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Technical Note

Performance Optimization of Overshot Water Wheels at High Rotational Speeds for Hydropower Applications

Emanuele Quaranta[1] and Roberto Revelli[2]

Abstract: Overshot water wheels are hydropower converters generally employed for head differences up to 6 m and maximum flow rates of 150–200 L/s per meter width. The hydraulic efficiency (80%–85%) is constant for rotational speeds below the critical speed, whereas the efficiency linearly decreases at higher rotational speeds due to the increase of water losses at the inflow. To improve the efficiency when the rotational speed is above the critical speed, an improved geometric design was investigated by implementing a theoretical model validated using experimental results. The new geometry consists of a circular wall around the periphery of overshot water wheels. The wall redirects into the buckets the water flow that is lost at the inflow, improving the efficiency up to 1.5 times at high rotational speeds. **DOI: 10.1061/(ASCE)HY.1943-7900.0001793.**

Author keywords: Low head; Hydropower; Microhydro; Overshot water wheel; Water mill.

Introduction

In irrigation canals and near old mills there is a great potential of low-head hydropower (ESHA ~~2004~~), in which head differences of a few meters and flow rates of few cubic meters per second are available. The installation of low-head hydroturbines could be a viable option to generate clean renewable energy below 100 kW (microhydropower) at these sites, with beneficial effects on the local economy and on the electrification of remote areas. In rural and decentralized areas, the installation of microhydro plants is considered one of the most economical options for rural electrification, especially when existing hydraulic structures are used to reduce infrastructure costs (Paish 2002; Bozhinova et al. 2013; ESHA ~~2004~~; Quaranta and Revelli 2018).

In this context, water wheels are considered more environmentally friendly and cost-effective than Kaplan and Francis turbines, due to their low rotational speeds, large buckets, and free-surface working behavior. Water wheels can be divided into stream water wheels and gravity water wheels; stream water wheels exploit the kinetic energy of flowing water (Quaranta 2018), whereas gravity wheels mainly use the water weight to generate energy (Quaranta and Revelli 2018). Among gravity water wheels, undershot, breastshot, and overshot water wheels can be identified. In overshot wheels (analyzed in this paper) the water enters the buckets from the top of the wheel; overshot wheels generally are employed for 2.5–6-m head and flow rates below 0.2 $\mathrm{m^3/s}$ per meter width, with maximum efficiencies of 85% (Williams and Bromley 2000; Dubas 2005; Pelliciardi 2015; Quaranta and Revelli 2015). Some existing overshot water wheels are shown in Fig. 1. Breastshot (Müller and Wolter 2004; Quaranta and Revelli 2016, 2017) and undershot water wheels (Von Harten et al. 2013; Quaranta and Müller 2018) generally are employed at sites with heads below 4 and 1.5 m, respectively, and have maximum hydraulic efficiency of between 75% and 80%.

The first systematic experiments on water wheels were carried out in the nineteenth century (Fairbairn 1864; Bach 1886; Chaudy 1896; Garuffa 1897; Weidner 1913; Meerwarth 1935), whereas the most recent hydraulic theory of overshot water wheels was presented and validated by Quaranta and Revelli (2015).

The tested water wheel was a 1:2 scale model of an existing overshot water wheel sited in an irrigation canal in North Italy [Fig. 1(b)]. The scale wheel was 1.46 m in diameter and 1 m wide, and had 24 blades that were 0.153 m deep [Fig. 2(a)] (Quaranta and Revelli 2015). A power loss model was developed and validated on experimental results with an average discrepancy between predicted and experimental power output of 8% [Fig. 2(b)]. Beyond a certain rotational speed, called the critical rotational speed, N_{cr}, the experimental power output P_o and the efficiency decreased linearly with the speed N. The efficiency decrease was due to the flow rate that was lost at the inflow (i.e. at the top of the wheel) when the flow impacted on the blades, forming splashes and water droplets [Figs. 1(a) and 3]. The critical speed was estimated as $N_{\mathrm{cr}} = 31.3/\sqrt{D}$, where D is the wheel diameter, in agreement with results reported by Williams and Bromley (2000). The results of Williams and Bromley (2000) and Quaranta and Revelli (2015) highlighted the importance of the inflow design of overshot water wheels, as also suggested for hydrodynamic screws (Lubitz et al. 2014; Straalsund et al. 2018) and breastshot and undershot water wheels (Quaranta and Revelli 2016; Quaranta and Müller 2019), both of which are classified as low-head hydraulic machines.

