RESEARCH ARTICLE OPEN ACCESS

# A comparative study on river hydrokinetic turbines blade profiles

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**ABSTRACT**
Diesel based electricity supply is the common practice in rural and isolated areas in the North of Brazil. The diesel fuel is usually transported from a nearby city as Manaus by river to these isolated communities. During wet seasons and inundations this means of transport is very risky and not usually safe. The hydrokinetic technology is among the promising technologies for most of the Amazon areas because of the large hydraulic capacity and low density population settlements. In this paper the authors propose a cheap hydrokinetic turbine system whose blades are easy to design, manufacture, replace when necessary and its operation is independent of flow direction. In this work CFD, RANS (Reynolds Average Navier Stokes) equations are used to characterize and develop a methodology of numerical simulation of a vertical axis hydrokinetic turbine. In the simulations, four blade profiles were investigated. The effects of the number of blades, blade profile and water flow velocity on the turbine torque and power coefficients were presented and discussed.
***Keywords –*** renewable energy, hydrokinetic turbine, vertical axis turbine, circular arc profile; flat plate profile; NACA profiles.

## I. INTRODUCTION

According to United Nations about 1.4 billion people or 20% of the global population – do not have access to electricity and a further 1 billion lack reliable access. Some 2.7 billion people – almost 40% of the global population – rely on traditional use of biomass for cooking. Specifically Brazil, the fifth most populated country in the world with about 200 million inhabitants, like other developing countries faces problems associated with the continuous increase of the population as well as the demand for additional energy supply to cope with the industrial growth and economic activities.

The continuous increasing demand for electric energy, the necessity of avoiding aggressions to the environment, geographic difficulties of extending electricity transmission lines across the dense Amazon forests and extensive number of rivers are some of the reasons for encouraging the use of renewable technologies especially for remote and isolated areas. Irrespective of the extensive governmental efforts and actions of non governmental institutions to extend energy to these isolated areas, the progress in this direction is marginal and extremely slow.

Among different renewable energy technologies, hydro-power generation seems to be the most adequate solution for providing energy on large and small scales. However, large-scale hydropower plants need large dams, huge water storage reservoirs and in most cases inundate large forest areas creating local dislodgements of the natives, animals and exterminating natural life in the area. Alternatively, the technology of small scale hydropower plants is diverse and different concepts have been developed and tried out with reduced impact on the environment. Hydro kinetic or in-stream turbines have received a growing interest in many parts of the world especially in relation to river applications.Hydrokinetic turbine transforms kinematic energy of water streams acting on a turning rotor coupled to a electric generator, into electric energy.

Although modern wind generators employ almost exclusively axial flow turbines due to their greater efficiency level at high values of tip speed ratio, vertical axis turbines have relevant advantages for hydrokinetic applications: with axis oriented vertically they can directly drive a generator above water level, suitable application at small depths, reliable application in lower flow rates,flexibility for operation in channels where the directionof the water current is difficult to characterize and change a lot because they are insensitive to changes in flow direction, easy stacking and enlargement of the generation unit. These factors overcome some of the issues of axial flow turbines intended for generating electric power from water currents, depicted by [1], expressly the issue regarding the flow rate, crash with debris at high tip speed ratio, fluctuation in current velocity and etc.

The Savonius rotor is simple in design and easy to manufacture at low cost. The basic driving force of Savonius rotor is the drag force difference between a concave surface facing the incoming flow and a convex surface of the same geometry facing the flow. This drag difference produces torque and causes rotation of the rotor.

Although, according to [4], Darrieus type hydrokinetic turbines (HKTs) with fixed pitch blades exhibit poor starting torque whether blades are straight, troposkein or helical and straight blade turbines have been observed to shake due to cyclic hydrodynamic forces on the blades, the manufacture simplicity of those hydrokinetic turbines has been relevant due to the present application.Some possibilities have been studied to reduce these problems, notably the profile of the blades and the increase of the solidity may decrease the fluctuations of the hydrodynamic forces on the blades. Blades with variable pitch angle are also analternative to reduce shaking of the blades and still maintain strong starting torque and high peak power coefficient. Turbines connected to generators which can also operate as motors can be motored up to speed, so the lack of low speed torque is not necessarily a serious problem, but the complexity of the system is increased.

The objective of this paper is to investigate possible blade profiles which are cheap, easy to manufacture, replace and are adequate for hydrokinetic turbines application in the Amazon rivers. Four geometries were investigated: flat plate profile, circular arc profile, NACA 0018 profile and NACA 1548 asymmetric profile. The influence of the blade profile, number of blades, tip speed ratio and vertical current gradients on the torque coefficient, power coefficient and fluctuation of these parameters are verified in order to define the best configuration regarding efficiency and cost.

## II. MATERIALS AND METHODS

### 2.1. Vertical Axis Hydrokinetic Turbine (VAHT)

The torque coefficient and the power coefficient, two important coefficients in the analysis of hydrokinetic turbines performance are determined by:

$$C_T = \frac{T}{\frac{1}{2}\rho {V_o}^2 S_{ref\ R}} \tag{1}$$

$$C_p = \frac{P}{\frac{1}{2}\rho {V_o}^3 S_{ref}} = \lambda C_T \tag{2}$$

where T is the torque developed by the device, ρ is the specific mass of water, $V_0$ is the upstream velocity of incoming water, $S_{ref}$ is the cross section area of the disc, R is the radius of the rotor. P is power developed by the device, while λ is the tip speed ratio.

