MODELING OF HYDROKINETIC TURBINE

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Abstract. *This paper presents an analytical theory for the behavior of free-flow hydropower turbine, also known as a hydrokinetic turbine. This kind of turbine is designed to generate electricity using only the kinetic energy of water flow in rivers and is used to supply energy for remote communities in the Brazil's inland. This technology needs to be robust and suitable for the extremely severe conditions of the remote and isolated villages, since it has been functioning uninterruptedly from several years with a minimum maintenance. An integral approach allows a formulation for power generation using conventional propeller rotors. It is shown that the mass and energy balance applied to the free rotors added to a radial equilibrium formulation for elements of the blade can be used to describe the increased performance of the machine with a good degree of accuracy. The hydrokinetic turbine needs a minimum river speed of 1,5 m/s, and a minimum depth of 1 m. From these conditions, it is possible to obtain 600 W of energy production. In the second part of the paper, CFD simulations of a four blade runner is performed and compared to the results of the zero dimensional model of the flow. The results agree well with the project condition of the machine, emphasizing the use of the proposed methodology for the hydrodynamic conception of free flow turbines.*

Keywords: *Hydrokinetic turbine, electric power, sustainability, remote communities.*

1. INTRODUCTION

For many centuries, the mankind has exploited the river streams by means of hydraulic wheels or other rudimentary types of machine. In the course of time, those equipments have been forgotten and the development of projects of larger and more efficient hydraulic turbines has composed the modern technological effort of the hydropower worldwide. However, considerable environmental and social impacts have occurred on this river basin.

Hydrokinetic turbines, or water currents hydroturbines, which convert the kinetic energy of rivers or tidal currents into electrical power, are alternatives for sustainable energy conversion. The exploitation of small hydrokinetic hydropower is not a new concept, but now the revisiting of this kind of energy conversion actually becomes an alternative for sustainable energy conversion (Pish, 2002). In developing countries, many small and remote communities are located beside river streams. Hydrokinetic power system, with weak environmental impacts of the energy conversion, should represent an excellent renewable energy resource.

The technical literature linked to the conception and design of hydrokinetic turbines (HKT) is not very large. Few articles are available and presented in restricted publications or conferences. Gorban et al. (2001) presents an analytical deduction based in a blockage modeling of hydrokinetic turbines rotors. This work explores a new concept of turbine called Gorlov Turbine (Gorlov, 1995). The positive contribution of Gorlov Turbine is symbolized by the circumvention of this hydrodynamical limit which is more restrictive than Betz limit. A helical design of the turbine blades is used.

Inagaki et al. (2004) and Kanemoto et al. (2002) proposes to use a vertical tilt axis turbine for shallow currents. The simplicity of this model as well as the results of performance of the prototype is the main goal of this kind of machine.

In Brazil, the experience of great success in the generation of electric energy using HKT technology is associated with the Department of Mechanics Engineering of the University of Brasilia-UnB, with the development of three generation of hydrokinetic turbines (Van-Els et al., 2003). The first machine developed for this research group, called Generation 1, was installed in July 1995 in Correntina-BA. This project presents some innovations as a frontal stator. Therefore, the incidence flow angle in the inlet plane of the runner is improved. The best results for this machine has been gotten with a river velocity of 2 m/s and six-blade propeller like turbine, with diameter of 80 cm (Tiago Filho, 2003).

The application of the concept of Second Generation diffuser-augmented is showed. It was installed in Correntina-BA about August 2005. The performance improvement was observed during the tests. However, the increase of dimensions in this machine becomes inadequate for use in some kind of rivers of low depth (Oliveira and Souza, 2006).

With the evolution of the project and the research for more compact and portable machine, the Third Generation of the hydrokinetic turbine was conceived. This machine possesses an external casing structure acting as a diffuser, forming a set with the rotor. A numerical simulation through CFD (Computational Fluid Dynamic) techniques shows a power of 1,5 kW for rivers with velocities of 2 m/s (Brasil et al., 2006).

