PERFORMANCE PREDICTION OF AXIAL FLOW HYDRAULIC TURBINES AND OPTIMAL DOUBLE REGULATING

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Abstract. *This article presents an off-design performance prediction methodology for axial flow hydraulic turbines (single or double regulated). The flow solver, developed in a previous work, includes the simplified radial equilibrium equation in order to evaluate the spanwise velocity distributions at the main sections of the turbine. The hydraulic losses are assessed by using empirical correlations. First, the regulating of the wicket gates opening and of the runner blades staggering is treated separately. Thence the coupling of these regulating mechanisms is carried out as an optimization problem, by maximizing the turbine efficiency in a given operating range. These less accurate but much faster simulations show the ability to predict the main flow patterns at various operating points. So, this code can be used during the selection of the best conceptual design parameters of the turbine.*

Keywords*: turbomachine, axial hydraulic turbine, performance prediction, optimal double regulating.*

1. INTRODUCTION

The performance prediction of a hydraulic turbine can be evaluated experimentally, by testing models in test rigs, or theoretically, by using computational flow simulations in a virtual test rig. Nowadays, the development of powerful parallel computation has made possible the use of complex numerical flow simulation methods for turbine analysis and design. 3D Euler codes and 3D viscous Navier-Stokes codes are already standard tools in the development of new water turbine units. Details of flow separation, loss sources, loss distribution in components, matching of components at design and off-design, and low pressure levels with risk of cavitation are now amenable to analysis with computational fluid dynamics – CFD. The application of these modern CFD techniques for predicting the flow through an entire turbine has brought further substantial improvements on its hydraulic design. Since the detailed understanding of flow phenomena is of great practical importance, it has a direct impact on the design, resulting in geometrical changes of existing components, the replacement of existing components by a completely new design and/or the use of new materials (Drtina and Sallaberger, 1999). The experimental tests with models, however, still remain as final results, but they are carried out only after enough numerical simulations, what reduces the number of these expensive tests.

Although 3D Navier-Stokes codes have shown reliable results, with accurate performance predictions and flow details, hence decreasing the costs with model tests, a considerable computational effort has to be spent with grid generation and grid modification in each numerical investigation. For instance, when a geometrical modification is made for accounting the regulating mechanism of the turbine (wicket gate and/or runner blade), the meshes must be recalculated and the flow solver — with its high computational cost — must be run again. Actually, a fast, simple and reasonable accurate analysis is still essential for assessing the off-design performance of preliminary design alternatives, when the geometry is not yet entirely defined and there are only conceptual designs (Oh and Kim, 2001; Yoon *et al*., 1998).

Intermediate performance prediction approaches are apparently scarce in the open literature for water turbines. One of the few examples is the paper of Parker (1996). However, in that work, the theoretical analysis of the turbine performance does not account for the effects of runner blade geometry changes in the flow field and requires some test information of a turbine model. Also, it is applied only to geometrically similar turbines.

Therefore it would be desirable to make available intermediate performance prediction methodologies — with low computational cost — for hydraulic turbines. In the present work, such a methodology is proposed for axial flow hydraulic turbines. First the regulating of the wicket gates and of the runner blades is considered separately. Thence the coupling of both regulating mechanisms is carried out as an optimization problem, by maximizing the turbine efficiency in a given operating range. From these results, one could try making conceptual design modifications in order to improve the efficient operating range of the turbine.

The performance prediction methodology described in this work uses as flow solver the calculation scheme reported in Albuquerque *et al.* (2007). The key point is that this solver is able to indicate the correct trends of loss variations due to geometrical changes, so that different designs can be judged in a comparative sense (Denton, 1993; Casey, 2003).

In section 2, a brief description of the flow model employed for the performance prediction calculations is presented. Section 3 deals with the statement of the single regulating problems proposed in this article. The double regulating task is presented in section 4, with the optimization scheme adopted for its execution. Application examples of these methodologies are given in section 5. Section 6 ends with some concluding remarks.

2. FLOW MODEL FOR AN AXIAL HYDRAULIC TURBINE

The water turbine considered in this study is a tube type axial turbine, sketched in Fig. 1. The distributor is cylindrical (non-conical) and the guide vanes are untwisted along the span. This configuration is applied to low head hydropower plants and is expanding worldwide due to the progressive exhaust of hydropower resources of high and moderate heads.

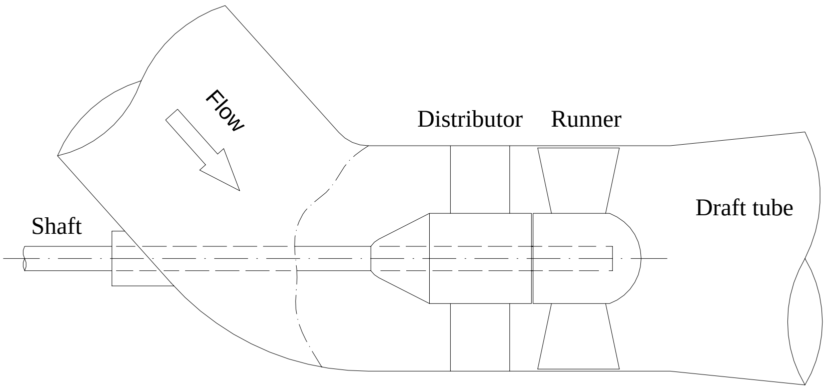


Figure 1. Sketch of the axial turbine water channel.