According to Betz Limit, which is independent of the turbine design, the maximum power coefficient developed by an ideal actuator disk in a open flow condition is 59.3%, for a speed ratio (ratio between downstream and freestream velocity)of 1/3. The power coeficient limit and its behavior with the speed ratio are usually taken as a baseline for the design of wind and hydrokinetic turbines.

Some mechanisms are used to achieve power coefficients higher than the limit determined by Betz. One of these mechanisms is the use of the so called ducted turbine, which increases the power density by means of a larger duct entrance and/or use deflector plates to avoid undesired water flow effects, such as net reverse torque acting on some of the blades.

Although the efficiency of vertical axis turbines is usually lower in comparison with horizontal axis turbines, the vertical axis rotor arrangement offers some advantages for hydrokinetic applications: vertical axis enables direct drive of a generator above water level, insensitive to changes in flow direction, a row of turbines on a common horizontal shaft can sweep a wide, shallow channel and they do not need to yaw in a reversing tidal flow, low cost untwisted and uniform blade cross section, less impact on the aquatic life due to its reduced rotational speed and larger internal empty space.

In general, the river currents offer considerably less energy than ocean currents and therefore, should have simple solutions for generators, and speed control. Power transmission and generator of vertical axis turbine can be assembled above water level and this facilitates the design, operation and maintenance of the system, [2].

Although VAHTs are relatively simple devices with fixed geometry blades rotating about a vertical axis, Fig. 1, the surrounding flow is complex. As the turbine rotates, the blade elements encounter their own generated wakes and those generated by other elements. The aerofoil section experiences a variation of incidence and strong unsteady effects in the flow field, [9]. These facts do not make possible the use of simpler methods to analyze in details the flow near the blades of a VAHT. Computational Fluid Dynamics is a viable tool to ensure accuracy and detailed characterization of VAHTs.

The drag and lift forces on the blade will generate the torque around the turbine shaft:

$$T = R(L\cos\alpha - D\sin\alpha) \tag{3}$$

where L is the lift force, D is the drag force and α is the angle of attack.

The lift (L) and drag (D) forces arise from the pressure and shear stress differences on the surface of the blade. The profile of the blade and its angle of attack determine the lift and drag coefficients.

Four geometries were investigated: flat plate profile, circular arc profile, NACA 0018 profile and NACA 1548 asymmetric profile each has a chord length of 180 mm while the flat plate and circular arc profiles have a thickness of 5 mm. To investigate the effect of number of blades on the performance of the hydrokinetic turbine three rotor configurations of three, five and seven blades were analyzed. For turbine radius of 0.5 m, the solidity for the case of seven blades is 0.4 against 0.17 and 0.29 for the cases of three and five blades, respectively. The investigated profiles are presented in Figs.2-5. The solidity is defined as a function of the number of blades, chord length and turbine radius:

$$\sigma = \frac{N_c}{2\pi R} \qquad (4)$$

where N is the number of blades, c is the chord length and R is the turbine radius.

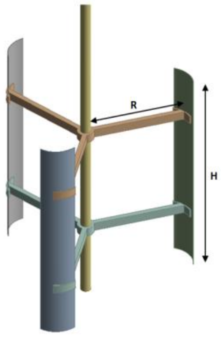



Fig.1-Vertical Axis Hydrokinect Turbine (VAHT)

### 2.2. Blade profiles

The flat plate profile, Fig. 2, is a simple geometry and although theoretically it has the same aerodynamic lift per unit angle of attack, in practice it is not efficient and stalls at relatively small angle of attack.. The circular arc profile, Fig.3, is also of simple geometry, can easily be manufactured. The NACA 0018 shown in Fig.4, has thickness of 18% and is well used in windmills, its geometry and manufacturing process are much more complex than first two profiles. The NACA 1548 profile, Fig.5, is asymmetric profile of complex geometry and is difficult to manufacture.

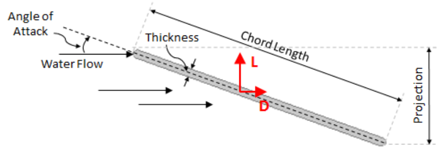



Fig.2-Flat plate profile

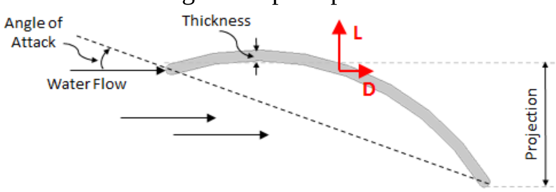



Fig. 3-Circular arc profile.

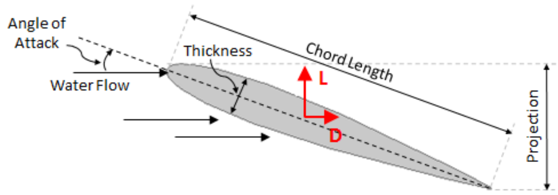



Fig.4-NACA 0018 profile

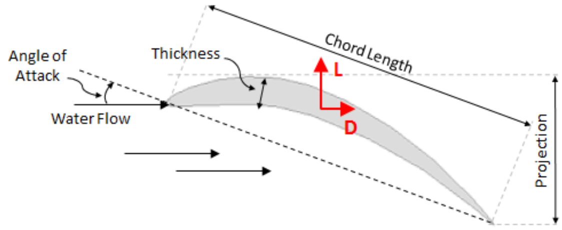



Fig. 5-NACA 1548 profile.

