Drivetrain Architectures for a Mechanically Decoupled Contra-Rotating Reversible Pump-Turbine

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Abstract—With the rise of renewable energy production in the pan-European grid, the need for flexible energy storage is experiencing a rapid increase. Pumped hydropower storage has proven viability due to its long lifespan and cost-effectiveness. The ALPHEUS project will implement pumped hydropower storage for flat topographies to augment grid stability in adjacent regions. To ensure optimal efficiency and fast switching times in these low head applications, a contra-rotating axial Reversible Pump-Turbine (RPT) is designed. The runners will be driven by two separate Axial-Flux Permanent Magnet Synchronous Motors (AF-PMSM) to ensure optimal efficiency and flexibility at variable speed and flow rate. In this new setup, great attention is needed for the drivetrain architecture. The AF-PMSMs can be placed either outside or inside the water tube, using respectively tube elbows or bulbs. Furthermore, coaxial shafts allow the machines to be placed together, on one side of the RPT. This paper proposes four drivetrain architecture concepts, which are evaluated qualitatively based on their influence on RPT and AF-PMSM performance as well as bearing arrangement.

Index Terms—Drivetrain, Bearing arrangement, Hydropower, Power take-off

I. INTRODUCTION

Pumped Hydropower Storage (PHS) is an established and cost-efficient technology. Conventional high head PHS uses Francis turbines, which can reach up to 70-80% efficiency. However, due to the high head range of these turbine types, low head PHS has long been omitted. In ALPHEUS, Reversible Pump-Turbines (RPT) and adjacent technologies will be improved to make low head PHS economically viable [1]. A 10 MW system will be designed that is highly efficient and has low mode switching times, to allow for grid balancing services through frequency containment reserve or frequency restoration. To accomplish this, a contra-rotating RPT is investigated. Here, the two runners have opposite rotational motion, which limits the creation of swirl behind the RPT, thus increasing efficiency and reducing switching times [2]. Furthermore, it can improve pumping stability at low flow rate and suppress cavitation at high flow rate [3]. To further enhance efficiency and flexibility, two separate electric machines are used to actuate the runners independently. This allows for a varying speed ratio between the runners to allow efficient and dynamic response to grid state changes. AF-PMSMs have high power densities and are suitable for low-speed-high-torque operations [4], [5]. They have a high diameter-to-length ratio and a small air gap to allow strong flux linkage. A high pole number averts reduction gearing and the accompanying friction losses [6]. An AF-PMSM with double rotor Yokeless and Segmented Armature (YASA) topology compensates undesired axial forces and has no yoke, thus reducing iron losses and weight [7]. To provide energy transmission between the RPT and electric machines, a drivetrain architecture needs to be designed. The drivetrain architecture includes the positioning and dimensioning of shafts, electric machines and machine components. While the RPT is submerged in water, the AF-PMSMs must be accessible for maintenance and thus must be placed out of the water stream. Furthermore, the drivetrain must manage the runner weight and torque, as well as the axial forces, while preserving optimal system performance. Therefore, the goal of this paper is to evaluate different drivetrain architectures based on different criteria. These criteria comprise the influence on flow uniformity, bearing arrangement and loads, as well as AF-PMSM construction and cooling options.

First, the different concepts will be presented and described in section II. In section III an evaluation will be made with respect to the different criteria. Finally, an optimal concept will be selected and further examined in section IV.

II. DRIVETRAIN ARCHITECTURE CONCEPTS

To design and validate a drivetrain concept, it is important to know the system’s loads and dynamics. Table I shows the relevant properties for the 10 MW contra-rotating RPT, where both runners are each coupled to a 6 MW AF-PMSM. Fig. 1 illustrates these parameters. Note that the RPT is positioned beneath both high and low reservoir and is thus always completely submerged. Flow direction in pump mode is from left to right as in Fig. 1. The nominal flow rate in pump mode is 130 m³/s. In this paper, the bearings positioned closest
To the runners are denoted the main bearings. In the concept figures, the main bearings are placed close to the runner in the water stream and the second bearings are incorporated in the AF-PMSM stator. This bearing arrangement is used throughout all concept illustrations, but is only indicative and not optimal for all concepts, as will be discussed in section III-B. Placing a bearing in water requires additional seals that increase friction. Note that alternative drivetrain concepts, where bevel gears redirect shaft direction, are excluded in this evaluation, since they restrict system efficiency due to friction [8]. Concept A is illustrated in Fig. 1 and shows the most straightforward concept. In this architecture, two 90° elbows are placed on either side of the RPT, allowing for the shaft to extend through the tube. However, to make sealing possible, the shaft has to be perpendicular to the tube. Therefore, the elbow is altered, as shown in Fig. 1. The AF-PMSM is placed outside of the tube, allowing for a large machine diameter as well as an optimal cooling structure. Note that also 45° elbows can be used instead of the 90° elbows, like is done in compact axial turbines. This limits the flow disturbance.

