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Coupled Hydraulic And Electronic Regulation For Banki Turbines

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Abstract

The potential benefit of coupling hydraulic and electronic regulation to maximize the energy production of a Bank turbine in hydraulic plants is analyzed and computed with reference to a specific case. Design criteria of the Banki turbine inside hydraulic plants are first summarized, along with the use of hydraulic regulation in the case of constant water head and variable discharge at the end of aqueducts feeding water distribution systems. Optimal turbine impeller rotational speed is derived and traditional, as well as innovative systems for electricity production according to controlled rotational speed of the generator are presented. The study case at the purification plant named Risalaimi, in Italy, is analyzed, and the potential production of energy along the year is computed according to the known monthly average demand and two possible choices: the choice of hydraulic regulation only, called CFT1, and the choice of coupled hydraulic and electric regulations, called CFT2. The Return time of Capital Investment (RCI) is then computed for both the CFT1 and CFT2 cases. The result is that the CFT2 choice provides an increment of the total produced energy, along with an increment of about 30% of the corresponding RCI.

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Keywords: Cross-Flow turbines; Banki-Michell; CFD analysis; distributed generation units; electrical drives

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1. Introduction

The biggest expenses for many water management companies are, after personnel, the pumping energy costs. On the other hand, the potential energy of rainwater at its impact on the soil is much larger than the energy needed for pumping and the energy cost in the company balance sheet could easily change in sign by recovering only part of that energy [1]. At the present time the potential gravitational energy of rain (or snow) water most of the times is totally dissipated in transportation from the catchment location (wells, natural or artificial basins, rivers) to the reservoirs located at the top of the pipe distribution networks. Losses can be split into three parts: continuous dissipations along the conduit, proportional to the pipe length according to a constant J called piezometric gradient; local energy losses Λ' due to abrupt geometry changes along the pipe; and local energy losses Λ due to valves placed at the end of the conduit in order to regulate discharge from zero to the maximum possible value (corresponding to fully opened valve and zero Λ). A recently popular idea has been to replace the regulation valves with micro-turbines and to convert into electrical energy the dissipation Λ required to limit the discharge carried by the conduit. This also add sits energy value to the domestic and agricultural value of the water, and makes the water distribution business more advantageous.

Assuming atmospheric pressure at the turbine outlet, the piezometric gradient J and the power P_T produced by the turbine are linked to the discharge q by the following relationships:

$$J = K_p \frac{q^2}{D^5} \tag{1a}$$

$$P_T = \eta \cdot \gamma \cdot q \cdot \left(\Delta H - JL - \Lambda'\right) \tag{1b}$$

where K_p is a parameter depending on the pipe material and its aging conditions, D is the pipe diameter, and ΔH is the total topographic jump between the aqueduct inlet and the turbine axis. Observe that coupling the Eqs. (1a) and (1b) we get a third order polynomial in the q variable, which is zero for both the zero and the maximum discharge values; the latter corresponds to the maximum discharge allowed by the conduit (without turbine or regulation valve) and to the condition $\Delta H = JL+\Lambda'$. The economic benefit of micro-turbines to be installed depends on two main parameters: 1) the small cost per unit power, 2) the possibility to maintain high efficiency values η in Eq. (1b) with different discharge and net head values, where the net head H_n is the hydraulic head at the turbine inlet, given by:

$$H_n = \Delta H - JL - \Lambda' \tag{2}$$

Observe that a turbine with a single characteristic curve, placed at the end of an aqueduct, cannot always provide the power given by the r.h.s. of Eq. (1b) for different discharges, because a one-to-one relationship exists between discharge q and net head H_n and this relationship is usually different from Eq. (2). In this case, for example if PATs (Pumps As Turbines) are installed, when the discharge changes from its design value part of the potential energy must be anyway dissipated by a valve placed immediately before the turbine or part of the discharge must be bypassed by means of a parallel pipe [2]. A possible alternative strategy is to provide the turbine with a regulation system, with the effect of changing the characteristic curve according to the net head measured at the turbine inlet. Many regulator systems are available to change the turbine characteristic curve according to the variation of the hydraulic working conditions. Recent power electronic devices make it possible to regulate the electrical voltage and frequency in order to vary also the generator speed [3]. In the following sections, after a short review of the Banki turbine (hereafter called CF as Cross-Flow) design and of its hydraulic regulation system, we shall show that the best global efficiency is obtained by coupling the hydraulic regulation with the electric regulation of the turbine impeller rotational velocity, but that electric regulation is only worthwhile if the net head variability exceeds a minimum value depending on the specific cost/benefit ratio. A study case is analyzed in order to validate the theoretical study.