### 2.3. Methods

To minimize the computational costs and at the same time capture the variation in the velocity and pressure fields in the fluid, the region ahead of the turbine was minimally extended while the computational domain behind is 30 R where R is the radius of the rotor. Numerical trials were realized to optimize the computational grid and as a result we used one million of elements where 600,000 elements were used for refinement near the blade and 400,000 elements were used in the general domain and in the layers near the blade wall. Numerical details were omitted for brevity.

### 2.4. Determination and comparison of lift and drag coefficient

Simulations were realized for the NACA 0018 profile and the results were compared with results of [7] for Reynolds number of 10000 and a range of angle of attack from 0$^o$ to 180$^o$. The agreement between the numerical CFD predictions and the results extrapolated by Sandia is reasonably good, Fig.6, showing an average total difference of 6.8% in case of the drag coefficient and 1.1% in case of the lift coefficient.

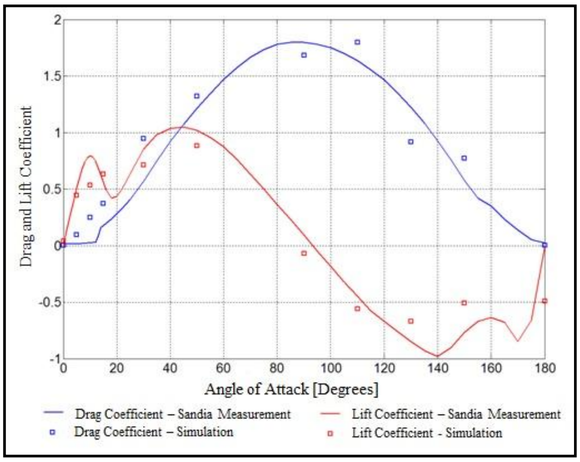



Fig. 6 Comparison of the numerical CFD predictions with the results extrapolated by Sandia National Laboratories (1981)

## III. RESULTS AND DISCUSSION

### 3.1. Simulations of rotor with single blade element

CFD simulations were realized for turbine configurations with single blade to establish the basic characteristics of the investigated profiles. For each simulation the profile was positioned at certain azimuth angle θ with respect to the reference position θ=0°.These simulations were realized in stationary regime at twelve equiangular positions, $\Delta\theta = 30^{o}$. The variations of coefficient of torque calculated during one revolution of the single blade rotor in terms of the azimuth angle (only six positions are presented because of symmetry) are plotted in Fig. 7 for the four blade profiles. The pressure distribution around the flat plate profile is nearly equal producing almost the same torque but opposite in direction which results in an average low torque coefficient and for this reason is considered inadequate profile for this application.

The circular arc profile, NACA 0018 and NACA 1548 show asymmetric torque coefficient behavior during one complete rotation of the turbine rotor producing a net torque coefficient.

In the case of the circular arc profile the hydrodynamic forces acting on the concave region of the profile are much more than the hydrodynamic forces acting on the convex region which results in increasing the average torque in comparison with the case of flat plate profile. Similar effects are found in the cases of the NACA symmetric and asymmetric profiles with resulting net torque smaller than in the case of circular arc profile.

The above comments can be easily confirmed from the flow and pressures distributions for each profile.

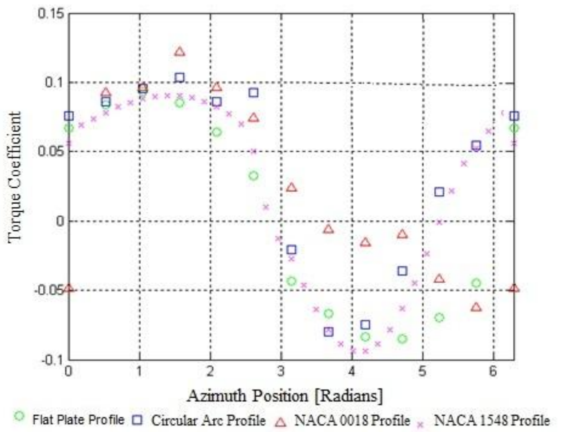


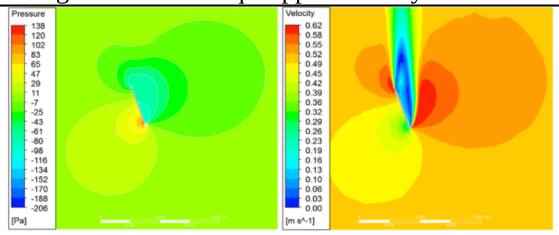

Fig. 7 Variation of the coefficient of torque with the azimuth angle for static single blade profiles.

The pressure distributions around the flat plate profile for azimuth angles and their corresponding opposite's angles produce torques of nearly the same magnitude and opposite in direction and consequently reducing the overall torque of the device. Pressure and velocity distributions around the flat plate profile are shown in Fig. 8 for two azimuth angles for brevity.

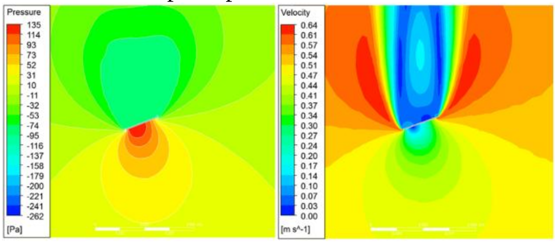
The circular arc, NACA 0018 and NACA 1548 profiles show different pressure distribution at opposite positions (for example 90º and 270º) during the period of one revolution and consequently there is an average positive torque. The hydrodynamic forces produced when the fluid is facing the concave region are considerably bigger in comparison with the opposite position when facing the convex region as can be seen from the pressure distributions around the circular arc profile shown in Fig. 9 (a) and (c).