![Fig. 1. Concept A: Elbows placed on either side of RPT to allow for the transmission shafts to extend out of water stream.](image1)

Fig. 2 shows concept B. Here, the elbows are replaced with bulbs. This concept is a promising alternative to concept A, since the bulbs divert the flow considerably less than elbows, which will be discussed in section III-A. The bulbs are supported by struts, through which the bulb can be accessed for machine and bearing maintenance. Note that the diameter of the bulb poses restriction on the AF-PMSM dimensions and corresponding cooling structure, as will be discussed in section III-C.

![Fig. 2. Concept B: Two accessible bulbs placed in the water stream, the AF-PMSMs are placed inside the bulbs.](image2)

Fig. 3 illustrates concept C, which utilizes two contra-rotating coaxial shafts. This design is derived from propulsion technology, which has matured within the maritime shipping industry [9]. This drivetrain design eliminates one elbow, which improves RPT operation. Since inlet flow in pump mode is most critical for the RPT, the left elbow is discarded and the coaxial shafts are directed towards the remaining right elbow.

![Fig. 3. Concept C: Inner shaft placed inside the hollow outer shaft, elbow on pump inlet is discarded.](image3)

Concept D combines concept B and C, making the coaxial shafts end up in a common bulb, located at the outlet of the RPT in pump mode. Concept D is shown in Fig. 4. This concept has the least impact on flow uniformity. However, the drivetrain is more complex and the bulb poses dimensional restrictions on the AF-PMSM.

![Fig. 4. Concept D: Coaxial shafts extended into one bulb at pump outlet side.](image4)

### III. Concept Evaluation and Comparison

In this section, the different proposed conceptual drivetrain architectures are evaluated based on water flow impact, bearing arrangement, electric machine restrictions and possible cooling structures.
A. Water flow impact

Flow uniformity is important for the RPT efficiency. All concepts have some form of impact on flow uniformity. Concepts A and C have 90° elbows that divert velocity profiles, inducing secondary flow across the tube cross section. The bulbs in concepts B and D also affect flow uniformity, although the resulting secondary flow is less prominent, depending on bulb diameter. One important difference is that the spatial velocity variations caused by the bulbs are constant for a given radius from the center of the tube cross section. Therefore, the axial forces on the RPT remain distributed equally among the runner blades.

In 90° elbows, a low pressure field is formed at the inner curve. As a result, a swirling flow is formed, which develops further towards the outlet of the bend. This results in spatial velocity variations, where the velocity on the outlet of the elbow is greater at the outer curve than at the inner curve [10], [11]. Furthermore, the velocities vary in time. This reduces the RPT efficiency and imposes unbalanced loading on the RPT and drivetrain. Reference [12] shows that a non-uniform velocity profile lowers pumping efficiency and causes vibration and wear. Furthermore, swirling flow can change the angle of attack on the runner blades, instigating cavitation in localized areas.

An experimental study [10] illustrates the velocity components along the horizontal and vertical centerlines of the tube cross section at different distances from the elbow. These lines are also called the cross and symmetrical lines respectively. The distances are expressed as a multiple of tube diameter $x/D$, which makes conclusions scalable for diverse tube diameters. The study shows considerable spatial velocity diversion at short distance from the elbow. The diversion is greatly diminished at $x/D = 5$, though not completely absent. A subsequent paper [11] extended the study with CFD calculations and compared the results with common prescribed pump inlet flow conditions, to find the minimum distance a 90° elbow should be placed from the inlet of a pump. The conditions are as follows [13]:

- Average swirl angle less than 5°
- Time-averaged velocities at any point within 10% of cross-sectional average velocity
- Temporal velocity fluctuations within 10% of time-averaged velocity at that point

It was found that for a short radius elbow ($R_e/D < 1.5$) and different turbulences, it takes between $x/D = 3$ to $x/D = 16$ for the flow to satisfy the first two criteria, while the last criterion was never met for the different setups. With the results these studies bring forward, it is clear that placing an elbow too close upstream from a pump is detrimental for pump and bearing operation. Even if the radius of the elbow was significantly increased, the flow uniformity will not be ensured. Since flow in concept Figs. 1-4 is from left to right in pump mode, this means that the left elbow in concept A has to be placed at least $x > 3 \cdot D$ from the RPT. This results in a shaft length of at least 18 m for a tube diameter of 6 m.