With the aim of reducing volumetric losses, Wahyudi et al. (2013) proposed converting volumetric losses from the buckets and from the channel into a high-velocity water jet by a nozzle, and squirting the jet on the lowest blade using a ram pump system. The efficiency increased from $\eta_o = 61.6\%$ to $\eta_{\mathrm{new}} = 73.5\%$, i.e., $\eta_{\mathrm{new}}/\eta_o = 1.2$.

The present study improved the performance of overshot water wheels at $N > N_{\mathrm{cr}}$ by converting volumetric losses into potential energy, rather than kinetic energy, by means of a wall installed around the periphery of the wheel. The theoretical model developed

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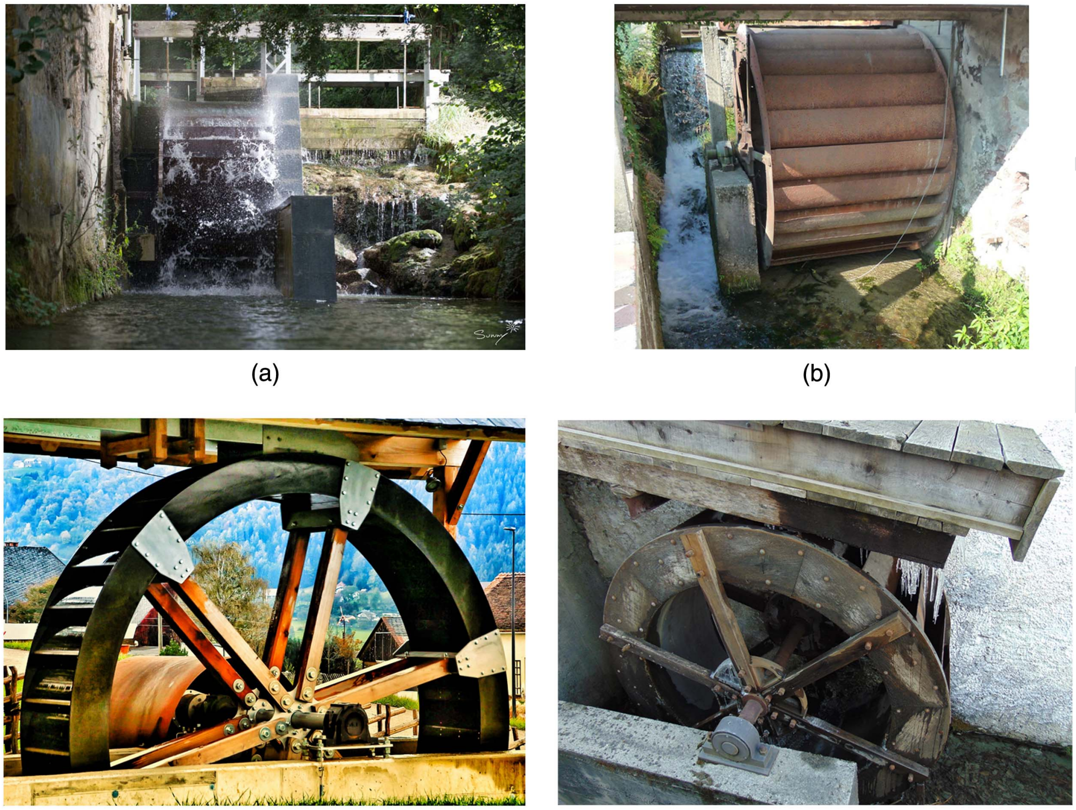
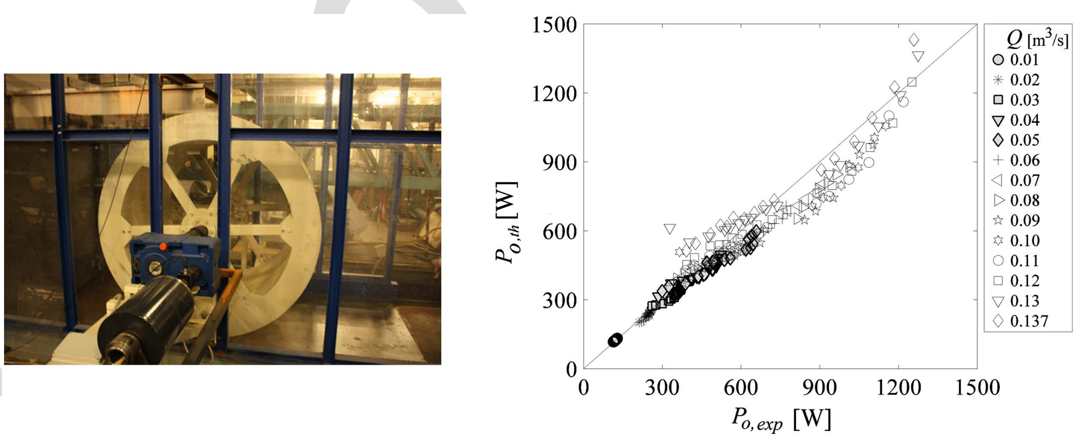