Considering the facts explained above, our methodology is deduced from the design approach and pertinent analysis of the axial hydraulic turbine and horizontal axis wind turbines. The proposed methodology involves the classical design methods of the propeller or Kaplan turbines concept, adjusting some parameters to the zero-head condition. This methodology is validated and optimized by performing commercial computational fluid dynamics (CFD). Numerical simulations are performed in order to verify some geometrical parameters.

2. AXIAL HYDROKINETIC TURBINE

A general layout of the proposed HKT (Hydrokinetic Turbines) can be observed in Fig. 1. It is a four-blade propeller like turbine, with external diameter D and a internal hub with diameter d. All this turbine assembly is evolved by an external casing structure, composed by an inlet nozzle, a lateral cylindrical case and a rear outflow diffuser.

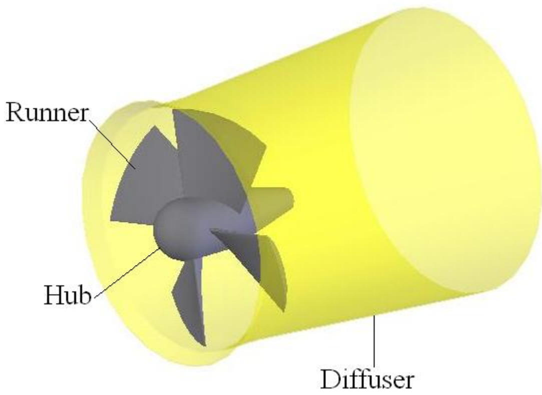

Figure 1. Axial hydrokinetic turbine

The proposed geometry is conceived to obtain an axial turbine with good hydraulic performance, as the conventional axial hydraulic turbines, which can obtain a hydraulic efficiency around 90%. On the other hand, as established by the Betz law for free flow machines, the maximum energy which can be converted from the kinetic energy in the projection runner area is 59,3 %(Cp,max=0,59). This lower performance is justified by the reduction of the velocity in the inlet plane due to the blockage of the flow by the turbine, inducing by this manner a lower exploitable kinetic energy for the runner.

To circumvent the Betz limit and augment the power coefficient of the axial rotor, the present turbine was conceived with an outflow diffuser. This design option has been analyzed for the enhancement of the performance of wind turbines by many authors (Badawi and Aly, 2000, Bet and Grassmann, 2004, Francovic and Vrsakovic, 2001, Grassman, 2005, among others).

To the numerical studies of hydrokinetic turbines can be developed, it needs of the definition of operational and dimensional parameters, which possible the verification of performance datas. Too, it makes possible the proposition of a methodology technology development of the runner hydrodynamic project.

3. GENERAL PARAMETERS

Considering a free stream flow with velocity V_0, where an axial machine will be installed, the kinetic energy in a cross flow surface with area given by $A = \dfrac{\pi D^2}{4}$, which can be converted by the hydraulic turbine, is given by:

$$E_0 = \frac{1}{2}\rho A V_0^3 \tag{1}$$

with ρ denoting the water density in Kg/m^3 .

For the study of free flow hydrokinetic turbines, the same parameters of wind turbines can be used (Burton, 2001). Therefore, considering that the turbine produces an effective power output $\dot{P}$, the machine power coefficient C_p can be defined as:

$$C_p = \frac{\dot{P}}{E_0} = \frac{\dot{P}}{\frac{1}{2}\rho A V_0^3} \tag{2}$$

This is normally used to express the dependency of this coefficient to the rotor speed ratio (λ), where:

$$\lambda = \frac{\omega R}{V_0} \tag{3}$$

with ω and R denoting the angular rotational speed (in rad/sec) and the turbine radius, respectively. The velocity in the turbine control volume is different from the free flow velocity. Generally, this change is taken into account by an additional factor "a", called axial flow induction factor.