Similar results are found for the symmetric profile NACA 0018 and the asymmetric profile NACA 1548 where the pressure distributions during one revolution are different resulting in a net positive average torque. The simulation results and the corresponding pressure distributions are presented in Figs.10-11.

Obviously, the pressure distribution on a flat plate profile will match for opposite positions, leading to resultant torqueapproximately null.

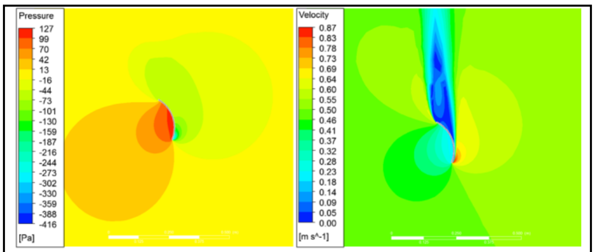



a) Distributions of pressure and velocity for the flat plate profile for = 0º.

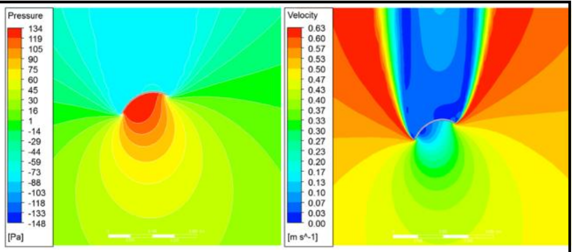



b) Distributions of pressure and velocity for the flat plate profile for = 90º.

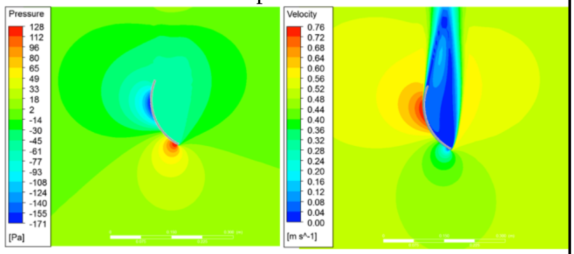
Fig.8 Pressure and velocity distributions around flat plate profile.




a) Distributions of pressure and velocity for the circular arc profile for = 0º

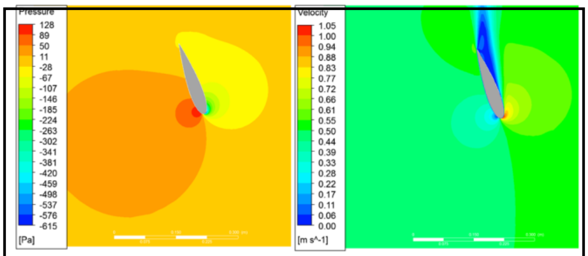


(b) Distributions of pressure and velocity for the circular arc profile for  = 90º.

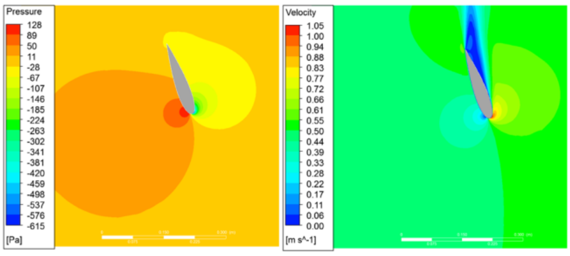


(c) Distributions of pressure and velocity for the circular arc profile for  = 180º.

Fig. 9 Pressure and velocity distributions around circular arc profile.

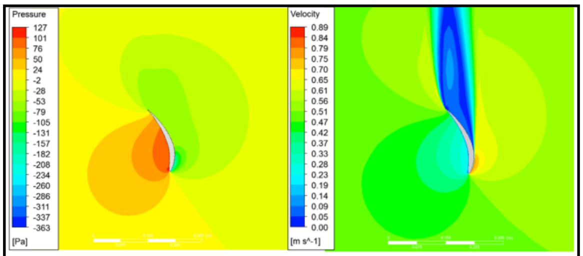


(a) Distributions of pressure and velocity for NACA 0018 when  = 0º.

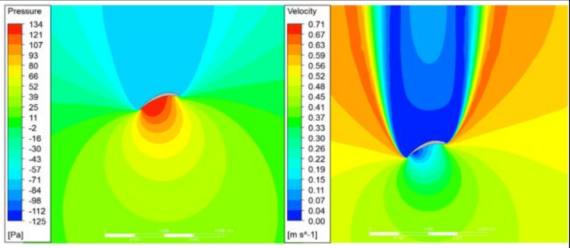


(b) Distributions of pressure and velocity for NACA 0018 when  = 90º.

Fig.10 Pressure and velocity distributions around NACA 0018 profile.



(a) Distributions of pressure and velocity for NACA1548 when  = 0º.



(b) Distributions of pressure and velocity for NACA1548 when  = 90º.

Fig.11 Pressure and velocity distributions around NACA 1548 profile.

The average value of the torque coefficient is evaluated by integrating the torque during one revolution of the turbine using

$$\overline{C_T} = \frac{1}{\theta_2 - \theta_1} \int_{\theta_1}^{\theta_2} C_T(\theta)\, d\theta \qquad (4)$$

It is found that the average torque coefficient for flat plate profile is 0.003, the circular arc is about 0.036 while the values for NACA 0018 and NACA 1548 are 0.027 and 0.020, respectively.