(a) (b) (c) (d)

Fig. 1. Overshot water wheels, with the conveying channel at the top: (a) overshot wheel of Gratia Hydro, 2.5 m in diameter and 1.5 m wide (image courtesy of Martin Eillebrecht); (b) wheel in Ciconio, 3 m in diameter and 2 m in width (image courtesy of Marco Gatta); (c) wheel in Judenburg, 4 m in diameter and 0.75 m wide (image courtesy of Helmut Mitterfellner); and (d) wheel in Bobbio Pellice, 2.6 m in diameter and 0.9 m wide (image courtesy of Emanuela Genre, Mulino di Bobbio staff).



(a) (b)

Fig. 2. (a) Tested water wheel in the laboratory (image by Emanuele Quaranta); and (b) predicted power output $P_{o,\mathrm{th}}$ versus experimental power output $P_{o,\mathrm{exp}}$ at different flow rates for the original configuration.

and validated by Quaranta and Revelli (2015) was implemented. As previously discussed, the model was proven to be reliable and accurate, and hence it can be used to explore different geometries in similar hydraulic configurations. The new design does not affect the upstream conditions (the upstream hydraulic depths and profiles do not change), which is an essential requirement in irrigation canals. The proposed method is expected to be more cost-effective than the conversion of water losses into kinetic energy.

Method

The theoretical model described by Quaranta and Revelli (2015), based on the estimation of power losses, was implemented. The model was based on the following assumptions:

1. The hydraulic behavior of overshot water wheels is very complex, because splashes, water droplets, and jets arise in the buckets during rotation. Such turbulent phenomena are neglected, so

