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Review

Stream water wheels as renewable energy supply in flowing water: Theoretical considerations, performance assessment and design recommendations

Emanuele Quaranta

Politecnico di Torino, DIATI (Department of Environment, Land and Infrastructure Engineering), Corso Duca degli Abruzzi 24, 10129 Torino, Italy

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ABSTRACT

Water wheels were the earliest hydraulic machines used in antiquity to convert water energy into mechanical one. Due to their simple installation, low maintenance costs, and thanks to the possibility to use local manpower and material for their construction, nowadays water wheels are again used as energy supply, especially in remote localities and emerging countries. In particular, stream water wheels are installed in flowing water where there are not head differences. The performance depends on the blockage ratio, so that they can be subdivided into three main categories: stream wheels in shallow subcritical flow, shallow supercritical flow and deep flow.

In this paper, experimental, theoretical and numerical data on stream water wheels were systematically collected from literature and analyzed. Guidelines for their design were discussed focusing especially on wheel dimensions, supporting structures, blades and speed. More light on their hydraulic behavior was shed, adopting the previous classification for a better explanation and understanding. Results showed that in shallow water an head difference can be generated by the wheel, increasing the power output. In deep flow, accurate hydrodynamic floating/supporting structures allow the hydrostatic force of water to be exploited in addition to the kinetic energy of the flow. As a consequence, power output can improve from 0.5 to more than 10 kW per meter width, so that stream wheels can represent an attractive energy supply in zero head sites.

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E-mail addresses: emanuele.quaranta@polito.it, quarantaemanuele@yahoo.it.

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Introduction

Nowadays, the increasing demand of energy and electricity, and the growth of population, pose two major challenges. The first one is the need of reducing greenhouse gas emission and environmental pollution, while satisfying the global energy demand. This challenge can be afforded by using large power plants based on renewable sources, like hydropower plants (Banja, Monforti, & Scarlat, n.d.; European Commission, 2009a, 2009b). The second challenge is to give equal energy access to all countries and people, since current energy access is neither universal nor guaranteed. The World Energy Outlook estimates that 1.2 billion people do not have access to electricity (Blodgett, Dauenhauer, Louie, & Kickham, 2017). Micro-grids (i.e. power output less than 100 kW) based on renewable energy are a promising option for this challenge, due to their low initial investment levels, scalability and suitability for rural areas (Blodgett et al., 2017).

Hydropower represents an interesting option to satisfy both the challenges. Hydropower plants are currently the most contributory renewable energy source worldwide (Laghari, Mokhlis, Bakar, & Mohammad, 2013), and they continue to be installed, especially in emerging countries (de Faria & Jaramillo, 2017). However, due to the need of large dams, large hydro plants generate some adverse effects on ecosystems: flooding of large areas upstream, interruption of the river longitudinal connectivity, changing in the hydrological regime and sediment transport processes of rivers, and, sometimes, social impacts (Kallis & David, 2001). Instead, micro hydro grids are more environmental friendly than big hydro plants. Sites suitable for micro hydro plants are present in almost all countries (Blodgett et al., 2017; Laghari et al., 2013), so that micro hydro plants could be a promising option both as energy supply and as easy access to energy.

Micro hydro plants are becoming very popular and attractive especially in rural and decentralized areas, and in developing countries, where the large distances usually require decentralized electricity production and off-grid power plants. Micro hydro plants can provide a simple energy access to small and local communities, or to remote industrial sites. Micro hydro schemes can use existing civil/hydraulic structures, for instance old water mills, so that total installation costs are minimized (European Small Hydropower Association (ESHA), 2014). Micro hydro is also of high importance in industrialized countries for meeting the non-fossil fuel targets, for satisfying the rising electricity demand and for new market opportunities (Paish, 2002; Quaranta & Revelli, 2017).

Turbines for micro hydro plants

The turbine is the component of the hydro plant that converts the power of the flow into mechanical one. Basically, turbines can be classified into action turbines, like Pelton turbines, and reaction turbines, like Kaplan and Francis turbines. Pelton turbines are used with heads up to 2000 m, Francis turbines up to 700 m, and Kaplan turbines up to 40 m, and a typical range of heads is tens/hundreds meters. Where heads of few meters are present (e.g. <3 m), Francis and Kaplan turbines can be scaled and installed, but they are not cost-effective: they are of not easy installation, investment costs are high and payback times of more than 20 years are expected. Environmental impacts are significant, because these turbines require pressurized pipes, draft tubes and screens to avoid passage of sediments and fish through the turbine (Bozhinova, Kisliakov, Müller, Hecht, & Schneider, 2013; Elbatran, Yaakon, Ahmed, & Shabara, 2015; Quaranta, 2017; Williamson, Stark, & Booker, 2014). Pelton turbines are not convenient due to the low flow rate that can be swirled.

As a consequence, in the last decades, new hydropower converters for low head sites have been introduced on the market, like Archimedes screws (Lubitz, Lyons, & Simmons, 2014; Lyons & Lubitz, n.d.; Waters & Aggidis, 2015), gravity water wheels and stream water wheels (Müller, Denchfield, Marth, & Shelmerdine, 2007; Müller & Kauppert, 2004; Quaranta, 2017) and hydrokinetic turbines (Vermaak, Kusakana, & Koko, 2014). These machines are more environmental friendly and cost-effective than typical action and reaction turbines (Bozhinova et al., 2013). Their rotational speed is slower, and they do not require pressurized pipes, so that risks imposed on fish and problems with trapped sediments are minimized (Bozhinova et al., 2013). Therefore, maintenance costs are reduced, and payback times are significantly lower than those of micro plants equipped for example with Kaplan turbines (Müller & Kauppert, 2004).

Archimedes screws (or hydrodynamic screws if the external supporting shroud does not rotate) and gravity water wheels are used in sites where there exists a drop in the channel bed, that hence creates an head difference. The pressure exerted on the blades is generated by the water weight, thus it is an hydrostatic pressure that only depends on the water depth over the blades. Therefore, such hydropower converters are called gravity machines, or hydrostatic pressure converters. Archimedes screws and gravity wheels are generally used from 0.5 m to 6–8 m head, and they are partially immersed in water. Archimedes screws rotate around an axis inclined of 22°–35° on the horizontal, while the rotational axis of gravity water wheels is horizontal. With regard to gravity water wheels, undershot wheels are used for head differences between 0.5 and 1.5 m (Quaranta & Müller, 2018-b; v. Harten, Paudel, & Saenger, 2013), breastshot wheels are usually employed for head differences between 1.5 and 4 m (Müller & Wolter, 2004; Quaranta & Revelli, 2015b), and overshot wheels are used for head differences between 2.5 and 6 m (Müller & Kauppert, 2004; Quaranta & Revelli, 2015a). Fig. 1a–c depict an example of Archimedes screw, undershot and overshot water wheel.

Hydrokinetic turbines were originally conceived like wind turbines. Nowadays they are also installed in flowing water, with zero head conditions and without drops in the channel bed, so that only the flow kinetic energy is employed for power production. Hydrokinetic turbines are completely immersed in flowing water, and they are typically built with vertical axis. Two types of hydrokinetic turbines can be identified: the drag type and the lift type. In the drag type, like the Savonius hydrokinetic turbine, the drag force exerted at the blades drives the turbine. In the lift type, like the Darrieus turbine, the turbine rotation is provided by the lift force at the blades (Anyi & Kirke, 2011; Vermaak et al., 2014). Fig. 1d depicts a Darrieus turbine. In Vermaak et al. (2014), hydrokinetic turbines have been deeply discussed.

Stream water wheels are used in the same hydraulic conditions of hydrokinetic turbines: flowing water with no head difference in the undisturbed flow regime (or very small so that the potential energy is less than the kinetic one). Differently from hydrokinetic devices, the rotational axis of stream wheels is horizontal, like gravity water wheels (Fig. 2), and it is installed over the free surface of water, so that only the lowest part of the wheel interacts with the water flow.

Fig. 1. (a) Three Archimedes screw in parallel (Enerca power plant, Crescenzago, Italy, photo of Emanuele Quaranta), (b) undershot water wheel (Germany, photo of Gerald Müller), (c) overshot water wheel, research project ORME (Ciconio, Italy) (Quaranta, 2017) and (d) Darrieus hydrokinetic turbine (installed in the Hydraulic laboratory of Politecnico di Torino).

