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Pump-Turbines in Conventional Hydropower Plants

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Abstract. This paper proposes an innovative approach to retrofit existing hydropower plants with suitable upper and lower reservoirs into pumped storage. The conventional methods are both comprehensive and expensive, mainly due to the increased risk of cavitation during pumping operation. In an existing facility, the installed runner is already sufficiently submerged to avoid unfortunate cavitation, but replacing the runner with a reversible pump-turbine demands an even further increase of the static pressure at the low-pressure side of the turbine. An alternative to the traditional method is discussed in this research by introducing the concept of a booster pump installed in the draft tube. Increasing the pressure at the inlet of the pump-turbine during pumping could eliminate the risk of cavitation and enable operation of a pumped storage plant without submerging the runner further. This conceptual paper discusses the motive for developing pumped storage and presents two uses for the proposed booster pump; eliminating cavitation and correcting for some of the necessary lifting head of the pump-turbine. The project is a part of the Norwegian Research Centre for Hydropower Technology, HydroCen.

1. Introduction

Recent years, efforts to reduce the greenhouse gas emissions have led to a massive development in renewable energy around the world. According to the Paris agreement, all Parties to the United Nations Framework Convention on Climate Change (UNFCCC) are committed to work out national climate actions to keep the global temperature increase below 2°C, and continue the effort to limit the increase to 1.5°C [1]. In accordance with the agreement, EU has decided that renewable energy should constitute at least 32.5 % of the total energy within EU border by 2030 [2]. The development of renewable energy is getting increasingly profitable, as the costs associated with solar and wind technology are falling for each year.

However, while Europe experiences a tremendous growth in renewable development, the demand for energy storage capacity to balance the grid is increasing. Although the battery prices are decreasing and their storage capacity is improving for each year, the capacity offered by pumped-storage hydropower is still superior, and the only large-scale renewable alternative in the immediate future.

Various power companies and research institutions have stepped up the effort to map the potential of pumped storage. According to the International Renewable Energy Agency (IRENA)[3], pumped storage accounts for 161 GW of today's global storage capacity and estimates that this number has to at least double by 2050 in order to follow the development of variable renewable energy. The eStorage project, supported by the European Commission, has prepared a report compiling the pumped storage potential by assessing the available water reservoir pairs in EU, Norway and Switzerland, concluding that the total realizable potential constitutes around 33 % of the theoretical potential of 6924 GWh [4]. The individual findings are summarized in Figure 1.

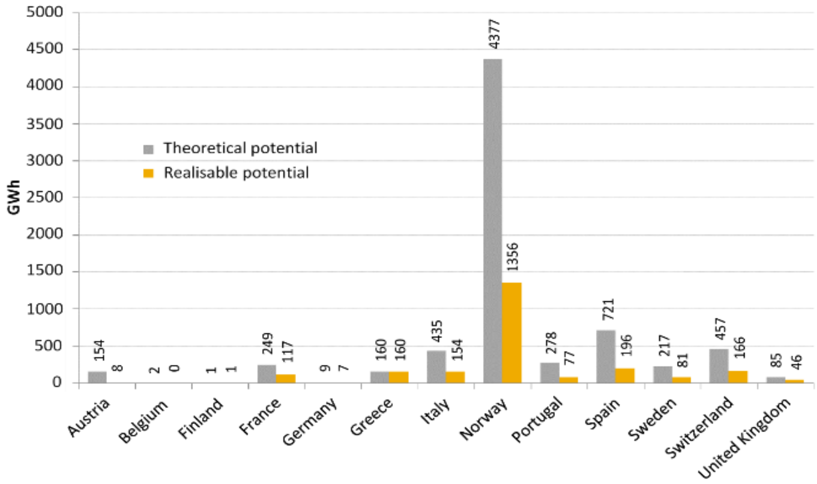

Figure 1: Theoretical and realizable potential of pumped storage in EU, Norway and Switzerland. Graph obtained from eStorage_D4.2: *Overview of potential locations for new variable speed PSP in Europe* [4].

