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CFD simulations to optimize the blades design of water wheels

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Abstract. In low head sites and at low discharges, water wheels can be considered among the most convenient hydropower converters to install. The scope of this work is to improve the performance of an existing breastshot water wheel changing the blades shape, using Computational Fluid Dynamic (CFD) simulations. Three optimal profiles are investigated: the profile of the existing blades, a circular profile and an elliptical profile. The results are validated performing experimental tests on the wheel with the existing profile. The numerical results show that the efficiency of breastshot wheels is affected by the blades profile. The average increase in efficiency using the new circular profile is about 4% with respect to the profile of the existing blades.

1 Introduction

Electricity production in large scale from renewable energy sources has become an important purpose in the European Commission legislations. Among renewable energy sources, hydropower is considered to be one of the most important ones (Bódis et al., 2014). However, large hydropower plants need the construction of large dams, buildings and installations for the generation, regulation and transmission of power, and the payback periods are generally long. In addition, there are often many adverse effects and drawbacks on the ecosystems, for example the flooding of large areas and the interruption of the continuity of the river. Micro-hydropower (net input power lower than 100 kW) is instead considered more eco-friendly. Therefore, the interest in micro hydropower is increasing. Most of low head and low discharge sites are still not exploited, since standard turbines cannot be employed economically in such conditions (Bozhinova et al., 2013; Müller and Kauppert, 2004).

In Bozhinova et al. (2013) a review of hydropower converters for very low heads has been presented, and an attractive opportunity in micro hydro field can be represented by gravity water wheels. Gravity water wheels exploit the potential energy of water and a portion of the kinetic energy. They can be classified in overshot wheels, where the water enters into the wheel from the top, and breastshot water wheels, where the water fills the buckets entering from the upstream side of the wheel. Breastshot water wheels can be divided in high, middle and low, depending whether the water entry point to the wheel is over the rotation axis (in the uppermost third of the wheel), near the axis (in the middle third of the wheel) or under the axis (in the lowest third of the wheel), respectively. In breastshot water wheels the upstream water level can be controlled by inflow structures. When there is an overflow weir or an undershot weir (with a sluice gate to regulate the upstream water level), breastshot wheels are called *slow* or *fast*, respectively, considering the higher flow velocity to the wheel occurring in the latter case (Quaranta and Revelli, 2016a). Low breastshot wheels for very low heads are called undershot wheels. *Zuppinger* and *Sagebien* undeshot water wheels are used in sites with very low heads (typically less than 1.5 m), and the upstream conditions can be controlled by an inflow weir, so that the approaching flow velocity is very low, generally less than 1 m/s (Quaranta and Müller, 2017). A particular kind of fast undershot water wheel which exploits well the kinetic energy of the water is the *Poncelet* wheel (Poncelet, 1843). *Poncelet* wheels are generally installed in straight channels, with no bed drops or geometric heads through the wheel. The channel drop is present downstream of the wheel, so that the blades do not interfere with the tailrace. The inflow is realized with a sluice gate which is very close to the wheel, in order to increase the flow velocity. The water jet exchanges its momentum with the wheel flowing along the blades.

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The maximum efficiency of water wheels can be higher than of 80% for overshot water wheels (Quaranta and Revelli, 2015b), 75% for breastshot water wheels (Quaranta and Revelli, 2015a, Quaranta and Revelli, 2016a, Vidali et al., 2016), higher than 80% for undershot *Zuppinger* and *Sagebien* water wheels (Quaranta and Müller, 2017) and approximately 55% for *Poncelet* wheels. Although water wheels are environmental friendly and efficient hydropower converters, only a small amount of research has been spent on their performance characteristics in the last century. There are now some companies and research centers which are currently dealing with water wheels, especially for electricity generation. Due to their several advantages over turbines (lower costs, shorter payback period, higher and simpler adaptability to the external conditions, but no simpler design), water wheels may constitute a suitable technology for the economic development, in particular in rural areas and developing worlds.

