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Cross-Flow turbine design for variable operating conditions

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Abstract

The potential energy hidden in water resources is becoming more and more a significant economic value. The value of the hydroelectric energy is often magnified by the proximity of the turbine to pumps or other energy sinks owned by the same water manager. Cross-flow or Banki-Michel turbines are a very efficient and economic choice that allows a very good cost/benefit ratio for energy production located at the end of conduits carrying water from a water source to a tank. In the paper the optimum design of a cross-flow turbine is sought after, assuming a flow rate variable in time.

Regulation of the discharge entering in the turbine is a key issue, which is faced adopting a shaped semicircular segment, moved inside the main case around the rotating impeller. The maximum efficiency of the turbine is attained by setting the velocity of the particles entering the impeller at about twice the velocity of the rotating system at the impeller inlet. If energy losses along the pipe are negligible, closing and opening the inlet surface with the semicircular segment allows always a constant hydraulic head and a constant velocity at the impeller inlet, even with different flow rate entering values. Observed reduction of the turbine efficiency along with the inlet surface reduction is first investigated; a design methodology, using also CFD simulations, is then proposed; finally, the same methodology is applied to a real site in Sicily, selected in the context of the HYDROENERGY P.O. - F.E.S.R. European project.

Keywords: Turbines; Cross-flow; Energy saving.

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1. Introduction

Hydropower has been for more than a century the major source of energy and is, still now, the major source of renewable energy worldwide. The number of new large hydro-plants per year is at present undergoing a strong reduction, due to an increased social sensitivity to the river ecological and transport equilibrium, which leaves a very limited number of sites available for new large hydro-plant construction.

On the other hand, the on-going transformation of the centralized system of energy production and transportation into a more flexible distributed system (smart grids) has given a strong input to the construction of micro and pico hydro-energy production devices (from few to 1000 kW). This type of turbine can be easily installed: 1) along small rivers, where it is possible to transform the potential hydraulic energy dissipated along a short reach between two river sections in electricity, without diverting the minimum flow rate required by the reach ecological equilibrium, 2) at the end of a water pipe delivering the water from a main source (spring, water well, natural or artificial basin) to a tank serving a city water district, 3) at the end of a sewer pipe delivering the treated waste water to its final receiving water body.

We define total efficiency the ratio between the output power of the turbine and the mechanical energy of the mass passing per unit time trough the upper inlet of the hydroelectric plant, measured with respect to the level of the turbine axis. This inlet is a river section in case 1) and a pipe section in cases 2) and 3). The total efficiency η_t can be estimated as:

$$\eta_t = \frac{P_{out}}{\gamma \cdot Q \cdot H_t} \quad \text{(1a),}$$

where P_{out} is the output power, Q is the discharge and H_t is the topographic jump between the inlet location and the turbine axis. Unless the source is an artificial basin and discharge regulation is carried out from the same hydraulic plant where the turbine is located, the discharge delivered to the turbine changes according to the river level or to the spring natural flow (Fig.1).

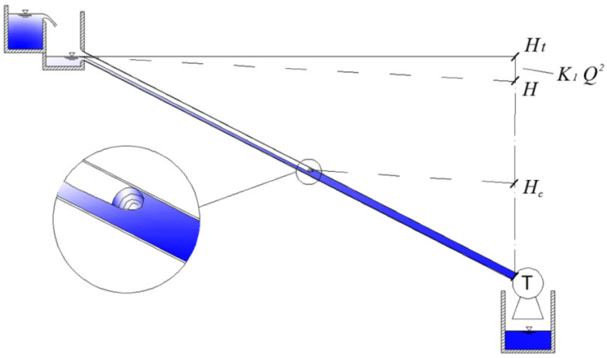

Fig.1. Turbine installed at an aqueduct end

If no dissipation valves are installed along the pipe, the pipe section will be only partially filled up by the water for some extension after the inlet. In this part of the pipe a free surface flow will occur, with abrupt transformation in pressurized flow at the intersection between the pipe and the energy line. The hydraulic head H_c immediately before the turbine depends on its characteristic curve (Fig. 2a), unless a specific discharge regulation system is installed inside the turbine.