A similar mapping of available reservoir pairs has been carried out by The Norwegian Water Resources and Energy Directorate (NVE), with the purpose of retrofitting existing hydropower plants into pumped storage in Norway[5]. Transforming hydropower plants with existing upper and lower reservoirs into pumped storage is less extensive than projecting entirely new pumped storage plants, as waterways, construction roads and power station are already established. However, the investment costs associated with such transformation are still assumed to be extremely high.

1.1. Scope and limitations

This paper introduces a concept removing the risk of cavitation in the hydraulic machinery when transforming existing hydropower plants into pumped storage. The scope includes an examination of the concept, some technical considerations regarding usage, design and installation, and an indication of possible economic benefits. It is necessary to emphasize that the concept is still within an initial stage and that no financial analysis has been carried out to evaluate its profitability. Still, the objective of the ongoing research is to provide a technical solution that is cost-wise competitive to other alternatives.

The review mainly concentrates on the turbine and the associated arrangement. The scope excludes possible necessary modifications of water tunnels, environmental and geological considerations and other elements affected by reversed and changed operational pattern. Alternative methods to transform conventional power plants into pumped storage have not been examined.

2. Presentation of concept

2.1. Background

It is a known problem that cavitation poses a major challenge to hydraulic machines, and it is even more severe and damaging in pumping units compared to ordinary turbines, given the unfavorable in-flow conditions on the suction side during pumping operation. Cavitation is highly undesirable given its ability to damage the runner and is therefore prevented by submerging the turbine arrangement to increase the static pressure in the draft tube. In existing power plants, the turbine is positioned sufficiently low relative to the tailwater level to avoid exposure to cavitation. If the turbine is replaced by a reversible pump-turbine (RPT) to give pump storage capabilities, different fluid pressures will be present during turbine and pump operation. Typical hydraulic grade lines (HGL) for both operational modes are illustrated by the dashed lines in Figure 2. Frictional losses in the waterways and changed premises for cavitation of the unit result in a low-pressure zone at the pump-turbine inlet during pumping, as seen by the HGL. Additional submergence of the RPT will obviously increase the static pressure, but such submersion will demand comprehensive civil works for tunnel excavation and relocation of the power station. The required civil work will involve substantial costs and lost income due to operational downtime. Allowing for pumping in existing facilities will therefore demand a lift of the HGL to prevent cavitation from occurring.

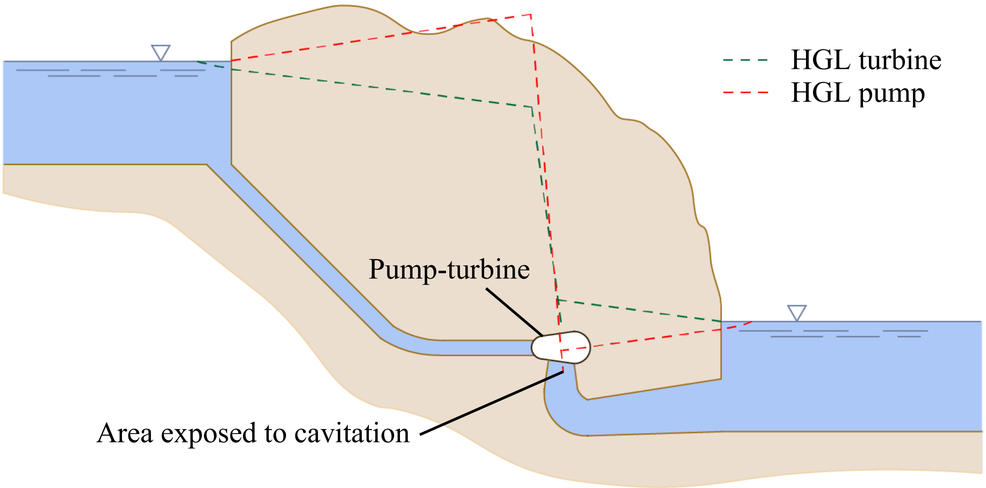

Figure 2: HGL for turbine and pump operation.