1.1 Scope of the work

In Quaranta and Revelli (2015a) a theoretical model has been proposed to estimate the power losses of breastshot wheels, and in Vidali et al. (2016) a dimensional approach was performed. In Quaranta and Revelli (2016b) the number of the blades has been investigated for breastshot water wheels. Concerning instead the blades profile, the general criteria that should be taken into account in the blades design are well established (Quaranta and Revelli, 2015a), whereas numerical or experimental investigations on the optimal profile of fast breastshot wheels' blades can be rarely found. The general design criteria for the blades profile are:

(1) the relative entry stream velocity in the impact point should be directed as the blade inclination, in order to reduce the inflow power losses;

(2) the uplift of water downstream of the wheel and the outflow power losses should be minimized. Hence the blades should exit at a normal angle with respect the free surface at the tailrace, or with a backward inclination in order to reduce the drag;

(3) the blades length should be long enough or curved in order to avoid losses of water at the root of the blades.

Therefore, the scope of this paper is to investigate by Computational Fluid Dynamic (CFD) simulations the effect of the blades profile on the performance of fast breastshot wheels. This is justified by the fact that, although the general criteria for the blades profile are well established, it is not so clear if the blades profile generates significant effects on the performance of this kind of wheel (as previously illustrated). Similar uncertainty has also been found for *Poncelet* wheels: in Weisbach (1849) and Faibairn (1864) the circular shape is suggested, while in Bresse (1869) the Author says that the blades curvature is a matter of indifference. This work is also led by the need to improve the performance of an existing wheel, acting on the blades shape. The existing wheel with its original blades profile was simulated and then two different profiles were also investigated. The 1:2 scale physical model of this wheel with the original blades profile has also been installed in the Hydraulics laboratory of Politecnico di Torino, both for studying in detail the performance of breastshot wheels (Quaranta and Revelli, 2015a), and to validate the numerical model.

CFD simulations for gravity water wheels have been already successfully used in Quaranta and Revelli (2016b), where the performance of the present breastshot water wheel has been investigated through CFD simulations for different blades numbers, and for overshot water wheels (Quaranta, 2017).

2 Method

The investigated breastshot water wheel is a 1:2 scale model (Froude similarity) of an existing one, sited in Verolengo, near Turin (Italy) (Fig.1); it is made of 32 blades and the diameter is 4 m. The scaled wheel is 2.12 m in diameter and the width of the installed wheel is b=0.65 m. The channel which conveys water to the wheel is 0.67 m wide; 0.7 m upstream of

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the entry point to the wheel there is a sluice gate. The geometric head (or channel drop, which is the difference between the elevation of the channel's bed upstream and downstream of the wheel) is 0.35 m, thus the wheel is a low breastshot wheel.

Figure 1: The existing breastshot water wheels, whose 1:2 scale model was investigated in this work. The original diameter is 4 m.

Furthermore, since the water flow accelerates passing under the undershot weir (through the sluice gate opening) and the blades are shaped in order that, at the beginning of the filling process, the jet flows along them (before becoming at rest in the buckets), the inflow process is similar to *Poncelet* wheels. In Fig.2 the sketch of the scaled wheel is reported.

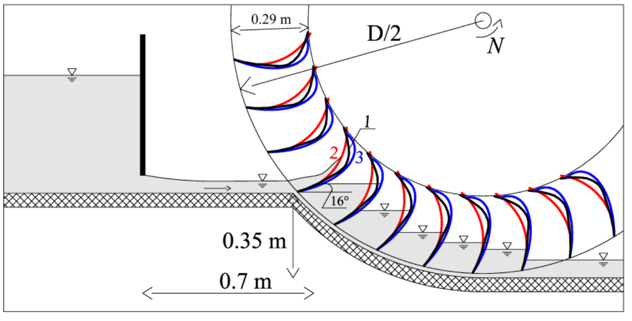

Figure 2: The sketch of the investigated breastshot water wheel, which is the 1:2 scale model of the existing one.

2.1 Blades profile

Three different shapes are here investigated by CFD analyses: the profile of the existing blades (1), a circular modified profile (2) and an elliptical profile (3) (Fig.3). The modified profiles (2) and (3) are designed with the same tip inclination of profile (1), which is 16° on the horizontal in the entry point, in order to compare objectively the effect of different profiles. The tip inclination of the profiles is almost parallel to the relative flow velocity, in order to minimize the impact power losses. In this case, the profiles also minimize the downstream power losses, since they exit from the tailrace approximately normally, without uplifting water. The angle between the profiles and the tailrace water surface is 83°; it is good to be smaller than 90°, since the slight backward inclination at the tip allows to reduce the drag.