2. CF turbines for energy recovery in hydraulic plants

The CF turbine is a simple and economic turbine appropriate for micro and mini hydropower plants. The peak efficiency of this turbine is somewhat less than that of 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 CF turbine has a drum-shaped runner consisting of two parallel discs connected together near their rims by a series of curved blades (see Fig. 1). The turbine has a horizontal rotational shaft, unlike the 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 twice goes along 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. A design methodology for the standard CF turbine has recently been proposed by some of the present authors [4, 5, 6, 7].

The CF turbine proposed by Czech engineer Miroslav Cink as a circular profiled segment (see Fig. 1), which makes it possible to get a different characteristic curve for each different segment position, without generating local energy dissipations. Sammartano el al. [4, 5] showed that in this turbine the inlet velocity V is a function only of the net hydraulic head, along with the velocity of the reference system at the impeller inlet. It is generally acknowledged that the efficiency of the turbine is mainly related to the velocity ratio, defined as:

$$V_r = \frac{V}{\omega \cdot R} \tag{3}$$

and that the maximum efficiency is attained for a velocity ratio almost equal to 2. If continuous energy losses in the conduit are negligible in Eq. (2), the regulation system adopted by Sammartano et al. [4] makes it possible to keep the velocity ratio constant, along with the impeller rotational velocity, when a change of the discharge occurs. If continuous energy losses are consistent, the same goal cannot be attained without changing the impeller rotational velocity, because the inlet velocity is directly related to the net head H_n. This implies that, in this second case, a velocity ratio close to its optimum value can be saved only if both nominator and denominator are changed in Eq. (3). In general, control of the rotational speed of the hydraulic turbine increases the system investment, and adds a new source of energy dissipation, given by the converters. This implies that the use of electronic regulation is only worthwhile if the increment of gross energy production is above a minimum value.

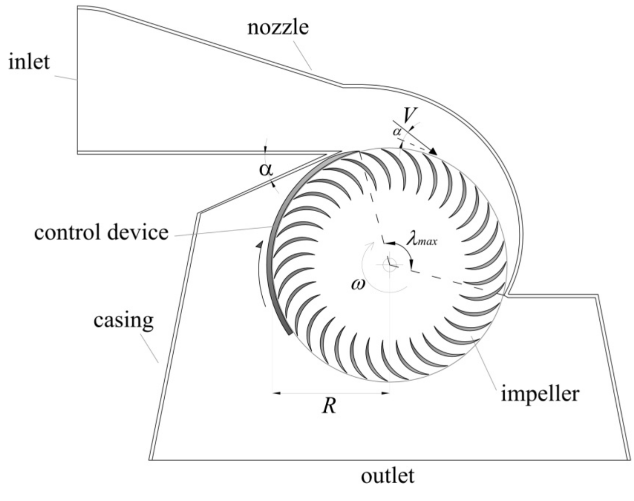

Fig. 1. Section of the CF turbine with the hydraulic control device.

3. Electrical energy production: optimal combination of hydraulic and electrical controls

A straightforward way to exchange energy with the main AC grid is to insert a three-phase induction machine (IM) between the impeller and the AC grid, which is used as a generator. The reactive power required by the electrical generator to properly operate is provided by a local capacitor bank. This solution is especially adopted for simplicity of construction, robustness of the generator, and ease of connection and disconnection from the grid. The main limit of this solution is that no speed control can be applied to the electrical machine and its rotational velocity is strictly related to the load torque and electromagnetic rotating field, which is established by the grid frequency (f_e): $\omega_e = 2\pi f_e$. A very different power conversion topology overcoming the previous limitation is displayed in Fig. 2. A back-to-back converter consisting of two conventional pulse width modulated (PWM) voltage source inverters (VSIs) is adopted to decouple the speed control of the electrical generator from the active and reactive power control implemented in the grid side converter. The latter converter is controlled in order to satisfy the grid regulations and maintain high quality electrical energy production, while the former ensures very flexible speed control of the electrical generator, tracking the maximum power point extracted from the hydraulic turbine, given by Eq. (3); furthermore, the input current harmonics and harmonic losses in the generator are mitigated thanks to the suitable modulation technique adopted in such converters. Although both asynchronous and synchronous electrical generators can be adopted in this power conversion topology, the permanent magnet synchronous generator (PMSG) has been recognized to be a promising technology because of its high efficiency, high power density, low inertia and high power factor [8, 9].