Although the inlet flow uniformity for pumping mode is most critical and the inlet flow conditions above are specified for pump operation, it can be deduced that the same principles apply for turbine operation. It was found that an elbow downstream from an axial pump also has a great impact on the pump under sub-nominal flow rates [14]. Furthermore, the flow crosses the shaft in elbow configurations, causing wake behind the shaft. This further reduces flow uniformity near the elbow. Therefore, it is concluded that the right elbows in concept A and B also abide a minimal distance from the RPT, thus increasing shaft length significantly.

Drivetrain architectures with longer shafts have drawbacks. They have greater weight, which increases shaft shear stress and radial bearing load, thus reducing lifetime. The rotational inertia in turn also increases, which limits mode switching and thus grid stabilizing abilities. Furthermore, long shafts are more prone to vibrations due to the flexibility, which causes fatigue loading and a great impact on bearing lifetime. The high axial expansion per temperature unit and great runner-to-machine distance also limit possible bearing arrangements, which will be discussed in the section III-B. For concepts B and D, the shaft length is much shorter. The minimal distance to the RPT is mainly imposed by the supporting struts, which also cause secondary flow. Therefore, positioning the struts too close would create a torque ripple every time a blade passes the nearby strut. However, the shaft length in these concepts is significantly smaller than in concepts A and C.

In the bulb concepts, there is a space between the bulb and the runner in which secondary flow can instigate. Therefore, ship thrust systems with contra-rotating propellers extend the bulb to the propeller [15]. This ensures a fixed cross sectional flow area from bulb to runner, thus reducing secondary flow. Here, great attention is needed for the seal between the bulb and the runners.

The coaxial concepts C and D are distinguished from respectively concepts A and B, since they only restrict flow uniformity on one side of the RPT. Furthermore, the inlet flow in pump mode, which is the critical point in RPT operation, remains uniform in these concepts, thus improving RPT performance.

B. Bearing arrangement

In most high capacity wind and water turbines today, a bearing arrangement with one locating bearing and one non-locating bearing is used [16], [17]. The locating bearing is exposed to both axial and radial loads, while the non-locating bearing accommodates radial loads exclusively. This arrangement allows for axial expansion, while keeping the shaft in place. The locating bearing can be a toroidal bearing or a tapered roller bearing, with the latter supporting higher axial forces but allowing little misalignment. For the non-locating bearing, a cylindrical or non-locating toroidal bearing are the main choice for their high radial load capacity. When applying existing bearing technologies to an RPT, the reversible nature of the axial forces on the runners must be noted. The locating bearing thus must support axial loads in both directions.
shows the discussed bearing arrangement with a double row tapered roller bearing as main bearing, which is placed closest to the runner. The radial cylindrical bearing is integrated in the AF-PMSM stator. Equation (1) shows the static balances of forces, divided into the vertical force balance V, the horizontal force balance H and the bending moment balance M. Here, the locating bearing is labeled A and the radial bearing is labeled B. Note that force $F_a$ can be positive or negative. $G_1$ is the combined weight of the runner and the part of the shaft left of bearing A. $G_2$ is the weight of the part of the shaft between both bearings, combined with the left machine rotor. $G_3$ is the combined weight of the right rotor with the right part of the shaft. For normal shaft lengths, $G_1$ is the largest out of the three and most important. The balance equations show that, in order to minimize bearing load, $L_1/X$ has to be sufficiently small.

$$\begin{cases}
V : & -G_1 - G_2 - G_3 + F_{r,A} + F_{r,B} = 0 \\
H : & -F_a + F_{a,A} = 0 \\
M : & -G_1 \cdot L_1 + F_{r,A} \cdot X - G_2 \cdot L_2 + G_3 \cdot L_3 = 0
\end{cases} \tag{1}$$

