$ (-) | 0,171 |
| ν (-) | 0,19 | | |
| $D_{\bar{1}e}$ (m) | 2,180 | f (Hz) | 50 |
| $D_{1i}$ (m) | 4,728 | n (t/s; t/min) | 7,14; 428,6 |
| $D_o$ (m) | 5,605 | $z_r$ (-) | 6 |
| $B_o$ (m) | 0,312 | $z_o$ (-) | 20 |
| | | $z_{avd}$ (-) | 10 |

**Bajina Basta - Turbine mode**

1. In Table 1, three different operating conditions are described: the minimum and the maximum power conditions, as well as the nominal condition. Compute the global machine efficiency of the three different operating conditions. Then, consider the condition with the highest efficiency as the best efficiency point (BEP), and use it for the next questions.

   *The hydraulic power is defined as* $P_h = \rho Q E$

   *The global machine efficiency in turbine mode is* $\eta = \dfrac{P}{P_h}$

   *The efficiency for the three different conditions is*

   $\eta_{\min} = 87.47\% \quad \eta_{nom} = 87.55\% \quad \eta_{\max} = 88.47\%$

   *The Best Efficiency Point (BEP) corresponds to the 315 MW power output, featuring the maximum head at the turbine inlet H = 600.1 m and a discharge Q = 60.5* $m^3s^{-1}$ *(which is, however, not the maximum discharge value). The fact that the BEP doesn't correspond to the nominal operating condition may be surprising. However, during the design phase of a power plants, a lot of variables have an impact on the attained efficiency throughout the entire range of operating conditions. Therefore, the correspondence between the BEP and the nominal operating condition can't often be ensured.*

2. Consider the velocity triangle at the <u>turbine outlet</u>.

a. Compute the meridional component of the absolute velocity and the rotating velocity. Neglect volumetric losses.

*To compute $Cm_{\bar{1e}}$, the area at the runner outlet must be computed:*

$$A_{\bar{1}} = \pi\left(\frac{D_{\bar{1e}}}{2}\right)^2 = 3.73\ m^2$$

*No volumetric losses, so the discharge at the turbine outlet is Q:*

$$Cm_{\bar{1e}} = \frac{Q}{A_{\bar{1}}} = 16.21\ m\ s^{-1}$$

*To compute the rotating velocity component, let's calculate the rotating velocity of the runner:*

$$\omega = 2\pi n = 44.86\ rad\ s^{-1}$$

*Therefore, the rotating speed component at the turbine outlet is:*

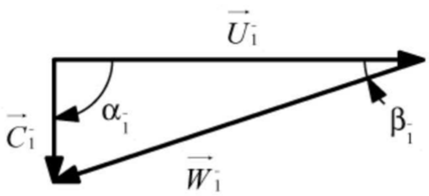
$$U_{\bar{1e}} = \omega\frac{D_{\bar{1e}}}{2} = 48.9\ m\ s^{-1}$$

b. Explain the impact that the hypothesis of neglecting volumetric losses has on the meridional component. Also, what can be said of the tangential component of the absolute velocity at this operating condition?

*As depicted in the previous answer, the meridional component is defined by the discharge going through the turbine. Higher volumetric losses imply lower meridional velocity component.*

*Moreover, the selected operating condition is considered as the BEP. The tangential component of the absolute velocity is $Cu_{\bar{1e}} = 0$.*

c. Draw the velocity triangle.




3. Let's now look at the velocity triangle at the <u>turbine inlet</u>.

a. Compute the meridional component of the absolute velocity and the rotating velocity. Assume $D_{1e} = D_{1i}$ and $B_1 = B_0$.

*Area at the runner inlet:*

$$A_1 = 2\pi\left(\frac{D_{1e}}{2}\right)B_1 = 4.63\ m^2$$

*Therefore:*

$$Cm_{1e} = \frac{Q}{A_1} = 13.05\ m\ s^{-1}$$

*The rotating speed component at the turbine inlet is:*

$$U_{1e} = \omega \frac{D_{1e}}{2} = 106.05 \;\; m\, s^{-1}$$

b. Consider now the Euler equation, $E_t = k_{Cu_{1e}} U_{1e} Cu_{1e} - k_{Cu_{\bar{1}e}} U_{\bar{1}e} Cu_{\bar{1}e}$. Assume uniform flow at the inlet, and 2% of bearings and disk friction losses. Compute the absolute and the relative flow angles, $\alpha_1$ and $\beta_1$, and draw the inlet velocity triangle.