Stream water wheels can be used for different purposes: power supply for local activities and mills (handmade works or crop grinding) (Fig. 2a) electricity (Fig. 2b) (Müller, Jenkins, & Batten, 2010), and as device for pumping water in irrigation canals, the so called spiral pumps (Kumara, n.d.) (Fig. 2c). For the generation of electricity, an electrical generator has to be mounted at the shaft. Instead, in spiral pumps, a spiral tube is wrapped around the central shaft of the wheel. Water of the river is collected by the tube external edge (located at the wheel circumference). Water flows along the pipe, from the pipe edge to the wheel shaft, where a pipe connected with the river side carries water to the end-user. Common spiral pumps are able to pump to a maximum height of 20 meters and a maximum flow rate of 43.6 m^3/day (Aqysta, n.d.; Kumara, n.d.).

Stream wheels are especially worthwhile in sites where local manufacture and materials can be employed for their installation, like in rural areas and emerging countries. They are of simple construction (little civil engineering work is required), with low installation costs, few maintenance problems and high cultural and aesthetic value (Müller et al., 2007, 2010). The full implementation of such technology can lead to the establishment of many small river hydroelectric power stations, that in turn will create sustainable development, manufactures and jobs (Akinyemi & Liu, 2015).

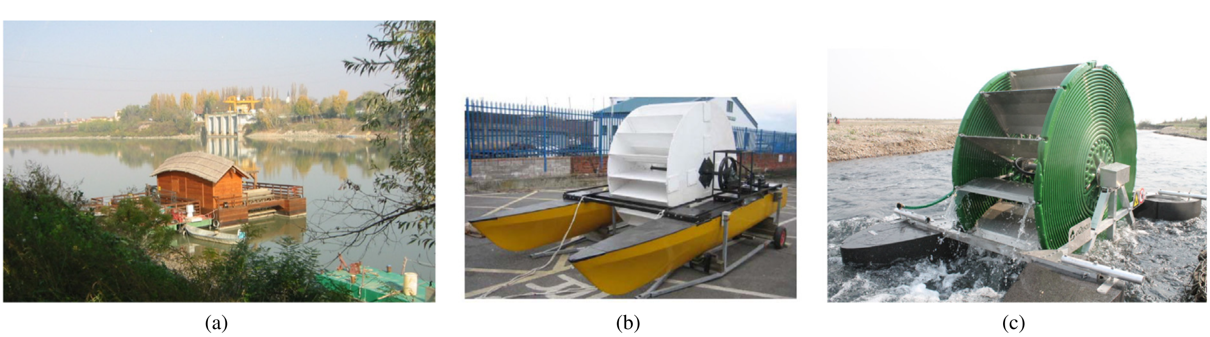

Fig. 2. Stream water wheels for different purposes: (a) mechanical power generation (photo courtesy of AIAMS Italy), (b) electricity production Turnock et al., (2007), and (c) pumping in irrigation systems (photo courtesy of Jaime Michavila (Aqysta, n.d.)).

(c) (d)

Fig. 3. Stream water wheels in: (a) subcritical shallow flow (Quaranta, 2017), (b) supercritical shallow flow (photo of Emanuele Quaranta), deep flow (photo courtesy of AIAMS Italy) (c), and Hydrostatic Pressure Machine (picture of Nick Linton) (Linton, 2013) with diagonal blades to reduce drag at the blades (d).

Stream wheels: types and scope of the work

Apparently, in flowing water only the kinetic energy of the flow could be exploited by a stream water wheel, but, actually, performance and hydraulic behavior of stream wheels depend on Froude number (subcritical or supercritical flow) and blockage ratio (Bahaj, Molland, Chaplin, & Batten, 2007; Müller et al., 2007). The blockage ratio is defined as B.R. $= A/A_c$, where A is the immersed blade/wheel area (measured orthogonally to the flow direction) and A_c is the wet channel cross section. At low B.R. the presence of the wheel substantially does not modify the flow field inside the channel, except very close to the wheel, due to the blade entry and exit process, and fluid-structure interaction. This is the typical case of stream wheels installed in deep flow and large rivers; such stream wheels are also called floating wheels and the power output depends on B.R. As B.R. increases a higher portion of flow is forced to pass through the wheel, increasing the power output. For example, at B.R. $= 0.2$ the increase in power output due to the blockage effect is 30% with respect to the undisturbed configuration, while it is less than 10% at B.R. ≤ 0.05 (Müller et al., 2010). As B.R. still increases (B.R. $\rightarrow$ 1), power losses in the river flow generated by the presence of the wheel would be so high that, in subcritical flow, a well identifiable backwater propagation arises, and the discharge that can pass downstream only depends on the wheel rotational speed (see Stream wheels in shallow supercritical flow for more details). This is the case of stream water wheels installed in shallow water, where the blade length is similar to the water depth, and an head difference is induced to drive the wheel. If the undisturbed flow regime is supercritical, the upstream flow can be converted into subcritical, or it can remain supercritical, depending on wheel tangential speed.

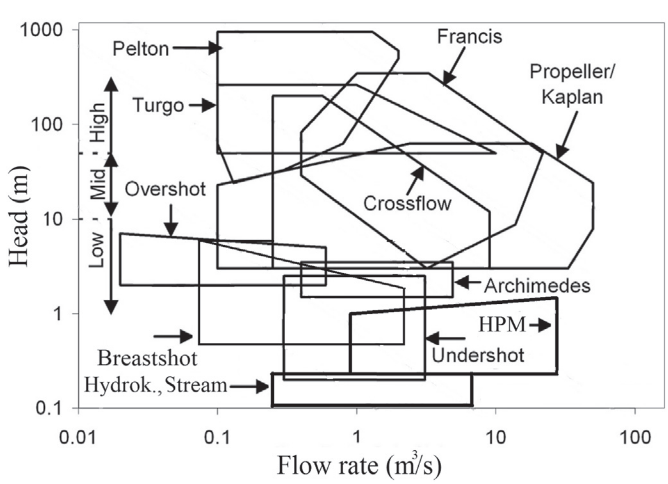

Fig. 4. Working conditions of hydropower converters (adapted from Williamson et al., 2014). Stream water wheels and HPM are highlighted with a thicker line, since they were discussed in this review.

In light of this, three types of stream wheels can be identified, as suggested in Müller et al. (2007): stream wheels in shallow subcritical flow, that are called Hydrostatic Pressure Wheels (HPW), stream wheels in shallow supercritical flow (kinetic wheels) and stream wheels in deep flow (floating wheels). The types of stream wheels are depicted in Fig. 3. The most optimized design of HPW (in terms of efficiency and smaller encumbrance) is the hydrostatic pressure machine (HPM) intentionally conceived to self generate the hydraulic head. In this way, maximum head differences are expected to be 2.5 m. The Dethridge water wheels is a machine similar to the HPM and used for discharge measurements (Paudel & Saenger, 2016). Classical HPW are hydropower machines where the working blade surface extends from the channel bed to the upstream water surface, while in HPM the working blade surface extends from the

Table 1
Scientific works on stream water wheels. The kind of investigated water wheel and the hydraulic configuration are also reported (shallow “S” or deep “D” water, subcritical “M” or supercritcal “H” flow). “T” means theoretical work, “E” means experimental work and “N” means numerical simulations. Results based on electrical components, structural analyses and fabrication issues are valid for all kinds of wheels.

Authors	Year	Type of work	Type of wheel	Type of analysis	Reference
Patent, De Borda, Smeaton	XVIII century	Hydraulic performance	–	T, E	Capecchi (2013), Smeaton (1759)
Bach, Weisbach, Busquet, Bresse	XIX century	Hydraulic performance and design	D	T	Bach (1886), Bresse (1896), Busquet (1906), Weisbach (1849)
Baddhadi and Mikhail	1985	Hydraulic performance and design	SH	E	Bagdhadi and Mikhail (1985)
Gotoh et al.	2001	Hydraulic performance and design	SM	E	Gotoh et al. (2001)
Domozhirov	2005	Structural analysis	D, SM, SH	T, E	Domozhirov (2005)
Müller et al.	2007	Hydraulic performance and design	D, SM, SH	T, E	Müller et al. (2007)
Turnock et al.	2007	Hydraulic performance and design	D	E	Turnock et al. (2007)
Li et al.	2008	Electrical analysis	D, SH, SM	E	Li et al. (2008)
Müller et al.	2010	Hydraulic performance and design	D	T, E	Müller et al. (2010)
Senior et al.	2010	Hydraulic performance and design	SM	E, T	Senior et al. (2010), Senior et al. (2008)
Batten and Müller	2010–2011	Hydraulic performance and design	D	T, E	Müller and Batten (2010), Batten and Müller (n.d.), Batten et al. (2011)
Dietz et al.	2011	Electrical equipment	D, SM, SH	T, E	Dietz et al. (2011)
Tevata et al.	2011	Hydraulic performance and design	SM	E	Anurat and Chainarong (2011)
Hadler and Broekel	2011	Hydraulic performance and design	D	T, E	Hadler and Broekel (2011)
Paudel et al.	2013	Hydraulic performance and design	SM	E	Paudel et al. (2013)
Luther et al.	2013	Hydraulic performance and design	SM	E	Sule et al. (2014), Sule et al. (2013)
Sonaje et al.	2013	Hydraulic performance and design	SH	N	Sonaje et al. (2013)
Kyaw et al.	2014	Structural analysis	SM	E	Kyaw et al. (2014)
Kumara	2014	Hydraulic performance and design	D	T, E	Kumara (n.d.)
Garcia et. al	2015	Hydraulic/structural analysis	SM	E, N	García et al. (2015)
Liu et al.	2015	Hydraulic performance and design	D	T, N	Akinyemi and Liu (2015), Yucheng (2012)
Khan et al.	2015	Hydraulic/structural analysis	D	T, E	Khan et al. (2015)
Dutta et al.	2016	Fabrication	D, SM, SH	T, E	Dutta et al. (2016)
Cleynen et al.	2017	Hydraulic performance	D	E	Cleynen et al. (2017)
Butera et al.	2017	Hydraulic performance and design	SM	E	Butera et al. (n.d.)