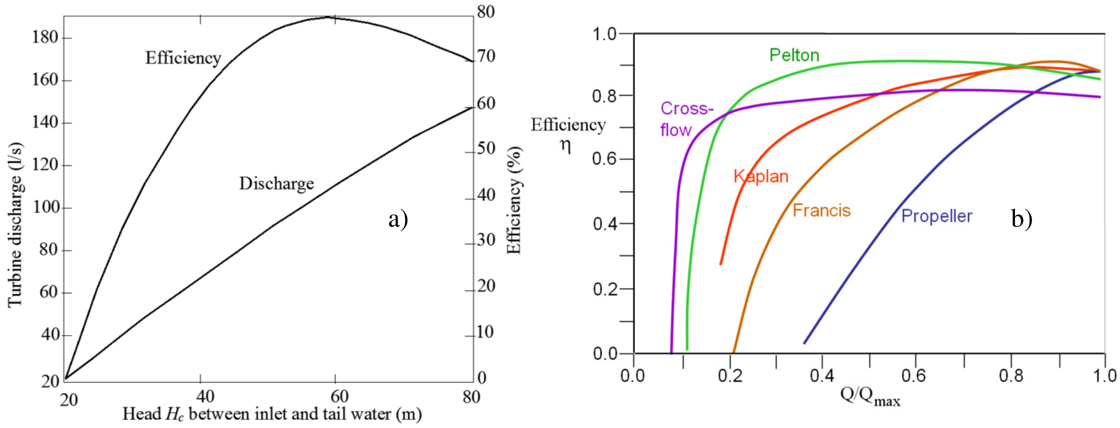

Fig. 2. a) Characteristic and efficiency curves for a generic turbine. b) Efficiencies versus discharges for Kaplan, Pelton, Francis, Cross flow and Propeller turbine

The efficiency of the turbine is defined as:

$$\eta = \frac{P_{out}}{\gamma Q H_c} \tag{1b}$$

H_c can be much smaller than both H and H_t, where H is the hydraulic head resulting at the aqueduct end in the case of pressurized flow along the entire pipe. H is related to H_t by the following curve of pipe energy losses:

$$H_t - H = K_1 Q^2 \tag{2}$$

where K_1 depends on the pipe size and material. In many cases we can neglect energy losses along the pipe and set $H = H_t$.

The output power of the turbine is given by:

$$P_{out} = \eta(Q) \cdot \gamma \cdot Q \cdot H_c(Q) \tag{3}$$

The turbine efficiency η and the hydraulic head H_c can be much lower than respectively η_{max} and H_t far from the optimal discharge value (where η_{max} is the maximum efficiency). Moreover, if we want to avoid any free flow condition inside the pipe, the mechanical energy corresponding to the head difference H-H_c must be dissipated by an extra valve that has to be installed along the pipe.

All this inconvenient can be avoided by the use of a discharge regulator inside the turbine. An optimum regulator should allow the turbine characteristic curve and the curve of the pipe energy losses, to match each other and should guarantee an efficiency η always close to the η_{max} value. If pipe energy losses are negligible, this is equivalent to keep a constant $H_c = H_t$ hydraulic head for all the possible discharge values.

2. Existing turbine regulation systems

Documentation provided by the companies that construct turbines usually includes the characteristic curve (for one or many possible regulation conditions) and the efficiency curve, defined as in Eq. (1b). Not many information are available about the attainable total efficiency, as defined in Eq. (1a). A good efficiency (as defined in Eq. 1b) under variable operating conditions can be attained with two different approaches: hydraulic and electrical regulation.

The hydraulic regulation, often more flexible and efficient than the electrical one (Carravetta et al., 2013), depends on the type of turbine: needle stroke for Pelton turbines, adjustable guide vanes for Francis turbine, fixed or adjustable guide vanes or adjustable runner blades for Diagonal or Kaplan turbine (Panish, 2002; Singh, 2009). Pelton turbines can have multiple needles, which can be set in on/off position according to the available discharge. The reverse pumps have no regulation system, so in order to obtain a flexibility of the turbine related to the variability of discharge and/or head drop, hydraulic regulation has to be done with a series/parallel hydraulic circuit (Carravetta and Giugni, 2006; Carravetta et al., 2013). This circuit consists of a by-pass conduit and of a pressure reducing valve in series with the turbine.