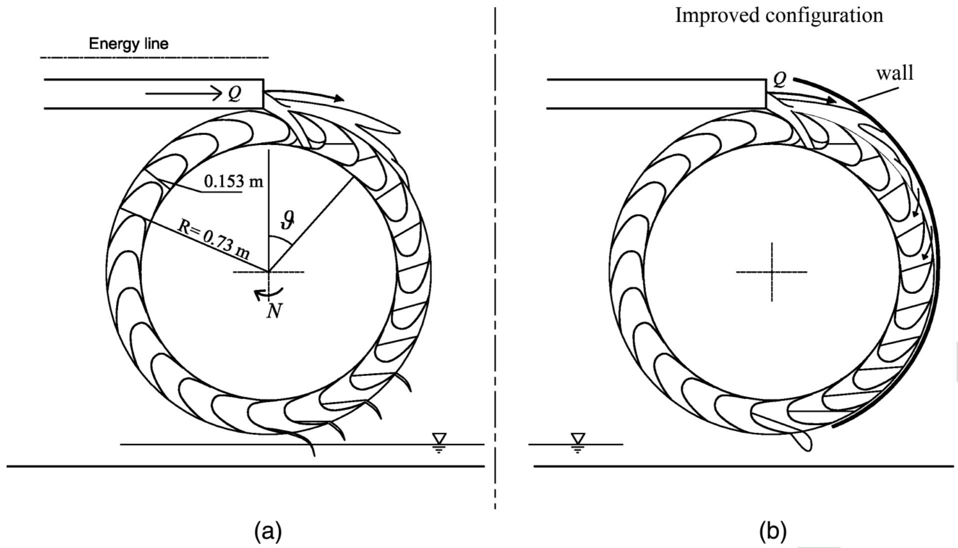


Fig. 3. (a) Original water wheel; and (b) modified configuration on the right, with additional wall to recover and direct the water into the buckets.

that the water jet is considered as a continuum and water in the buckets was considered to be at rest.

2. The flow behavior of water wheels is two-dimensional, i.e., velocity components parallel to the rotation axis are negligible and the water depth inside each bucket is uniform along the direction of the rotation axis. This assumption is well satisfied observing well designed water wheels.
3. The kinetic energy of the water flow which is redirected into the bucket is lost when the flow enters the bucket, because for most of the time the water flow impacts on the water volume already inside the bucket, dissipating its kinetic energy. Therefore, the additional impulsive force of the water impact is neglected.

With the previous assumptions, the mechanical power output P_o of a generic overshot water wheel can be estimated by the following equation (Quaranta and Revelli 2015):

$$P_o = P - \sum \text{Losses} = P - L_{\text{imp}} - L_t - L_g - L_{Q_u} - L_{Q_r} \qquad (1)$$

where L_{imp} = kinetic power loss occurring at inflow; L_t = impact loss of blades on tailrace (if blades are submerged in tailrace); L_g = mechanical friction loss at shaft supports; L_{Q_u} = volumetric loss at inflow; and L_{Q_r} = volumetric loss during rotation, due to the emptying process of the buckets. Inflow volumetric losses were quantified by the following equation:

$$\frac{L_{Q_u}}{P_{\text{in}}} \simeq 0 \quad \text{for } N < N_{\text{cr}}$$
$$\frac{L_{Q_u}}{P_{\text{in}}} = 2.2 \cdot \left[\frac{N}{N_{\text{cr}}} - 1\right] \quad \text{for } N > N_{\text{cr}} \qquad (2)$$

where L_{Q_u} = volumetric power loss; and $P_{\text{in}} = \gamma Q H$ = power input, where $\gamma = 9810\ \text{N/m}^3$ = specific weight of water, Q = total flow rate (m^3/s); and H = head difference (m).

This model was implemented and applied to the new geometric configuration depicted in Fig. 3. The new design consists of a wall located around the periphery of the wheel, used to redirect into the buckets the water which is lost at the inflow. No additional moving part is added, and thus the modification is cost-effective. A similar geometry was suggested (but not tested, neither experimentally nor theoretically) by Weisbach (1849), but with some conceptual differences. In our design, the water above the wheel and at the inflow is at atmospheric pressure; otherwise, the gravity wheel would become a pressurized turbine, which would be beyond the purpose of the present study. The diameter of the wheel is almost equal to the head difference, and the wall does not affect the upstream conditions (Fig. 3). In Weisbach (1849), the headrace was dammed at its end by a weir, creating a water basin above the wheel, upstream of the weir. The pressurized water entered into the wheel at high velocity through a nozzle on the bed of the basin. Therefore, a portion of the potential head was converted into kinetic energy. This solution can be useful for head differences above 6 m to avoid very large wheel diameters, and where the increase in the upstream water level (due to backwater propagation) is not a problem; this design was used in the last century in the Black Forest and in the Swiss Alps in Europe (anonymous reviewer, personal communication, 2020). A similar idea also was tested by Ikeda et al. (2010), although it must not be confused with the wall investigated herein, because in Ikeda et al. (2010) the role of the wall was to direct the water of a waterfall into an action turbine, with the aim of controlling the flow direction at variable flow rates.

