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# Design of a reversible pump turbine

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**Abstract.** The design of reversible pump turbines for the purpose of retrofitting existing hydropowerplants has been investigated. A preliminary design for further analysis with computational fluid dynamics has been made. As this is current work, the project has not yet been finished, and the author does not yet know if the resulting pump turbine has good efficiency or not.

## 1. Introduction

In recent years, Europe as a whole has started to rely more and more heavily on renewable energy sources like wind and solar power, as shown in figure 1. The drawback with these sources

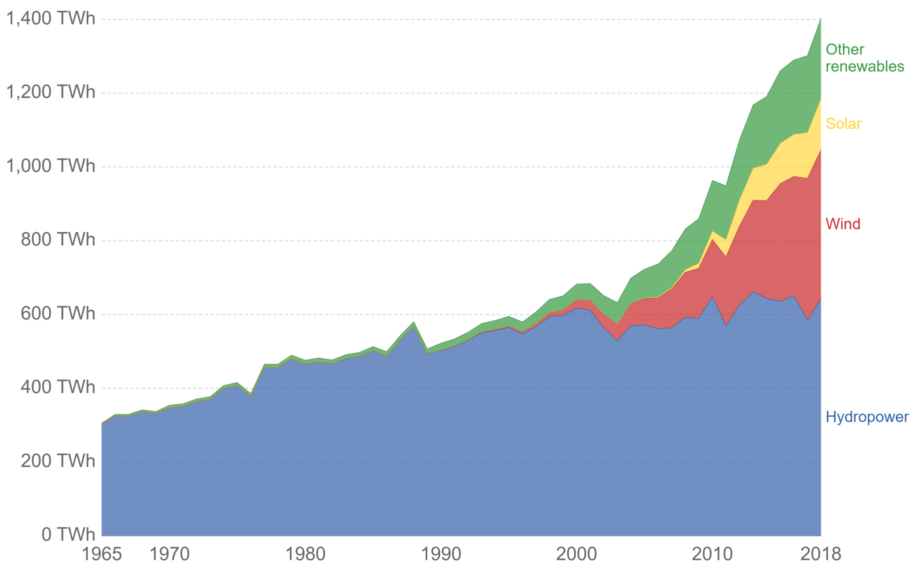


**Figure 1.** Amount of renewable energy sources in Europe, collected from [1]

is that they are directly influenced by the weather, and thus cannot be used on demand. This necessitates good techniques for energy storage. The most economical technology for storing energy is the application of pumped storage technology[2], where excess energy is taken from the grid and used to pump water to a higher reservoir. When there is a demand for more energy in the grid the same water is run through a turbine to regenerate the energy. However, to be able to use pumped storage technology, one needs a reservoir pair and a height difference.

Norway has very good conditions for the application of pumped storage technologies. The country already has approximately 50% of Europes hydropower reserves and has utilized water as its primary energy source for a long time, as shown in figure 2. Therefore most of the