Considering the axial averaged velocity through the turbine runner noted by V, the following relationship yields:

$$V = V_0(1-a) \tag{4}$$

For free flow machines, without augmented-diffuser, $a \in [0,1]$; for machines with augmented-diffuser, this induction factor is near of zero. A local rotor speed ratio is also considered:

$$\lambda' = \frac{\omega R}{V} = (1-a)'\lambda \tag{5}$$

Alternatively, a power coefficient based in the velocity V can be proposed. It is written as:

$$C_p' = \frac{\dot{P}}{\frac{1}{2}\rho A V^3} \tag{6}$$

It is important to remark that this coefficient is intrinsic to the runner hydrodinamical performance, which is related to the effective flow rate in the turbine. It is always convenient to express $C_p' = C_p'(\lambda')$. In a different way, the standard power coefficient C_p, given in Eq. (2), takes into account the behavior of the machine in a free stream and its efficiency to extract the power of the stream kinetic energy.

The design methodologies and performance analysis of conventional hydraulic turbines runner often use dimensionless variables involving both plant duty parameters and scale variables. The flow rate Q, the water head H, the angular rotational speed ω and the diameter D are grouped in classical dimensionless variables given by:

$$\phi = \frac{8Q}{\pi\omega D^3} = \text{ Flow coefficient} \tag{7}$$

$$\psi = \frac{8Hg}{\omega^2 D^2} = \text{ Head coefficient} \tag{8}$$

$$N_s = \frac{\phi^{\frac{1}{2}}}{\psi^{\frac{3}{4}}} = \text{ Specifc speed} \tag{9}$$

For the hydrokinetic turbines, otherwise, a zero-head situation occurs with no conversion of potential energy in work. If the conventional methodologies for design of axial hydraulic turbines were used, the dimensionless coefficients on Eq. (7)-(9) have to be re-written. Therefore, for HKT the equivalent head can be expressed by the conversion of all flow kinetic energy in potential, i.e. $H = \dfrac{V^2}{2g}$. Hence, all dimensionless variables can be related to the parameter λ', as follows:

$$\phi = \lambda'^{-1}; \psi = \lambda'^{-2}; N_s = \lambda' \tag{10}$$

From the last description of the turbine coefficients, all the machine behavior is parameterized only by the variable λ'. This allows to estimate other equivalent dimensionless parameters, which can be used in empirical equations for the design of axial turbines, in particular for propeller and Kaplan turbines. It will be considered that a free (or augmented-diffuser) propeller turbine, working in an operational condition , will extract the same power of a conventional casing machine in equivalent (ψ, ϕ) point.

4. LUMPED MASS AND MOMENTUM BALANCE

When the water current crosses any permeable obstacle, its free velocity is changed. A part of the water flow is percolated through the obstacle and another important amount of fluid is redirected to the lateral of the blockage volume, reducing the velocity in the inlet surface of the obstacle control volume. It is the case of the free flow turbine, where the turbine assembly restricts the flow, as show in the Figure 2. The classical deduction of Betz law (Betz 1926) is based on the description of the fluid flow (Figure 2). The mass conservation and the momentum for free flow runner can be evaluated to obtain global relations for this flow.

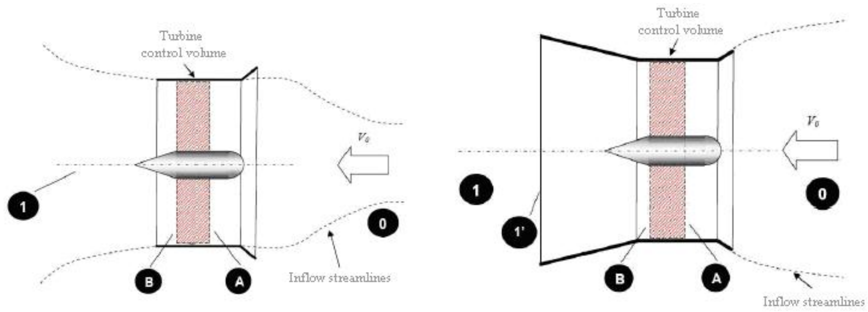

Figure 2. Flow through hydrokinetic turbines with and without augmented-diffuser

When the concept of diffuser-augmented is applied, the flow is captured from a larger cross flow area due to the pressure suction in downstream diffuser. Sometimes, as a consequence of mass conservation and of the coarse dependency of the diffuser geometry, the velocity could be greater then V_0, which makes the axial flow induction factor $a < 0$.