For the drivetrain of the described system in this paper, another constraint rises. Opposed to traditional radial electric machines, axial movement of the rotors relative to the stator of an AF-PMSM has an impact on the air gap between them. Furthermore, the air gap in an AF-PMSM is small, down to 1 - 2 mm. Therefore, the bearing arrangement must ensure minimal axial position variation of the shaft at AF-PMSM position. Since radial bearing B in Fig. 5 only supports radial loads, axial expansion of the shaft can cause the air gap to vary. For the arrangement illustrated in Fig. 5, this imposes the following restriction:

$$\Delta L = X \cdot \alpha \cdot \Delta T < l$$ \tag{2}

where $X$ is the distance from the AF-PMSM to locating bearing, $\alpha$ is the shaft material linear expansion coefficient, $\Delta T$ is the maximal yearly temperature fluctuation of the water at the system site and $l$ is the maximum slack between rotor and stator. For a typical construction steel shaft in a site at the North Sea, a maximum slack of $l = 1.0 \, \text{mm}$ gives $X < 5.3 \, \text{m}$. Although this gives a good indication of the maximum allowed distance between machine and main bearing, any decentralization of the stator relative to the rotor introduces flux linkage variation and thus efficiency reduction [18]. Therefore, $X$ should be as small as possible, simultaneously increasing factor $L_1/X$. As discussed in section III-A, shaft lengths for concept B and D are relatively small, thus allowing both minor slack and allowable bearing load and shaft bending tension. Because of their small shaft lengths, it is possible to place the main bearing inside of the bulb for these concepts. This allows for easier bearing sealing and accessibility for maintenance. However, for concepts A and C, the shaft length cannot satisfy the slack constraint while preserving permissible moment load on the bearing.

Fig. 6 (a) shows a bearing arrangement where the locating bearing is integrated in the AF-PMSM, with the non-locating bearing near the runner. The relative axial movement of the rotors in now fully restricted, ensuring constant air gap, while moment lever to runner weight is minimal. One drawback in this arrangement is that the axial position of the runner is now subject to the axial expansion, which might alter RPT performance. Fig. 6 (b) shows a bearing arrangement with two locating bearings. In this topology, a ball spline tolerates axial movement caused by expansion. To ensure a large contact surface, grooves are made into the inner shaft. The balls transmit torque and can move freely in axial direction. The bearings at runner and machine side ensure axial location of the runner and constant air gap respectively. The use of a ball spline introduces a disadvantage in lifetime and reliability. The ball spline is not a rigid body, making it prone to torque vibrations. Furthermore, the relatively small contact surfaces limit its load capacity. However, it is a viable alternative for the arrangements above. Implementation of this shaft architecture is more complex for coaxial shafts as in concepts C and D.

It can be concluded that for concept A, only the arrangements in Fig. 6 are eligible. For concept B, all proposed bearing arrangements are suitable. For concepts C and D, the same conclusions can be found as respectively concept A and B, however with the exclusion of the ball spline arrangement. Next to the proposed bearing arrangements, there is an additional possibility for the coaxial shaft concepts C and D. In these concepts, the main bearing of the inner shaft can also be placed between the inner and outer shaft, thus transmitting its load to the outer shaft instead of the system housing. Positioning of this bearing in the outer shaft instead of in the inlet pipe of the RPT in pump mode further increases pumping efficiency. However, this approximately doubles the radial and axial load experienced by the outer shaft and its bearings.
Fig. 6. (a) Arrangement with non-locating bearing near runner and locating bearing integrated in AF-PMSM. (b) Arrangement with two locating bearings and a ball spline to allow axial expansion.

C. Machine Restrictions and Cooling

As previously discussed, the AF-PMSM has a high diameter-to-length ratio. This can be seen in (3), where machine power is proportional to the third power of outer diameter $D_o$ of the permanent magnets on the rotor:

$$ P_{\text{module}} = \frac{\pi^2 \Omega_{\text{nom}} k_D k_w B_{m0} A_m D_o^3}{60} \quad (3) $$

In this equation $k_D = 0.081$ is the diameter ratio factor, $k_w = 0.96$ is the winding factor, $B_{m0} = 0.8$ $T$ is the peak value of magnetic flux density in the air gap and $A_m$ is the linear current density in the air gap. $A_m$ can range from $30 - 80$ kA/m for air cooling to $150 - 200$ kA/m for direct water cooling [19]. Meanwhile, the axial length of the machine is virtually independent of machine power and is $0.6 - 0.8$ m. If the power for one runner has to be supplied by one AF-PMSM unit, $D_o$ would be between $5 - 7$ m and $4 - 4.5$ m for air and water cooling respectively. These large diameters pose no problems for concepts A and C, where no machine restrictions are present. However, in concepts B and D, the bulb diameters restrict the machine diameters. If the bulb diameter is restricted to hub diameter, being $3.5$ m, the flow impact is minimal. However, 2 water cooled AF-PMSMs would be needed to supply the power of one runner. This leads up to 5 modules using air cooling. The bulb diameter needs to be increased to allow for the electric actuation for one runner to be provided by one AF-PMSM module. However, increasing this diameter impacts flow uniformity.