channel bed to the downstream water surface. The buckets of the HPM are completely filled, and the discharge that can pass through the HPM does not depend on the upstream water depth, but only on wheel dimensions and rotational speed. Operational ranges of stream wheels, HPM and HPW are plotted in Fig. 4 and compared to other turbines. Common flow rates vary between 0.5 and 8 m^3/s.

However, design prescriptions and efficiency estimates of stream water wheels often did not consider the flow regime (shallow/deep water, sub/super critical flow), so that literature data are often quite confusing, fragmented and of difficult comparison. Hence the first aim of this manuscript was to collect and organize literature data on stream wheels on the basis of the flow regime, supported by complementary material on gravity water wheels.

With regard to the power output, two non-dimensional variables can be identified. Where the head difference is the main driving torque (HPM and HPW), the power output P from tests/simulations can be normalized to the power input P_{in} of the flow based on the head difference upstream-downstream, i.e. $P_{in} = \rho g Q \Delta H$ (ρ is water density, g is the gravity, Q is the flow rate and ΔH is the head difference). Such normalized variable is called efficiency η. Instead, when head differences are small and the kinetic power of the impinging flow gives the predominant driving contribute, the power coefficient C_p can be calculated normalizing the power output to the kinetic power of the flow $P_{kin} = \frac{1}{2}\rho Q {v_1}^2$ ($Q = A \cdot v_1$ is the flow rate, A is the blade area and v_1 is the approaching flow velocity). The coefficient C_p is typically used in wind and hydrokinetic turbines (Vermaak et al., 2014). However, several studies called this coefficient not C_p, but η, so that misleading results are present in literature. In this review, η and C_p were distinguished.

Furthermore, a survey conducted in UK found that there was no correlation between the design parameters of historic water wheels (Turnock et al., 2007), although diameters were generally included between 3 m and 4 m, rotational speeds lower than 12 rpm and widths lower than 2.5 m, independently from the flow regime. This occurred because water wheel designs were chosen based on designs that already existed or were proven to work, although they were not optimal (Turnock et al., 2007). The third aim of this paper was thus to propose guidelines for the engineering design of stream wheels, based on literature results and considering the flow regime. Finally, existing gaps and open questions in the engineering design were highlighted, establishing research objectives to be addressed in the future.

In particular, in General considerations general data valid for all kinds of stream wheels were discussed, independently from the flow regime, like literature works on electro-mechanics equipment, materials and fabrication processes. After General considerations, performance characteristics, hydraulic behavior, strategies to improve the efficiency and design guidelines of stream wheels were presented for stream wheels in shallow subcritical flow, stream wheels in shallow supercritical flow and stream wheels in deep flow. At the end of the paper, an all-inclusive discussion section was provided, where future research items were also discussed.

General considerations

Research conducted on stream wheels are summarized in Table 1. The oldest scientific works on stream wheels were conducted during the XVIII century (Capecchi, 2013). By the time, theories and manufacturing methods of stream (and gravity) water wheels improved (Bach, 1886; Bresse, 1896; Busquet, 1906; Weisbach, 1849). The scientific interest in water wheels declined in the twentieth century, although water wheels continued to be in operation, to restart again at the beginning of the twenty-first century. As it can be seen, the most of the works dealt with the investigation of hydraulic performances (power output and optimal rotational speed). A little research has been spent on structural and electrical issues. While the hydraulic performance depends on the flow regime, data on structural and electrical components can be considered of general validity.

Structural analyses of the wheel components can be found in García, Sola, and de la Morena-de la Fuente (2015) and Kyaw, Kyaw, and Aye (2014). In García et al. (2015), a study has been carried out for a stream wheel in shallow water. A design methodology based on three aspects was proposed: 3D geometric modeling, analysis with computational fluid dynamics tools and finite-element analysis on tension forces caused by the fluid dynamic interaction. In Kyaw et al. (2014), the global design of a water wheel was suggested, recommending to take into account structural aspects, load torque, driving torque, power and efficiency, shaft and bearing design (Kyaw et al., 2014). Fabrication considerations can be found in Khan, Ahmed, Khan, and Haider (2015), while in Domozhirov (2005), a fatigue analysis has been conducted for the blades. In Turnock et al. (2007), structural analyses and fabrication suggestions were discussed. It was found that great investments are required with regard to the materials. Indeed, although stream wheels were typically built with wood, nowadays innovative materials have been introduced, like High Density Polyethylene (HDPE), that is lighter and stronger (Dutta, Shrestha, Shahi, Chaudhary, & Lal Shrestha, 2016).

Stream wheels rotate very slowly, and they require gearboxes of multiplication ratio of approximatively 100 times for the production of electricity with induction generators. In this regard, useful information can be found in Laghari et al. (2013), where descriptions of electro-mechanics equipments and power take-off systems of mini hydro plants were reported. In Li et al. (2008), analyses on the electric output results were discussed specifically for a stream wheel equipped with gearbox and three-phase permanent-magnet generator. However, as it will be highlighted in Gaps and future works, the power take off system represents the most serious deficit is stream wheels operation for electricity generation. New induction generators and intelligent controllers, despite their complexity, can successfully make micro and mini hydro schemes more economical and cost-effective options (Dietz, Groeger, & Klingler, 2011; Fergnani, Silva, & Bavera, 2016).

Concerning with costs, payback periods were estimated in two years for modern stream wheels (when using an ad hoc floating structure hydrodynamically shaped Turnock et al., 2007), and between 4 to 7 years for the traditional stream wheel (Drews Wasserrad, 2017). When local resources and manufacture are used, installation costs can be significantly reduced: this is for example the case of emerging countries.

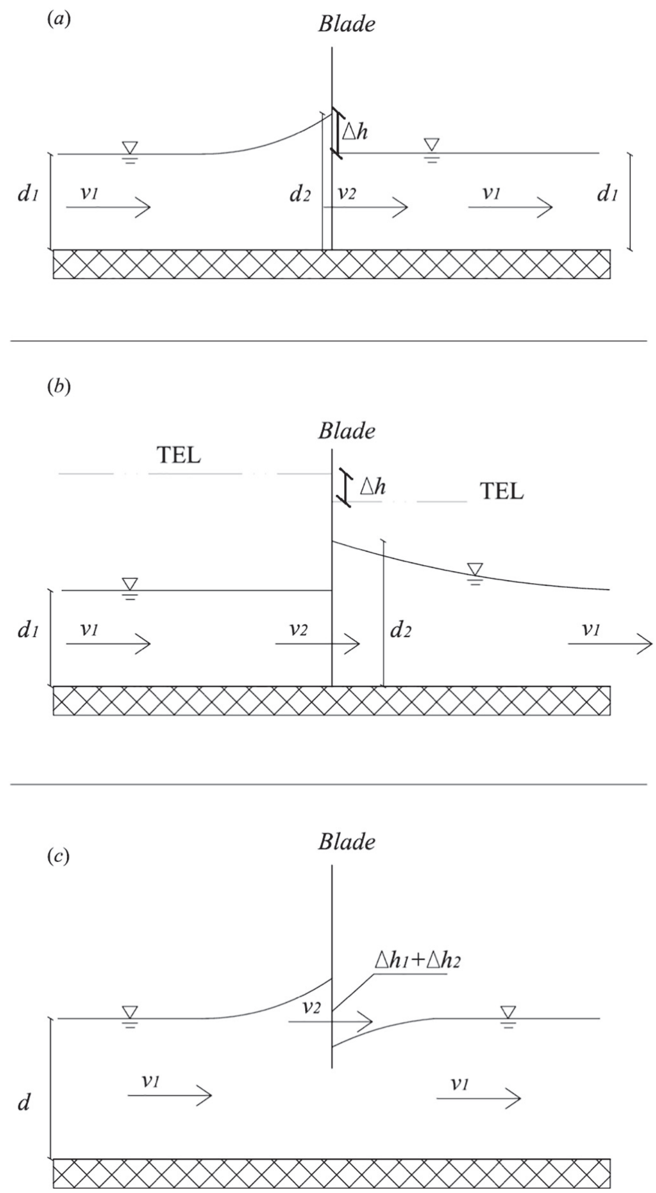

Fig. 5. Hydraulic behavior of stream wheels and theoretical energy transfer between the water flow and the blades of the wheel. (a) Stream water wheel in subcritical shallow water (b) stream water wheel in supercritical shallow water, (c) stream water wheel in deep (subcritical) water. v_1 is the undisturbed flow velocity, while v_2 is the blade velocity

Stream wheels in shallow subcritical flow

In this paper, water flows are considered to be shallow when the blade length is very similar to the water depth, and the presence of the wheel generates a high blockage effect (B.R. is almost 1). Shallow flows are subcritical when the water depth is higher than the critical depth, or supercritical when the water depth is lower than the critical depth. In Fig. 3a–b two examples of wheels in shallow flow are depicted.