The flow model adopted in this work is the one reported in Albuquerque *et al.* (2007). In summary, the flow model away from the blade rows is considered axisymmetric, steady and with cylindrical stream surfaces. The flow at the cross sections behind the distributor and behind the runner is treated by means of the simplified radial equilibrium equation. The flow losses and deviations are assessed by using empirical correlations. Although simplified, the model allows the consideration of non-free vortex analysis at an early design stage. For reducing the set of parameters used in the geometry definition, the runner blading stagger angle, camber and chord-pitch ratio are parameterized in terms of their values at the hub, mean and tip stations. The parameterization is an efficient way to make geometrical changes even when using grids in CFD.

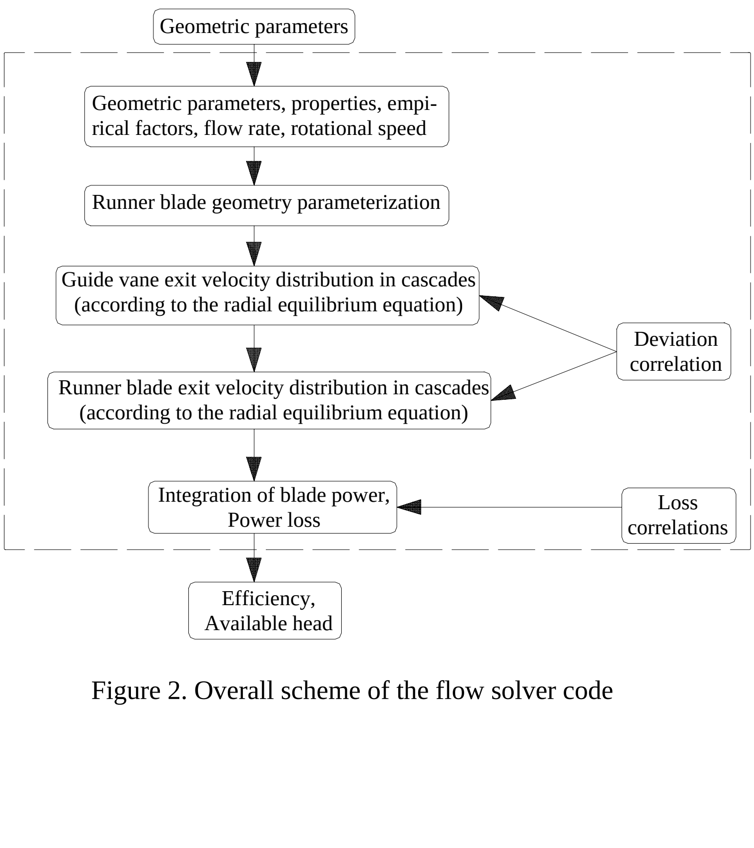


Figure 2. Overall scheme of the flow solver code

The simplified radial equilibrium is a topic that can be found in many textbooks. However, the differences between the flow conditions downstream a stator (where there is no work transfer) and downstream a runner or impeller (where a spanwise distribution of work transfer occurs) often are not clearly pointed out. Commonly, the actual energy balance in the upstream blade rows is not explicitly accounted for. So, in Albuquerque *et al.* (2007), suitable formulations of the radial equilibrium equation were developed for the flow downstream the distributor and downstream the runner. The basic procedure consists in applying integral energy balances through the corresponding upstream blade cascades and introducing the results in the simplified radial equilibrium equation. An iterative scheme is then carried out to calculate the meridional and circumferential velocity components downstream the distributor and downstream the runner.

Once determined these velocity profiles, the hydraulic losses are assessed by using empirical correlations and the blade specific work is calculated by means of the Euler work equation. The sum of the blade specific work with the hydraulic losses gives the turbine available energy (head). The turbine efficiency can then be calculated as the ratio between the blade specific work and the turbine available energy (head). Figure 2 shows the overall scheme of the flow solver code. This solver was coded in MatLab™ language since the MatLab™ optimization toolbox will also be employed.

3. SINGLE REGULATING MECHANISMS: WICKET GATES OR RUNNER BLADES

The purpose of the regulating mechanisms is to adjust the flow rate of the turbine, and thus its output, to the power demanded by the consumers. For instance, in synchronous hydroelectric generation — the most common application of water turbines, also assumed in this work —, the rotational speed of the shaft is kept constant due to the synchronism between the electric generator and the electric network. The turbine available head presents just small variations and thus can be assumed as a constant during the operating. Therefore, the adjustment of the turbine power output must be done by regulating its discharge.

3.1. Wicket gates

The regulating of the flow rate in propeller turbines is normally done by varying the guide vanes opening (wicket gates). This type of regulating is preferred when the power output is small or medium and presents just small perturbations, such that the flow rate is also little variable. This mechanism varies the area and the angles of the fluid flow through the distributor, what changes the flow rate. The two basic mechanisms are the Fink's ring (for small units) and individual servomotors (for medium and large turbines).