In traditional hydropower plants, the water stream is used to cool the machines through a simple air-water heat exchanger. However this method does not extract the heat directly from its origin. Reference [20] shows that most heat is generated and captured in the stator windings, in the stator core and in the air gap. Therefore, a more refined cooling structure, integrated in the AF-PMSM stator shows great results [20], [21]. Using the system water in these circuits would require excessive filtering. Water also has a low evaporating point, reducing its effectiveness in extendedly high temperature regions. Furthermore, if the system is placed at a coastal site with salt water, the small tubes would have to be designed to withstand corrosion. However, the system water can be utilized as a secondary cooling circuit, in which the heat exchanger of a primary glycol-water cooling circuit is placed.

IV. Concept selection and discussion

In section III-A, it was discussed that in order for the flow to be uniform at RPT in and outlet, the elbows in concept A and C would have to be placed at a significant distance. Therefore, long shafts with their corresponding issues are the only option for these concepts. Alternatively, the bulbs in concepts B and D do not cause the same impedance on flow uniformity. Subsequently, the bulbs can be placed considerably closer to the RPT, ensuring shorter shaft lengths. Furthermore, using coaxial shafts discards flow diversion on the pump inlet flow, thus increasing RPT performance in pump mode. Next, the possible bearing arrangements were discussed in section III-B, taking into account the shaft lengths, runner weight and electric machine air gap. Because of the short shaft lengths in concepts C and D, the main bearing can be placed inside the bulb. This further reduces flow influence and allows for easier bearing seals and maintenance accessibility. In section III-C it was found that the outer diameter of an AF-PMSM module is up to respectively $7$ m and $4.5$ m for air and water cooling. With all criteria being evaluated for the different concepts, it’s clear that for optimal operation and reliability, concept D is most suitable because of its minimal flow influence, reasonable shaft lengths and bearing loads. However, because of the high initial cost of the cooling system and coaxial shaft, future hydropower projects might investigate how the decreased initial cost of other concepts weighs up against the reduced RPT and drivetrain performance and reliability.

Fig. 7 shows the derived full conceptual drivetrain. Here, the non-locating bearings and cooling circuits are integrated in the AF-PMSM stator. The locating bearings are placed inside of the bulb. A bulb and machine diameter of $4.5$ m is chosen. This increases flow impact slightly, but allows each runner to be actuated by one water cooled AF-PMSM module, which reduces cost.
In this paper, four drivetrain concepts are investigated and evaluated qualitatively based on different transdisciplinary criteria. Concept A has two 90° elbows, which allow the shaft to be extended out of the tube in these bends. Concept B instead makes use of two bulbs on either side of the RPT, limiting the influence on water flow. Concept C and D are variants on respectively concept A and B, making use of coaxial shafts to eliminate the flow interference on pump inlet side. The evaluation criteria comprise impact on water flow uniformity, bearing arrangements and electric machine dimensioning and construction. It is found that 90° elbows introduce spatial velocity variations, causing i.a. unbalanced forces on the RPT and thus on shaft and bearings. It is found that in order to limit the velocity variations, the shaft length has to be three to sixteen times the tube diameter. Next to the added weight, inertia and vibrations, these shafts also need alternative bearing arrangements, due to the air gap of the AF-PMSM. However, due to the high diameter-to-length ratio of the AF-PMSMs, it was found that the bulb diameters have a great influence on the power of the AF-PMSM modules.

It is concluded that concept D is optimal, since it imposes the least restrictions on RPT performance and has reasonable shaft lengths and bearing loads, and thus high bearing lifetime. However, the other concepts can be economically attractive because of their lower initial cost. Future research includes quantitative analysis of the proposed drivetrain architecture to result the optimal dimensions and topologies of shafts, bearings and machines, in order to limit wear and fatigue, while allowing optimal efficiency.

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REFERENCES