2.2. Proposed booster pump solution

Installation of a booster pump in the draft tube is proposed as an alternative to the traditional runner submergence. Booster pumps are characterized as pumping units aiding to lift the pressure of a system, typically to improve the required net positive suction head (NPSH) [6]. In this particular case, the booster pump will boost the pressure at the RPT inlet in pumping mode to lift the available HGL of the system. As the induced pressure from the booster pump also affects the overall pump characteristics of the serial connected pumping units, the booster pump can be utilized to handle some of the required lifting head of the pump-turbine. The following sub-sections discuss how a booster pump affects the hydraulic grade lines and the corresponding required lifting head of a system, and how it in practice can handle two problems related to turbine retrofit.

2.3. Booster pump compensating for submergence

In order to compensate for the required submergence, the booster pump must be able to provide a head that lifts the hydraulic grade line sufficiently, as shown in Figure 3. The figure illustrates a fluid head slowly decreasing from the lower reservoir towards the pump-turbine, marked by a red dashed line. At the position of the booster pump, it experiences a sudden lift before entering the draft tube and pump-turbine. As distinct from Figure 2, it is now possible for the pump-turbine to operate in an adequate pressure zone. The decreasing pressure corresponds to the head losses in the waterways, which are losses due to friction, change of flow pattern, flow mixing etc in the tunnels, and the booster pump and the pump-turbine must overcome these losses to ensure a sufficient HGL over the whole system.

The risk of cavitation can be evaluated by means of the $NPSH$, defined as the difference between the absolute pressure and local vapor pressure downstream the turbine. The following equation, derived from the energy equation ([7], [8]), estimates the necessary submergence of the runner, i.e. the required difference between the lower reservoir and the runner:

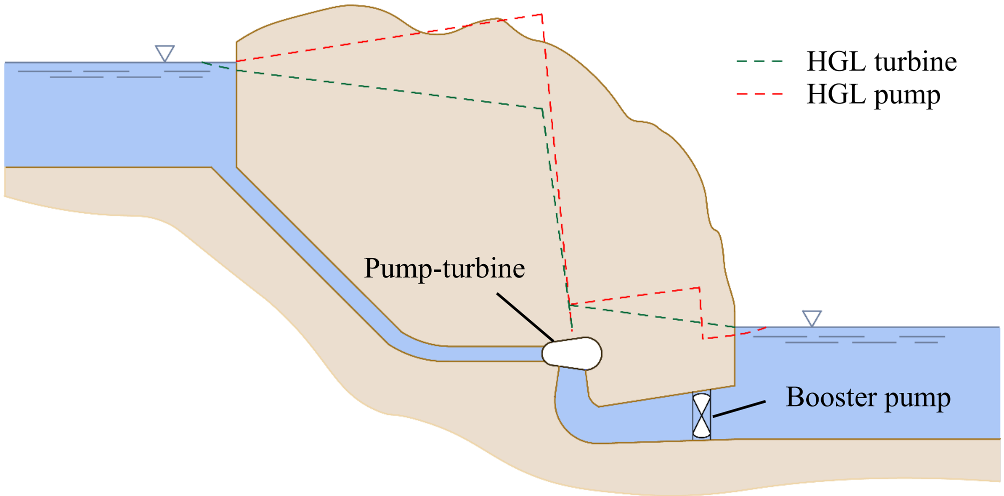

Figure 3: HGL of a pumped storage plant where a booster pump compensates for submergence.

$$H_s \leq h_b - h_{va} - NPSH_R \ [m]. \tag{1}$$

h_b denotes the barometric pressure, h_{va} the vapor pressure and $NPSH_R$ the required NPSH, which is dependent on the inlet velocities in pumping mode of operation and the circumferential speed of the turbine. The booster pump must be dimensioned for a pressure head managing both the change in submergence ΔH_s and the tailrace head losses to be able to handle the cavitation issues of the RPT in reversed mode.

2.4. Booster pump enabling a new RPT design

A further utilization of the booster pump concept regards the possibility of maintaining most of the existing turbine arrangement. In general, a new pump-turbine for retrofitting purposes demands a larger outlet diameter than the original turbine to achieve a higher lifting head, and the associated components, such as the guide vanes, draft tube and spiral casing, must be replaced to fit the new dimensions and to handle the changed pressure conditions during pumping. This involves demolishing large parts of the existing power station, extensive civil works and costs hardly justifying a booster pump solution only assuring cavitation free operation of the RPT.