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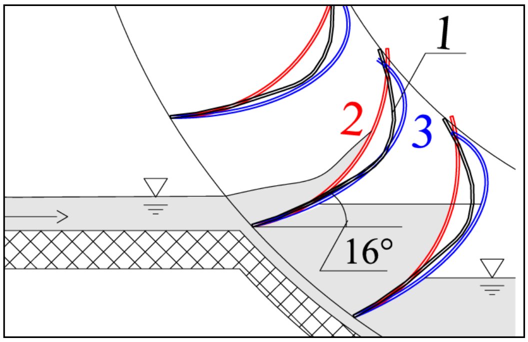

Figure 3: The zoom on the blades investigated in this work. (1) is the original profile, (2) the circular one and (3) the elliptical profile.

Profile (1) is 0.40 m long, and it can be considered as composed of three parts. The first part of the profile (which immediately starts to interact with the flow) is a circular arc 0.22 m long and 0.60 m in radius. This part of the profile seems to be quite flat. The third part (the internal one) is flat and 0.1 m long. The previous two parts are connected by a circular arc 0.08 m long and 0.11 m in radius. The external part of the profile, which is the part that interacts with the flow mostly, is similar to the profile that would be obtained following the design procedure described in Weisbach (1849) for *Poncelet* wheels. In order to apply the design procedure described by Weisbach (1849) to the present wheel, the tip inclination of the blade in the impact point and the depth of the blades are required, which are 16° and 0.29 m, respectively; thus the tip inclination on the horizontal is 62° under the wheel axis. The circular profile that would be obtained using the *Weisbach* procedure would have a radius of 0.62 m, very close to the real one (profile (1)) of 0.60 m. However, in our case we do not deal with an original *Poncelet* water wheel, because *Poncelet* wheels are generally installed in straight channels, with no geometric heads or channel's bed drops through the wheel. Therefore, the radius of curvature of the procedure suggested by Weisbach (1849) for the blades design, which is similar to the existing profile, may not be the optimal for this breastshot water wheel. Therefore, two different profiles were also investigated.

Profile (2) is a circular arc. A circular profile was studied both to make the manufacture process easier, and because also Weisbach (1849) and Faibairn (1864) suggest a circular shape. The shorter the radius of curvature, the more the deviation that the jet, flowing along the blade, undergoes, to which corresponds a change in its momentum. The change in momentum leads to a force on the blade, pushing the wheel more than what would occur using a straight blade or a bigger radius of curvature. But if the radius of curvature is too short, the jet may not be able to flow along the blade, since it would separate from the blade surface. Furthermore, the blade may uplift water from the tailrace, generating power losses. In the present case, an optimal radius can be considered r=1/4R=0.25 m, where R=D/2 is the wheel radius.

Profile (3) is an elliptical shape: the major axis is again 0.25 m and the minor one is 0.14 m. Also this profile can be considered a good one, since it exits at an optimal angle from the water surface of the tailrace.

2.2 Numerical model: geometry and mesh setup

The computational domain of the numerical model was divided into the stationary domain of the channel, which supplies water to the wheel, the rotating domain of the water wheel and the stationary air filled domain outside of the wheel. The stationary air domain is subdivided in an internal domain, directly in contact with the wheel, and an external domain (Fig.4).

The channel and the wheel are meshed with tetrahedral elements. In order to check the mesh independence of the solution, the buckets were meshed both with elements of 0.02 m, and then with the elements of 0.01 m near the blades. The stationary air domain is meshed with tetrahedron and cubic elements, with dimension of 0.02 m near the wheel and the

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channel, up to 0.1 m at the boundaries of the external domain. In order to save further time, half of the domain of the wheel (π rad instead of the whole 2π rad domain) was simulated with blades, while the other half portion of the wheel was simulated with a coarser mesh and without blades (Quaranta and Revelli, 2016b).