The lumped analysis of the flow through the HKT in both situations presented in Fig. 2 will consider that the velocity and pressure at the axial position 0 (zero) in the free flow is denoted by V_0 and p_0, respectively. The velocity and pressure change through the axial positions A, B, 1' and 1, which denote respectively the planes in inlet and outlet of stator-rotor control volume, the outlet of diffuser and far downstream.

The mass conservation through axial flow at the positions 0-A-B-1'-1 can be formulated as follows. Considering the incompressibility of the water flow, the mass conservation is given by:

$$A_0 V_0 = A V_A = A V_B = A_{1'} V_{1'} = A_1 V_1 \tag{11}$$

With $V_A = V_B = V$:

$$A_0 V_0 = A V_A = A V_B = A V \tag{12}$$

After the axial momentum balance in turbine control volume between the positions A-B, the fluid axial forcing the turbine can be expressed by:

$$F = (p_A - p_B) A \tag{13}$$

If the first law of thermodynamics is also applied to the same control volume, considering an adiabatic flow, the following equation is obtained for the turbine power extraction:

$$\dot{P} = V A (p_A - p_B) \tag{14}$$

Using the power coefficient definition presented in Eq. (6), the pressure drop in the turbine can be rearranged to:

$$(p_A - p_B) = \frac{1}{2}\rho C_p' V^2 = \frac{1}{2}\rho V_0^2 (1-a)^2 C_p' \tag{15}$$

Now, using a larger control volume between 0-1 and considering that $p_1 = p_0$, the momentum conservation can be written as:

$$F = \dot{m}(V_0 - V_1) \tag{16}$$

where $\dot{m}$ is the mass flow in the turbine. Replacing Eq. (13) in (16), another equation for pressure drop in the turbine can be obtained by:

$$p_A - p_B = \rho (1-a) V_0 (V_0 - V_1) \tag{17}$$

An equation for V_1 can be obtained by replacing Eq. (15) in (17):

$$V_1 = V_0 [1 - \frac{1}{2}(1-a) C_p'] \tag{18}$$

The entrance flow is analyzed by using Bernoulli equation between 0-A:

$$p_A = p_0 + \frac{1}{2}\rho (V_0^2 - V^2) = p_0 + \frac{1}{2}\rho V_0^2 [1 - (1-a)^2] \tag{19}$$

Applying also Bernoulli in the outlet flow between 1'-1:

$$p_{1'} = p_1 + \frac{1}{2}\rho(V_1^2 - V_{1'}^2) \tag{20}$$

Using the continuity to express the velocity in 1' in function of V, this equation can be rewritten as:

$$\begin{aligned} p_{1'} &= p_1 + \frac{1}{2}\rho(V_1^2 - V^2 n^2) \\ &= p_1 + \frac{1}{2}\rho V_0^2 \left[\frac{V_1^2}{V_0^2} - n^2(1-a)^2\right] n = \frac{A}{A_{1'}} \end{aligned} \tag{21}$$

To obtainan equation for the pressure drop in the diffuser, a modified Bernoulli formulation is used. Then, the flow in the diffuser is formulated by:

$$p_{1'} - p_B = \frac{1}{2}C_{pr}\rho V_2 \tag{22}$$

where C_{pr} is the diffuser pressure coefficient estimated for the conical diffuser, as follows:

$$C_{pr} = \eta_d(1 - n^2) \tag{23}$$

where η_d denotes the diffuser efficiency, computed by an empirical relation as:

$$\eta_d = 0.058\theta - 0.148 \text{ with } \theta = 2\arctan(\frac{D-d}{2L}) \tag{24}$$

and L denotes the diffuser length.