Instead of increasing the diameter of the RPT, the additional required head could be handled by the proposed booster pump. The booster pump must then be able compensate for the head losses in the whole system (Figure 4), in practice assuring a significant part of the overall required lifting head. Alternatively, the rotational speed of the RPT can be increased to obtain a higher lifting head. This will obviously require a new or modified generator.

Allowing a new RPT design to, more or less, fit into the existing turbine arrangement enables new design considerations for pumped storage plants. The pump-turbine can maintain the original turbine inlet height, inlet and outlet diameters, and the remaining turbine arrangement can be retained with some modifications. Moreover, as the pump-turbine now may be designed with a pump net head close to the original turbine net head, the pump-turbine will be able to operate more advantageous in turbine mode. This is due to the fact that pump-turbines generally are dimensioned for a higher head than what the turbine runs at to overcome the friction losses in pumping mode of operation, resulting in a poorer turbine efficiency. Instead of operating outside its optimal operational area, the turbine can, with the proposed booster pump modifications, operate around the intended head, increasing the performance of the turbine. On the other side, some drop in the overall pump efficiency is inevitable

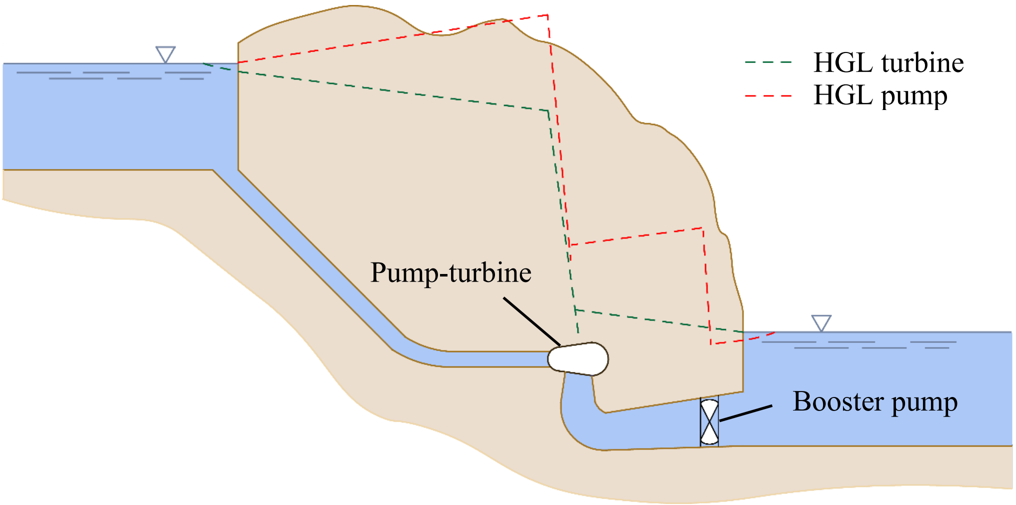

Figure 4: HGL of a pumped storage plant where the booster pump overcomes the frictional losses in the waterways.

when including the booster pump efficiency. Still, a booster pump correcting for the necessary pump-turbine head can potentially be a cost-effective alternative to a larger turbine and the associated arrangement. It is therefore a highly interesting concept for further evaluation.

3. Conceptual layout

The following section discusses a proposed layout of the booster pump and presents some of the previous work on the topic carried out at the Norwegian University of Science and Technology (NTNU). As the author's research is ongoing, design details are not definite, but an axial counter-rotating booster pump configuration is thought to hold certain advantages given its ability cancel out the rotating flow leaving the pump and be positioned directly into the tunnel, as sketched in Figure 5.