Recent power electronic devices allow regulating electrical voltage and frequency in order to vary the generator speed (Joshi et al., 2005; Ramuz et al., 2005). When turbines are coupled with asynchronous generators, like it happens in most of the hydropower plants, the electronic regulation allows an almost constant ratio between the inlet particle velocity and the impeller rotational speed, for different discharge values. This leads also to a constant efficiency, because efficiency is mainly related to the above mentioned velocity ratio, but has little effect on the turbine characteristic curve and H_c can be much smaller than the total head H_t.

The cost of the regulation system is of course a central issue. Adjustable runner blades or guide vanes have a cost that is proportional to their number, which usually is very large (30-40), multiple needle strokes for Pelton turbines require also a continuous switching on/off the single strokes and some of them can remain unused most of the time. Electronic devises are still very expensive. See in Fig. 2b the efficiency curves of several types of turbines.

2.1. The traditional Ossberger cross-flow turbine

The Banki-Michell turbine is a simple and economic turbine appropriate for micro-hydropower plants. The peak efficiency of this turbine is somewhat less than a Kaplan, Francis or Pelton turbine, but its relative efficiency is close to one within a large range, especially above the optimum discharge value.

The Banki-Michell turbine has a drum-shaped runner consisting of two parallel discs connected together near their rims by a series of curved blades (see Fig. 3a). The turbine has an horizontal rotational shaft, unlike Pelton and Turgo turbines, which can have either horizontal or vertical shaft orientation. The water flow enters through the cylinder defined by the two disk circumferences (also called impeller inlet) and it crosses twice the channels confined by each blade couple. After entering the impeller through a channel, the particle leaves it through another one. Going through the impeller twice provides additional efficiency. When the water leaves the runner, it also helps to clean the runner of small debris and pollution. So the cross-flow turbines get cleaned as the water leaves the runner (small sand particles, grass, leaves, etc. get washed away), preventing losses. Other turbine types get clogged easily, and consequently face power losses despite higher nominal efficiencies. The edges of the blades are sharpened to reduce resistance to the flow of water and the blades are welded to the disks.

A design methodology for the standard Banki-Michel turbine has been recently proposed by the authors (Sammartano et al., 2013). The most important parameter for the turbine efficiency is the angle α between the particles trajectories along the circumference of the impeller inlet and the tangent to the same circumference. The optimal efficiency is obtained when the shape of the nozzle allows a constant velocity norm V along the impeller inlet and a very small α value. The lower limit for α is given by the need to limit the impeller width and by the additional resistance caused by the consequent small channel cross-section area existing between each couple of blades. The optimal efficiency is obtained for a ratio between the inlet particle velocity and the machine rotating velocity equal to approximately two. The efficiency drops slowly for larger ratios and more quickly for lower ratios, up to a theoretical zero value for a ratio equal to one.

The Banki-Michell turbine is often called cross-flow, due to special geometry of its impeller, or also Ossberger, from the name of the industry that carries on its production since 1933. The efficiency of the traditional Ossberger cross-flow turbine is well documented in the case of constant discharge. See for example the historical overview of cross flow turbine described in Khosrowpanah et al. paper (1984).

In the Ossberger cross-flow turbine the hydraulic head can be regulated by using a hydraulic flap that is easily installed into the nozzle upstream the impeller (Fig. 3a). In this system the flap is rotated around its axis according to the actual discharge, as it is simply shown in the same Fig. 3a, with the aim to reduce the inlet cross-section and to keep an inlet velocity norm close to the optimal value. On the other hand it is easy to recognize that the flap leads to: 1) local particle deceleration inside the nozzle downstream the flap, and corresponding energy losses, 2) irregular velocity distribution along the impeller inlet and corresponding departure from the optimal norm and direction values. The result is a relatively poor efficiency curve, especially for partially closed positions of the hydraulic flap.