Using Eq. (21) and (22), an equation for the pressure in B can be obtained:

$$p_B = p_1 + \frac{1}{2}\rho V_0^2[\frac{V_1^2}{V_0^2} - (1-a)^2(n^2 + C_{pr})] \tag{25}$$

Subtracting Eq. (19) from Eq. (25) yields:

$$p_A - p_B = \frac{1}{2}\rho V_0^2[1 - \frac{V_1^2}{V_0^2} + (1-a)^2(n^2 + C_{pr} - 1)] \tag{26}$$

Replacing the equation for the pressure drop in the turbine as function of power coefficient and the equation for downstream velocity (Eq. (18)) in the last equation yields:

$$\frac{4a}{1-a} = \frac{(C'_p - 1 + n^2 + Cpr)^2}{C'_p + 1 - n^2 - Cpr} \tag{27}$$

Using the definition of C_{pr} given in Eq. (23), the last equation can be rewritten as:

$$\frac{4a}{1-a} = \frac{(C'_p - 1 + n^2 + (\eta_d(1-n^2)))^2}{C'_p + 1 - n^2 - (\eta_d(1-n^2))} \tag{28}$$

This equation establishes a relationship between the induction factor "a" and the other operational parameters of the turbine, explicitly the diffuser area ratio, n, and its efficiency η_d . In fact, a non-linear problem causes the dependency of C'_p on V as well as on "a". By numerical solution of this nonlinear problem, and considering the scale and operation parameters (n, η_d, V_0 and ω), the axial induction factor "a" and C'_p can be obtained for a specific operational point.

Some important points must be considered:

- For free flow turbines, without diffuser, it is simple to be shown that Eq. (28) is reduced to the classical relations:

$$C'_p = \frac{4a}{1-a} = 4a(1-a)^2 \tag{29}$$

- For free flow turbine, the Betz limit can be deduced only by computing $(dC'_p/da) = 0$, which gives:

$$a = \frac{1}{3}; C_{pmax} = \frac{16}{27} \tag{30}$$

- Considering a stream velocity V_0 and the geometrical and hydrodynamical characteristics of the turbine, the solution of Eq. (28) extracts the pair (a, C'_p). Finally, the power coefficient of the shrouded machine can be determined by:

$$C_p = (1-a)^3 C'_p \tag{31}$$

At this point, it is necessary to develop a description of the turbine set (stator + rotor) internal hydrodynamics. Therefore, an equation for $C'_p = C'_p(\lambda')$ can be evaluated.

5. ROTOR FLOW MODELING

The simplified model for the hydrodynamics behavior of the rotor is based on Blade Element Theory. This kind of methodology is largely applied in eolic rotor as Mesquita et al. (1999) shows for the studies of hydrokinetics machines, with some adaptations. In this paper, this theory is applied considering a correction made by using a finite number of blades, reaching for the design characteristics that are associated to the geometry of the rotors.

To develop the calculation of the load in one blade, first this blade is divided into a series of infinitesimal elements dr. Then, for each element, a hydrodynamics load balance is performed in different directions.

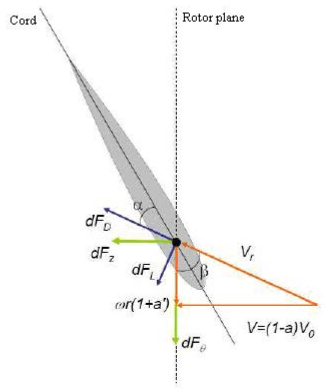

Figure 3. Hydrodynamic loads at the blade element

Figure 3 shows different load components at the blade element. In the axial direction V_z, the absolute velocity of the fluid is given by the velocity decay governed by the induction factor, as expressed at Eq. (4), i.e., $V = V_0(1 - a)$. In the circumferential direction, the tangential velocity of the fluid is given by:

$$V_t = \omega r(1 + a') \tag{32}$$

where a' is the tangential induction factor. The velocity triangle defines the relative behavior V_t and the inlet angle of flow at the blade β. They are written as:

$$\beta = \tan^{-1}(\frac{V}{V_t}) = \tan^{-1}(\frac{1}{\lambda_r(1 + a')}) \text{ where } \lambda_r \equiv \frac{\omega r}{V} \tag{33}$$

$$V_t = \sqrt{V^2 + (\omega r(1 + a'))^2} = V\sqrt{1 + \lambda_r^2(1 + a')^2} \tag{34}$$