*The transferred turbine specific energy is computed as:*

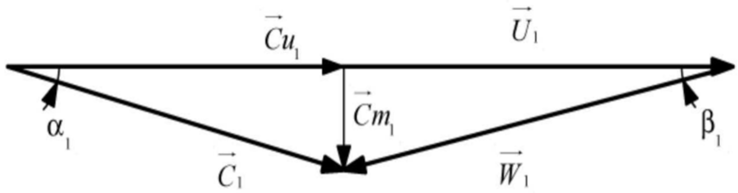
$$E_t = \frac{P}{\rho Q \eta_{rm} \eta_m} = \frac{315 \times 10^6}{998 \cdot 60.5 \cdot (1 - 0.02)} = 5324 \;\; J\, kg^{-1}$$

*With* $k_{Cu_{1e}} = 1$ *due to the uniform flow at the inlet, and* $Cu_{\bar{1}e} = 0$ *at BEP, we can compute:*

$$Cu_{1e} = \frac{E_t}{U_{1e}} = 50.20 \;\; m\, s^{-1}$$

*Therefore :*

$$\alpha_1 = \tan^{-1}\left(\frac{Cm_{1e}}{Cu_{1e}}\right) = 14.57° \;\; and \;\; \beta_1 = \tan^{-1}\left(\frac{Cm_{1e}}{U_{1e} - Cu_{1e}}\right) = 13.15°$$




**Bajina Basta - Pump mode**

4. Compute the different IEC Factors for speed, discharge, torque and power at <u>nominal</u> operating condition, considering 1% of bearing power losses.

*With data of Table 1, the rotational speed* $\omega = n \cdot 2\pi = -44.86 \;\; rad\, s^{-1}$ *and the torque computed as* $T_m = \frac{P \cdot \eta_m}{\omega} = 6.2 \times 10^6 \; N\, m$:

$$n_{ED} = \frac{nD_{1e}}{\sqrt{E}} = -0.4393 \quad Q_{ED} = \frac{Q}{D_{1e}^{\;2}\sqrt{E}} = -0.0243 \quad T_{ED} = \frac{T_m}{\rho D_{1e}^{\;3} E} = 0.01$$

$$P_{ED} = \frac{P_m}{\rho D_{1e}^{\;2} E^{1.5}} = -0.028$$

5. Compute the energy and discharge coefficients for the nominal operating condition. Then, compute the specific speed and compare it with the value provided in Table 1.

*Energy and discharge coefficients:*

$$\psi = \frac{2E}{\omega^2 \left(\frac{D_{\bar{1}e}}{2}\right)^2} = 4.9386 \quad and \quad \varphi = \frac{Q}{\pi\omega\left(\frac{D_{\bar{1}e}}{2}\right)^3} = 0.229$$

*Therefore the specific speed is computed as:*

$\nu = \frac{\varphi^{\frac{1}{2}}}{\psi^{\frac{3}{4}}} = 0.144$ *which verifies the specific speed value* $\nu = 0.14$ *in Table 1.*

*Side note:*

*In hydraulic turbines, the definition of the reference diameter for computing dimensionless numbers can vary depending on the norm. In turbine mode, the reference diameter is by default always the outlet diameter,* $D_{\bar{1}e}$*, but in pump mode the choice of reference diameter can vary between* $D_{1e}$ *and* $D_{\bar{1}e}$*.*

*The most important is that you stay consistent with your definition. If you choose a reference diameter, stick with it for all your computations, unlike we did in this exercise. In turbine mode, use the outlet diameter as reference, and in pump mode choose between the inlet and outlet diameter and stay consistent through the whole exercise.*

**Pumped storage power plant – FMHL+**

Let's now consider the Veytaux II powerhouse of the FMHL pumped storage power plant, already studied last week in the framework of the exercise series focused on industrial pumps. A cut-view of one of two ternary units, each one featuring two hydraulic machines (a Pelton turbine and a centrifugal multistage pump) and an electrical machine on the same shaft, is given in Figure 3.