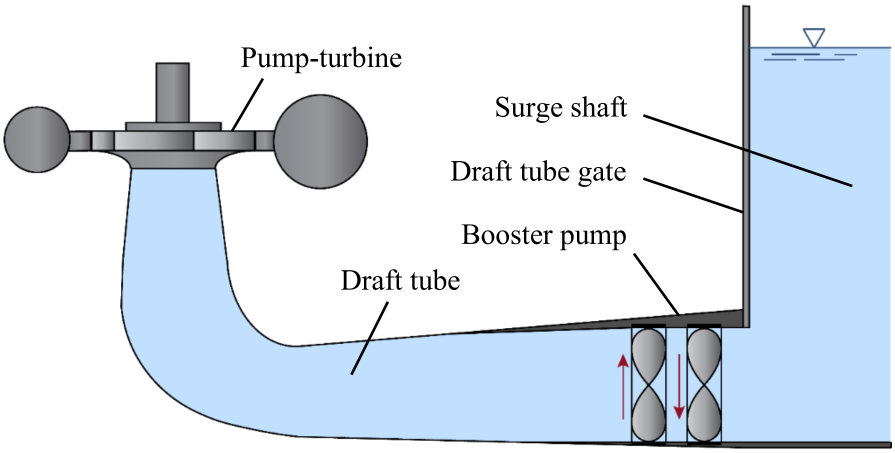

Figure 5: Proposed booster pump configuration.

A tool for designing counter-rotating pumps for booster pump purposes by means of the lifting line theory has previously been developed[9], but the final performance of the proposed design is not very promising for the intended usage, even though it was concluded that the overall efficiency of the system is fairly good. Given that the developed tool is based on thruster theory, the applied technique is not necessary completely transferable for pumping theory and it is therefore interesting to continue the search for a satisfactory counter-rotating design to obtain a favorable in-flow for the pump-turbine.

On the other side, a booster pump adding rotation to the flow could in fact benefit the pump-turbine to some extent. Theoretically, a pre-rotation induced at the pump-turbine inlet should contribute to an increased lifting head of the unit. Pre-rotation induced by a booster pump has also been investigated[10], but the research concluded that the potential benefits are almost insignificant as no increase in efficiency is achieved. Pre-rotation during off-design operation will most likely complicate the performance of the RPT, as it is unlikely that a rotating flow bent by the draft tube will be able to create a fairly uniform inflow at the RPT over the whole operational range.

Other pump alternatives, such as semi-axial pumps and water jet pumps with stationary guide vanes, have also been considered. However, practical integration of such pumps is more challenging, as their space-requiring arrangements limit possible pump locations. Higher velocities might also be unfavorable with regards to cavitation. Still, such arrangements are not entirely excluded for further evaluation, but in the initial stage of the booster pump development it is interesting to investigate which possibilities an axial counter-rotating configuration may offer. A single stage axial pump with stationary guide vanes will most likely not provide the necessary head contribution with a sufficient hydraulic efficiency.

One of the main challenges with the proposed solution, is ensuring a stable operation of the pumps in series over the entire operational range. Previously 1-D simulations of a booster pump interacting with a RPT has shown a promising cooperation between the pumps with regards to both steady-state and transient operation[11]. Suter characteristics curves were implemented into a method of characteristics model to investigate the cooperation between two pumps in a potential Norwegian retrofit project, resulting in a system quickly achieving steady state and handling disturbances forced on the system. Similar results were achieved with a dynamic pump model, which has a higher level of accuracy compared to Suter curves. On the other side, 1-D simulations cannot handle the actual flow patterns and pressure gradients present. Also, the proposed model includes several considerable simplifications and is not giving a just picture of the pump stability. It is, however, an interesting starting point for a more complex analysis of the pump stability.

With two different sized pumps operating in series, some caution must be carried out with regards to the design of the pumps, as the two individual pump curves are added together to a common curve. If the pumps are dimensioned for two different operational ranges, there will be flow rates only one of the pumps can handle. The characteristics of the booster pump and the RPT should cover the same range, producing a new pump characteristic similar to the illustrated pump curve in Figure 6.