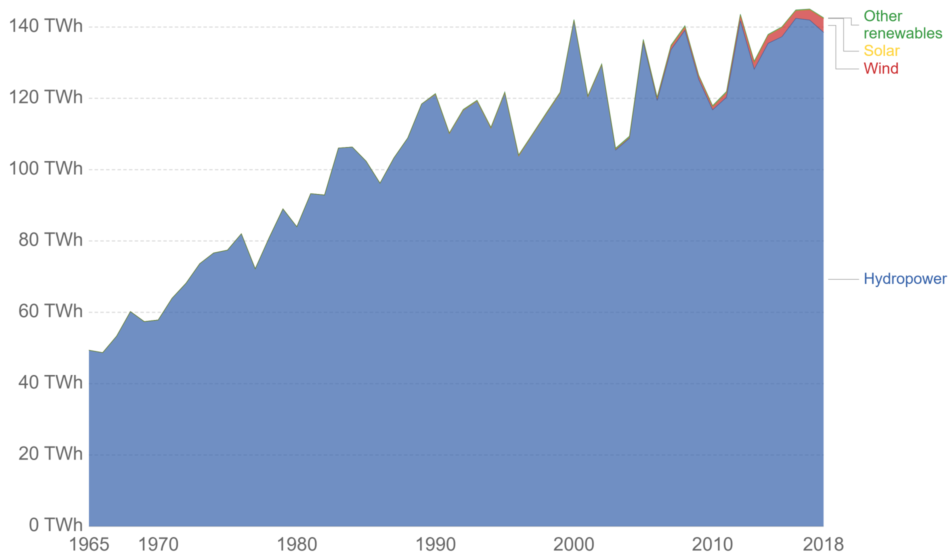


**Figure 2.** Amount of renewable energy sources in Norway, collected from [1]

available hydropower resources are either already utilized as regular hydropower plants or are protected due to environmental concerns. There are only 9 powerplants with pumping capability in Norway[3]. Therefore there is room for more pumped storage powerplants in the Norwegian grid.

One idea to make new pumped storage hydropower plants at low costs is to reuse existing powerplants and retrofit them with reversible pump turbines (RPTs) which are to function both as pumps and as turbines.

There are some problems connected to this approach. One is that pumps need larger submergence than turbines to avoid the fluid from evaporating due to low-pressure zones, so-called cavitation. A possible solution for this may be to insert a booster pump downstream of the RPT. Booster pump technology is currently under development in a project parallel to this one.

The other problem is that a Francis turbine requires a lower radius to operate at best point efficiency (BEP) for a given upstream water height than a radial pump requires to pump water up to the same height, with the same volumetric flow rate. Additionally, there is a pressure height difference created by the friction in the penstock which reduces the pressure height experienced by a turbine and increases the pressure height a pump has to deliver.

In this paper, Roskrepp powerplant, owned by the Sira-Kvina Power company will be used as a baseline for the design of an RPT for retrofitting. The turbine will be designed for $H = 59\,\mathrm{m}$. This is not the maximum difference between the reservoirs but assumed to be a reasonable height for the pump to operate in. While the maximum volumetric flow rate of the turbine is $Q = 70\,\mathrm{m^3\,s^{-1}}$, the actual design flow rate is $Q = 50\,\mathrm{m^3\,s^{-1}}$. The current rotational velocity is 250 rpm.