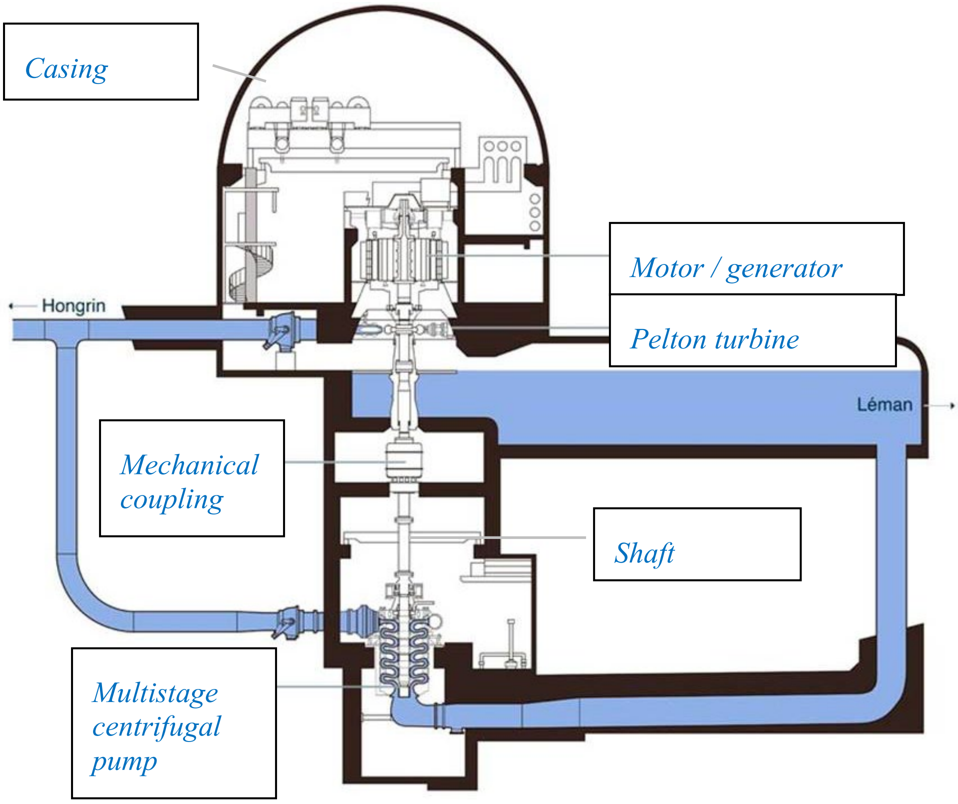



*Figure 3 – Veytaux II ternary unit cut view. Retrieved from www.alpiq.com*

6. Complete the nomenclature in Figure 3 and specify what type of hydraulic machines are used in this ternary unit.

   *See Figure 3.*

7. In pumping mode, are the turbine and the multistage storage pump coupled by mean of the mechanical coupling? Why?

   *In reason of the layout of this ternary unit, namely with the Pelton turbine which is located on the shaft between the pump and the electrical motor, the turbine and the multistage pump must be mechanically coupled during pumping operation to allow the transmission of power from the motor (which is taking the power from the grid) to the pump through the shaft. Therefore, in pumping mode, the turbine rotates in air, with a small amount of water used to cool down the structure heated because of the frictions.*

   *Moreover, in case of operation as hydraulic bypass (also known as hydraulic short circuit mode), a fraction of the pumped discharge passes through the Pelton turbine, to balance the grid power. This occurrence is needed when the centrifugal pump is a ON/OFF machine: its operating condition cannot be adjusted, so it operates consuming its rated power value. During this operating mode, the pump and the turbine must also be coupled by mean of the mechanical coupling.*

8. Which locations of the hydraulic circuit inside the pump are particularly concerned by the risk of cavitation phenomena inception? Does each stage feature the same risk of experiencing cavitation?

   *The blades of the impeller, especially at the leading edge, and the volute can experience damages due to cavitation more frequently. Since cavitation occurs at low-pressure locations, the first stage of the pump is the one mainly concerned by these damages. Given that each stage of the* *pump contributes to increase water pressure, the risk of cavitation inception in the following stages is strongly reduced.*

Suppose now that we are facing a situation of power excess in the grid. The power plant is therefore required to consume power from the grid, and for this reason we need the multistage pump to be in operation. However, the pump is designed for a rated power of 120 MW, whereas the excess power in the grid is way lower. Since the pump is an ON/OFF machine, its operating condition can't be adjusted, and the ternary unit is operating as hydraulic bypass.

$\rho = 998$ kg m$^{-3}$

$g = 9.81$ m s$^{-2}$

9. The pump is working at $Q$ = -11.5 m$^3$s$^{-1}$, $E = 9000$ J kg$^{-1}$ with a global machine efficiency of $\eta^P = 0.89$. Calculate the input power required by the pump.

   *The hydraulic power of the pump is calculated as follows:*

   $$P_h^P = \rho Q^P E^P = -103.29\ MW$$

   *The input power required by the pump is:*

   $$P^P = \frac{P_h^P}{\eta^P} = -116.06\ MW$$

10. Assuming an electrical efficiency of $\eta_{el} = 0.99$ for the motor/generator, and that the excess power from the grid is $P_{grid} = -80$ MW, calculate the output power provided by the Pelton turbine.

    *Considering the losses in the electrical machine, the power at the shaft is:*

    $P_{TOT} = P_{grid} \cdot \eta_{el} = -79.2$ *MW. The negative sign means that the ternary unit is net consumer of power.*

    *The output power of the Pelton turbine can therefore be deduced:*

    $$P^T = P_{TOT} - P^P = -79.2 + 116.06 = 36.86\ MW$$

11. The Pelton turbine is working at a specific energy $E^T = 8927$ J kg$^{-1}$ with a global machine efficiency of $\eta^T = 0.91$. Compute the discharge $Q$ pumped in the headwater reservoir. Neglect volumetric losses.

    *Let's compute the hydraulic power of the Pelton turbine:*

    $$P_h^T = \frac{P^T}{\eta^T} = 40.51\ MW$$

    *The discharge which is going through the Pelton turbine is calculated as:*

    $$Q^T = \frac{P_h^T}{\rho E^T} = 4.55\ m^3 s^{-1}$$

    *The discharge pumped in the reservoir is therefore:*

    $$Q = Q^T + Q^P = 4.55 - 11.50 = -7.05\ m^3 s^{-1}$$