The lift and drag forces at the blade elements can be written as:

$$dF_L = \frac{1}{2}\rho V_t^2(dr.Lc)C_L; dF_D = \frac{1}{2}\rho V_t^2(dr.Lc)C_D \tag{35}$$

In this equation, $C_L = C_L(\alpha)$ and $C_D = C_D(\alpha)$ represent the drag and lift coefficients, which are functions of the attack angle α and Reynolds number L_C is the cord length of the blade element.

The attack angle is related to the inlet angle of the flow in the blade:

$$\beta = \alpha + \gamma \tag{36}$$

where $\gamma = \beta_m(r) - \alpha_0$ is the blade setting angle.

The load components of lift and drag can be projected in the axial and circumferential directions Z and θ:

$$dF_Z = dF_L \cos\beta + dF_D \sin\beta; dF_\theta = dF_L \sin\beta - dF_D \cos\beta \tag{37}$$

Integrating the load in each blade element and using Eq. (35), the relationship for thrust and torque are determined by:

$$F_Z = \frac{N_p \rho L_C}{2} \int_{d/2}^{D/2} V_r^2(C_L \cos\beta + C_D \sin\beta)dr \tag{38}$$

$$T = \frac{N_p \rho L_C}{2} \int_{d/2}^{D/2} V_r^2(C_L \sin\beta - C_D \cos\beta)dr \tag{39}$$

Therefore, the power output can be obtained by:

$$P = \omega T \tag{40}$$

Using Eq. (39) and the power coefficient definition intrinsic in Eq. (2):

$$C_p' = 2\sigma_R \lambda' \int_{R_0}^{1} [1 + \lambda_r^2 (1 + a')^2](C_L \sin\beta - C_D \cos\beta) r^* dr^* \tag{41}$$

where $R_0 \equiv (\frac{d}{2R})$; $r^* = \frac{r}{R}$ and $\sigma_R = \frac{N_p L_C}{2\pi R}$ is the solid angle of the blade.

5.1 Angular Momentum Balance

At this point, an additional equation is necessary to determine the tangential induction factor a'. Then, for each radial position "r", an angle β and a relative velocity V_r are calculated, making possible the full resolution of the integral, given in Eq. (41).

According to the tangential velocity in the rotor, the balance of the angular momentum along the fluid, which passes through the actuator disc formed by the blades of rotor, can be consider. The angular momentum is given by the product between the mass flow rate, the angular velocity variation and the radius. The result is equivalent to the torque generated by the dr element of all blades. Then, taking the following torque expression as stepping stone:

$$\delta T = \rho A_r V 2a' \omega r^2 \tag{42}$$

where the area is given by $A_r = 2\phi R dr$. The equation above becomes equal to Eq. (37), yielding:

$$\rho(2\phi r dr) V 2a' \omega r^2 = N_p (dFL \sin\beta - dFD \cos\beta) r \tag{43}$$

Reorganizing the terms:

$$\frac{a'}{1 + a'} = \frac{\sigma_r}{4 \sin\beta \cos\beta} (C_L \sin\beta - C_D \cos\beta) \tag{44}$$

The drag and lift coefficients, used at the load formulation, are obtained by empiric curves from hydrofoil experimentation. Those coefficients are corrected by relations who consider the grid arrangement. The present work uses NACA 0012 profiles.