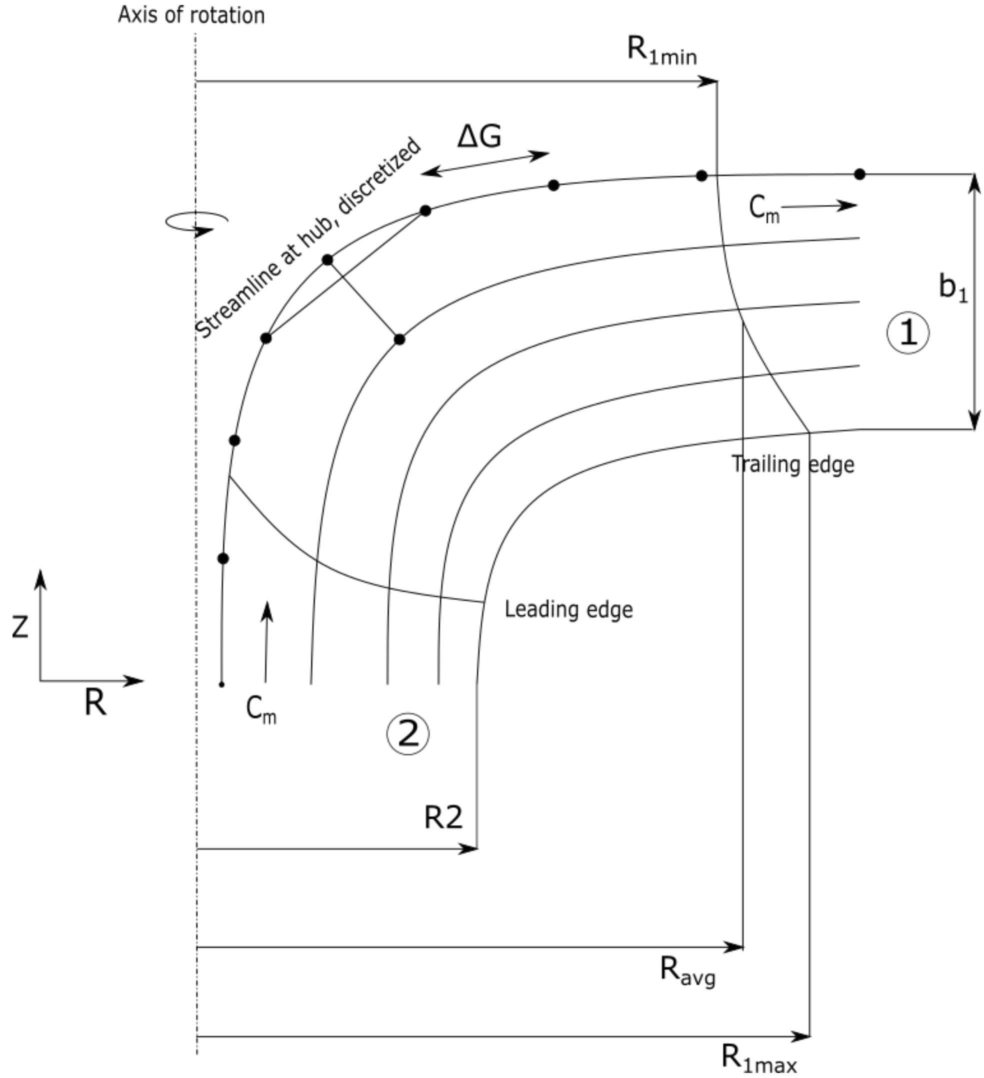


**Figure 3.** Projection of the R-Z plane in a radial turbomachine

## 2. Theory and methodology

In the remainder of this text, the subscripts related to location in the RPT will be as shown in figure 3. Inlet and outlet will be the inlet and outlet in pump mode, so 2 and 1, respectively.

### *2.1. Fundamental idea*

In the context of turbomachines, pressure is usually called head and is given in meters of water column [mWc]. The lifting head of a pump is given by the Euler equation for turbomachines:

$$H_{th} = \frac{u_1 c_{u1} - u_2 c_{u2}}{g} \qquad \text{[mWc]} \quad (1)$$

where $g$ is the gravitational acceleration, $H_{th}$ is the theoretical possible pumping head, $u$ is the peripheral velocity of the runner and $c_u$ is the peripheral velocity of the fluid[4]. A similar equation can be used to find the power generated by a Francis turbine.

The main idea when designing an RPT is as follows: A radial pump can function as a Francis turbine when put in reverse. While the power production of a pump put in reverse will be sub-optimal, it will still produce some power. A Francis turbine will also work as a pump if it is reversed, but there is nothing to guarantee that it will produce the required amount of head. For this reason, an RPT will always be designed as a pump, to ensure that it behaves satisfactory both in pump and turbine mode.

Because the main dimensions of the RPT are predefined by the current arrangement, these cannot be freely designed and optimised as one would do if one were to design an entirely new powerplant. If the booster pump works very well, it may make up for the lack of head on behalf of the RPT, but as it is still under development, the booster pump performance is unknown.

The other possibility to increase the head of the pump is the internal design of the runner, more precisely the design of the runner vanes.

By application of trigonometry to the velocity-diagram at the outlet and the assumption of inlet flow without swirl, equation (1) may be rewritten to

$$H_{th} = \frac{u_1^2}{g} - \frac{u_1 c_{m1}}{g \tan \beta_1} \qquad \text{[mWc]} \quad (2)$$

where $c_m$ is the meridional velocity component of the flow and $\beta_1$ is the angle of the blade relative to the radial direction.