The main disadvantage of this type of regulating is that the efficiency curve presents a peak, and quickly decreases as the flow rate becomes distant from the design point. So, the resulting operating range is narrow in comparison with other types of regulating. The energy production over the operating range is also the smallest. These issues are due mainly to the changes in the runner incoming flow angles, imparted by the guide vanes. The propeller runner is designed assuming an incoming flow from the distributor. When the wicket gates are moved, these incidence angles are considerably changed, producing high shock losses at the runner inlet. Also, the runner turning angles are designed assuming a small swirl for the exit flow (which is the draft tube inward flow). With a different inlet flow, a considerable exit swirl can be produced, what reduces the pressure recovery in the draft tube, and thus the turbine efficiency.

In the present work, the wicket gates regulating problem is stated as follows: given the turbine design, the rotational speed, the designed available head, H_{design}, and the wicket gates opening range, find the flow rate, Q, that leads to H_{design} for any value of guide vanes opening angle. This problem was coded as a nonlinear equation:

$$H(Q) - H_{design} = 0 \tag{1}$$

where $H(Q)$ is the resulting turbine available head for the flow rate Q and for the particular value of guide vanes opening angle. $H(Q)$ is calculated by the flow solver described in section 2 and sketched in Fig. 2.

The Eq. (1) is solved by using a bisection-based algorithm (implemented in the `fzero` function of MatLab™). Each time the flow rate that solves Eq. (1) is found, the real turbine efficiency is also stored. After solving Eq. (1) for all the values selected in the wicket gates opening range, it is plotted: flow rate *versus* guide vanes opening angle, efficiency *versus* flow rate and output *versus* flow rate. These plots can be done for different conceptual designs for the turbine and the resulting off-design performance predictions can be judged in a comparative sense.

3.2. Runner blades (Kaplan mechanism)

When the power output is higher and more variable during the turbine operating, it is preferable to regulate the discharge by varying the runner blades staggering position. This is known as the Kaplan mechanism or Kaplan runner.

With this regulating mechanism, the peak in the efficiency curve is attenuated and a flat region arises, i.e., a high efficiency is achieved over a wide operating range. Now, with the blade staggering accompanying the flow rate variation, the overall incidences at the runner inlet and the swirl at the runner exit are less variable and so the shock and the draft tube pressure recovery are less prejudiced.

In the present work, the runner blades regulating problem is stated similarly as in the previous subsection: given the turbine design, the rotational speed, the designed available head, H_{design}, and the blade staggering range, find the flow rate, Q, that leads to the designed available head for any value of blade staggering. This problem was coded as the same nonlinear equation — Eq. (1).

Again, after solving Eq. (1) for all the values selected in the runner blades staggering range, it is plotted: flow rate *versus* blade staggering, efficiency *versus* flow rate and output *versus* flow rate. These plots can be compared with those ones related to the wicket gates regulating and so the suitability of each type of single regulating mechanism of a turbine can be investigated.

4. DOUBLE REGULATING: AN OPTIMIZATION PROBLEM

It is not surprising that the double regulating, i.e., wicket gates *and* runner blades, brings the most favorable conditions for the flow field through an axial turbine. With the double regulating, good incidence levels at the runner inlet and small swirl levels at the runner exit can be achieved in a wide operating range, so that the efficiency curve is even flatter than that for the runner blades regulating only. These full Kaplan turbines present a normalized efficiency curve better than that of Francis turbines and compete with the Pelton turbines — these ones having the flattest normalized efficiency curve among the water turbines.

The Kaplan turbines, as they are commonly known, present the double regulating mechanism. Actually, the regulating of the runner blades only (single regulating) is used in few situations. Nevertheless, it is not a pleonasm to say "full Kaplan turbine" when considering the double regulating. This type is the one preferred in most situations, mainly in the high hydropower plants.

Since the designed available head can be achieved with different configurations when using the double regulating, the idea is to couple the wicket gates opening with the runner blades staggering in order to maximize the turbine efficiency for any flow rate of the operating range. In this way, the double regulating task is stated as a constrained nonlinear optimization problem:

$$\begin{aligned} &\text{maximize } \eta(\mathbf{x}) \\ &\text{subject to } H(\mathbf{x}) - H_{design} = 0 \end{aligned} \tag{2}$$

where η is the resulting turbine efficiency, $\mathbf{x}$ is the two dimensional vector of regulating parameters (guide vane opening angle and blade stagger angle), H is the resulting turbine available head and H_{design} is the designed available head. Again, η and H are furnished by the flow solver code described in section 2 (see Fig. 2).

This optimization problem is performed for each chosen value of flow rate on the operating range. Each time a solution for the problem stated in (2) is found, the real turbine efficiency and the regulating parameters are stored. After solving the problem (2) for all the values of flow rate, it is plotted: blade staggering and flow rate *versus* guide vanes opening angle, efficiency *versus* flow rate and output *versus* flow rate. Now, these plots can be compared with those ones related to the single regulating mechanisms and so the suitability of each type of regulating (single or double) can be studied for a particular turbine design. Note that the plot "blade staggering *versus* guide vanes opening angle" is the envelope of the turbine and the control unit should be programmed to couple the regulating mechanisms following this curve.