6. NUMERICAL COMPUTATION

The numerical computations of the model, described in the previous sections were performed. The main goal of this algorithm is to evaluate the effective power output produced by the hydrokinetic turbine considering river velocity, angular rotation speed and geometry of the rotor blades. For the torque and power calculation, the force vector decomposition for each blade element is made. The decomposition is made at the tangential direction, which is the responsible for torque application, and at the axial direction, responsible for thrust application. The following steps describe this algorithm:

- Given a river velocity V_0, geometry of the rotor blades and the rotation of the machine ω, the velocity triangles are determined on the rotor inflow and outflow sections.
- The pressure recovery coefficient and ratio area of the diffuser are computed using the Eq. (23).
- The velocity V is computed through the Eq. (4), giving an estimative of the power coefficient and axial flow induction factor "a".
- The rotor speed ratio (λ) can be obtained through the Eq. (5).
- Using the velocity V, calculated in the previous steps, and considering the blade element theory, for each radial blade position, the relative velocity (V_r), given by Eq. (34) and the incidence angle α, given by Eq. (36), are calculations.
- Considering the incidence angle on the inflow section, the lift and drag coefficients are obtained by interpolation from the experimental data.
- The torque generated by this machine is thus estimated by Eq. (39), making possible the estimated of power output, Eq. (40).

- Using this power output, the power coefficient (C'_p) is calculated based in the velocity V, Eq. (6) and the local rotor speed ratio (λ'), Eq. (5).
- A relationship between the induction factor "a" and the other operational parameters of the turbine, given by Eq. (28), is solved iteratively with the last computation of (C'_p), in order to obtain a equilibrium state of the turbine operational point (C'_p, a).
- The power coefficient (C_p) and the rotor speed ratio (λ) are calculated, using the Eq. (31) and the Eq. (3), respectively. This way, the power output (P), are obtained multiplying the torque, Eq. (39) by the machine rotation.

7. RESULTS AND DISCUSSIONS

The simulations are performed for a prototype of hydrokinetic turbine with diffuser, four-blade propeller like turbine and external diameter of 0,6 m and an internal hub with diameter of 0,3 m. The rotor blades of this machine have been constructed using the NACA 0012 family of hydrofoils.

The intrinsic power coefficient (C'_p), presented in the Fig. 4, using the nonlinear model, reproduces the real behavior of the axial rotor. This simulation was accomplished through geometric data of rotor, certain mass flow conditions and rotation. Which were obtained for a rotation rated varying between 30 and 160 RPM.

To verify the performance of the turbine. The variation of the power coefficient (C_p) for different rotation of the machine, which characterizes different values of λ, is presented in Fig. 5. It can be verified that the best results of the power coefficient of this turbine with diffuser are in the interval between $\lambda = 0,9$ to $\lambda = 1,5$, with the maximum (C_p) of 0,54.

It is possible to observe the difference of the values among the intrinsic power coefficient (Figure 4) and the power coefficient (Figure 5). This difference is related to the fact that in the second graphic the power coefficient of the entire machine is considered. Here, all the losses are considered, as for instance, the velocity correction through the turbine for the axial induction factor, Eq. (4). It is noted the difference from the first graphic, where is considered the power coefficient only in the rotor.

The predicted power output for different river stream is plotted in Fig. 6. It can be observed that power output is near the project of this machine, around 600 W. It is observed when the rotation is increased to 80 RPM, the power also is increased at the maximum. This maximum value of the power is related with the ideal values of speed rotation of the machine, taking into account the different flows speeds.

The flat pattern observed at the power and power coefficient graphics and explains its application with variable rotation, since that the power is maintained around of maximum at a rotation interval from 60 to 90 RPM.