Experimental and numerical studies (Costa Pereira and Borges, 1996; Kokubu et al., 2011) show that the use of hydraulic flap inside the nozzle could increase the efficiency of the turbine for given discharge, but this device cannot be used to guarantee a constant H at the turbine inlet. Other researchers tried to improve the efficiency of the cross-flow by inserting some kind of guide inside the impeller (Kokubu et al., 2011; Haurissa et al., 2012) or adding a draft tube to redesign the shape of the guide vanes (Kaniecki, 2002).

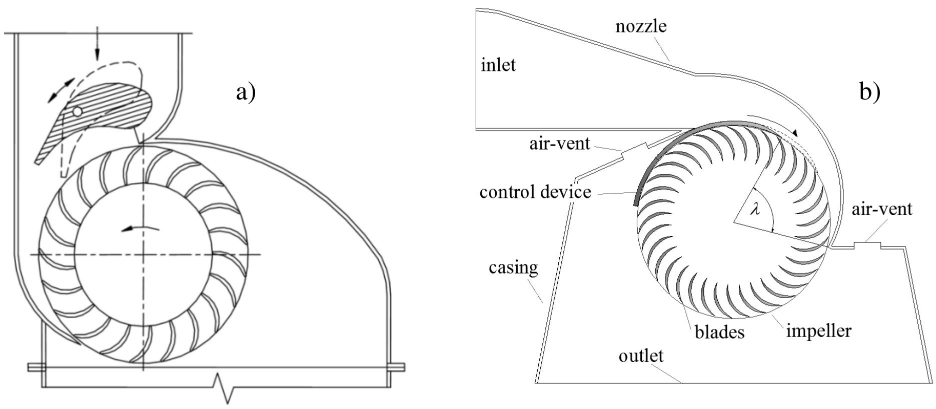

Fig. 3. a) Hydraulic Flap in an Ossberger cross-flow turbine. b) Section of the cross-flow with the Cink control device

The Czech engineer Miroslav Cink introduced a further development of the cross-flow radial turbine. This cross-flow turbine has a circular profiled segment for inflow regulation (see Fig. 3b). The main difference between the Cink and the original Ossberger systems is that in the first one the particles immediately enter the impeller after the restriction given by the circular segment, without any preliminary deceleration, and exchange their energy with the impeller. There are no particular information about the fluid-dynamic efficiency of the turbine, based on experiments or CFD analysis, but the Cink regulation system seems the most promising one for the goal of achieving high total efficiency values, because it provides a variable reduction of the impeller inlet surface, without any additional dissipation inside the nozzle. Moreover, it will be shown in the next sections that its design fits quite well with the cross-flow design recently proposed by the authors (Sammartano et al., 2013).

3. Energy transfer optimization

The relationship between the hydraulic head immediately before the turbine and the norm V of the inlet velocity is given by:

$$V = c_{in}\sqrt{2gH_c} \quad (4),$$

where c_{in} is a coefficient that would be exactly one if the pressure along the inlet circumference were exactly zero, along with the energy losses inside the nozzle.

The actual c_{in} can be estimated “a posteriori” by running simulations where the hydraulic head H_c is given as upstream boundary condition. Preliminary numerical tests suggest that its value changes between 0.85 and 0.7, according to the actual size of the turbine, but it remains almost constant by changing the impeller inlet closure. This implies that, assuming a constant value for different segment rotation, both velocity norm V and head H

remain constant when the inlet surface area is reduced proportionally to the actual discharge by rotating the circular segment. Using a simple pressure control at the end of the pipe, immediately before the turbine, it is possible to regulate the closure of the inlet area in order to keep constant the H_c value, as well as the optimal particle velocity (Sammartano et al., 2013).

In the above mentioned paper authors have provided simple criteria for the design of a cross-flow turbine without discharge regulation. If we include the regulation system in a cross flow turbine designed only for the maximum discharge according to these criteria, and carry on CFD simulations with different impeller inlet closures, we observe that the efficiency drops along with the closure extension. To better understand the reason of such reduction, we need to remind that the energy exchanged between the water particles and each channel of the rotating impeller along a time interval Δt is given by the Euler's equation only if all the particle streamlines enter and leave the channel along Δt. In this case the Euler's equation is written as:

$$E_{\Delta t}^{c} = \rho \cdot Q_c \cdot \left(\underline{V_1} \cdot \underline{U_1} - \underline{V_2} \cdot \underline{U_2} \right) \cdot \Delta t \quad (5),$$

where ρ is the water density, Q_c is the discharge entering and leaving the single channel, $\underline{V}_1$ and $\underline{V}_2$ are the velocities of the particles respectively entering and leaving the channel, $\underline{U}_1$ and $\underline{U}_2$ are the corresponding velocities of the rotating reference system. The energy exchanged inside a single channel along Δt is different from the l.h.s. of Eq. (5) if, along that time, the particle streamlines only partially cover the channel extension, which is the case of channels crossing the beginning or the end of the impeller inlet along Δt.

For a given blade channel and within a fixed Δt, three cases can occur: a) within Δt time fluid particles are entering along all the inlet channel section and are leaving along all the outlet channel section (which is the case of fully developed exchange, when Eq. (5) holds); b) within Δt time no particles are entering along all the inlet channel section or are leaving along all the outlet channel section (no entering or leaving flux and zero exchanged energy); c) at least for a fraction of Δt particles are entering/leaving in/out the blade channel along a part of the channel inlet/outlet section. This case will hold close to the inlet initial and final points (see Fig. 4). To compute the energy exchange in the case c) we can divide the total flow inside the channel in an infinite number of stream tubes with infinitesimal section, each one with its specific discharge q per unit length and initial and final sections located inside the channel (see Fig. 4), but changing along the time. The energy exchanged along time Δt in case c), according to this simplified scheme, is equal to:

$$E_{\Delta t}^{c} = \iint_{\Delta t, \Delta L} \rho \cdot q \cdot \left(\underline{V_i} \cdot \underline{U_i} - \underline{V_f} \cdot \underline{U_f} \right) \cdot dsdt \quad (6),$$

where ΔL is the blade distance measured along the inlet arc, $\underline{V}_i$ and $\underline{U}_i$ are respectively the particle and the reference system velocity at the first section of the stream tube, $\underline{V}_f$ and $\underline{U}_f$ are respectively the particle and the reference system velocity at the final section of the stream tube. $\underline{V}_i$ and $\underline{V}_f$ are equal to $\underline{V}_1$ and $\underline{V}_2$ only if the first or the final stream sections are respectively on the inlet or the outlet channel section.

We investigate step 1 of the energy transfer, when the flow enters inside the impeller and most of its energy is exchanged with the same impeller. Observe that, if the last particle of the stream tube is located before the channel outlet section (Fig. 4a), the second scalar product in Eq. (6) is greater than the corresponding scalar product in Eq. (5), because of the smaller angle β_f between $\underline{V}_f$ and $\underline{U}_f$. Also, if the first particle of the streamline is located after the channel inlet section (Fig. 4b), the first scalar product in Eq. (6), for given particle and reference system velocity norm, is smaller than the corresponding scalar product in Eq. (5), because of the larger angle β_i between $\underline{V}_i$ and $\underline{U}_i$. The integral in Eq. (6) will be, for these reasons, always smaller than the result obtained in case a) and given by the r.h.s. of Eq. (5).

The efficiency of the overall mechanism will increase, of course, as much as the number of channels falling in case c) is small with respect to the channels falling in case a). To do that, assuming a given rotational speed, it seems important to increase the λ angle and to reduce the channel extension by increasing the D_2/D_1 ratio. The effect of the increment of the D_2/D_1 ratio is, for given number of blades, a smaller curvature of the trajectories most internal to the channel with respect to the trajectories of the particles moving close to the blade surface.