The Sequential Quadratic Programming Method (SQP) is one of the most efficient optimization techniques for solving constrained nonlinear problems, being suitable for the present application (Nash and Sofer, 1996). The `fmincon` function from MatLab™ optimization toolbox was chosen in this work. This is an efficient implementation of the standard SQP using the BFGS formula for approximating the Hessian matrix (Nash and Sofer, 1996). In the present application, the option for evaluating the directional derivatives by finite differences was set.

5. APPLICATION EXAMPLES

The application examples presented here use as geometrical reference the model of tube type propeller turbine designed and tested by Souza (1989). Table 1 gives some basic features of this turbine. The relevant geometric parameters of the runner are reasonably reproduced by the parameterization proposed in Albuquerque *et al.* (2007). This parameterized approximation of the previous design (Souza, 1989) is referred here as the *design A* and will be compared with another design obtained in Albuquerque *et al.* (2007) by using optimization techniques, the *design B*.

In all results, the rotational speed and available head were kept constant equal to 1145 rpm and 4.0 m, respectively. Observe that the design flow rate is 0.267 m^3/s.

Table 1. Turbine main features

Flow rate	0.267 m³/s
Rotational speed	1145 rpm
Head	4.0 m
External diameter	280 mm
Internal diameter	112 mm
Number of guide vanes	8
Number of runner blades	4
$n_{qA} = 1000\,n\,Q^{0.5}\,/\,(gH)^{0.75}$	629

Figure 3 shows the results of the performance prediction procedure described in this article for the design *A* considering the wicket gates regulating only (like in the propeller turbines). The Fig. 3a represents the regulating key curve of this turbine, showing which discharge occurs for any wicket gates position. In Fig. 3b, one can observe that, in this design *A*, the maximum efficiency point occurs in partial load operating (Q < 0.267 m³/s). Also, the peak of power output occurs almost for the maximum wicket gates opening (see Fig. 3c) what is a common feature in propeller turbines.

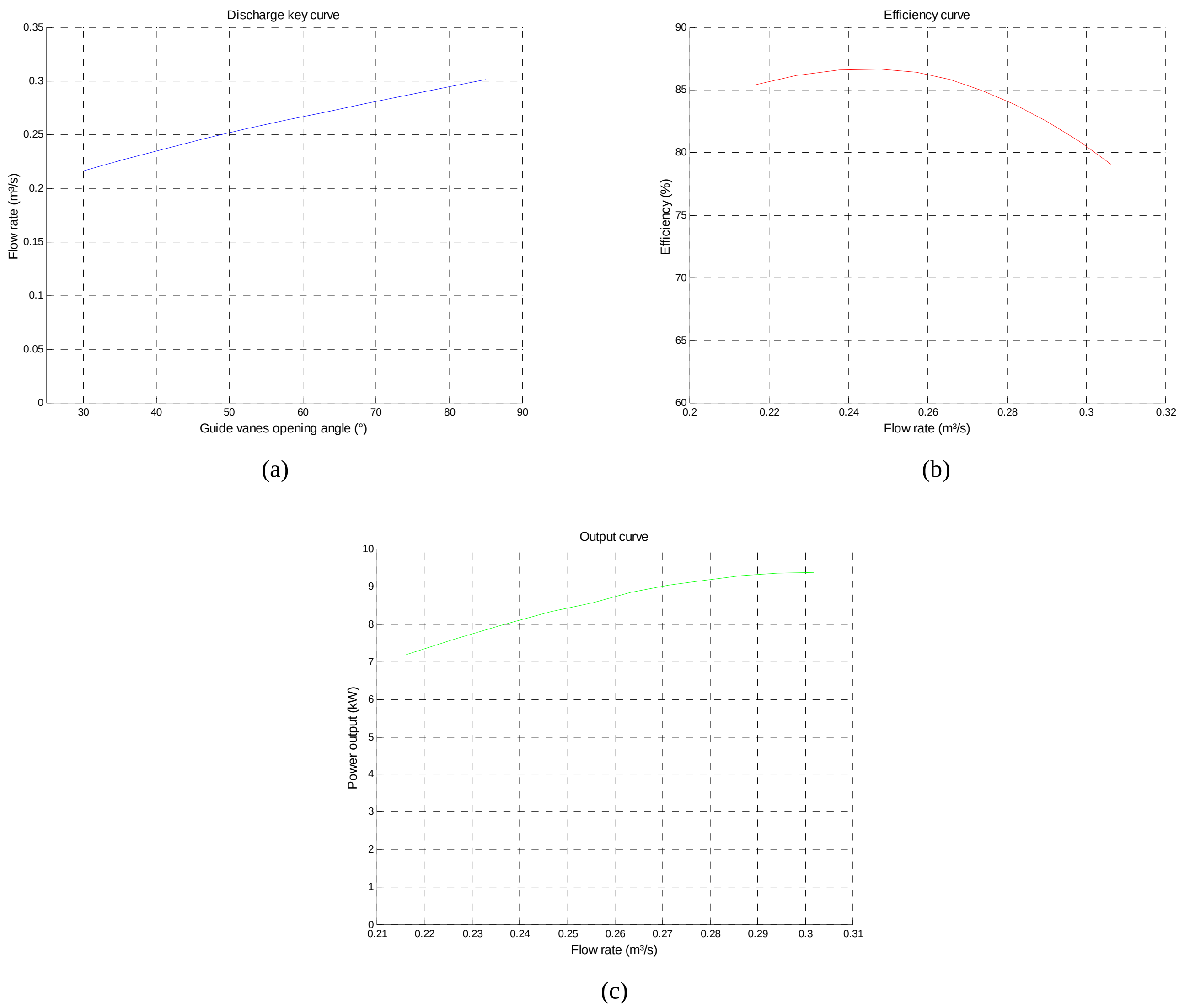
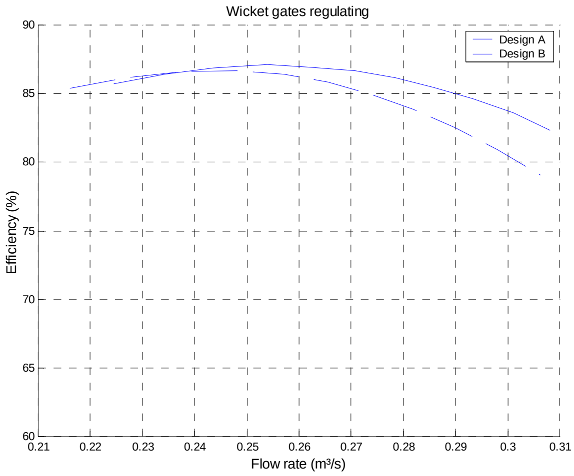
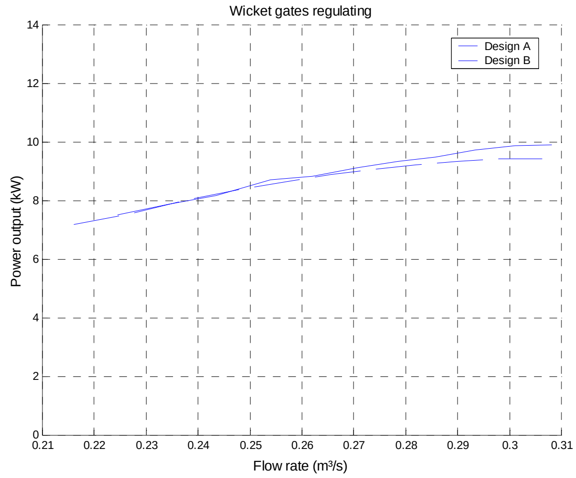