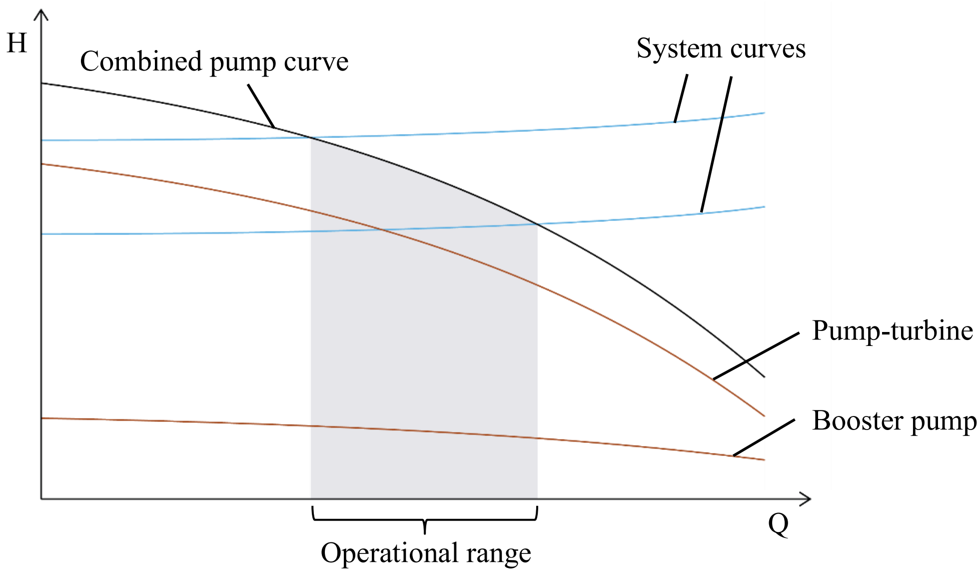

Figure 6: Potential pump curve of booster pump and RPT operating in series.

4. Discussion and further work

With the increasing need for electrical energy storage, pumped storage will play an important role worldwide in the years to come. Pumped storage offers a unique solution for the future power grid, given its storage capacity and its ability to respond rapidly between the different modes of operation. The entry of renewable energy will gradually replace parts of the coal-fired industry, which today are indispensable for grid stabilization. Coal-fired power plants are unfortunately slowly regulated and are therefore mostly continuously operated. Re-adjustment of the industry demands a replacement of the non-renewables and pumped storage must constitute a part of the solution. Investments in wind and solar power are becoming more and more profitable, but there is no corresponding profitability for new pumped storage projects, postponing the necessary development. Thus, it is essential to consider other methods to carry out an expansion of the pumped storage industry.

The economic aspects of developing pumped storage plants have, among others, been examined by Iberdrola Renewals, who concludes that reusing existing facilities give the lowest capital expenditures, CAPEX[12]. Still, there are several technical challenges complicating a profitable investment, including limiting the costs related to civil works and ensure a new pump-turbine managing the entire head range without being exposed to cavitation. The proposed booster pump configuration can possibly be a solution to some of the challenges and enable a further reduction in the assumed CAPEX. As the method does not require excavation of new tunnels and relocation of the turbine position, the need for civil works is heavily reduced. A possible re-use of the existing generator and parts of the turbine arrangement decreases the CAPEX further. Obviously, additional construction work is required for the booster pump installation and its accompanying supplement, for the time being representing a technical and cost-wise uncertainty to the concept.

Considering the remark about the required pump-turbine performance, the proposed solution could be capable of handling the entire head range, given that the available head range is sufficiently low and that the booster pump provides the required head. Higher head ranges could be manageable if unfavorable operation on the outer operational areas is accepted.

The various technical possibilities related to the booster pump solution depend on the possible pump performance. Cavitation of the pump itself will be a problem if dimensioned for an unfavorable large head. Also, given that axial pumps are most efficient for lower heads, high efficiencies might not be attainable for higher heads. The formerly discussed counter-rotating solution will, however, enable a pressure rise divided on two steps, enabling a more efficient machine.

Other design solutions of the booster pump are naturally possible, but initially, the proposed layout is considered best suited with regards to usage and position during both modes of operation. Immediately, the alternative of bypassing parts of the incoming flow to boost the pressure close to the pump-turbine inlet might appear as a possible way of preventing cavitation. However, such configuration has been examined and will not have the desired effect as this will only induce a negative flow in the main pipe. A complete review of this alternative is found in Appendix A.

A short summary of the potential booster pump benefits is listed below:

- The booster pump can prevent cavitation of the pump-turbine.
- It enables a more optimal pump-turbine design for turbine operation.
- It enables reuse of the original turbine arrangement.
- Reduces civil works costs.
- Reduces the overall operational downtime of the hydropower plant during the construction phase.

Accordingly, possible challenges and uncertainties of the proposed solution:

- The booster pump, generator and associated arrangement represent unknown costs, and the technology is highly immature.
- It could be difficult to accomplish an RPT design that fits into the existing arrangement with only minor changes to accompanying parts.