A graph relating the volumetric flow rate and the head of a pump is commonly called a pump characteristic. as seen in equation (2) is particularly important for the shape of the characteristic, as it is related to the slope and the lifting height. The inlet angles also play an important role, to make the swirl-free inlet flow shockless.

The peripheral outlet velocity is also influenced by the phenomenon of slip, which is deflection of the outgoing flow due to pressure differences between the suction side and the pressure side of the blade, as illustrated in figure 4. Slip is only a relevant phenomenon in pump mode. During turbine operation, all pressure differences between the pressure side and the suction side should be transferred to the runner in the form of energy, and so the outgoing flow should have uniform pressure distribution. Another aspect influencing the pumping head is the hydraulic efficiency,

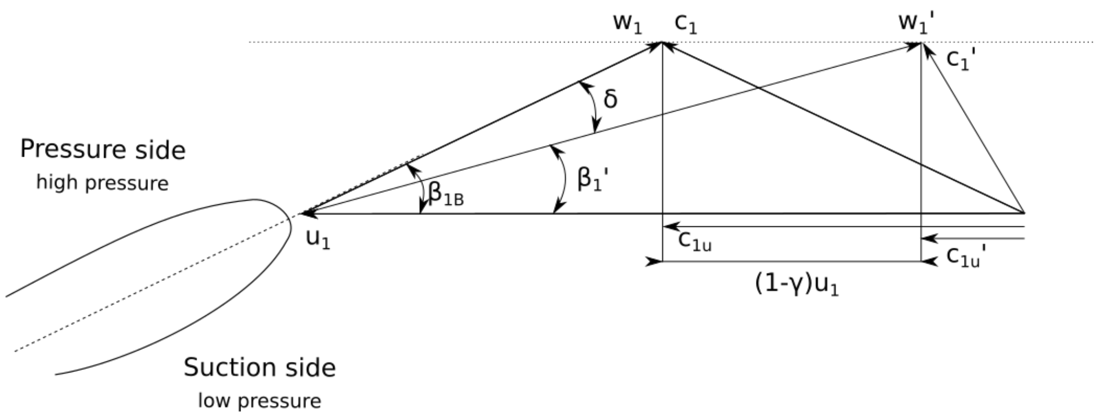


**Figure 4.** illustration of slip at the outlet of a pump, illustrated by use of velocity diagrams

$\eta_h$. Hydraulic efficiency is the ratio of the energy which may be utilized to the energy input to the pump[5]. It can be written in terms of head as:

$$\eta_h = \frac{H}{H_{th}} \qquad \text{[-]} \quad (3)$$

where $H$ is the actual lifting head and $H_{th}$ is the theoretical possible lifting head, assuming no hydraulic losses between the suction and discharge nozzle of the pump. Hydraulic losses are losses related to skin friction and turbulent dissipation[6],[7]. Some sources include the slip in the models for slip factor, as incidence losses[8], while others use it as a separate value[5]. The approach utilized in this paper applies a model for a slip factor, $\gamma$, to estimate the slip. $\gamma$ is shown in figure 4.

There are also hydraulic losses in the penstock of the powerplant. Those are not incorporated in the hydraulic efficiency of the pump but are added to the system curve of the powerplant. The system curve is the pump head as observed by the turbine at any volume flow rate, $Q$. Thus the lifting height the pump has to overcome at the design conditions, $H_{required}$, is actually given by:

$$H_{required} = \Delta H_{reservoir} + H_{loss}(Q) \qquad [\text{mWc}] \quad (4)$$

where $\Delta H_{reservoir}$ is the difference in height between the upper and the lower reservoir at the design point, and $H_{loss}(Q)$ is the head loss in the penstock as a function of $Q$. $H(Q)$ has previously been measured for the current powerplant.