Figure 3. Performance prediction of design *A* with the wicket gates regulating: (a) discharge key curve, (b) efficiency curve and (c) power output curve.

The same analysis can be done for the design *B*. Figure 4 presents a comparison between the off-design performance predictions of designs *A* (dashed line) and *B* (continuous line) considering the wicket gates regulating. Observe that the design *B*, that was obtained in Albuquerque *et al*. (2007) after an optimization of the design *A* in the design point, presents higher efficiencies over a reasonable operating range, mainly for the higher loads. The design *A* is a little better just in the region of small loads.



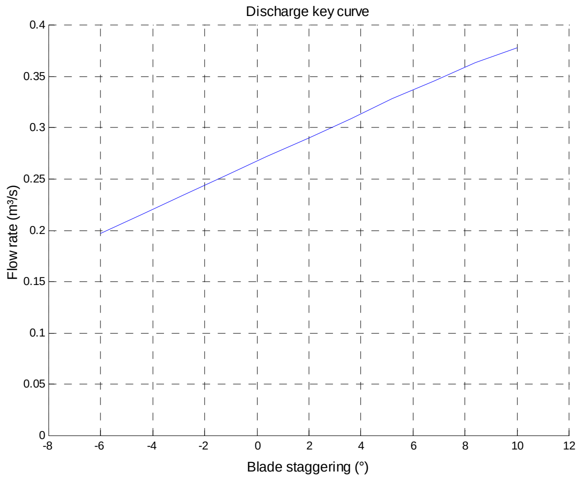
(a)



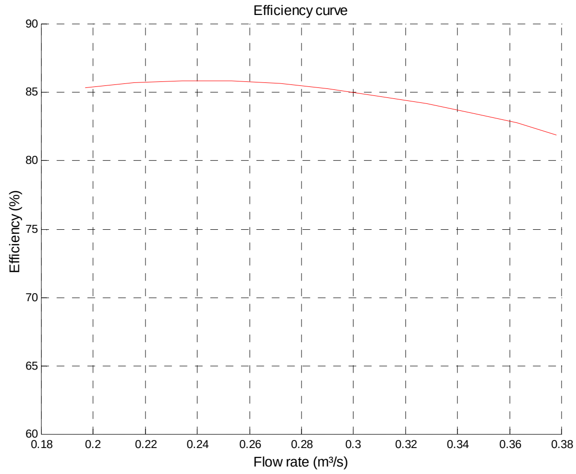
(b)

Figure 4. Comparison between (a) efficiency and (b) output of the designs *A* and *B* with the wicket gates regulating