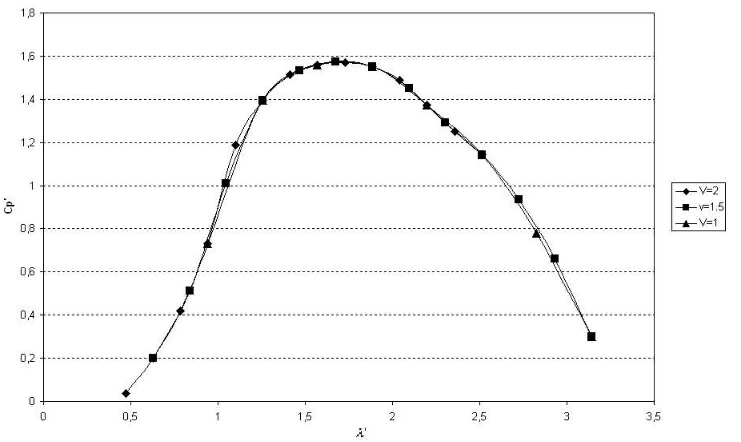

Figure 4. Intrinsic power coefficient

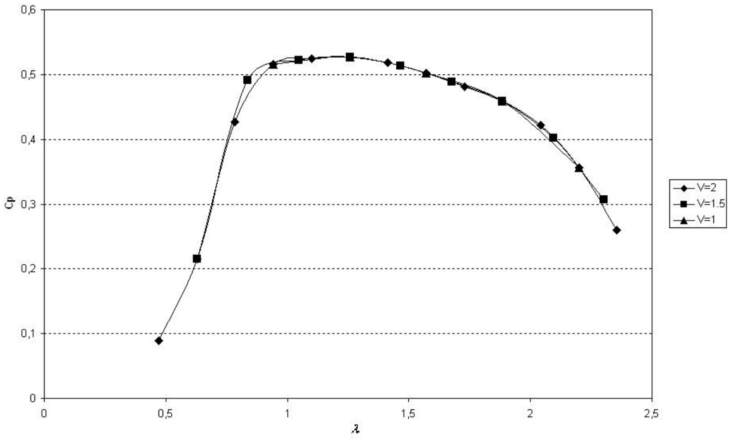

Figure 5. Power coefficient

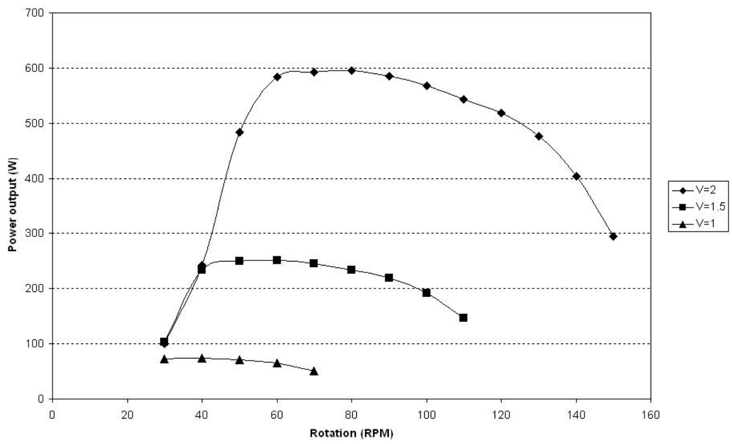

Figure 6. Power generated

8. CONCLUSION

A simplified hydrokinetic turbine was simulated according with the presented theory. The results obtained through the implementation holds a good agreement degree with the project conditions of the machine. This fact demonstrates the viability of the methodology used for the behavior modeling of this turbine.

These results also demonstrate the viability of the use of this machine type to provide electricity for isolated communities located beside the river. Therefore, this technology presents as main characteristics the low cost, high reliability and mainly low environmental impact.

According Fig. 5, it is possible to visualize a same tendency for the curves in different river flows. This phenomena shows that the hydrokinetic turbine can be operated in rivers with speeds of intervals from 1 m/s to 2 m/s, assigning the versatility for most of the existent rivers in Brazil.

The simulations presented a power output of 600 W. With that energy, the turbine can supply, during the period of one hour, a house with two bedrooms, room, cooks and bathroom. They can also be added in the energy consumption, some appliances (refrigerator, blender and other house instruments).

With the objective of increasing the capacity of energy generation of this hydrokinetic turbine, for future works, an optimization of the blade profile is proposed. Nowadays, the optimization of the blade profile is accomplished by the attempt and error methodology. An optimization proposal can be accomplished through the application of artificial intelligence techniques, with the use of genetic algorithm.

9. ACKNOWLEDGEMENTS

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11. Responsibility notice

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