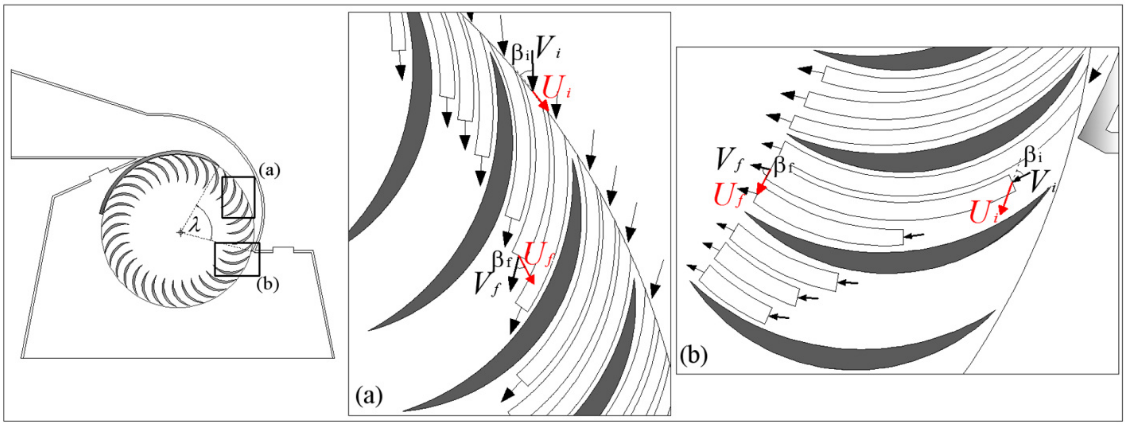

Fig. 4. a) larger angle β_i between $\underline{V_i}$ and $\underline{U_i}$; b) smaller angle β_f between $\underline{V_f}$ and $\underline{U_f}$

To avoid this reduction, it is possible to increase the number of blades N_b (and corresponding channels) and to reduce the thickness of each blade. Because the strains in each blade are proportional to the momentum entering in each channel, the increment of the blade number and the corresponding momentum reduction allows also a reduction of the blade thickness.

4. Turbine design

The turbine designer has to select the best values of the following parameters (see Fig. 5): 1) the α angle between the particle velocity direction and the tangent direction at the impeller inlet, 2) the λ_{max} angle facing the fully opened circular segment, 3) the outer impeller diameter D_1, 4) the inner impeller diameter D_2 and the N_b blades number. The width B of the impeller, the initial β_1 and final β_2 angles between the blade tangent and the reference system velocity directions, as well as the nozzle shape, can be derived according to Sammartano et al. (2013) given the previous parameters. The thickness of the blades can be chosen according to the structural strains computed in preliminary tests. A possible strategy for the choice of the primary parameters 1-4 is:

1) Select α. It has been shown in Sammartano et al. (2013) that the optimum α value, according to Euler's equation, is zero, but this value leads to an infinite B impeller width. Other lower limits for α are given by the corresponding high velocity variability along each channel inlet, as well as constructive difficulties in matching the corresponding angles. A largely accepted compromise is $\alpha = 15°$.

2) According to the observations in section 3, the λ_{max} impeller inlet angle should be as large as possible to get a constant efficiency for increasing closure values. On the other hand, because the extension of the semicircular segment has to be the same of the impeller inlet (to be able to get a complete closure) it is easy to realize (see Fig. 5) that an excessive length would stop the second flow channel crossing, in the case of fully opened segment, with consequent reduction of the produced energy. According to preliminary testing this maximum angle is 120°.

3) Apply the strategy proposed in Sammartano et al. (2013) to compute the D_1 value, which leads to the relationship:

$$V_{opt} = \frac{\omega D_1}{\cos\alpha} \tag{7}$$

where the norm V_{opt} of the velocity at the impeller inlet is selected according to Eq. (4), adopting an initial c_{in} =

0.85 coefficient; the ω angular velocity of the reference system depends on the number of electrical poles of the asynchronous generator and on the electric network frequency.

4) The remaining D_2 and N_b parameters are chosen according to the following iterative computational procedure:

Given a known discharge versus time curve $Q(t)$, along with a constant hydraulic head $H = H_t$ immediately before the turbine, select the maximum Q_{max} and the modal Q_{mod} value. Compute the impeller width B (Fig.5) corresponding to Q_{max} and $\lambda = \lambda_{max}$ according to Sammartano et al. (2013). Select the $\lambda = \lambda_{mod}$ inlet angle corresponding to Q_{mod} and to the V velocity norm given by Eq. (4), that is:

$$\lambda_{mod} = \frac{\lambda_{max} \cdot Q_{mod}}{Q_{max}} \quad (8),$$

Select an initial N_b number and an initial D_2/D_1 ratio. According to the results of section 3, this ratio should be larger for smaller λ_{mod} angles. Compute the corresponding efficiency. For given impeller inlet angle and $Q = Q_{mod}$, optimize N_b and D_2/D_1 with the help of CFD simulations.