### *2.2. Design*

In the following, a process for the design of the blade is outlined[9].

One starts by designing the radial projection of the turbine, which is one blade in the $Z$-$R$ plane, as shown in figure 3. The outlet height is defined as $b_1$, and is given by the existing geometry, the same goes for the outlet radius, $R_1$. This is used to define the projection of the streamline along the hub. Initially, the hub is chosen to be shaped like part of an ellipse. then the shape of the shroud is found by defining a number of streamlines, starting at the outlet, and extending them to the inlet by using the hub as a reference point. When one does this, one utilizes that the mass flow rate between two streamlines is always constant[10]. For water, which is incompressible, this translates to a constant volume flow rate between the streamlines.

In regular pumps and turbines, one usually designs for the flow to accelerate through the runner. This is to avoid separation in the flow due to adverse pressure gradients, so due to flow which flows from a low-pressure area to a high-pressure area. Separation leads to stall in a radial turbine, in the same way as it would lead to stall in regular airfoils.

In an RPT, however, the fluid has to flow in both directions. Therefore there can be no significant acceleration of the flow through the runner in either direction, as this would lead to retardation in the opposite direction. For a flow with constant volume flow rate, and constant meridional velocity, the meridional area between the streamlines has to be constant along each streamline. So:

$$c_{m1} = c_{m2} = c_m \qquad \text{m}\,\text{s}^{-1} \quad (5)$$

$$\Rightarrow A_{m1} = A_{m2} = A_m \qquad \text{m}^2 \quad (6)$$

where $A_m$ is the meridional area. The new streamlines can, therefore, be found by discretizing the initial streamline along the hub and defining the start of all the other streamlines along the outlet. Then one requires the meridional area between the streamlines to be constant and thus the distance between two streamlines at any given point can be found.

When the streamlines are found, the blades trailing edge (TE) and leading edge (LE) can be defined. These are also projected in the R-Z plane. The shape of these are very much up to the designer, and are initially chosen to be in the form of two Bezier curves, see figure 3

Now that the shape of the blade in the R-Z plane is fully designed, the distribution of the angles throughout the blade may be defined.

First of all, the inlet and outlet angles have to be found. This is done by using a model for the hydraulic efficiency and applying equations (2) and (3) to find a suitable value for the outlet blade angle.

Requirements for the blade angle are that the desired pump head is achieved at the desired volume flow rate and that the pump characteristic is as steep as possible. Steepness is desired to be able to operate at many different head values while changing the volume flow as little as possible. This is illustrated in figure 5. Because the steepness of the pump characteristic