One more important parameter is the amplitude of variation of the torque coefficient. From Fig.12 the amplitude of variation of the torque coefficient for the case of flat plate profile is approximately 0.178 while those of the circular arc, NACA 0018 and NACA 1548 profiles are 0.183, 0.185 and 0.185, respectively. These values together with the average torque coefficient can be used to characterize the investigated profiles.

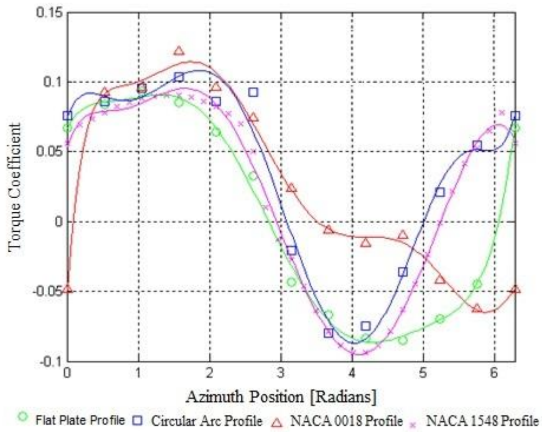



Fig.12 Torque coefficients for single blade rotor of different profiles.

### 3.2. Prediction of rotor performance from the superposition of single blade element

The torque produced by a rotor composed of multi blades can be approximately estimated by the superposition of the results of single blade element calculated before. This approximation neglects the interaction and mutual interference effects between blades, leading to a overestimated result.

Fig.13 shows the results of simulations of the four investigated profiles where a turbine with three blades is determined by the superposition of the results of the single blade element to predict the performance of the rotor with three blades elements. As can be seen the flat plate profile shows an average torque coefficient of 0.005 and amplitude of variation of torque coefficient of 0.076 while the circular arc, NACA 0018 and NACA 1548 profiles show average torque coefficient values of about 0.100, 0.087 and 0.060 and amplitude of variation of torque coefficient of about 0.041, 0.109 and 0.031, respectively. This result confirms that the circular arc profile has better performance than any of the other profiles.

The same tendencies are found in the case of rotors of five blades determined by the superposition of the single blade element results, shown in Fig.14. As can be seen the flat plate profile is found to have a torque coefficient of about 0.009; while the circular arc, NACA 0018 and NACA 1548 have 0.167, 0.144 and 0.100 successively. Following the same order, the amplitude of variation of the torque coefficient is found to be 0.050, 0.034, 0.078 and 0.027. Fig.14 shows higher torque coefficient and smoother cyclic variation in comparison with Fig.13 due to the increase of the number of blades

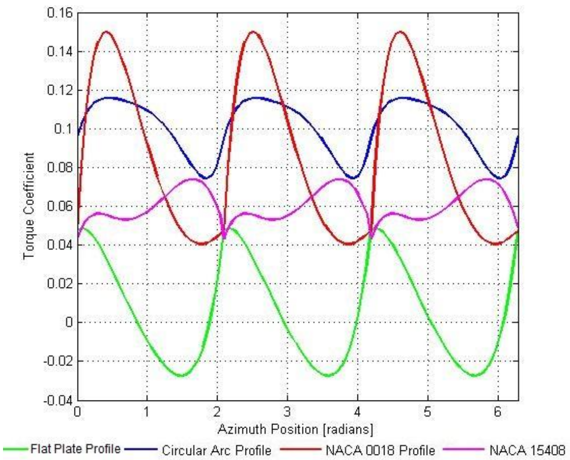



Fig.13 Torque coefficients for rotors with three superimposed blades for all investigated profiles.

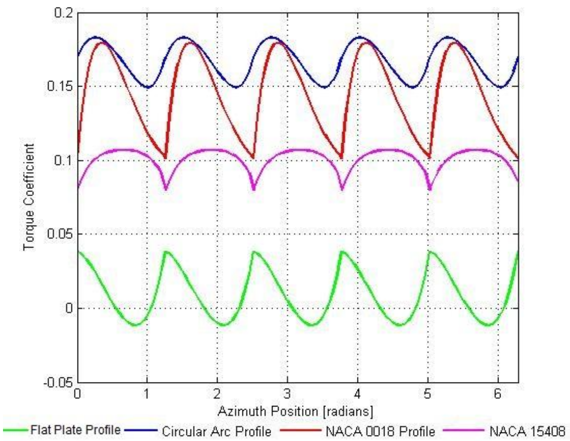



Fig.14 Torque coefficients for rotors with five superimposed blades for all investigated profiles.

Similar results are found in the case of rotors of seven blades, Fig.15, where the flat plate profile is found to have a torque coefficient of about 0.013; while the circular arc, NACA 0018 and NACA 1548 have 0.234, 0.202 e 0.141, respectively. The amplitude of variation of the torque coefficient is 0.035 for the flat plate profile, 0.020 for the circular arc profile, 0.061 for NACA 0018 profile and 0.025 for the NACA 1548 profile.

The increase of the number of blades, as shown in Fig.16, causes increasing the average torque coefficient and decreases the amplitude of variation of the torque coefficient and consequently the dynamic and vibration effects on the rotor blades are reduced.