Observe that c_{in} coefficient (Eq. 4) has been initially arbitrary set. This choice can be tested by comparing the value adopted in step 3) with the value returning from Eq. 4 using the actually computed velocity. If the test is negative, all the procedure has to be repeated with the value given by Eq. 4, until convergence is attained.

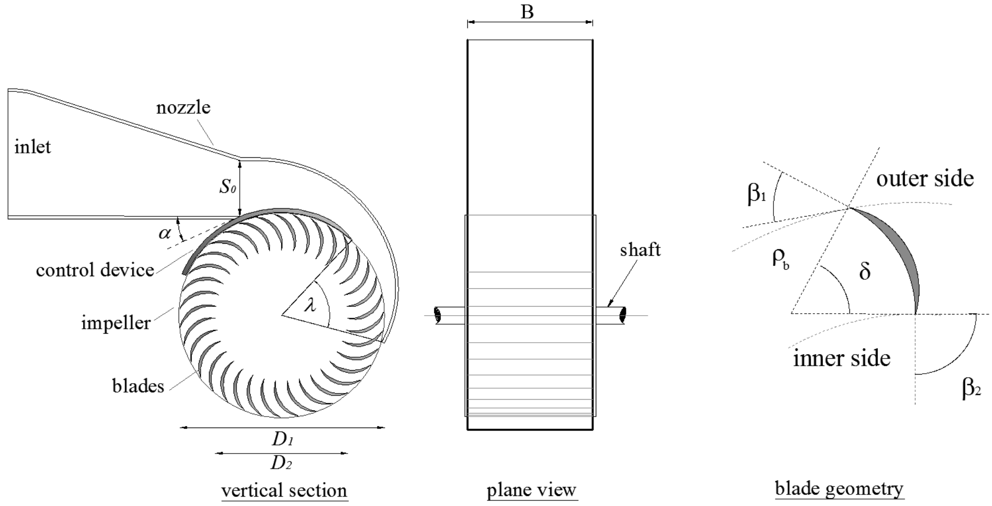

Fig. 5. Vertical section and a plane view of the cross-flow turbine

5. Application to a real site

The following case study investigates the economic benefit of the replacement of a flow control valve with the proposed cross flow turbine in the outlet node of an oversized water pipeline.

The pipeline, 10 km long, supplies fresh water to the drinking water plant of Palermo town (Italy), named “Risalaimi”.

The free surface of the lower tank is located at an altitude of 211.3 m a.m.s.l. and the upper reservoir, named “Cozzo Tondo” is located at an altitude of 244.0 m a.m.s.l. The tanks are connected by a cast-iron pipeline with diameter 900 mm. In the pipe, near the lower tank, is installed a flow control valve.

The upper reservoir is fed by a pumping station composed of four parallel pumps, with a nominal flow rate of 0.2 m^3/s each. Even if the reservoir has some capacity, the water manager wants to maintain an almost constant water level and the entire pumped discharge has to be delivered to the lower tank. In order to do that, the flow in the downstream pipe is controlled by different opening degrees of the flow control valve that have to be automatically selected according to the measured trend of the upstream pressure. Fig. 6 shows the monthly mean flow distribution, measured in the year 2010, which can be assumed also unvaried in the next subsequent years. The peak demand attains in the original discharge hydrograph a maximum value of 0.820 m^3/s.