In subcritical shallow flows, the flow rate that can flow downstream through the wheel depends on wheel dimensions and rotational speed. For HPW, the swirled flow rate depends also on the upstream water level (that is affected by the wheel, because the wheel creates a backwater propagation). Instead, for HPM, the upstream water level does not affect the swirled flow rate, since buckets are already filled with water completely. Therefore, when the flow rate in the channel is higher than the flow rate that can pass through the HPM, the water level upstream would theoretically approach infinite. However, its increase will not be unlimited, because at a certain water level the wheel will become submerged. Hence upstream hydraulic structures, like outflow weirs, are needed to maintain the desired upstream water level, as well as the head difference to drive the wheel. This is also valid for HPW if a certain upstream water depth is desired.

Theoretical considerations

In this section, theoretical equations to estimate the power output of stream water wheels in shallow water are presented. In the equations found in literature, the following hypothesis were generally assumed, although not always specified: (1) the behavior is one-dimensional and a steady state is considered; (2) one blade only interacts with the flow; (3) the blade is perpendicular to the flow velocity.

The most simplified attempt to estimate the power output P was that reported in Müller et al. (2007). The momentum theory was applied on a blade of velocity v_2 in unconfined flow, neglecting the hydrostatic force generated by the head difference at the blades:

$$P = \rho A(v_1 - v_2)^2 v_2 \quad (1)$$

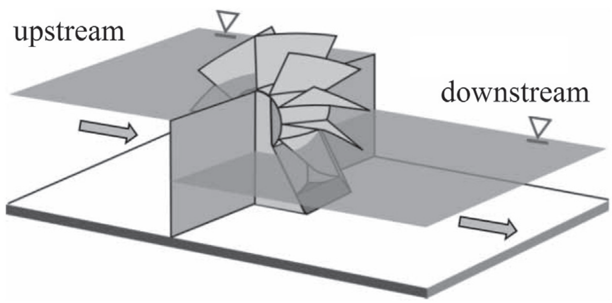

Fig. 6. Schematic sketch of the hydrostatic pressure machine (HPM). The free surfaces upstream and downstream are depicted (Senior et al., 2010).

where A is the submerged area, v_1 and v_2 are the flow and blade velocity, respectively, and ρ is the water density. The theoretical model shows maximum power when $v_2/v_1 = 0.33$. Normalizing Eq. (1) to the kinetic power of the approaching flow $P_{kin} = \frac{1}{2}\rho A v_1^3$ ($A \cdot v_1 = Q$, where Q is the approaching flow rate), the power coefficient is obtained. Its maximum value is $C_p = 0.296$, obviously at $v_2/v_1 = 0.33$. Therefore, as extension of Eq. (1), the power of a stream wheel in shallow flow can be estimated as:

$$P = \frac{1}{2}\rho A C_p v_1^3. \tag{2}$$

The value of C_p is a function of v_2/v_1, and its theoretical maximum value is $C_p = 16/27$ (Betz limit), while $C_p = 0.296$ is the maximum value from the momentum theory.

Since the head difference is not considered, Eq. (2) is valid for stream wheels whose dimensions are smaller than the channel ones (low blockage ratios), i.e. stream wheels in deep water, or when $v_2 \simeq v_1$ (in the latter case $C_p \to 0$). This is because when $v_2 \le v_1$ and in shallow water, the head difference is not negligible (due to the high blockage ratio), and the theoretical process depicted in Fig. 5a has to be considered to estimate the power output. Since subcritical flows are downstream governed, just upstream of the blade the flow velocity is the same of the blade tangential speed, which is lower than the undisturbed flow velocity. Therefore, the water level upstream increases as a consequence of the full blockage effect. Hence an head difference arises, although the original situation would be a zero head condition. Downstream, the flow velocity is the undisturbed flow velocity, since subcritical flows do not depend on the upstream boundary conditions. Therefore, from a theoretical point of view, the previous considerations have to be taken into account. Considering an HPW with an infinite diameter, and neglecting turbulent, leakages and inflow/outflow power losses, the efficiency can be calculated by Eq. (3) (Senior, Saenger, & Müller, 2010):

$$\eta = \frac{1}{2}\left(1 + \frac{d_1}{d_2}\right) \tag{3}$$

where d_1 is the water depth in the undisturbed configuration (downstream water level) and d_2 is the upstream water depth ($d_1 \le d_2$). Eq. (3) was derived by normalizing the power output (caused by the hydrostatic forces acting on the blade due to the head difference $\Delta H = d_2 - d_1$) to the power input $P_{in} = \rho g Q \Delta H$. The efficiency estimated by Eq. (3) was slightly higher than experimental results for $0.6 \le d_1/d_2 \le 0.9$. Turbulent losses can be estimated by $P_{turb} = \frac{1}{2}\rho A C_d v_b^3$, where A is the blade area, $C_d = 1.5$ is the drag coefficient and v_b is the blade tip speed ($v_b \simeq v_2$).

The optimized design of the HPW is the Hydrostatic Pressure Machine (HPM), intentionally designed to behave like a weir, and with blades conceived to undergo the minimum drag while rotating in water. Standard HPM dimensions are with external diameter (maximum diameter) three times larger than the downstream water depth, while the central hub diameter is as large as the downstream water depth (Fig. 6). This means that the hub has a diameter equal to the head difference, and blades with a depth similar to the downstream depth. The upstream water depth ranges up to the hub top level. The blades are mounted diagonally to reduce the drag (see Fig. 3d). The HPM has a smaller diameter than common HPW, and it is more compact (Senior et al., 2010). If leakages and turbulent losses are considered, the power output can be estimated by Eq. (4) (Linton, 2013):

$$P = \eta_{th} P_{in} - \gamma Q_l \Delta H - K v_m^3 \tag{4}$$

where the power input is $P_{in} = \rho g Q(d_1 - d_2)$ and η_{th} is the efficiency estimated by Eq. (3), but corrected in order to consider that the wheel diameter is not infinitely long. Leakages can be estimated as $Q_l = Q(1 - dh/\Delta H)$, with dh the kinetic pressure head (Senior et al., 2010). K is the turbulent loss factor, while v_m is the average blade tangential speed. K is an empirical coefficient that includes all the power losses occurring in a HPM (excluding leakages). K can be expressed as $K = 1/2 f \rho b l$, where b is the blade width, l the blade linear length and $f = 2.5$ (Senior, 2009).

In Linton (2013), a theory based on continuity and momentum equations was developed, where each power loss (indirectly included in the coefficient K in Eq. (4)) was estimated. By this theoretical model, efficiency could be estimated with a discrepancy of 5% from experimental data.

Efficiency assessment and performance improvement

In literature, some experiments were carried out with the aim of determining and improving the power output of stream wheels in shallow water (Gotoh, Kowata, Okuyama, & Katayama, 2001; Paudel, Linton, Zanke, & Saenger, 2013; Senior et al., 2010).

In Senior et al. (2010), an HPW with 1.8 m diameter has been physically tested, and found to have maximum efficiencies between $\eta = 0.8$ and $\eta = 0.9$ for $0.6 \le d_1/d_2 \le 0.9$. The HPW performed better with a curved bed section below it, and the optimal efficiency occurred at $d_1/d_2 = 0.8$. The optimal tip speed was identified in $v_2 = 0.20\sqrt{2g\Delta H}$. In Sule, Wardana, Soenoko, and Wahyudi (2013, 2014), the effect of the blades number n and their shape has been investigated for a HPW with B.R. almost 1. In Sule et al. (2014) the tested stream wheel had a maximum diameter of 0.60 m, with curved blades 0.20 m wide and 0.15 m long. The power increased passing from 6 to 10 blades. In Sule et al. (2013) the power of a straight blade stream wheel of 0.6 m in diameter increased from 4 to 8 paddles, with power coefficient between 0.30 and 0.40 (stream velocities between 0.15 to 0.6 m/s). The performance depended thus on the blade design, but a general rule can be anyway drawn for estimating the optimal number of blades; this aspect will be discussed in Discussion and design suggestions in shallow flow.