2.2 Numerical model: simulation setup and boundary conditions

The flow field was modeled by the 3D Reynolds Averaged Navier Stockes (RANS) equations, thus by a continuity and three momentum equations for the time averaged pressure and velocity of the mixture. In order to solve these equations, the turbulent viscosity μ_t is introduced for modeling the Reynolds stresses. The turbulent viscosity μ_t was modeled using the shear-stress transport (SST) k-ω closure turbulent model, where the turbulent viscosity is expressed as a function of the turbulent kinetic energy k and the specific dissipation rate $\omega=\varepsilon/k$, where ε is the turbulence kinetic energy dissipation. Hence two additional equations are solved, one for k and the second for ω, determining the turbulent viscosity μ_t.

The Volume of Fluid method (VOF) was used for the multiphase problem, with an implicit interpolation scheme and a level-set method, which is a well established interface-tracking method for dealing with two-phases flows with topologically complex interfaces. The Turbulence Damping option was included in the interface area to model such flows correctly; indeed, in free surface flows, a high velocity gradient at the interface between two fluids may generate high turbulence, both in water and air. The Curvature Correction was also enabled to sensitize the model to the system rotation and streamlines curvature.

The pressure-velocity coupling was solved by the PISO scheme and the spatial discretization was made by the PRESTO scheme for pressure and the Second Order Upwind for momentum and turbulent kinetic energy. The modified High Resolution Interface Capturing scheme (HRIC) was used for computing the volume fraction. The mass flow rate was imposed at the channel inlet, specifying also the free surface level, the value of the turbulence intensity I=0.05 and a fixed value of the turbulent viscosity ratio μ_t/μ=10, with μ the water dynamic viscosity. At the outlet of the channel and at the external surfaces of the external air domain, the pressure outlet option was adopted. At the top of the external domain the symmetry boundary condition was imposed (it gives more stability to the solution, with no effects on the interaction between water and wheel). Since the wheel is symmetric with respect to a vertical plane perpendicular to the rotation axle, the symmetry boundary condition was imposed on this surface, and only one half of the wheel was simulated, saving computational time. The no slip boundary condition was imposed at the walls (blades' surfaces, channel's walls and wheel), as showed in Fig.4. The detailed numerical model is described in Quaranta and Revelli (2016b), where the same water wheel has been investigated for different blades numbers.

The opening of the sluice gate was set at 0.075 m; flow rates of 0.05-0.06-0.07 m^3/s were adopted and optimal rotational speeds were chosen, based on the experimental results described in Quaranta and Revelli (2015a). Table 1 reports the investigated working conditions. In these cases the downstream water depth was 0.07 m, 0.085 m and 0.095 m, depending on the flow rate, respectively.

Once the numerical model with profile (1) was validated, it was hence possible to obtain a performance optimization by changing the shape of blades in the geometry of the numerical model.

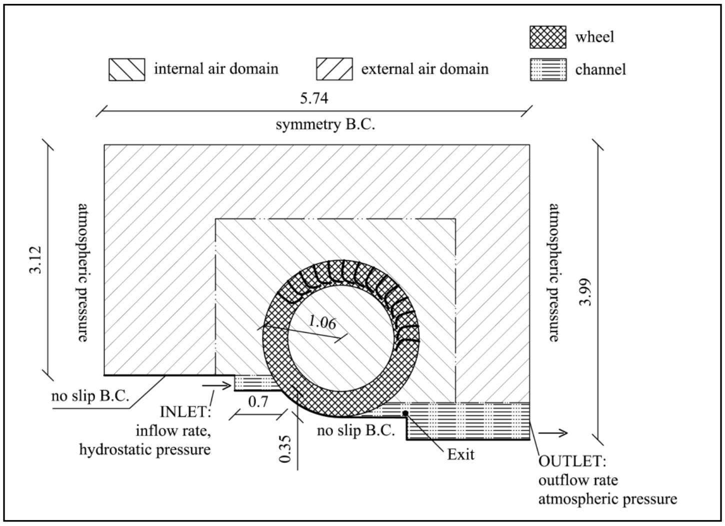

Figure 4: The numerical domain and the boundary conditions for the CFD model (Quaranta and Revelli, 2016b).

3 Results

The time step chosen for the unsteady simulation was 0.0008 s, that sometimes needed to be reduced to 0.0005 s. The Second Order Implicit scheme in time was used; 20 inner iterations were carried out between two consecutive time steps for the pressure-velocity solving. Each time step took approximately 2 min to be solved in a processor at 2.40 Ghz with 8 Gb RAM, for a total time of approximately 7 days for each simulation.