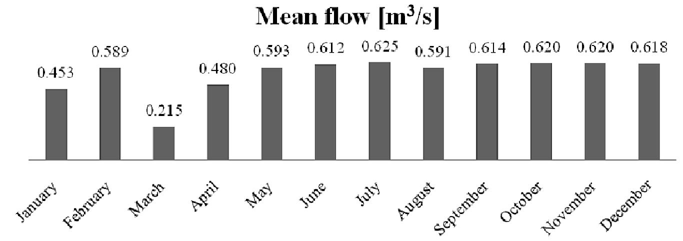

Fig. 6. Monthly mean flows at the year 2010

The procedure described in section 4 has been applied for the turbine design. In order to ensure the water demand even in exceptional conditions, the maximum discharge $Q_{\max}$ selected in step 4) of the design procedure is equal to 0.820 m^3/s. The rotational speed has been set to a fixed value of 360 RPM. In steps 1-3 of the design procedure, an outer diameter D_1 = 385 mm has been computed. The Q_{mod} discharge used to optimize the N_b and D_2/D_1 ratio in step 4) is equal to 0.62 m^3/s; in this condition the control device reduces the impeller inlet section up to a corresponding λ_{mod} angle equal to 90.7° (Eq. 8). The width B of the impeller, computed according to Sammartano et al. (2013) is equal to 530 mm.

The blade and the circular segment thickness have been assumed equal to 5 mm. The optimum design carried out by means of 2D simulations of the CFX-ANSYS code (Sammartano et al., 2013), and outlined in point 4), required three iterations to finally converge to a c_{in} coefficient equal to 0.75.

In this case the proposed procedure led to a D_2/D_1 optimal ratio equal to 0.65 and a number of blades equal to 50. Observe that the optimal ratio D_2/D_1 is consistent with the same value already found by many authors (Aziz and Desai, 1993; Aziz and Totapally, 1994) for the case of a simple cross-flow turbine without discharge regulation, and a fixed angle equal to 90°. Fig. 7 shows the efficiency curve of the designed turbine. The characteristic curve is horizontal ($H_c = H_t$) for all the plotted discharge range, and would increase for larger values.

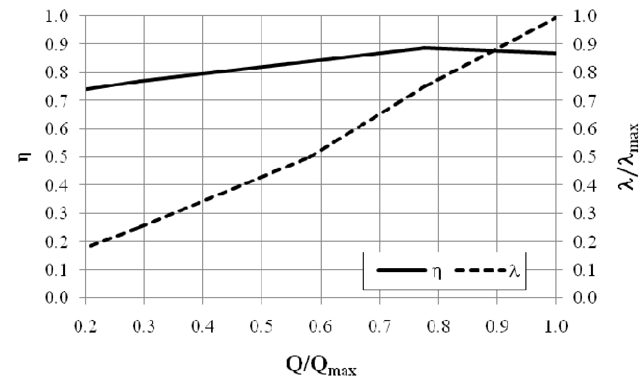

Fig. 7. Efficiency and λ curve of the designed turbine

Observe in Fig. 7 that the efficiency is almost constant despite the large variation of the discharge. The maximum efficiency is equal to 0.89 at 80% of $Q_{\max}$ (that is for $Q = Q_{\mathrm{mod}}$) and the minimum efficiency is equal to 0.73 at 20% of $Q_{\max}$. Assuming an 8 poles asynchronous generator coupled to the turbine, with efficiency equal to 0.93, and a belt drive efficiency equal to 0.95, the average annual electricity production of "Risalaimi" plant amounts to 722 MWh, divided as shown in Fig. 8. Taking into account that the cost of electricity for the water treatment plant is equal to 0.22 €/kWh, the installation of the cross flow turbine would allow an annual benefit of about € 160,000.

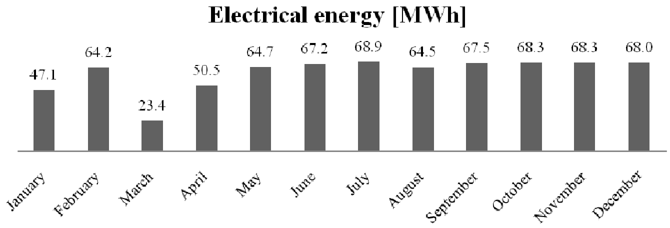

Fig. 8. Averaged monthly energy production at the year 2010

6. Conclusions

The paper has outlined a simple but rigorous procedure for the design of a cross-flow turbine with discharge regulator, which allows a constant upstream hydraulic head, if the head losses from the water source to the turbine are negligible. The construction and the management cost of the regulation system, including automation, are very low (no more that 1 €/W). In the specific test case the discharge variability is significant only for few months a year, but the use of regulation is still convenient, also because it avoids the cost of dissipation.

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