To avoid backwater propagation, the wheel and the wall at the inflow should not interact with the upstream water flow; hence the distance between the wheel and the wall was set at 0.15 m at the inflow. In the lowest half of the wheel, the clearance between the wall and the wheel must avoid friction between the wall and the wheel, and must minimize the flow through the gap. The gap value of 0.01 m was chosen because this value generally is adopted between undershot/breastshot water wheels and their curb and lateral shrouds for manufacturingconsiderations (Quaranta and Müller 2018). Therefore, the curvature radius R of the wall in the lowest half of the wheel was $R_{\text{wall}} = R_{\text{wheel}} + 0.01$ m. The wall in the upper part of the wheel was shaped to gradually connect the superior part of the wall (where there is clearance of 0.15 m) to its inferior portion. The wall has to be interrupted at $\theta \leq \pi$ (Fig. 3) for a correct emptying process and to avoid counteracting torque due to residual water that may be carried out upstream of the wheel.

Considering the presence of the wall, the following assumptions have to be taken into account in the theoretical model:

1. The amount of water that cannot enter into the buckets runs through the gap between the wall and the wheel. When the flow reaches $\theta = \pi/2$, i.e. one-half the wheel diameter D (Fig. 3), it enters the bucket. This means that the water flow which is lost

from the top of the wheel does not lose the full hydraulic head H (as in the original configuration), but it loses a potential head equal to $D/2$. Therefore, the inflow power loss L_{Q_u} [Eq. (2)] must be multiplied by $(D/2)/H$ (generally in overshot water wheels, $D \simeq H$). The position $\theta = \pi/2$ was chosen for two reasons: at lower values of θ, the water jet may impact on the outer surface of the blade (the downstream face), generating power losses; and at $\theta = \pi/2$, the tangent at the wheel circumference is vertical, and thus the wall is easier to build in practical applications.
2. Once the water is redirected into the bucket, it remains inside the bucket, acting by its weight. When the bucket water volume equals the maximum storable volume, the bucket starts to empty; the emptying process is not affected by the wall, as in the original theoretical model. Therefore, the calculation method for the power losses L_{Q_r} does not change (Quaranta and Revelli 2015). However, in this case the bucket is filled with a larger amount of water, because it also contains the water that is lost at the inflow.
3. The critical rotational speed N_{cr} is the same, and it depends only on the wheel diameter [Eq. (2)]. This is because the inflow volumetric losses start at the same rotational speed (i.e., N_{cr}), so the inflow not affected.

Results and Discussion

The modified theoretical model was applied to 256 operative conditions: the flow rate was in the range $Q = 0.01$–$0.137\ \mathrm{m^3/s}$ and the wheel rotation speed was in the range within $N =$ 0.5–3.5 rads/s. The upstream flow velocity varied between 0.5 and 1.5 m/s, whereas the water depth at the end of the conveying channel (i.e. just upstream of the wheel) varied from 0.02 to 0.09 m. The velocity of the water jet during the impact with the blade ranged between 1.5 and 2.4 m/s (Quaranta and Revelli 2015). The average discrepancy between theory and experiments was 8% [Fig. 2(b)].

Fig. 4 depicts the efficiency in the original configuration η_o and the efficiency in the modified configuration η_w with the wall. The efficiency is defined as the ratio of the power output to the power input, i.e. $\eta_o = P_o/P_{\mathrm{in}}$ considering the original configuration, and $\eta_w = P_w/P_{\mathrm{in}}$ considering the modified configuration, where P_w is the power output with the wall.