Instead, the first insight into the operating principle of the HPM was conducted in Gotoh et al. (2001), where it was shown that a water wheel (0.5 m in diameter, 0.23 m in width, and with twelve blades) set closely along the channel performed like a weir. The hydrostatic force, due to the difference of water level between upstream and downstream, acted in addition to the flow kinetic energy, increasing the power output.

The HPM was investigated deeper in Butera, Fontan, Poggi, Quaranta, and Revelli (n.d.); Linton (2013); Senior et al. (2010); Senior, Wiemann, and Müller (2008), finding a maximum efficiency of $\eta = 0.65$ at the optimal tangential speed $v_2 = (0.25-0.3)\sqrt{2g\Delta H}$ (result post-processed in this review). Furthermore, in Paudel et al. (2013) one HPW was investigated with blades tip made of flexible rubber, and the results can be extended to HPM. Different channel widths were also tested. The water wheel was 0.15 m in diameter and 0.25 m wide, with 12 blades, 0.10 m long and 0.25 m wide. The use of flexible rubber on the blade edges improved sediment transport and reduced damage to fish, as well as increased the efficiency of the water wheel thanks to the decrease of leakages. Significant improvements were observed by reducing the channel width, with maximum efficiency between 0.50 and 0.70. Moreover, the divergent downstream channel created supercritical flow immediately downstream of the water wheel; this resulted in an increased net head acting on the wheel (see also Efficiency assessment and performance improvement). The number of blades was tested in Linton (2013), finding an efficiency improvement passing from $n = 12$ to $n = 6$.

Stream wheels in shallow supercritical flow

Theoretical considerations

In supercritical flow, the velocity of the flow is higher than in subcritical flow. Theoretically speaking, the upstream flow is not affected by the blade. The flow velocity becomes the same of the blade velocity after the impact on the blade, and, since supercritical flows are affected by the upstream conditions, also just downstream of the blade the flow velocity is supposed to be the same of the blade tangential speed. The energy exchange depends especially on the high kinetic energy of the flow upstream, so that this wheel can be called kinetic wheel. The higher downstream water depth could generate an opposing hydrostatic force. The water depth downstream of the wheel assumes the critical depth for a ratio $v_2/v_1 = 0.56$, where v_2 is the wheel speed and v_1 is the velocity of water upstream in the undisturbed configuration (with water depth d_1).

The few available theories are dated back to the XVIII century, when in 1704 Antoine Parent published his theory on jets with high velocity. He calculated the efficiency of stream wheels assuming that the force F exerted by the water flow on the blades was proportional to the square of the relative velocity between the blade and the water flow. Therefore, $F \propto (v_1 - v_2)^2$, where v_1 and v_2 are the flow and blade velocity, respectively. With this assumption, the maximum power coefficient should have been $C_p = 8/27$ for $v_2/v_1 = 1/3$, but Parent limited the hydraulic efficiency of stream water wheels to just 4/27 (Capecchi, 2013). In 1767, de Borda published his theory and corrected Parent analysis, assuming $F \propto v_1(v_1 - v_2)$, and not $F \propto (v_1 - v_2)^2$ to calculate the force on the paddles. The maximum power coefficient became $C_p = 1/2$, when $v_2/v_1 = 1/2$. De Borda had good reasons to accept his theory, as it was in good agreement with experience (Capecchi, 2013). John Smeaton published then experimental data demonstrating a maximum power coefficient of $C_p = 1/3$, higher than that provided by Parent ($C_p = 4/27$) but lower than that provided by de Borda ($C_p = 1/2$) (Capecchi, 2013; Smeaton, 1759). However, a more complex theory, as well as a better understanding of the hydraulic behavior, is required for an adequate assessment of this type of energy converter (Müller et al., 2007).

Efficiency assessment and performance improvement

Supercritical flows exploitable by water wheels are difficult to be found, hence fewer data are available in literature than those for subcritical flow.

One basis work was that developed in Bagdhadi and Mikhail (1985), where a water wheel close to the river bed was investigated. Water was forced through the narrow opening of the upstream sluice gate (ensuring the supercritical flow). The paper estimated that efficiencies of up to $\eta = 0.6$ were possible with this design.

Physical model tests on a water wheel with flat blades conducted in Germany (flow rate of 8 m^3/s, water depth of 0.47 m and flow velocity of 5 m/s) indicated that a 22.5° forward inclination of the blades (with respect to the radial direction) gives the maximum power output (Müller et al., 2007). The results showed that a mechanical power of 38–40 kW could be produced, i.e. power coefficient of $C_p = 0.4$. The case of an existing wheel built in 1892 was also described: this wheel was used to power a paper mill in Switzerland, with power output of 26–33 kW.

In a recent work, numerical simulations have been performed to investigate different blade profiles (changing blade curvature in the horizontal plane). The power output achieved using a semicircular shape was almost twice than that reached with a flat blade, and higher with respect to the power output achieved by using a V-shaped blade. This was due to the lower drag encountered during rotation and to the better exploitation of the flow momentum (Sonaje, Karambelkar, Hinge, & Sathe, 2013).

Discussion and design suggestions in shallow flow

Based on the results achieved in previous works, it is possible to draw some strategies for the design of water wheels in shallow water.

HPW can generate head differences of 0.2–1.0 m. Power outputs ranged between 1.6 and 9.2 kW/m, while nowadays power output can reach 20 kW/m. Instead, HPM can be used with flow rates of 1.5–2.5 m^3/s per meter width, and induced head differences between 1 and 2.5 m. The rotational speed of HPM is higher than that of standard water wheels, thus reducing the demand of high gearboxes for electricity production. Exit flow velocities of HPM and HPW are quite small, in the range of 1–1.5 m/s (Senior et al., 2010). The position inside the canal has to be accurately chosen (Butera et al., n.d.). The approaching flow velocity is nearly negligible. An optimal tangential speed for HPM can be $v_2 = (0.25 - 0.3)\sqrt{2g\Delta H}$, higher than the tangential speed of HPW.

Stream wheels in supercritical flow could produce 10–13 kW per meter width (Müller et al., 2007). However, flow velocities higher than 3 m/s are required to produce appreciable power output. A study aimed at specifically determining the whole characteristics curves of such machines has not been found in literature. Furthermore, based on a personal experience of the author, full scale wheels in supercritical flow exhibit high power losses due to the fast flow and wheel speed (Fig. 3b). The tangential speed v_2 of kinetic wheels is as a function of the approaching flow velocity v_1, and ratios $v_2/v_1 = 0.30–0.55$ are suggested. This is because the momentum exchange between the flow and the blades depends on the relative velocity of the flow with respect to the blade speed. This consideration has been confirmed also in historic books (Bach, 1886; Bresse, 1896; Busquet, 1906; Weisbach, 1849).

Stream wheels in shallow water were generally constructed in the past with straight blades mounted in radial direction, or with a slight forward inclination, in order to reduce the inflow power losses. Based on the results achieved in modern times, it was confirmed that the recommended blade inclination is 22.5° forward, as suggested historically. This result can be generalized by saying that the blade inclination should be parallel to the relative approaching flow velocity, which is the vectorial difference between the approaching flow velocity and the wheel tangential speed. This is also a common approach in the design of gravity water wheels where the flow kinetic energy is significant (Quaranta & Revelli, 2017), in Sagebien undershot wheels to minimize inflow power losses (Quaranta & Müller, 2018-b) and in vertical axis water wheels (Pujol, Vashisht, Ricart, Culubret, & Velayos, 2015). A semicircular shape could lead to an higher power coefficient (Sonaje et al., 2013), but maybe to higher costs of fabrication. Flexible blades could provide better behavior in relation to sediment transport (Paudel et al., 2013). A curved shroud on the channel bed would be useful to reduce gaps and leakages.