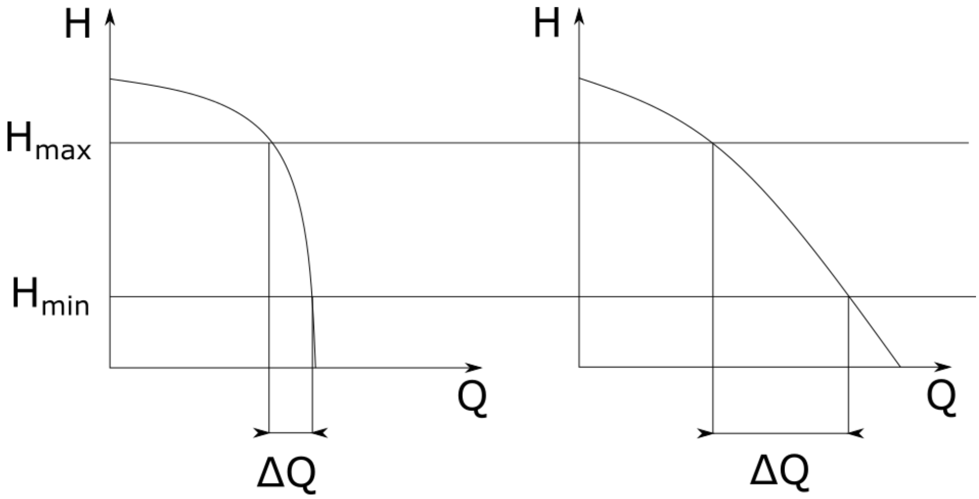


**Figure 5.** Relation between steepness of pump characteristics, and change in $Q$ to obtain the desired value for $H$.

decreases with increasing blade outlet angle, $\beta_{1B}$, the desired outlet blade angle will always be the lowest angle which provides sufficient head. When finding $\beta_{1B}$ one uses values at an average outlet radius, $R_{1,avg}$. However, when the radius varies along the trailing edge, so will the blade angle. To find the corresponding blade angles for a varying blade, one uses $R_{1,avg}$ and the corresponding blade angle as reference values and applies free vortex theory to find the remainder of the angles. A free vortex is defined by

$$c_u = \frac{K}{R} \qquad [\mathrm{m\,s^{-1}}] \quad (7)$$

where $K$ is a constant.
The values for the blade angles can be found by

$$\beta_B = \arctan\left(\frac{c_m}{u - c_u}\right) \qquad [^\circ] \quad (8)$$

Due to the requirement of no acceleration of the flow, $c_m$ will remain constant. $u$ and $c_u$ will both be dependent of the radius.

The angles at the inlet are defined by no pre-swirl in the incoming flow, i.e. $c_{u2} = 0$. The angles are found by use of equation 8 there as well.

To find the angles from LE to TE for the initial guess, the only requirement for the distribution of the angles along the blade is that they are changing smoothly from inlet to outlet.

To be able to transform the distribution of $\beta$ along the blade into points in the radial plane, an additional plane is defined to simplify the transformation, namely the $G$-$H$-plane. $G$ is the length of a streamline in the $R$-$Z$ plane and $H$ is the length of the streamline in the $R$-$\theta$ plane. the length of $G$ between two discrete points along a streamline is shown as $\Delta G$ in figure 3 and

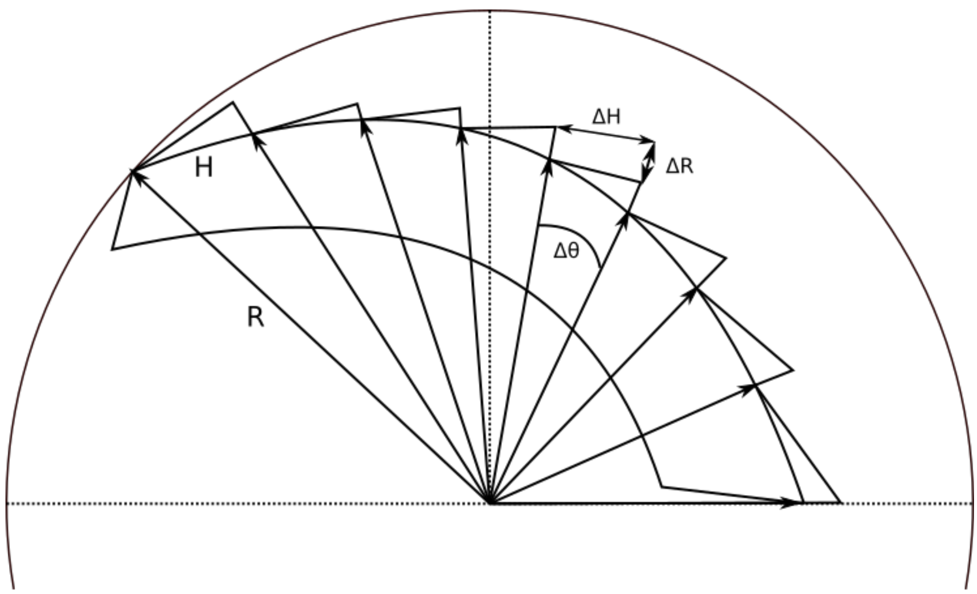


**Figure 6.** Schematic of the $R$-$\theta$ plane.

the length of $H$ between two discrete points is shown in figure 6. The values of $G$ can be found from the streamlines in the $R$-$Z$ plane, and the values of $H$ can be found as by using the values for $\beta$ along the blade[11].