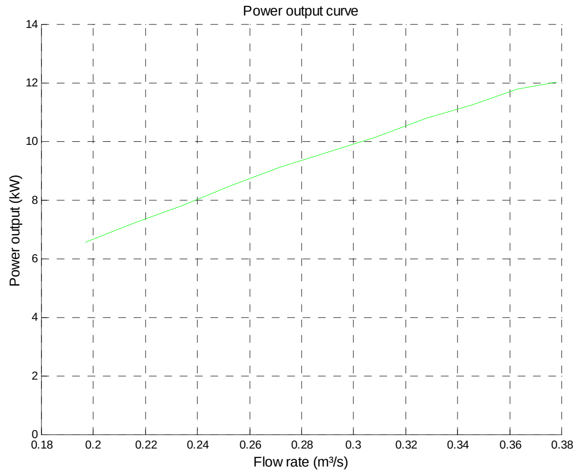
Figure 5 shows the results of performance prediction for the design *A* considering now the runner blades regulating mechanism (Kaplan runner and non-adjustable guide vanes). The Fig. 5a represents the regulating key curve of this turbine, showing which discharge occurs for any blade staggering position. It can be observed that a wider discharge range can be achieved with the runner blades regulating in comparison to that due to the wicket gates regulating.



(a)



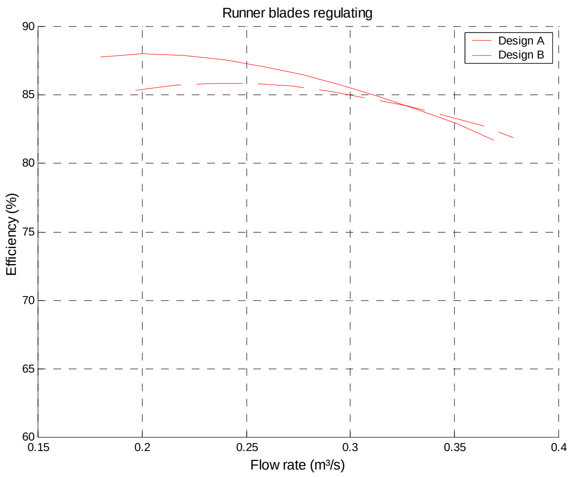
(b)



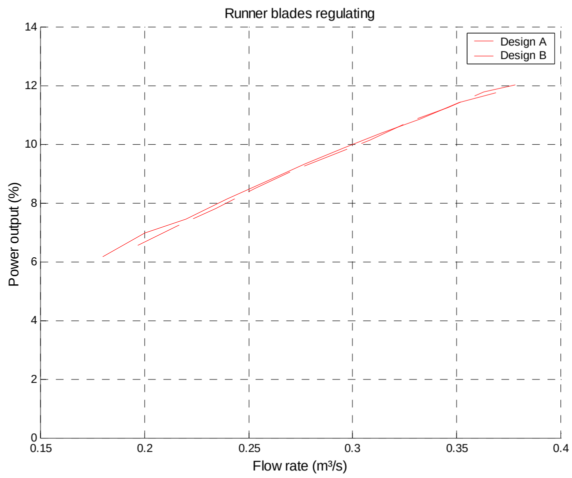
(c)

Figure 5. Performance prediction of design *A* with the runner blades regulating: (a) discharge key curve, (b) efficiency curve and (c) power output curve.

The same analysis can be done for the design *B*. Figure 6 presents a comparison between the off-design performance prediction of designs *A* (dashed line) and *B* (continuous line) considering the runner blades regulating. Now, the design *B* shows a superior performance mainly in the partial load operating (Q < 0.267 m³/s). Also, the efficient operating ranges of both designs were improved in comparison to those with the wicket gates regulating (note the wider flow rate ranges). This is due to the best fluid flow conditions that can be obtained when the regulating is done by varying the blades staggering instead of varying the guide vanes opening.