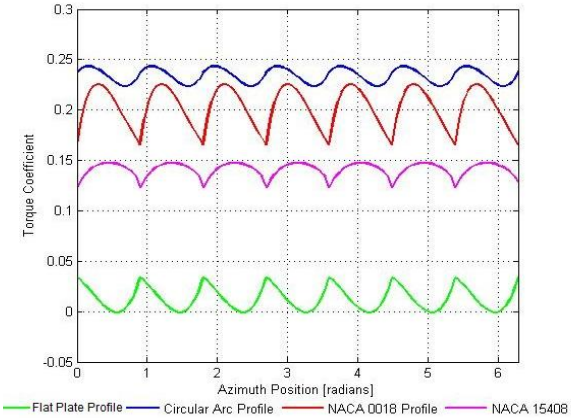



Fig.15 Torque coefficients for rotors with seven superimposed blades for all investigated profiles.

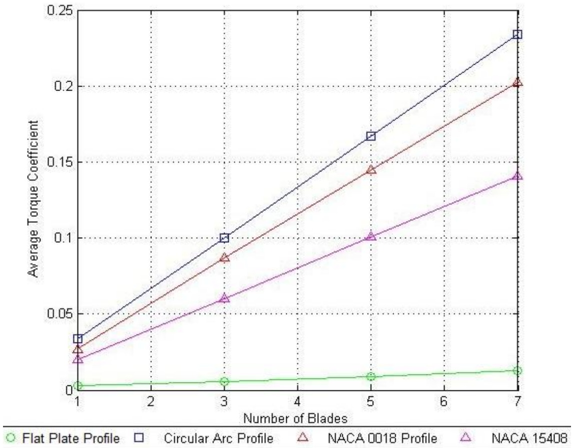



Fig.16 Variation of the mean torque coefficient with the number of blades for the investigated profiles.

### 3.3. Static simulation of rotor with three, five and seven blades

The approximate method used to predict the performance of the rotor composed of multi blade elements based on the results of single blade can reduce the computational time but unfortunately is not accurate enough since it does not include the interaction and interference effects caused by the wakes of the preceding blades. A better way of estimating the rotor performance is to use in the simulations rotors having the required number of blades so that the effects of interference and interaction are included in the analysis. Due to the high computational time and costs the cases of flat plate and the NACA 1548 profiles were not included in the simulation since they did not present satisfactory torque coefficient in comparison with the circular arc and NACA 0018 profiles.

Fig.17 presents a simulation for a three blades rotor when θ=0, where blade number 3 affects the flow conditions of blade number 2 and consequently the torque produced by it. The simulations were realized for an interval corresponding to 120$^o$ and for increments of $\Delta\theta = 5^o$. As can be seen from Fig. 18 the rotor with circular arc blades shows better performance than that of the NACA 0018 profile which confirms earlier findings. The average torque coefficients calculated for the circular arc profile is 0.070 and that of NACA 0018 profile is 0.026.

As is expected the average torque coefficient for multi blade rotors predicted from the method of superposition are different from the static simulation results due to interference effects caused by other blades provoking reduction in the average torque. Figs.19 and 20 show comparison between the predictions of the simulation of the complete rotor with the predictions from the method of superposition for the cases of circular arc and NACA 0018 profiles. As can be seen the superposition method predicts higher average torque because of not including the interaction and interference effects.

In the case of simulating a turbine with five blades the interference effects are more severe due to the larger solidity as can be seen in Fig.21. The average torque coefficient in the case of circular arc profile is 0.111 while that of the NACA 0018 is 0.037 as in Fig.22.

In a similar manner the rotor with seven blades of circular arc profile is superior to the rotor with seven blades NACA 0018 profile. The average torque coefficient is 0.186 for the circular arc rotor against 0.031 for the NACA 0018 profile rotor as can be verified from Fig. 23.

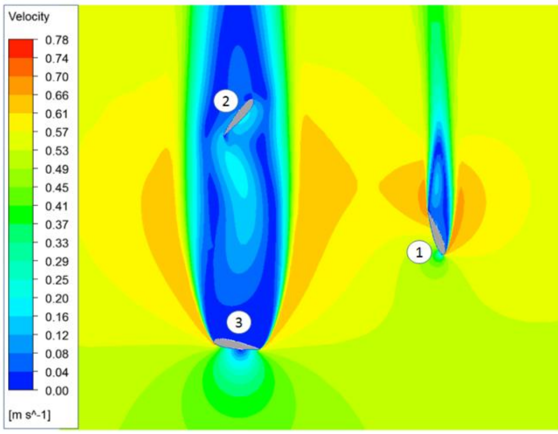



Fig. 17 Velocity distribution for a rotor with three blades.

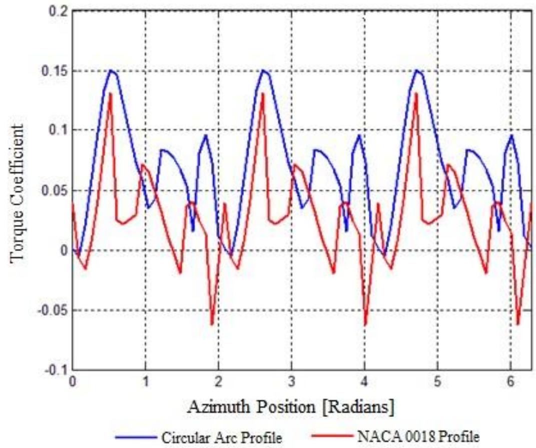



Fig. 18 Variation of torque coefficient with the azimuth angle for two rotors with three blades.