Speaking about the number of blades n, three experimental studies were found in literature (Linton, 2013; Sule et al., 2013, 2014) on

(a) (b)

Fig. 7. (a) The floating water wheel with the hydrodynamically shaped floating structure, that can convey water to the wheel with minimum head loss (experimental installation, W.M.J. Batten Batten et al. (2011)); (b) example of floating wheel in deep water, real installation (photo of Salmini Santino Elettromeccanica, Italy)

HPW and HPM. Most studies identified a performance decrease by increasing n. Furthermore, the efficiency of one HPM with six blades was higher than the HPM with twelve blades, due to the lower drag globally undergone by the blades (Linton, 2013). The performance improvement as n decreased was reasonable because in shallow and subcritical water it was the head difference that mainly contributed to the power. Hence a reduction in the blades number determined a drag reduction, so that the lower n the higher the power. But below a minimum blades number $n = n_{\min}$ volumetric losses and leakages increase. Therefore, the drag reduction and leakage increase with n decrease cast a trade-off effect. The minimum value $n_{\min}$, below of that volumetric losses would increas too much, can be calculated considering that when one blade is fully submerged (under the rotational axis), the upstream and downstream ones have to be in contact with the free surface of water (Weisbach, 1849). In this way, when one blade is fully submerged in its optimal position, the other blades do not interfere with it, while reducing volumetric losses during rotation. As a consequence, from geometric considerations applied to such situation, $n_{\min}$ only depends on the ratio $\frac{l}{D}$, where l is the depth of the immersed blade. For example, when $\frac{l}{D} = 0.2$, that is a common ratio suggested in literature (Müller et al., 2010; W. Müller, n.d.), $n_{\min} = 8.1 \rightarrow 8$.

Elaborating literature results, for the HPM with six blades $\left(\frac{n_{\min}}{n}\right)_{opt} = 0.88$, (although twelve blades are generally adopted to avoid too much impulsive and oscillatory forces at the submerged blade), while the ratio ranged between 0.8 to 1.1 for the wheels tested in Sule et al. (2013, 2014). The value $\left(\frac{n_{\min}}{n}\right)_{opt} \simeq 1$ can be thus considered a design guideline. However, the ratio has been only validated at $\frac{l}{D} = 0.33$ for the HPM and at $\frac{l}{D} = 0.25$ for the HPW. CFD simulations or experiments would be hence necessary to validate this guideline at different $\frac{l}{D}$ values.

Stream wheels in deep flow

Deep water flows are those with low B.R., so that the presence of the wheel does not create a backwater propagation. Indeed, at low blockage ratios a portion of flow rate passes also around and below the wheel, depending on the upstream water level. Therefore, as the upstream water depth increases, also the portion of flow passing around the wheel increases, until the equilibrium is reached.

When there are boats that support the wheel on its sides, the wheel is called floating water wheel (Fig. 7) (Müller et al., 2010) . Floating wheels were generally built in large open channel flows, so that wall and blockage effects were negligible.

The large water depths and the low blockage effects make floating wheels very eco-friendly machines, with low environmental impacts with regard to fish and sediments passage. Floating wheels do not need of dams and weirs, so that impacts on fish migration are minimized. However, the main disadvantage of a floating wheel is that to generate the same amount of energy of a stream wheel in shallow water, it has to be bigger. This is caused by the low flow velocities encountered in deep water. Therefore, in recent times, floating wheels have been investigated with the aim of improving their potential. Similar strategies used for stream wheels in shallow flow have been applied, transforming floating wheels into floating HPW, by using hydrodynamically shaped floating structures. In the next sections, both floating wheels and floating HPW will be discussed.

Theoretical considerations

One of the most simplified and oldest formulations to estimate the power coefficient in deep water was that developed by Weisbach, that took into account of the number of submerged blades n_{sub}. He gave the equation $C_p = 16/81 \cdot n_{sub}$, valid for wheels in deep water and when the stream velocity approaches the tangential speed of the wheel (Müller et al., 2010). Considering the Betz limit ($C_p = 16/27$), the maximum theoretical number of submerged blades has to be 3.

In order to achieve a more accurate estimation, the power output can be calculated considering the ideal process depicted in Fig. 5c, that depends on the force exerted on the immersed blade by the momentum exchange and by the small head difference. In this way, the power output can be estimated by Eq. (5) (Müller et al., 2007):

$$P = \rho g \frac{b}{2}\left[(d_1 + \Delta h_1)^2 - (d_1 - \Delta h_2)^2\right] v_2 + \rho b\,(d_1 + \Delta h_1)\,(v_1 - v_2)^2 v_2 \tag{5}$$

where b is the blade width, v_1 and v_2 the upstream and blade velocity and the other variables are depicted in Fig. 5c. The undisturbed water depth is d_1. Experimental tests found $\Delta h_1/\Delta h_2 = 3/2$, that was independent from the flow velocity (Müller et al., 2007). The theoretical power values differed slightly from the experimental measurements, in particular for slow flow velocities (Müller et al., 2007). Eq. (5) supposes the wheel to be large, so that the small head difference at the submerged blade can be maintained.

In Müller et al. (2010), the following simplified model was proposed to estimate the power output:

$$P = \frac{1}{2}\rho C_d A (v_1 - v_2)^2 v_2 \tag{6}$$

where C_d is the drag coefficient that, similarly to a flat plate, could be taken as equal to 2. If Eq. (6) is normalized to the kinetic power of the flow to obtain the power coefficient, the theoretical maximum would be $C_p = 0.296$ at $v_2/v_1 = 0.33$ (Müller et al., 2010). Eq. (6) is practically the same of Eq. (1). It underestimates the maximum power and the optimal rotational speed (see experimental results in Efficiency assessment and performance improvement). This was probably due to the fact that the head difference at the blade is not taken into account in Eq. (6). It is also worthwhile to note that when the area ratio is higher than 5%, the blockage effect of the wheel has to be considered (Bahaj et al., 2007; Müller et al., 2010).

In Yucheng (2012), a power losses theoretical model has been proposed to estimate the power output. The flow angular momentum provided the driving torque to the wheel, while the resistance force was caused by the friction force of water on the back of all submerged paddles. The analytical results have been then validated by comparing them to results obtained from computational fluid dynamic simulations. The theoretical results well predicted the power output when the wheel rotated at rotational speeds lower than the optimal one (wheel tangential speed around one half of the approaching flow velocity), while they were slightly overestimated for rotational speeds bigger than the optimal one.

Efficiency assessment and performance improvement

During the years, experiments have been performed for quantifying the performance of stream wheels in deep water. Blade numbers and shapes have also been investigated.

In Müller et al. (2010), a floating wheel of 0.50 m in diameter, with blades 0.25 m wide and 0.05 m long was tested. The water depth was kept constant at 0.215 m, and flow velocities ranged between 0.2 to 0.59 m/s. The blockage ratio was 0.2. Power output results were corrected, so that presented results referred to the analogous situation in undisturbed flow. The maximum power coefficient was $C_p = 0.42$ for $n = 24$. If friction losses at the shaft and turbulent losses would be null, probably the maximum power coefficient would be $C_p = 0.5$ for $v_2/v_1 = 0.5$, as discovered by de Borda (Capecchi, 2013). Instead, for lower blades numbers ($n = 8$ and 12), the maximum power coefficient was $C_p = 0.25$ and $C_p = 0.35$, respectively. The optimal velocity ratio was $v_2/v_1 = 0.4 - 0.55$. A small increase of C_p appeared possible by using a U-shaped cross section (Müller et al., 2010), like results found in Sonaje et al. (2013) for supercritical flow. Considering the meaning of $n_{\min}$ discussed in Discussion and design suggestions in shallow flow, with such dimensions $n_{\min} = 10.7 \rightarrow 10$, so that $\frac{n_{\min}}{n} = 0.45$ at $\frac{l}{D} = 0.1$.

In Kumara (n.d.) (not peer-reviewed), a floating wheel of 0.60 m in diameter and 0.2 m wide has been tested in a flow of velocity 0.54–0.6 m/s. Straight blades, inclined and curved blades, with different numbers of blades, were investigated (blades were 0.2 m wide and 0.1 m long, i.e. $\frac{l}{D} = 0.16$, but it was not possible to calculate the B.R.). The power output and the efficiency increased from $n = 6$ to $n = 12$ ($\frac{n_{\min}}{n} = 0.96$), and passing from the straight shape to the inclined and curved one, in agreement with results found in Müller et al. (2010). Power coefficients were however less than 0.4.

In Anurat and Chainarong (2011), a water wheel with a diameter of 0.4 m and straight blades has been tested at 0.3 m/s flow velocity. The optimal blade immersion was 0.1 m, i.e. $\frac{l}{D} = 0.25$ and B.R. $= 0.15$, and the power output decreased passing from 6 to 12 blades, thus $\frac{n_{\min}}{n} = 1.6$. The maximum power coefficient was slightly less than $C_p = 0.6$. This value is higher than that predicted by Eq. (1), since the theoretical equation does not consider the blockage effect.