- Losses are likely to occur in the original parts during pump mode of operation, as most turbines are designed to accelerate the flow in the opposite direction. Without a proper pump diffusor, less energy will be converted into pressure energy.
- Avoiding cavitation of the booster pump and provide the required pressure head with an acceptable efficiency could be difficult to achieve and is highly dependent on the available submergence. The booster pump configuration will not suit all potential retrofit locations.
- Operating the booster pump and pump-turbine over the whole range and simultaneously have an acceptable efficiency, stability between the units and no exposure to cavitation can be difficult to achieve.

It is so far difficult to state to which extent the booster pump can be integrated in existing power plants. Given the tight optimization gap, the proposed method will most likely not be applicable for all potential locations. An individual analysis must be conducted to evaluate to which extent such arrangement is realistic and how the associated parts are affected. The ongoing research aims to uncover some of the listed uncertainties and investigate if the proposed concept can represent an efficient and technical feasible alternative to conventional methods. The current research stage involves design and simulation of a booster pump and a pump-turbine for a potential power plant to examine their interaction, performances and cavitation characteristics during pumping, where the final objective is to provide a technical proof of concept.

Appendix A
The alternative of pressurizing a bypass flow close to the pump-turbine inlet is evaluated by a simplified system, illustrated in Figure A1:

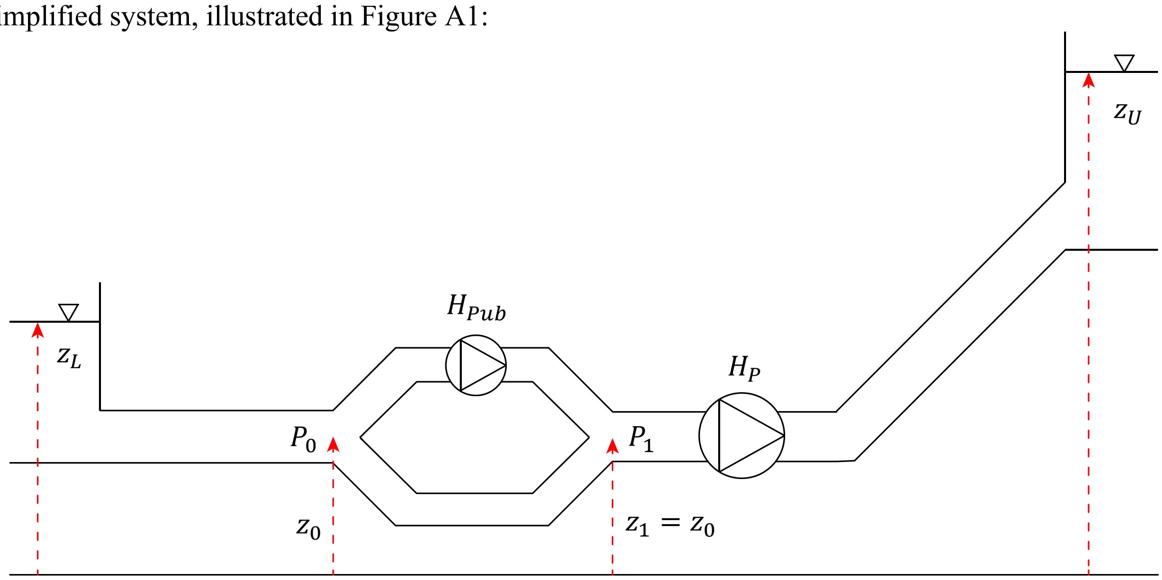

Figure A1: Bypassing pressure close to pump-turbine.