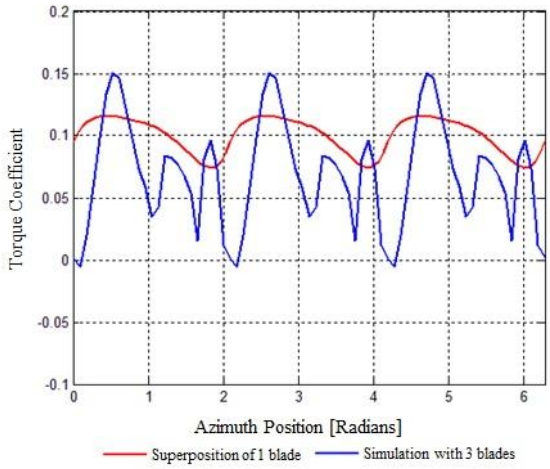



Fig.19 Comparison of the three imposed blade rotor with the three blade arrangement rotor (circular arc profile).

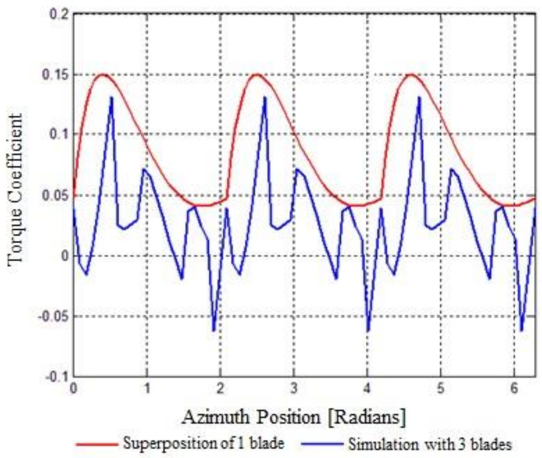



Fig.20 Comparison of the three imposed blade rotor with the three blade arrangement rotor (NACA 0018 profile)

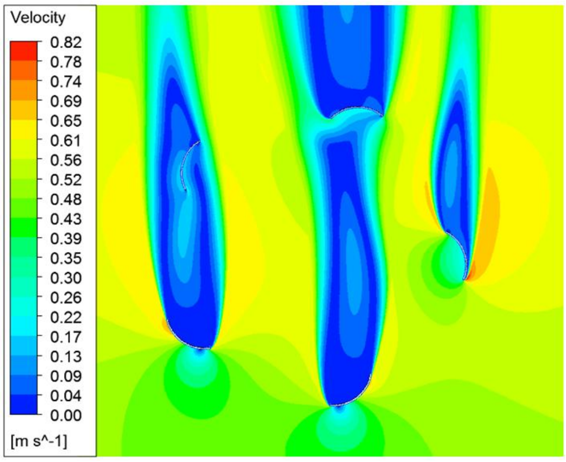



Fig.21 Velocity field around a rotor with five blades.

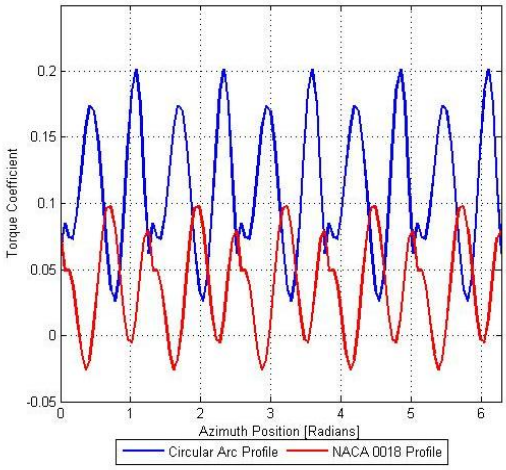



Fig. 22 Comparison of the predicted torque coefficient for rotors with five blades of circular arc and NACA 0018 profiles.

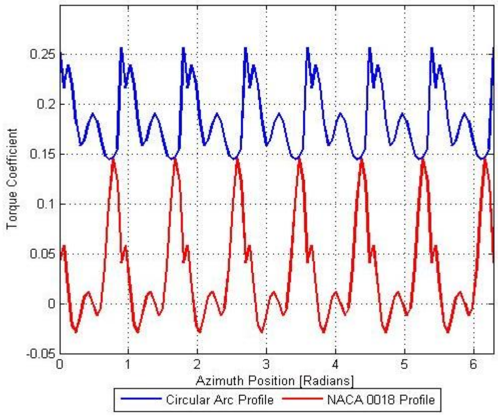



Fig.23 Comparison of the predicted torque coefficient for rotors with seven blades of circular arc and NACA 0018 profiles.

### 3.4. Effect of the vertical gradient of water velocity

It is known that there are vertical velocity gradients in the Amazon Rivers. If a hydrokinetic turbine is installed in these rivers, its blades will be subject to velocity distribution in the vertical direction and consequently will produce global average torque different from the case of uniform velocity distribution.

To investigate this condition and its effect on the performance of the turbine and the average torque coefficient a linear vertical velocity gradient is assumed for simplicity, as in Fig.24. The elementary torque generated by an element of the blade is given by $dT = C_T.(\rho V_o^2 R^2 dh)$ and the total torque is obtained by integrating the elementary torque over the blade length to obtain the final expression as

$$T = C_T \rho R^2 \left( a^2 H + abH^2 + \frac{b^2 H^3}{3} \right).$$

This formulation was used to simulate a seven blade rotor of circular arc profile of 500 mm radius, 1000 mm length subject to a velocity gradient varying from 1.6 to 2.0 m/s along the blade length. As a result, the simulation the average torque coefficient was found to be 0.186 or 150 Nm, while if the incoming velocity was constant at 2.0 m/s the average torque would be 185 Nm.