From previous studies it emerged that an accurate blade design was not enough to increase significantly the power output. This fact stimulated studies focused on the floating and supporting structure of the wheel (Fig. 7) (Batten & Müller, n.d.; Batten et al., 2011; Cleynen, Kerikous, Hoerner, & Thevenin, 2017; Hadler & Broekel, 2011; Müller & Batten, 2010; Turnock et al., 2007). The floating structure had a contraction region upstream of the wheel which was designed for the development of an head in front of the turbine. A downstream expansion section was provided so that the flow could exit at a shallower depth and with higher velocity. Additional scoops upstream of the structure enhanced the inlet constriction. Downstream separators were installed to provide a region of low pressure downstream of the model; the water level downstream of the wheel was hence reduced, facilitating the discharging process. A base plate was installed under the wheel, that improved the power output significantly with respect to the configuration without base plate (Batten & Müller, n.d.; Cleynen et al., 2017; Müller & Batten, 2010). The generation of an head difference (with a backwater propagation mostly maintained inside the floating structure) and the base plate made the floating wheel operate like a HPW in shallow water. In Batten et al. (2011), a stability analysis of such floating structure was conducted. Despite the presence of the head difference, Eq. (2) has been used to estimate the power output in this configuration. The maximum power coefficient was found to be $C_p = 0.7$–0.8 (due to the head difference), twice that achievable without such a floating structure. This kind of floating stream wheel was considered as the most promising type of stream wheel both in terms of energy production and number of possible locations (Batten & Müller, n.d.; Müller & Batten, 2010).

An inclined plate under the wheel was investigated by CFD in Akinyemi and Liu (2015), without an hydrodynamically shaped floating structure; the plate improved the wheel power output by 2.7 to about 4 times. Furthermore, the blades inclination was changed from radial to forward. The most effective improvement was achieved by the bottom plate, confirming that the shape of the blade, despite its importance, cannot provide an optimal design without acting on the surrounding structure of the wheel.

Discussion and design suggestions in deep flow

Stream wheels in deep flow exhibit maximum power coefficient $C_p = 0.3$–0.4, generally when the tangential speed is one half of the approaching flow velocity, similarly to stream wheels in supercritical shallow flow. Power output is typically 0.5–2 kW per meter width (Müller et al., 2010). However, in deep flow, the wheel tangential velocity v_2 can be increased up to $v_2 = (0.6 - 0.7)v_1$ (v_1 is the approaching flow velocity) when ad hoc floating structures are used, increasing the power coefficient up to $C_p = 0.7$–0.8. Such a floating

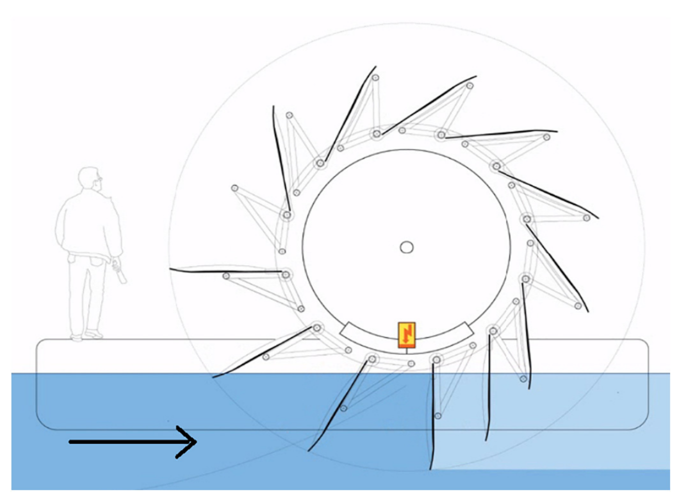

Fig. 8. Floating wheel with adjustable blades inclination, so that inflow and outflow losses are both minimized.
Source: Photo courtesy of Hartmuth Drews.

wheel starts to behave like a HPW and the estimated power output becomes 5 kW/m (Batten et al., 2011).

The number of blades affects the performance of the wheel. Historically, design prescriptions about water wheels in deep water suggested 10–12 blades and slightly inclined forward (about 10°–20° Weisbach, 1849). Bach (1886) and Busquet (1906) suggested to use 18–24 blades, while Chaudy (1896) suggested 6 to 10 blades. However, historic prescriptions were only empirical ones, while the optimal number of blades depends on the ratio $\frac{l}{D}$, where l is the blade length and D is the wheel diameter. For example, the optimal ratio $\frac{n_{\min}}{n} = 0.45$ was found for $\frac{l}{D} = 0.1$ (Müller et al., 2010), $\frac{n_{\min}}{n} = 1.62$ for $\frac{l}{D} = 0.25$ (Anurat & Chainarong, 2011) and $\frac{n_{\min}}{n} = 0.96$ for $\frac{l}{D} = 0.16$ (Kumara, n.d.), so that a practical rule can be $n_{\min}/n = 7.76\frac{l}{D} - 0.31$, with $R^2 = 0.998$.

The power can be improved acting on the shape of the blades by using curved forward blades, to minimize inflow power losses and to better exploit the approaching flow kinetic energy. When outflow losses are also desired to be minimized, adjustable blades could be used, where the blades root is hinged to the rotor instead of being fixed (Fig. 8). In this way, blades are free to adjust their inclination automatically. After passing under the wheel shaft, blades automatically assume a backward inclination. Therefore, at the outflow they dispose normally to the water surface, minimizing water uplift downstream (Drews Wasserrad, n.d.).

However, although an optimal blade design could improve wheel behavior, the power output of stream wheels in deep water can only be significantly increased acting on the supporting structure of the wheel. The floating structure has to be designed with an hydrodynamic shape, with the aim of carrying water flow to the wheel with minimum head losses. Such a structure allows for the exploitation of the hydrostatic force of water, by increasing pressure upstream of the wheel and decreasing it downstream. A curved bottom shroud would be useful to reduce volumetric losses, but the use of a simpler inclined plate was also found to be effective (Akinyemi & Liu, 2015).

Discussion

General considerations

Stream water wheels can be considered an interesting technology in flowing water, due to their competitive costs and simplicity in construction. Three flow regimes were identified to describe stream water wheels: shallow supercritical flow, shallow subcritical flow and deep flow. Instead, if the wheel geometry is considered in addition to the flow regime, five kinds of stream wheels can be identified: the hydrostatic pressure wheel (HPW) in subcritical shallow flow, the hydrostatic pressure machine (HPM) as evolution of HPW in subcritical shallow flow, the kinetic wheel in supercritical flow, the floating wheel in deep water and the floating HPW in deep water, as evolution of the traditional floating wheel in terms of potential. Such classification was useful to better clarify the operating principle and to better determine preliminary design guidelines, especially on the rotational speed and number of blades.

Table 2 summarizes, for each wheel type, the achievable maximum power output, efficiency and/or power coefficient, the optimal tangential speed v_2 and number of blades n. The HPM and the kinetic wheel are the most productive stream wheels: the HPM is intentionally designed to be driven by an head difference, while the kinetic wheel is used in very fast flows, so that the available power input is significant. The less productive stream wheel is the floating type, because it is used in slow flows. However, the maximum power coefficient of floating wheels is similar to that of kinetic wheels, where high power losses occur due to the fast flow and high turbulence. The power output of floating HPWs is more than twice the power output of the original floating wheel; in this case a small hydraulic head is self generated, so that the hydrostatic force is employed for power production. The power coefficients are higher than those of other hydrokinetic devices, which are typically less than $C_p = 0.35$ (Vermaak et al., 2014). By using Table 2, it is possible to determine the achievable power output of each stream water wheel, depending on the hydraulic conditions.

Design procedure

In this review, data were presented and discussed with the aim of achieving design guidelines of stream water wheels. Theoretical results, experimental/numerical data and equations here presented can be considered an adequate tool for a preliminary estimate of the hydraulic behavior of stream wheels, although the optimal design can only be achieved by ad hoc experiments and CFD simulations. In the following lines, design suggestions are briefly summarized and contextualized inside the design methodology. Refer to Discussion and design suggestions in shallow flow and Discussion and design suggestions for details.

The most important parameter to be chosen for the wheel operation is its tangential speed v_2, that can be chosen from Table 2. In some cases, the optimal tangential speed v_2 is a function of flow velocity v_1, and the ratio v_2/v_1 is similar for kinetic wheels and floating wheels. But, in kinetic wheels, due to the supercritical flow, approaching flow velocities are faster, typically higher than 3 m/s, so that kinetic wheels rotate more than three times faster than floating wheels. Tangential speeds of HPW (and floating HPW) are higher with respect to floating wheels. The optimal tangential velocity of HPM depends on the head difference: in such conditions, the upstream flow velocity is negligible, and the aim of the speed design is to discharge a certain flow rate.