Since the shaft torque (exerted by the water on the blades of the wheel) could be easily monitored, and it represents a direct measurement of the water wheel performance, the torque was chosen as control parameter of the simulations. Therefore, during simulations the shaft torque C_j (with j the blades profile) due to the water-blades interaction was monitored. When the blades began to interact with the stream, the torque started to increase. Due to the wheel radial symmetry, after the transitory time the torque trend oscillates periodically around the average value $\hat{C}_j$ with a period of $T=\beta/N$ (where β is $2\pi/n$, and n the number of blades). The average value $\hat{C}_j$ was then compared with the experimental measured one C_{exp}, testing the accuracy of the solution. Once the torque was calculated, the mechanical power output could be easily obtained multiplying the torque by the rotational speed.

A mesh sensitive analysis was performed to check the mesh independence of the solution and discussed in Quaranta and Revelli (2016b), showing that both the meshes are fine enough to capture the mean flow field and to calculate the wheel performance. The accuracy of the numerical model was determined by calculating the discrepancies between the numerical and the experimental solution, using the finer mesh. The numerical model underestimates the torque, but the accuracy of the numerical shaft torque prediction is very good, with the average discrepancy between the numerical and experimental torques lower than 5% (Tab.1). At the flow rate of 0.05 m^3/s, the discrepancy is 1.16%, while the discrepancy settles around 5.5% for the higher flow rates, and it is practically the same for discharges of 0.06 m^3/s and 0.07 m^3/s.

Table 1 illustrates the performance of the wheel for different blades shapes. As it can be seen, the second profile is the optimal one, while the elliptical profile is the worst. The circular profile allows to reduce the power losses at the inflow, since the momentum of the flow is better exploited. It is also optimal for the downstream conditions, since a circular profile can exit the free surface at a better angle during its rotation. Table 1 shows also the percentage increase of the two new profiles with respect to the real blades profile. The increase is between 2% and 5.6% for the modified circular one, and between -1.7% to -8.3% for the elliptical profile. For detailed information on the hydraulic behavior of the wheel, refer to Quaranta and Revelli (2016b). Concluding, the achieved results shows that the profile of the blades affects the performance of the wheel, thus their curvature is not a matter of indifference as expressed in Bresse (1869). In this case the circular profile is to prefer to the elliptical one, in order to increase the wheel efficiency.

Table 1: Investigated working conditions and torque results.

Q (m^3/s)	N (rad/s)	C_{exp} (Nm)	$\hat{C}_1$ (Nm)	$\hat{C}_2$ (Nm)	$\hat{C}_3$ (Nm)	$\frac{\hat{C}_1-C_{exp}}{C_{exp}}$	$\frac{\hat{C}_2-\hat{C}_1}{\hat{C}_1}$	$\frac{\hat{C}_3-\hat{C}_1}{\hat{C}_1}$
0.05	0.78	175	173	180	170	-1.16%	+4.0%	-1.7%
0.06	0.79	223	211	226	205	-5.4%	+5.7%	-2.8%
0.07	0.89	253	239	244	219	-5.5%	+2.1%	-8.4%

4 Conclusions

Water wheels are an environmental friendly and efficient technology to produce energy, but only a small amount of research has been spent on their performance characteristics in the last century. Due to their several advantages over turbines, water wheels may constitute a suitable technology especially in rural areas and developing worlds.

In the present paper a study of three blades shapes of a breastshot water wheel is reported, in order to improve its performance. Numerical CFD analyses are performed to deal with the 3D turbulent multiphase problem. Numerical results show that the blades profile affect the performance of the wheel; the circular profile is better both than the elliptical profile and than the existing profile. Using the circular profile, the performance of the wheel was improved of an average value of 4% with respect to the existing one.

Therefore, for a practical application of similar breastshot water wheels, the Authors recommend to use a circular profile, considering that the profile should be designed in order to have the tip inclination parallel to the relative entry stream velocity and to exit approximately at a normal angle from the tailrace. Simultaneously, the profile should be able to exploit the momentum of the water flow, while avoiding separation phenomena.

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