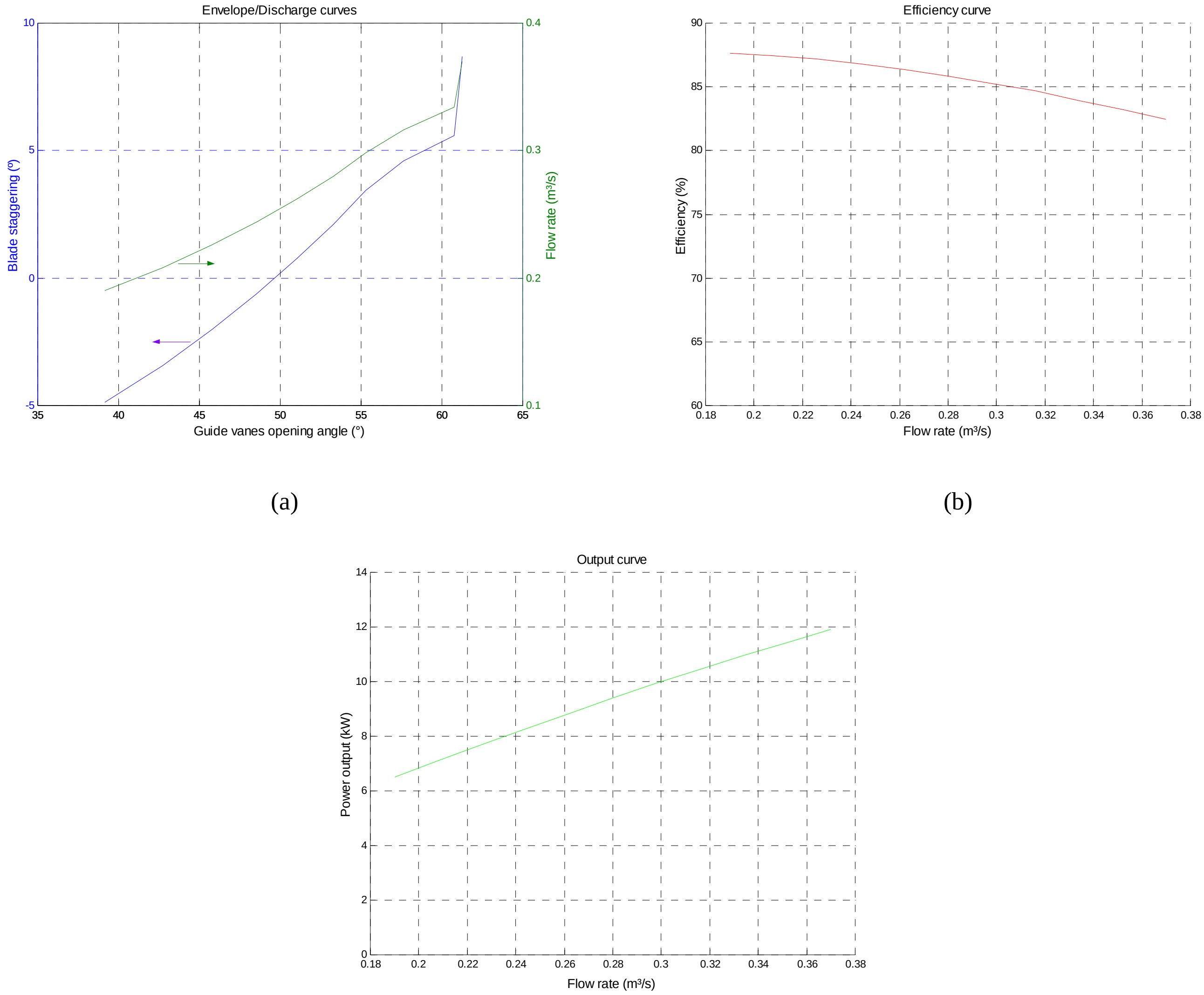
(a)



(b)

Figure 6. Comparison between (a) efficiency and (b) output of the designs *A* and *B* with the runner blades regulating

Figure 7 brings the optimal double regulating calculated for the design *A* as a full Kaplan turbine. Figure 7a is the envelope of this design, i.e., the control unit of the runner blades/guide vanes mechanisms should be programmed to follow this curve. In this way, the turbine efficiency is the maximum for any operating condition. The flow rate is also shown as a function of the guide vanes position.



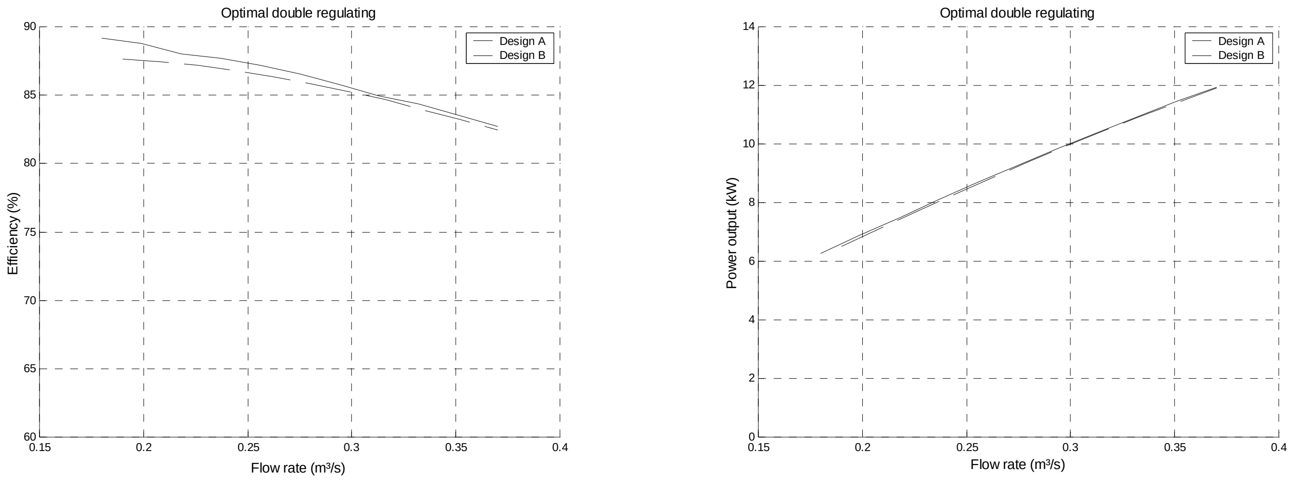
(a) (b) (c)

Figure 7. Optimal double regulating calculated for the design *A*: (a) envelope curve, (b) efficiency curve and (c) power output curve.

The same optimization can be done for the design *B*. In Fig. 8, the optimal double regulating performances of both designs are compared. Now, the design *B* (continuous line) shows a superior performance in the whole operating range (partial load, full load and overload). The optimal double regulating in fact couples the good characteristics of both single regulating mechanisms (see Figs. 4a and 6a). This is also shown in Fig. 9, where the three regulating mechanisms are compared for each turbine design. The coupling of the guide vanes opening with the runner blades staggering yields the most favorable flow conditions at any operating point. This double regulating increases the flexibility of the turbine in absorbing the flow energy with high efficiency, i.e., without high shock losses at the runner inlet and small swirl at the runner exit (draft tube inlet) in a wide discharge range. Therefore, the turbine efficiencies with the double regulating are always higher than those with the single regulating mechanisms.