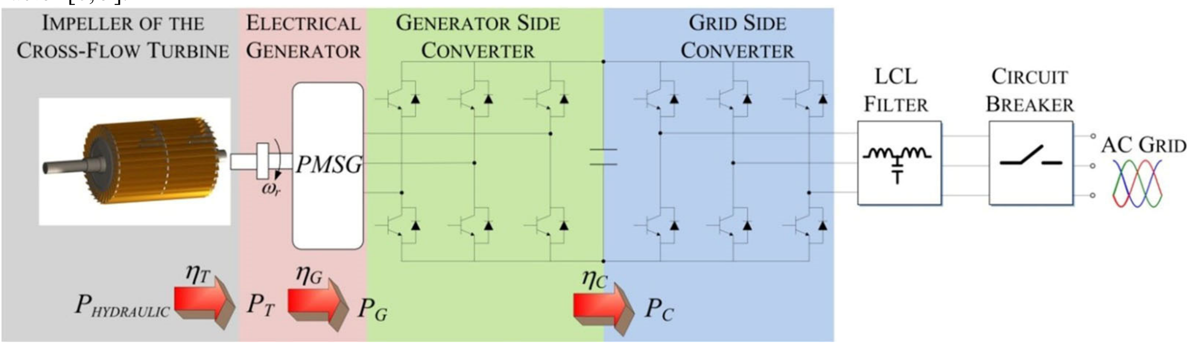

Fig. 2. Block diagram of a direct drive power conversion unit using a back-to-back power converter.

From the efficiency point of view, this energy conversion configuration includes an additional source of losses due to the use of the power converter. Hence, P_{out} in this case is obtained as:

$$P_{out} = P_G = \eta_C \eta_G \eta_T P_{Hydraulic} \tag{4}$$

where η_C, η_G and η_T are the efficiency of respectively the power converter, the electrical generator and the turbine. The first quantity, as well as η_G, changes as a function of the power delivered to the AC grid. Observe that, due to the additional losses given by the converter, the global efficiency could drop or the additional cost of the electronic regulation may not be recovered by the larger energy production. In order to determine the effectiveness of the latter power conversion unit, a study case is presented in which the design constraints and investment costs are compared to those of the traditional power conversion carried on with asynchronous generator. Basically, the focus of the study is to analyze how far in terms of cost and energy production the suboptimal operating condition given by the use of constant speed operation is from optimal operation of the hydraulic turbine obtained using the variable speed control method provided by the solution in Fig. 2.

4. Study case: purification plant at Risalaimi (Italy)

The Rosamarina artificial basin is one of the major water sources for the city of Palermo (Sicily). The water is delivered first to the Monte Cozzo disconnection tank and then to the downstream purification plant. The total hydraulic jump ΔH between the Monte Cozzo tank and the purification plant is about 36.34 m and the installation of a CF turbine for energy recovery at the inlet of the purification plant is planned. The cast iron conduit is about 10 km long and has a nominal diameter of 900 mm, delivering a maximum discharge of about 1 m^3/s. The upper tank at Monte Cozzo is fed by a pumping station made up of four centrifugal pumps. In order to maintain a constant water level in the lower tank, the water manager controls the flow rate by different opening degrees of a control valve placed in the downstream section of the pipe. The peak demand in the hydrograph has a value of q_{max} = 0.800 m^3/s and the modal water discharge has a value of q_{mod} = 0.600 m^3/s. The total head losses along the conduit are consistent with respect to the total jump and the net head immediately before the turbine follows the head-discharge relationship shown in Table 1.

Table 1. Conduit discharge-net head values at Risalaimi water drinking system (Palermo, Italy).

Water Discharge q (m^3/s)	Net Hydraulic Head H_n (m)
0.8	18.60
0.6	29.20
0.4	31.65
0.2	34.90

The CF turbine was designed according to Eqs. (2) - (3) and applying the four-step procedure described in the numerical study by Sammartano et al. [6]. In the case of constant ω impeller rotational speed (no electric regulation) the rotational velocity of the impeller was set to a fixed value of 500 rpm in order to be directly coupled to a 12-pole IM whose synchronous frequency coincides with that of the main grid, f_e = 50 Hz. The nameplate of the three phase induction machine (IM) is reported in Table 2 [10].

Table 2. Nameplate of the induction machine IM for the CFT1 configuration.

Rated power [kW]	Rated voltage [V]	Rated frequency [Hz]	Rated speed [rpm]	Rated torque [Nm]	Rated efficiency [%]	Pole Pairs	Rotor Inertia [kgm^2]
145	400	50	495	2826	91.7	6	15.8

The other geometrical parameters of the CF turbine, the number of blades N_b and the diameter ratio D_2/D_1, were estimated performing computational fluid dynamics (CFD) analyses using the ANSYS CFX commercial code, solving the Reynolds-averaged Navier Stokes Equations [11]. In this case study the procedure led to the optimal geometry of the CF turbine reported in Table 3.

Table 3. Geometrical parameters of the CF turbine.