Flow is pumped from a lower reservoir to a higher reservoir by a pump providing a lifting height H_p. A second pump, H_{Pub}, pressurizes a bypass flow close to the inlet of the first to reduce the risk of cavitation. Applying the Energy equation (Bernoulli with losses) from the lower surface to the bypass inlet (position 0), gives the following expression:

$$\frac{P_0}{\rho g} = z_L - z_0 - \frac{\left(\frac{Q}{A}\right)^2}{2g} - h_L \tag{A.1}$$

The pressure P_0 at position *0* is then defined by the difference in elevation between L and *0,* the velocity (V=Q/A) and the head loss h_L at the given distance. Similarly, the pressure P_1 at position *1* can be defined by applying the Energy equation between *1* and the upper reservoir:

$$\frac{P_1}{\rho g} = z_U - z_1 - \frac{\left(\frac{Q}{A}\right)^2}{2g} - H_P + h_L \tag{A.2}$$

Thus, the pressure P_1 depends on the difference in elevation between *U* and *1*, the velocity, the contribution from the main pump and the head loss at the given distance. From this, it can be concluded that the boundary conditions at the inlet and the outlet of the parallel pipes, *0* and *1*, are not defined by the conditions within the pipes, only by other parts of the system. Therefore, at a given discharge, all operations within the parallel branches will only cause internal energy conversions.

Applying the Energy equation between *0* and *1* at the upper branch, *ub*, gives the following expression:

$$\frac{P_0}{\rho g} - \frac{P_1}{\rho g} = h_{Lub} - H_{Pub} \tag{A.3}$$

The elevations, z_0 and z_1, are assumed identical. Further, the head losses, h_{Lub}, are described by the Darcy-Weisbach equation:

$$\frac{P_0}{\rho g} - \frac{P_1}{\rho g} = \left[f_{ub} \frac{L_{ub}}{D_{ub}} \frac{\left(\frac{Q_{ub}}{A_{ub}}\right)^2}{2g} \right] - H_{Pub} \tag{A.4}$$

Where f_{ub} denotes the friction factor in the pipe. The discharge is assumed to always flow in the same direction. Likewise, for the lower branch, *lb*, the equation becomes:

$$\frac{P_0}{\rho g} - \frac{P_1}{\rho g} = h_{Llb} \tag{A.5}$$

Applying the same head loss conditions for the lower branch as for the upper, (A.5) becomes:

$$\frac{P_0}{\rho g} - \frac{P_1}{\rho g} = sign(Q_{lb}) \left[f_{lb} \frac{L_{lb}}{D_{lb}} \frac{\left(\frac{Q_{lb}}{A_{lb}}\right)^2}{2g} \right] \tag{A.6}$$

$sign(Q_{lb})$ ensures that the expression changes sign if the flow reverses. Combining (A.4) and (A.6), gives the following expression for the aiding pump in the upper branch:

$$H_{Pub} = \left[f_{ub} \frac{L_{ub}}{D_{ub}} \frac{\left(\frac{Q_{ub}}{A_{ub}}\right)^2}{2g} \right] - sign(Q_{lb}) \left[f_{lb} \frac{L_{lb}}{D_{lb}} \frac{\left(\frac{Q_{lb}}{A_{lb}}\right)^2}{2g} \right] \tag{A.7}$$

From this consideration, the lifting height of the pump is only covering the head losses in the two branches. Further, assuming that the pipe branches are identical and that the combination of the flow in each of the branches must equal the discharge in the system, the expression becomes:

$$H_{Pub} = f \frac{L}{2gA_2D} \left((Q - Q_{lb})^2 - sign(Q_{lb})(Q_{lb})^2 \right) \tag{A.8}$$

If there is no pump in the upper branch, $H_{Pub} = 0$:

$$Q = Q_{lb} \pm Q_{lb} \tag{A.9}$$

Giving:

$$Q_{lb} = Q_{ub} = \begin{cases} \frac{Q}{2} \\ 0 \end{cases} \tag{A.10}$$

which is the precise solution of a flow in parallel operation where both pipes are identical. If the flow Q_{lb} in the lower branch is zero in Equation (A.8), all flow must go through the upper pipe and the pump must supply the needed head loss corresponding to the full system flow Q. Increasing the lifting head additionally will only require that the flow in the lower branch becomes negative. Thus, attempting to pump in the upper branch will only initiate a flow going in a hydraulic short-circuit through the parallel pipes. Including a valve in the lower branch to prevent a reversed flow will of course force all flow to go through the upper branch. However, this causes a serial pumping situation with a pump positioned very close to the main unit with unfavorable cavitation conditions. The proposed serial pumping is therefore evaluated as a better option for avoiding cavitation of both units.

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