When $N > N_{\mathrm{cr}} = 2.7$ rad/s, η_o and η_w decrease due the the increase of volumetric losses at the inflow with N (Fig. 4). However, η_w is higher than η_o, because the inflow water losses are redirected into the bucket at the middle of the wheel. When $N \leq N_{\mathrm{cr}}$, improvements are negligible, because inflow volumetric losses are negligible. The ratio $\eta_w/\eta_o = P_w/P_o$ increases with the rotational speed (Fig. 4) according to the following dimensionless equation:

$$\frac{P_w}{P_o} \simeq 1 \quad \text{for } N < N_{\mathrm{cr}}$$
$$\frac{P_w}{P_o} = 1 + \left(1.54 \cdot \left[\frac{N}{N_{\mathrm{cr}}} - 1\right]\right) \quad \text{for } N > N_{\mathrm{cr}} \tag{3}$$

Once P_w is estimated and normalized to P_o [Eq. (3)], the power output improvement G_w strictly related to the wall can be calculated as $G_w = P_w - P_o$. If G_w is normalized by P_{in}, it can be added to the power loss terms in Eq. (1) to theoretically take into account of the wall benefit (the other power loss terms can always be estimated as in the original configuration); G_w/P_{in} can be expressed as a function of the normalized speed N/N_{cr}

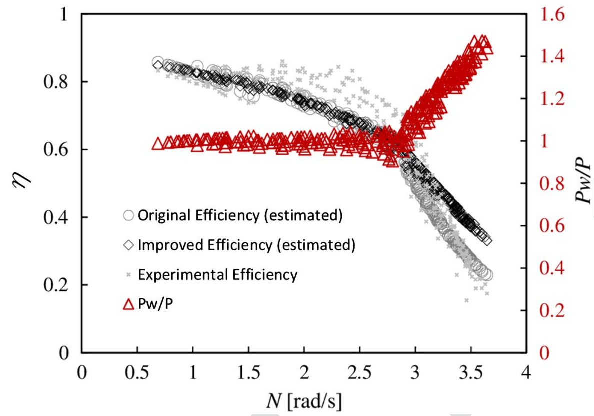


Fig. 4. Original efficiency (theoretically estimated and experimentally measured), η_o, improved efficiency, η_w, and $P_w/P_o = \eta_w/\eta_o$ versus rotational speed.

$$\frac{G_w}{P_{\mathrm{in}}} = -0.94\left(\frac{N}{N_{\mathrm{cr}}}\right)^2 + 2.57\left(\frac{N}{N_{\mathrm{cr}}}\right) - 1.65 \tag{4}$$

Based on the investigated conditions, the terms calculated in Eqs. (3) and (4) are valid when $1 < (N/N_{\mathrm{cr}}) < 1.35$, and can be applied in practical applications.

In our new configuration, water losses are converted into hydrostatic pressure, instead of velocity, generating a more effective [with η_w/η_o up to 1.5, whereas the ratio was 1.2 in Wahyudi et al. (2013)] and cheaper system. Furthermore, the additional cost of the wall is negligible with respect to the total cost of the installation, although some sediments may be trapped between the wheel and the wall, interfering with the rotation. A trash rack is required in this case, which, in any case, also is recommended for traditional water wheels.

Conclusions

A modified design of overshot water wheels was theoretically investigated with the intent of improving the performance at rotational speeds higher than the critical one. The modified design consists of a lateral wall around the width of the wheel. The wall recovers the inflow volumetric losses and redirects them into the wheel without affecting the upstream conditions. The modified design is a cost-effective improvement of overshot water wheels, and equations were proposed for use in practical applications to quantitatively estimate the benefits of the wall. This optimization can lead to the exploitation of higher discharges with higher efficiency. Future developments of this concept could lead to the increase of the operational speed of water wheels (with a consequent reduction of the electromechanics costs), improving the efficiency of the wheel up to 1.5 times with respect to the original efficiency.

Data Availability Statement

All data, models, and code generated or used during the study appear in the published paper.

Acknowledgments

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