While the width of the water wheel is strictly related to the desired power output, and it is limited by the channel width, the choice of diameter of the water wheel still represents a challenge. The optimal wheel tangential speed $v_2 = ND/2$ can be calculated quite easily from Table 2, so that the product between the wheel rotational speed N and the wheel radius (i.e. the diameter) $D/2$ is known. Therefore, different combinations of wheel speed and diameter are possible. The range of wheel speeds should be chosen in order to obtain reasonable diameters for engineering applications, like between 1 m and 5 m. The higher the diameter, the higher the costs, but the lower the outflow power losses downstream, due to the more favorable blades inclination with respect to the free surface of water. The final solution will depend on the best compromise. At the moment there is not enough supporting literature on this interaction, that should be investigated in the future.

Instead, the preliminary value of n can be determined using the equations reported in Table 2 and obtained by literature results interpolation. However, only few $\frac{l}{D}$ ratios have been investigated in literature, and further experiments or CFD simulations are needed. Speaking about the shape of the blades, a curved blade shape would be useful to reduce drag and increase power output, due to the better exploitation of the flow momentum. But no information has been found on the choice of the blade curvature from an analytical point of view. Finally, literature results highlighted the importance of an ad hoc supporting structure of the wheel and a curved bottom shroud.

Gaps and future works

In this review, the performance of each wheel kind was discussed distinguishing between power coefficient and efficiency. More light was also shown on stream wheels hydraulic behavior, and general guidelines for their design were achieved. Therefore, the gaps presented in Stream wheels: types and scope of the work have been addressed, so that results here presented will be useful to support the engineering design of stream wheels. Nevertheless, further gaps

Table 2
Summary of hydraulic characteristics of stream wheels for practical applications. For each wheel type the maximum achievable performance is reported, expressed by the normalized power output, i.e. efficiency η or power coefficient C_p, depending on the wheel kind. Additional information: range of producible power output per metre width, optimal speed v_2 and number of blades n (v_2 is typically related to the approaching flow velocity v_1 or to the head difference ΔH). D is wheel diameter, P.T.O. is power take off system. Data of this table were obtained by experimental tests.

Wheel type	Flow regime	Max. η, C_p	kW/m	Speed	$\frac{n_{min}}{n}$	Fig.	Section	Future works
HPW	Sub. shallow	$\eta = 0.8–0.9$, $C_p = 0.4$	< 20	$\frac{v_2}{\sqrt{2g\Delta H}} = 0.20$	1	3a	Efficiency assessment and performance improvement	D, n, P.T.O.
HPM	Sub. shallow	$\eta = 0.60–0.65$	> 10	$\frac{v_2}{\sqrt{2g\Delta H}} = (0.25–0.3)$	1	3d	Efficiency assessment and performance improvement	Geometrical ratios, blades shape, P.T.O.
Kinetic wheel	Sup. shallow	$\eta = 0.4$, $C_p = 0.4$	10–13	$\frac{v_2}{v_1} = 0.3–0.55$	–	3b	Efficiency assessment and performance improvement	C_p, whole design
Floating wheel	Deep flow	$C_p = 0.4$	0.5–2	$\frac{v_2}{v_1} = 0.4–0.55$	$7.76\frac{l}{D} - 0.31$	3c	Efficiency assessment and performance improvement	D, n, P.T.O.
Floating HPW	Deep flow	$C_p = 0.7–0.8$	5	$\frac{v_2}{v_1} = 0.6–0.8$	1	7	Efficiency assessment and performance improvement	D, n, P.T.O.

were identified. These gaps need to be addressed in the future, and they are discussed in the next lines.

1) *Wheel diameter*. From a theoretical point of view (especially for wheels in deep water), efficiency and power coefficient depend on the tangential wheel speed $v_2 = ND/2$, hence on the product between N and D, independently from the proper value of rotational speed N and diameter D. But, from a practical point of view, this may not be true, because the diameter affects the inclination of the blades at the outflow, if they are fixed to the wheel. Research should be carried out in this sense.
2) *Blade design*. The longer the blade, the higher the exploitation of the flow kinetic energy, but also the higher the resistive drag. The blade length $l = 0.2D$ suggested from literature is empirical and dated back to one century ago. Furthermore, the blade length l affects the optimal number of blades, because here it was found that $\left(\frac{n_{min}}{n}\right)_{opt} = f\left(\frac{l}{D}\right)$. As a consequence, since the higher l the higher the blockage ratio, it can be deduced that n depends on the blockage ratio B.R. When B.R. is maximum (like for the HPM), the optimal normalized number of blades is $\frac{n_{min}}{n} = 1$. However, very few B.R. values have been investigated, and further studies are necessary to identify B.R. effects both on the wheel behavior and to achieve more accurate engineering tools for the design of the number of blades. The contribution of flow momentum and head difference to the power output at different B.R. must be made more clear. Instead, for what concerns with the HPM, different dimension ratios should be investigated, like width and blade length as a function of the shaft diameter. Further studies would be worthwhile to achieve better design rules on the shape of the blades, for example relating the blade curvature to the flow velocity. The shape of the HPM blades still needs to be optimized, by choosing an ad hoc curvature that allows to reduce the drag. As a supporting reference, the effects of the number of blades have been tested in Quaranta and Revelli (2016a) for a breastshot water wheel.

3a) *Power take off*. One other important issue highlighted in this study, that is worthwhile to be further investigated, is the power take off system. Table 2 shows that nearly all optimal speeds are proportional to the approach velocity which means that generator speed must change with the flow velocity. The changing in flow velocity is generally due to the flow rate variation. It can be a major challenge to make induction generators work efficiently over a significant range of speed and power (Fergnani et al., 2016). Even permanent magnet generators can find this difficult, because they require reasonably sophisticated power electronics to achieve this variation (Dietz et al., 2011). One solution to overcome the need of a variable speed of operation is to use adjustable inflow structures. For example, sluice gates and inflow weirs are commonly installed upstream of gravity water wheels to keep constant the flow velocity or the water depth in variable flow rate situations (Quaranta & Müller, 2018-a; Quaranta & Revelli, 2016b).

3b) Furthermore, the rotational speed of stream wheels is generally low, so that gearboxes are needed to match the generator frequency. Preliminary works have been conducted at Southampton University to overcome this deficit (Quaranta, Müller, Butera, Capecchi, & Franco, 2018). The central shaft was removed and the wheel was supported on two rollers in contact with the wheel periphery. Therefore the rollers had the same tangential speed of the wheel, but being their diameter smaller than the wheel diameter, their rotational speed was higher, reducing the multiplication ratio of the gearbox.

4) *Blockage effect*. Stream wheels in deep water have been tested in the following two configurations: floating wheels laterally confined by channel walls, and floating HPW. It would be interesting to test floating wheels with the aim of understanding the minimum channel width to avoid wall effects on the wheel performance (see Butera et al. (2018) for a similar work on HPM). Furthermore, an all-inclusive equation for the estimation of the power output should be achieved. In this equation, the contribution of head difference and flow momentum should be added together by means of blending functions, where the blending functions are a function of the blockage ratio.

Conclusions

Stream wheels are micro hydropower converters installed in flowing streams. Due to their simple construction, they were the first kind of water wheel to be used. Nowadays, thanks to the renewable energy targets set in worldwide legislations, and the need of providing energy also in remote localities, micro-hydropower is becoming very attractive, and stream wheels can constitute an interesting technology in this context. Stream wheels are very cheap, making them a suitable option especially in emerging countries.

Theoretical, experimental, numerical results and literature data were here collected and discussed. The literature results show that the efficiency and hydraulic behavior of stream wheels is affected by the wheel geometry and hydraulic conditions. The maximum power output occurs at a certain wheel speed, that is a function of the hydraulic conditions.

Successful studies to improve the power output have been performed. The most efficient achievement is the awareness that the optimal design can be obtained not acting only on the wheel rotational speed and blade design, but especially on the surrounding structure of the wheel. This consideration is valid for all kinds of stream water wheels. In shallow water, curved bed sections are required to minimize gaps and leakages, while, in deep water, ad hoc floating structure hydrodynamically shaped can double the power output.

Anyway, also the shape and number of blades play a significant role in the achievement of the optimal efficiency. For example, at least three blades should be simultaneously in contact with water; furthermore, a forward flat blade, or a semicircular shape, could improve the efficiency noticeably. This was a common result found in several works.

Further research is needed to better understand the performance of stream wheel in shallow supercriticial flow. Additional research is required to solve some gaps in the engineering design of stream wheels, like those concerned with the number of blades, their geometric dimensions and wheel diameter. The electro-mechanics equipment is an important aspects to be investigated. This review defines the state of the art of stream water wheels, and guidelines were presented to achieve a good preliminary design of such hydropower converters.

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