Parameter	N_b [-]	D_1/D_2 [-]	R_1 [mm]	R_2 [mm]	B [mm]	λ_{max} [degree]	α [degree]
Description	Number of blades	Diameter ratio	Outer impeller radius	Inner impeller radius	Impeller width	Maximum Inlet discharge angle	Attack angle
Value	50	0.75	196	147	450	120	15

As explained in the previous section, the CF turbine was coupled with two different power conversion topologies between the impeller and the AC grid: 1) a three-phase induction machine; 2) a permanent magnet synchronous generator coupled with a back-to-back converter. In the first solution, called CFT1, the generator works approximately at a constant rotational speed, while in the second configuration, called CFT2, the generator (PMSG) has the possibility to change the rotational velocity according to the optimal operation of the hydraulic turbine. In Table 4 the characteristics of the selected permanent magnet synchronous generator are summarized [12].

Table 4. Nameplate of the PMSG generator for the CFT2 configuration .

Power [kW]	Voltage [V]	Frequency [Hz]	Speed [rpm]	Torque [Nm]	Efficiency [%]	Inertia [kgm^2]
214	400	60	600	3400	96	6.76

In order to estimate the energy production, the costs and the economic benefit of the two configurations, CFT1 and CFT2, the efficiency curve of the designed CF turbine was previously estimated by means of CFT simulations. For system CFT1, a series of runs was carried out for each of the four different hydraulic boundary conditions reported in Table 1, by changing the opening degree λ_i of the discharge regulator until the value q, given in the first column of Table 1 was attained. In this case the rotational speed ω was set constant and equal to the design value (500 rpm). For the CFT2 system similar computations were carried out, but two parameters were changed for each boundary condition: the opening degree λ_i and the rotational velocity ω, in order to match both the required discharge qi and the optimal velocity ratio V_r = 2. The global efficiency η_{tot} of the designed turbine coupled with the selected generator was estimated by using Eq. (4), with $\eta_C = 1$ in the case of CFT1. In these equations the efficiency η_{tot} is the product of the turbine efficiency η_T, of the generator efficiency η_G and of the converter generator η_C (only in the case CFT2). The plot in Fig. 3 reports the global efficiency η_{tot} curves for the final configurations. As shown in Fig. 3, the performances are generally worse in the CFT1 system than in CFT2 system: the highest differences occur when the flow discharge regulator is fully opened (λ = 120 degree) or is almost closed (λ = 27.7 degree).

The gain of the investment of the energy-recovery configurations can be estimated starting from the average annual energy production of the CFT1 and CFT2 configurations. Thus, according to the discharge duration curve of the plant and to the efficiency curves obtained, the annual energy production was calculated using the following expression:

$$E_i = \sum_{j=1}^{12}\left[E_{im,j}\right] = \sum_{j=1}^{12}\left[\gamma \cdot \eta_{tot,j}(\lambda) \cdot q_j \cdot H_{n,j} \cdot \Delta t_j\right] \tag{5}$$

where $E_{im,j}$ is the monthly energy production and Δt_j is the number of seconds per month for the investment considered. The result is that, in this case study, the CFT2 configuration generally has a higher monthly production than the CFT1 configuration, particularly when the flow rate reaches the highest value (q = 0.800 m^3/s). To define the cost of each investment C_i, the authors took into account the following economic components: the manufacturing cost of the turbine C_t, the cost of the electric equipment C_e, and the maintenance costs C_m (estimated as 10% of the sum of C_t and C_e). The final result is that the energy-recovery CFT2 configuration, even if it has higher installation and maintenance costs, has a very similar return time of the invested capital to that of CFT1, with a difference of only 3 months.

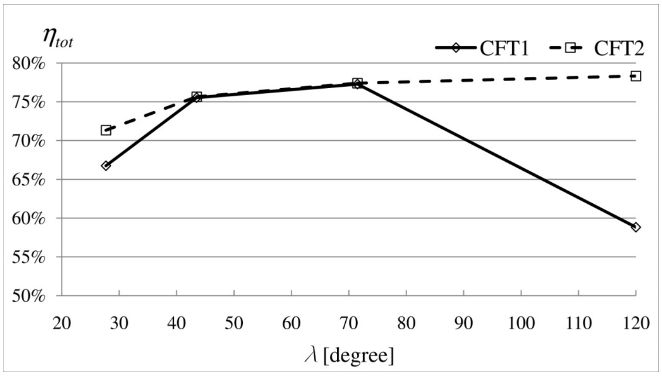

Fig. 3. Plot of the global efficiency η_{tot} curves for the final configurations.

4. Conclusions

The potential benefit of the impeller velocity electronic regulation in CF turbines, coupled with hydraulic regulation, was analyzed and evaluated with reference to a specific case. In the specific study case electronic regulation provided an increment of the energy production, with a return time increment of the invested capital of about 30%, but the benefit can be of course much worse or much better. On the other hand, smaller dissipations could be planned in the future for new aqueducts. As the velocity ratio is proportional to the root of the net hydraulic head and cross-flow turbines maintain high hydraulic efficiency within a large range of relative velocities, a strong reduction of head variability could lead in the next few years to a smaller benefit of electronic regulation for new plants.

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