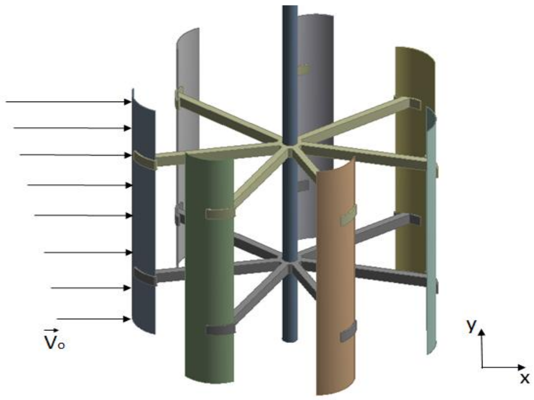



Fig. 24 A hydrokinetic turbine in a flow with vertical velocity gradient.

### 3.5. Dynamic simulation of a rotor with seven circular arc blades

Simulations realized until the moment did not account for the rotor rotation with reference to the incoming fluid flow. The inclusion of rotor rotation adds some difficulties and needs longer computational time. To investigate the dynamic effects due to rotor rotation, simulations were realized for the case of circular arc profile with tip velocity ratio of $\lambda$=1.25 and for the cases of rotors of three and seven blades. Fig. 25 shows the velocity field for the rotor of seven blades. The torque coefficient for the two rotors is shown in Fig.26 for rotor tip speed ratio of $\lambda$=1.25. Comparison of the results of Fig. 26 with the simulation results of the static rotor, shows differences of over 35%. This shows that in order to obtain a realistic estimate of the rotor performance it is important to realize dynamic simulations where the rotor rotation with reference to incoming fluid flow is taken into account.

Additional simulations were realized for the rotor with seven circular arc blades for different values of tip velocity ratios as shown in Fig. 27. As can be seen the maximum power coefficient occurs at tip velocity ratio of 2.5.

As an example, a turbine of radius of 500 mm, 1000 mm blade length, seven blades of circular arc profile, tip velocity ratio of 2.5 and free water velocity of 2.0 m/s will produce a power of about 1.6 kW.

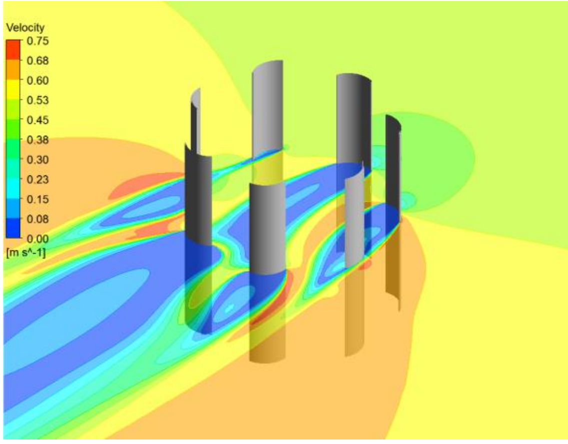



Fig.25 Velocity field around a rotor with seven blades.

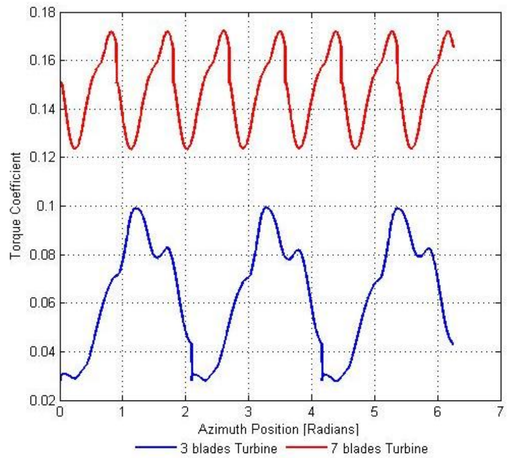



Fig. 26 Comparison of the torque coefficients for rotors of three and seven blades when $\lambda$ =1.25.

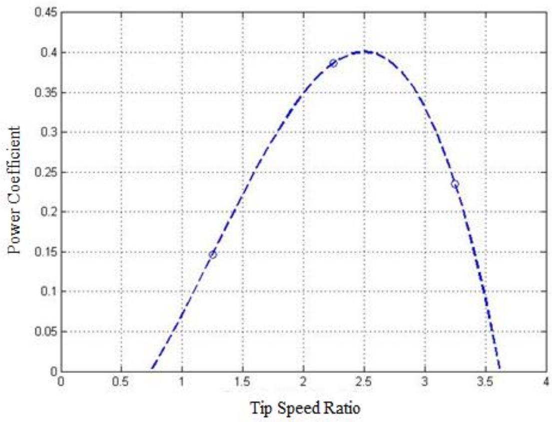



Fig. 27 Variation of the power coefficient the tip speed ratio.

## IV. CONCLUSIONS

The main objective of the paper is to investigate possible blade profiles adequate for hydrokinetic turbines for utilization in the Amazon rivers. The hydrokinetic turbine must have easily manufactured blades, easy to install, replace and maintain. Four blade profiles were investigated, that is, flat plate, circular arc, NACA 0018 and NACA 1548. CFD simulations showed that circular arc profiles are more efficient and produce more power. Rotors with three, five and seven blades were simulated and it was found that the seven blades arrangements can produce about 1.6 kW sufficient for the consumption of seven average Brazilian homes. It is hoped that this preliminary investigation can stimulate further and deep investigations and can be useful for decision makers.

## Acknowledgements

The authors wish to thank the CNPQ for the PQ Research Grant to the first author.

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