When the streamlines are defined on the $G$-$H$ plane, they can be defined on the $R$-$\theta$ plane by using the knowledge about the values of $R$ and $\theta$ for each point, as in figure 6. By further applying the known values of $Z$ from the $Z$-$R$ projection, the blade is modelled in three dimensions.

Now all that remains is to add thickness to the blade. This is done by adding a layer on both sides of the blade, offset normally from the surface created by the streamlines. These two layers are further connected at both the LE and the TE with an elliptical profile. While a regular pump or a regular turbine would have an elliptical-shaped LE and a sharper TE, this would be unfavourable for an RPT, as it would lead to less streamlined design in one operation mode.

## 3. Results

When assuming the design conditions to be $Q = 50\,\mathrm{m^3\,s^{-1}}$, $H = 59\,\mathrm{mWc}$ and $n = 250\,\mathrm{rpm}$, and by following the steps outlined in the paper, the geometry for one blade was as shown in figure 7. The blade has the modelled characteristic as shown in figure 8. In the applied model the hydraulic efficiency and the slip are found separately. The distribution of $\beta$ along the blade is shown in figure 9.

## 4. Discussion

There are many different philosophies for the design of a blade. The one outlined in this paper is a very basic one, which takes hold in theoretic, idealized design methods, and utilizes models for the slip and the hydraulic efficiency. The reason for this is that retrofitting RPTs has not been done before, so empiric data for other kinds of pump turbines may not apply to the case. Instead of empirical data, the further design of the RPT will be based on CFD analysis for optimization.

Models for slip are usually quite uncertain. Therefore it might be more reasonable to make the initial design of the turbine without accounting for slip and adjusting the outlet angle according to the results from a CFD-analysis. The same may be valid for the efficiency.

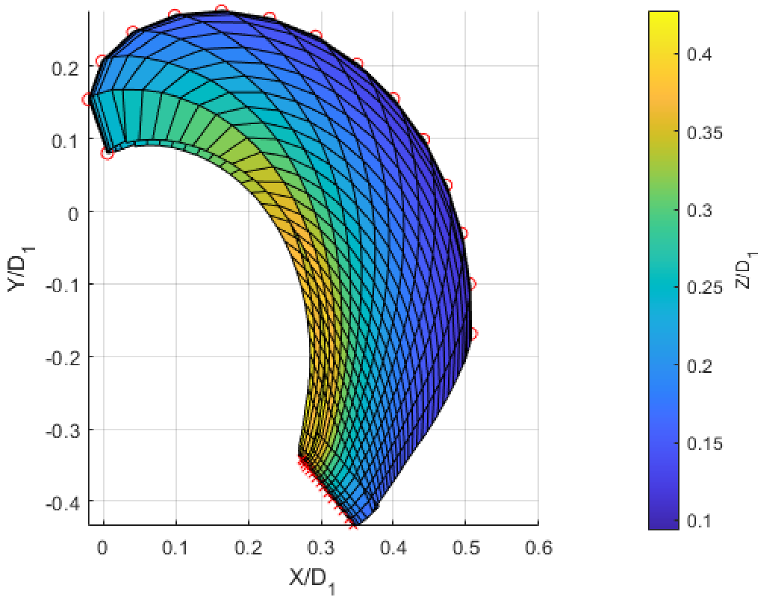


**Figure 7.** The geometry of one blade. The red crosses denote the trailing edge, and the red circles denote the leading edge. The values have been made dimensionless by dividing all by $D_1$