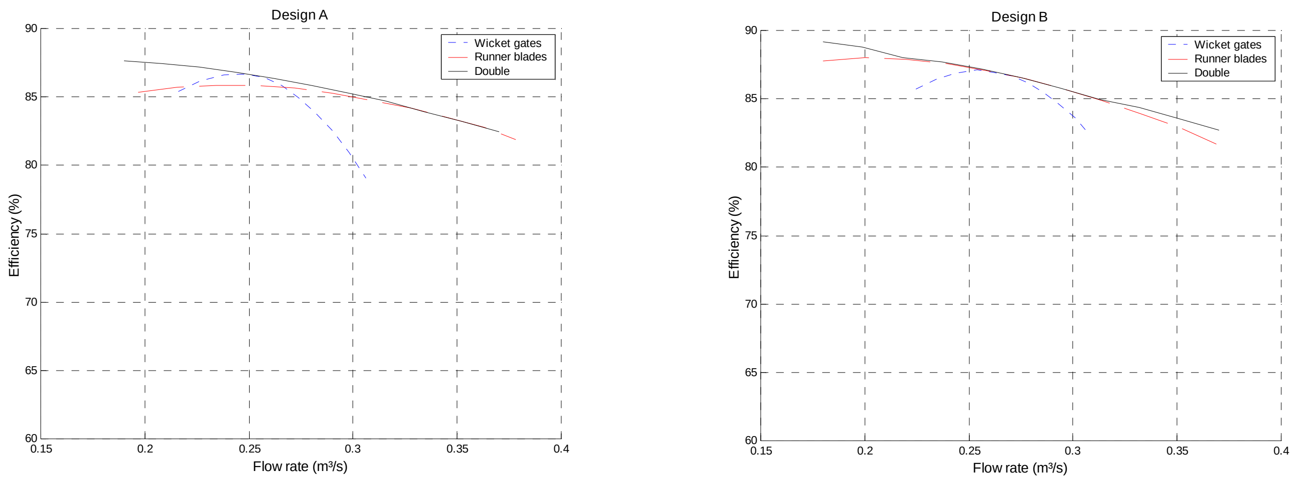
For the design *B*, the runner blades regulating could be a good alternative instead of using the more expensive double regulating. As can be seen in Fig. 9b, the double regulating provides further substantial improvements only in the limits of the turbine operating range. If the demanded load presents only small variations around the design point, the single regulating by the blades staggering gives a performance very close to that achieved by the double regulating.

Another point that should be commented is a general trend observed in hydraulic turbine design (Ueda, 1982): the efficiency improvement at the design point (generally, the maximum efficiency point), leads to performance improvements also in a considerable operating range around this design point. As it was said, the design *B* was obtained after an optimization of the design *A* in the design point only. Nevertheless, the off-design performance of *B* is better than that of *A* in a reasonable range for all types of regulating mechanisms (see Figs. 4a, 6a and 8a), confirming this statement of Ueda (1982).



(a) (b)

Figure 8. Comparison between (a) efficiency and (b) output of the designs *A* and *B* with the optimal double regulating



(a) (b)

Figure 9. Comparison between the three regulating mechanisms in designs (a) *A* and (b) *B*

6. CONCLUDING REMARKS

An off-design performance prediction methodology for axial flow hydraulic turbines has been presented. The flow solver employs a simplified modeling that includes the radial equilibrium equation in order to evaluate the spanwise velocity distributions at the main sections of the turbine. The hydraulic losses are assessed by using empirical correlations.

The wicket gates opening and the runner blades staggering were firstly considered as single regulating mechanisms. The discharge key curves were plot and the performance predictions of two different designs – *A* and *B* – were compared.

The double regulating, i.e., the optimal coupling of the wicket gates opening and the runner blades staggering, was carried out as an optimization problem. By using sequential quadratic programming, the turbine efficiency was maximized for some discharge values in a given operating range. So the envelope curve of the full Kaplan turbine can be determined. Again, the performance predictions of the designs *A* and *B* were compared in this double regulating mode.

The most favorable flow conditions at any operating point were achieved by the optimal double regulating. This mechanism increases the flexibility of the turbine in absorbing the flow energy with high efficiency, i.e., without high shock losses at the runner inlet and small swirl at the runner exit (draft tube inlet) in a wide discharge range. Therefore, the turbine efficiencies with the double regulating are always higher than those with the single regulating mechanisms.

A general trend in hydraulic turbine design was observed: the efficiency improvement at the design point generally leads to improvements in the performance also in a considerable operating range around this design point. The design *B*, that was obtained after an optimization of the design *A* in the design point only, presented an off-design performance better than that of *A* in a reasonable range for all types of regulating mechanisms, confirming this trend.

These less accurate but much faster simulations show the ability to predict the main flow patterns in various operating points. So, this code can be used during the selection of the best conceptual design parameters of the turbine.

6. ACKNOWLEDGEMENTS

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