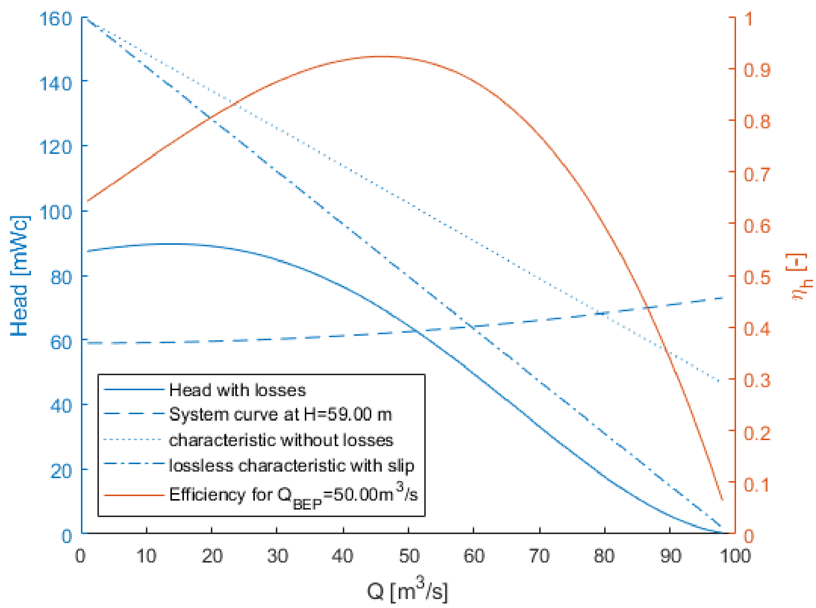


**Figure 8.** Theoretical pump characteristic for the current design and modelled efficiency curve

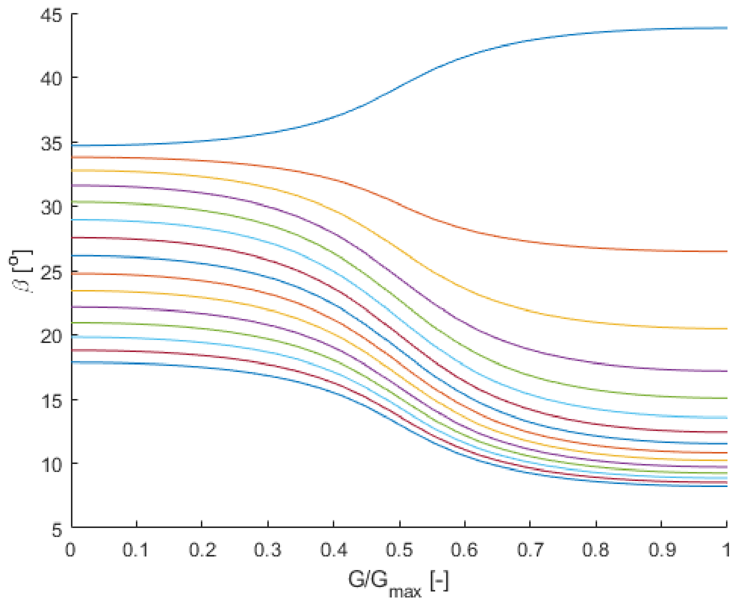


**Figure 9.** Distribution of the values for $\beta$ along the blade, plotted against dimensionless values of $G$. Here G starts at the outlet and reaches $G_{max}$ at the inlet

## 5. Further work

With the initial geometry of the blade ready, it remains to test it with Computational Fluid Dynamics (CFD).

Once the CFD results for the initial design are ready, the blade should be adjusted according to the results. If the blade does not produce enough head, the angle of $\beta_1$ must be increased. If the blade has pressure zones below the critical pressure, these have to be removed by smoothing out the $\beta$-distribution.

The behaviour of the RPT should also be examined at off-design conditions.

When a satisfactory design for the blade has been reached, it has to be simulated together with the booster pump, to ensure good cooperation between the two machines, both in pump mode and in turbine mode.

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